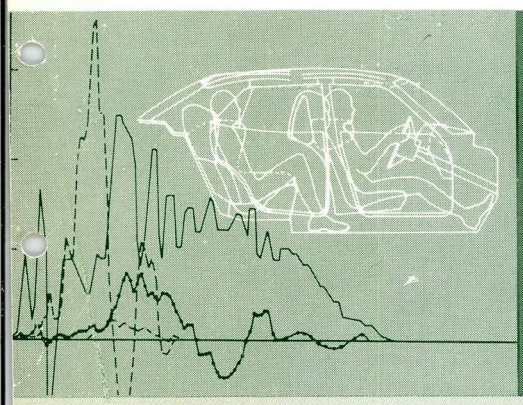
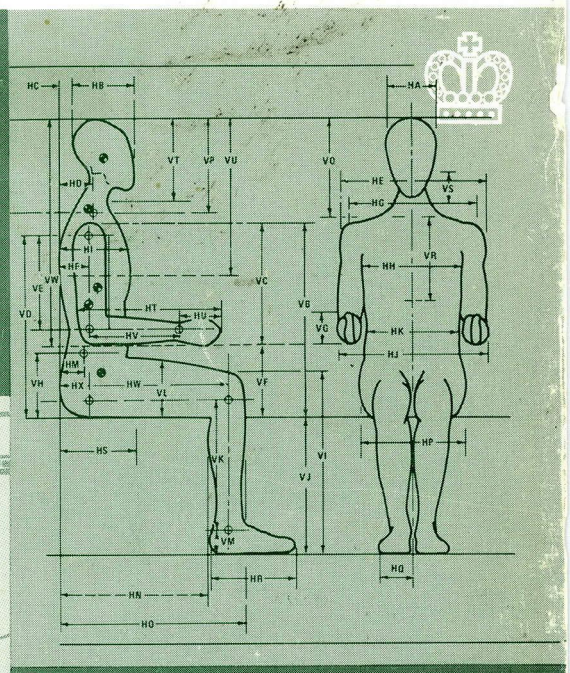
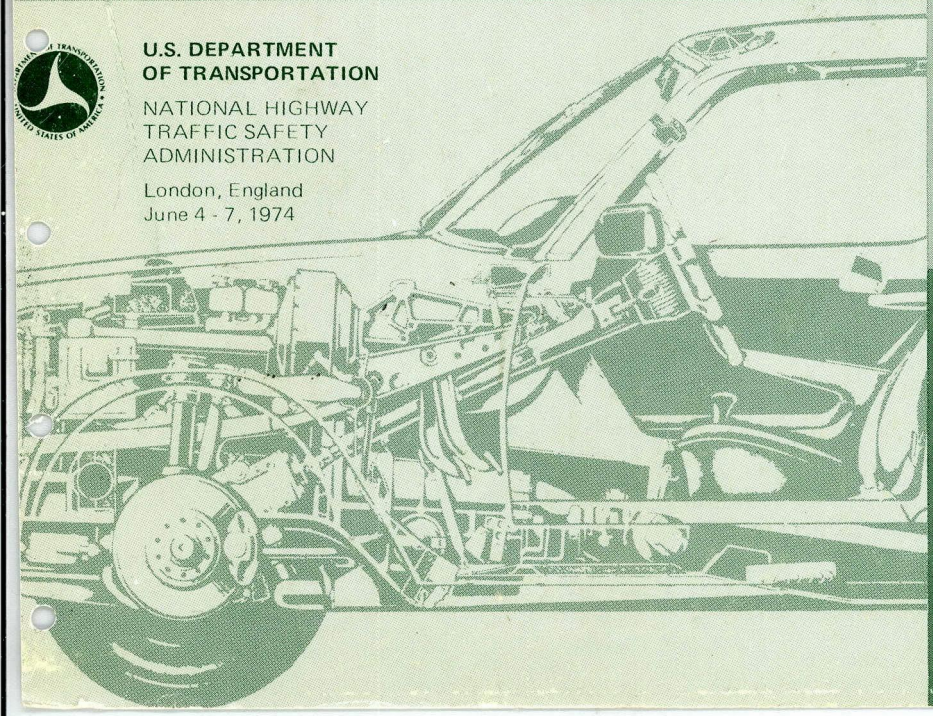
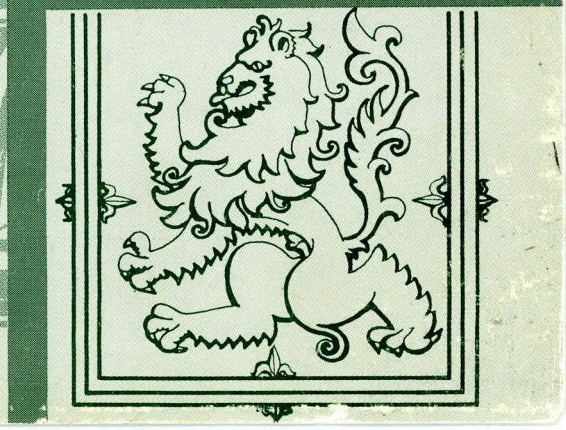
Report On The Fifth International Technical Conference On Experimental Safety Vehicles









U.S. DEPARTMENT OF TRANSPORTATION

NATIONAL HIGHWAY TRAFFIC SAFETY ADMINISTRATION

London, England
June 4 - 7, 1974

Report On The Fifth International Technical Conference On Experimental Safety Vehicles

London, England
June 4 - 7, 1974

Sponsored by:
The U.S. Department of Transportation

Hosted by:
The U.K. Department of the Environment
and
The Society of Motor Manufacturers and Traders, Ltd.

Held at:
London, England
June 4 - 7, 1974

FIFTH INTERNATIONAL TECHNICAL CONFERENCE ON EXPERIMENTAL SAFETY VEHICLES

U.S. DEPARTMENT OF TRANSPORTATION
NATIONAL HIGHWAY TRAFFIC
SAFETY ADMINISTRATION

For sale by the Superintendent of Documents
U.S. Government Printing Office, Washington, D.C. 20402
Price $10.20 Stock Number 050-003-00210

Foreword

The report of the proceedings of the Fifth International Technical Conference on Experimental Safety Vehicles was prepared by the Office of Vehicle Safety Research, National Highway Traffic Safety Administration, United States Department of Transportation.

The report includes the conference opening remarks, status reports by governmental representatives and their automotive industries, technical papers presented and the discussions held during the seminars on "Accident Analysis," "Human Factors," "Vehicle Structural Properties," "Accident Avoidance," and "Overall Car Requirements," a discussion period on the "Application of Research Results to Standards and Future Activities," and concluding remarks by the United Kingdom and the United States.

For clarity and because of some translation difficulties, a certain amount of editing was necessary. Apologies are, therefore, offered where the transcription is not exact.

Participants

FIFTH INTERNATIONAL TECHNICAL CONFERENCE
ON EXPERIMENTAL SAFETY VEHICLES

AUSTRALIA

Department of Transport
J. A. McKenzie

AUSTRIA

H. Buchinger
Loba Chemie

BELGIUM

Commission of the European Communities
Dr. H. Henssler
Dr. D. Verdiani

Ministry of Communications
P. Nicolas

C.C.M.C.
F. Barthel
D. G. Bissell
M. D. Quin

CANADA

Canada Ministry of Transport
E. R. Welbourne

FEDERAL REPUBLIC OF GERMANY

Adams Opel AG
E. S. Kiefer
L. Kussman
F. Lohr
G. J. Pelz

Allianz-Versicherungs AG
J. M. Danner

A.T.Z.
Prof. W. Rixmann

Audi N.S.U. AG
D. Banholzer
J. Frings
Dr. R. Loebich

B.M.W.
J. K. P. Elsholz
W. Hofmeister
H. J. Kraft
B. Osswald
Dr. K. Radermacher
W. Stork

Robert Bosch, GmbH
Dr. H. Manger
H. Mohl
A. Hammerstein

Daimler-Benz AG
W. Breitschwerdt
Dr. K. Enke
K. Goeschel
W. Heiss
G. Hespeler (U. S. A.)
Dr. W. Reidelbach
Prof. H. Scherenberg
J. H. Sorsche
F. H. Van Winsen
P. Von Manteuffel
Dr. K. Wilfert
W. K. Schmidt

Federal Ministry of Transport
K. D. Burchard
Prof. B. Friedel
A. Gulich
H. Jungblut
Dr. H. Wagner

Dynamit Nobel AG
Dr. K. E. Flach

Flachglas AG Delog-Detag
Dr. G. Rodloff

Ford Werke AG-Cologne
W. Ebers

Porsche AG
U. Bez
H. Bott
Dr. H. H. Braess
W. Eyb
Dr. R. Weber

R.E.P.A. Alfdorf
A. A. Föhl

Sekurit-Glas Union, GmbH
Dr. H. Kunert

Teldix, GmbH
R. O. Peter

Verband der Automobilindustrie E.V.
Dr. G. W. Brenken

Vd. TUV
D. Kleinsteuber

HUK Verband
K. Langwieder

Volkswagenwerke AG
A. Bauer
J. P. E. Bruss
Prof. E. Fiala
J. Haenchen (U.S.A.)
J. Kennebeck (U.S.A.)
A. Konrad
Dr. W. Lincke
A. R. Railton (U.S.A.)
Dr. D. Schade
Dr. H. Schimkat
A. Schuttemayer (Canada)
Dr. U. Seiffert
R. Weissner

FRANCE

Automobiles Peugeot
G. Boschetti
N. Bureau
C. Chillon
J. Genestier
J. Hamon
G. Oberle
Dr. C. Tarriere

Bureau Permanent International des Constructerurs d'Automobiles, (B.P.I.C.A.)
J. L. Phelps

Cibie Projecteurs
J. Rit

Citroen
H. S. Allera
S. R. Bohers
M. F. Clavel
A. Garry
R. Larousse
Dr. M. Palfer-Sollier

Chambre Syndicale des Constructeurs des Automobiles
M. Martin

Dunlop
A. Christin
M. Couturier

Federation Des Industries D'Equipement Des Vehicules (F.I.E.V.)
A. Duchange
R. J. A. Guasco

Institut de Recherche des Transports
M. Frybourg

Kleber-Colombes
R. P. Monier

M.S.C.A.
J. F. Quilici

Ministere de l'Amenagement du Territoire
B. Gauvin
A. Osselet

O.N.S.E.R.
M. Biecheler
J. Bluet
D. Cesari
M. Halpern-Herla
J. Leroy
N. Muhirad

Renault
A. Chenevez
P. H. M. Eelsen
Y. Georges
F. Hartemann
C. Prost-Dame
H. Seznec

P. Ventre
J. C. Maroselli

Saint Gobain Industries
R. Orain

S.A.F. du Ferodo
J. P. Leruez

Societe Nationale des Poudres et Explosifs
P. O. Cord
P. Jenoc
J. P. Thomas

U.T.A.C.
E. Chapoux
H. R. Le Guen
J. P. Thiry

L.E.A.D.
Prof. F. P. Mayer

ITALY

Alfa Romeo, S.P.A.
D. Black (U.S.A.)
Dr. A. Cacciabue
Dr. D. Chirico
Dr. M. Fogliata
Dr. E. Landsberg
Dr. R. Luraghi
Dr. L. Maestri-Pieri
Dr. F. Masciadri
Dr. V. Pagliarulo
Dr. L. Rosti-Rossini
Dr. F. Surace
Dr. L. Vincenti

AVIC S.P.A.
Dr. A. Rispoli

Fiat S.P.A.
Dr. P. L. Ardoino
Dr. M. Barile
Dr. F. Canavesi
Dr. F. Cornacchia
E. Franchini
Dr. G. Gastaldi
V. Montanari
Dr. P. P. Messori
Dr. O. Montabone
Dr. R. Porro
Dr. G. Puleo
Dr. R. Scalet
Dr. L. Zandona

I.E.N.G.F.
P. Soardo

Instituto Sperimental Auto e Motori (I.S.A.M.)
Dr. P. Casucci
Dr. F. Moscarini
G. Rontani

Ministry of Transport
Dr. G. Danese
Dr. G. Pocci
Dr. A. Sirignano

Pinninfarina
Dr. A. Cogotti

Pirelli
Dr. A. Angioletti

Saga
Dr. G. Barone

JAPAN

Asahi Chemical Industry Co., Ltd.
Dr. S. Ikeda
I. Maeda
Y. Okamoto

Bridgestone Tire Co., Ltd.
T. Aoki

Fuji Heavy Industries, Ltd.
R. Matsumoto
S. Momose

Hitachi, Ltd.
T. Tanigami

Honda Motor Co., Ltd.
K. Higuchi
A. Meganck
Y. Nakamura
M. Otabe
S. Sano
T. Sumida

Japan Automotive Parts Industry Association (Ikeda Bussan Co.)
Y. Hamano
Y. Matsuo

Japan Automobile Manufacturers Association
M. Shizume

Japan Automobile Research Institute Inc.
Y. Okami
H. Sakai
M. Sakimura
K. Yokoi

Japan Seat Belt Council (NSK Warner Co.)
J. Banahoya
H. Tachiki

Japan Trade Center, London
T. Ueda

Jidosha Denki Kogyo Co., Ltd.
K. Kobayashi

Ministry of International Trade and Industry
N. Miyake
I. Noguchi

Ministry of Transport
N. Uno

Mitsubishi Electric Corporation
Y. Asahi

Nissan Motor Co.
Dr. Fleming (U.S.A.)
S. Kawazoe
Y. Serizawa
M. Takei

Sumitomo Metal America Inc.
Y. Shibata (U.S.A.)

Takata Kojyo Co., Ltd.
J. Takada

Toyo Kogyo (Brussels)
K. Akahige

Toyo Kogyo Co., Ltd.
S. Mochizuki
M. Yorizane

The Toyo Rubber Industry Co., Ltd.
Y. Shimabara
Y. Takamatsu

Toyota Motor Co., Ltd.
T. Iida
S. Inui
K. Iwata
J. Kawano
E. Kumabe
M. Noguchi
S. Okawa

National Police Agency
S. Nagakura

THE NETHERLANDS

Coenen-Wingard BV
M. J. W. Coenen

Van Dome's Automobile Manufacturers, Inc.
F. Christophe
A. Hoynk van Papendrecht
P. Van Baardwijk

Research Institute for Road Vehicles
H. S. T. Brockhoff
G. Tanis

Road Transport Service
A. Edelman
J. G. Kuiperbak
H. G. Paar
M. P. B. Van Kampen Walraven

SPAIN

Ministry of Industry
A. Gobianas
G. Oistrach
E. Tervel

SWEDEN

Association of Swedish Auto. Mnfrs. & Wholesalers
C. Nordgren

British Leyland, Sweden
G. Elmgren

Department of Traffic Safety
Prof. B. Aldman

National Swedish Road and Traffic Research Institute
O. Nordstrom

Saab-Scania, AG
B. H. Gustavsson
L. E. Nilsson
S. Rundkvist
H. Tornqvist
P. U. Unosson

Swedish Road Safety Office
G. Ekberg

AB Volvo
R. Almqvist
Dr. A. Asberg
Dr. E. G. A. Elgeskog
K. Goranson
P. E. Groenvall
A. Jonason
M. Kling
Dr. L. S. Larsen
R. Mellde
R. A. Rocke
C. Wilen

UNITED KINGDOM

Abacus Industrial Marketing Services
G. A. King

Armstrong Patents, Ltd.
S. J. Twidale

Association of Chief Police Officers
E. H. Post

The Automobile Association
M. A. I. Jacobsen

Automotive Products Co., Ltd.
W. N. Bainbridge
D. Parsons
Dr. D. Stuart

Auto Restraint Systems, Ltd.
R. A. Ashworth
R. Clifford
S. Meisner
A. G. Plackett

British Leyland Motor Corporation
K. Barnett
D. Blume
G. C. Boshier
R. E. Coles
W. J.Daniels
C. J. Drury
Dr. W. C. Emmerson
N. D. Farr
P. Firch
M. Fitzpatrick
J. E. Fowler
D. R. Gebbels
R. W. Goff
C. J. Goode
C. A. Griffin
J. J. Haley
J. R. Hartshorn
C. J. Hawkesford
A. C. Hill
A. A. Hodgetts
D. F. Howes
I. C. C. Johns
G. Jones
W. E. Keatley
C. W. Linder
A. R. Long
J. V. Mapson
J. B. McWilliams
D. P. Peck
L. J. Salkeld
D. W. Seymour
R. G. Shirley
B. Stockdale
C. J. Sykes
H. R. Wall
H. G. Webster
R. D. White
C. E. Wilday
C. L. Woodcock

Cam Gears, Ltd.
F. J. Adams

Cranfield Institute of Technology
Prof. J. R. Ellis

Chrysler International SA
H. J. C. Weighell

Datsun UK, Ltd.
O. C. Botnar
M. J. Hunt

Department of the Environment
W. L. Baxter
H. H. Calvert
N. S. Despicht
G. Donald

J. W. Furness
D. Jennings
J. K. Martin
R. J. A. Sharp
D. J. Spragg
S. M. Stearn
B. Stock

Department of Industry
F. Munns

Department of Trade
J. G. Bennet

Dow Chemical Co., Ltd.
C. Simey

DuPont (UK), Ltd.
G. Sweet

Dunlop, Ltd.
Dr. D. A. Newton

Fort Dunlop
T. French
T. Holmes
I. C. Mills

Enersorb, Ltd.
I. MacArthur

Ford Motor Co., Ltd.
D. B. Burland
F. D. Cooper
E. J. Cutting
M. Rodger
K. Stone
G. D. Suthurst
K. J. B. Teesdale

Girling, Ltd.
P. Oppenheimer

Goodyear Tire & Rubber Co.
J. A. Smith

Hatfield Polytechnic
G. F. Moseley
H. Skelton
B. G. Watson

I.C.I. Fibres
G. Chilton
Dr. P. C. Rhodes

I.C.I. Organic Division
K. A. W. Williams
G. Woods

Imperial Metal Industries, Ltd.
K. Deutsch
M. G. J. Elson
Dr. K. Saxl

Lucas Electrical Co., Ltd.
Dr. J. H. Nelson

Joseph Lucas, Ltd.
W. Waller

3M (UK), Ltd.
R. Waters

Motor Industry Research Association
S. Andrew
M. D. Croker
C. M. Haslegrave
Dr. M. A. Macaulay
Prof. R. H. Macmillan
Dr. J. A. Searle
R. E. Stirley

National Research Development Corp.
Dr. J. R. Cain

Ogle Design
P. S. Warner

Pressac, Ltd.
J. Cullis

Road Research, Ltd.
G. Persicke

Rolls-Royce Motors, Ltd.
Dr. D. G. C. Bacon
H. J. Burnham
J. S. Hollings
J. B. H. Knight

Shell Research, Ltd.
Dr. C. Bonsall
D. G. Jordan

Society of Motor Manufacturers & Traders, Ltd.
K. Barnes
F. Denmee
A. J. Smith
J. Weinthal

Telmar Retarder, Ltd.
L. Hall

Transport and Road Research Laboratory
F. Avann
R. R. Boyce
Dr. J. Fitchie
E. Grattan
Dr. L. G. Griffin
V. J. Jehu
R. D. Lister
I. D. Neilson
J. Prentice
K. Russam
B. E. Sabey
A. Silverleaf
H. Taylor
J. G. Wall
P. M. F. Watson

Triplex Safety Glass Co., Ltd.
A. Cleminson

Kings Norton
S. E. Kay
J. Pashley
J. Pickard

University of Birmingham
S. J. Ashton
P. Bates
Dr. A. B. Clayton
P. F. Gloyns
Dr. H. R. M. Hayes
Dr. M. Mackay
T. Suyama

Vauxhall Motors, Ltd.
J. H. Alden
D. H. Bratton
J. Curtis
R. North
J. Waller

Wolfson Institute of Interfacial Technology
Dr. R. S. Millman

UNITED STATES OF AMERICA

American Safety Equipment Corp.
C. Henderson

Automobile Importers of America
D. Busby

Allstate Insurance Co.
J. E. Martens

Automotive News
H. Kahn

A.M.F., Inc.
A. Roth
W. J. Rup

Automotive Research Co.
P. O'Shea

The Budd Company
H. W. Wessells

E.I. du Pont de Nemours & Co., Inc.
R. E. Knox

Environmental Protection Agency
J. J. Brogan

Eyring Research Institute
Prof. C. Warner

Energy Absorption Systems, Inc.
P. E. Rolthaus, Jr.

Calspan Corporation
J. Garrett

Ford Motor Company
K. H. Arning
C. R. Briggs
W. E. Brown
J. A. Edwards
E. Hellriegel (F.R.G.)
S. E. Henson
J. Starkey (Belgium)

General Motors Corporation
Dr. L. C. Lundstrom
G. Smith

The Goodyear Tire & Rubber Co.
J. Huchinson

Highway Safety Research Institute
J. O'Day
Dr. J. W. Melvin

Humanoid Systems Inc.
S. W. Alderson

Insurance Institute for Highway Safety
A. B. Kelley
B. O'Neill

Minicars Inc.
N. Di Napoli
L. Friedman
Dr. D. Struble

Motor Vehicle Manufacturers Association
A. C. Doty

National Motor Vehicle Safety Advisory Council
D. C. Hogue
R. T. Mullet

Sierra Engineering Co.
L. W. Popp

Subaru of America Inc.
P. Utans

Texas Transportation Institute
G. G. Hayes

Ultrasystems Inc.
S. Davis
R. A. Rockow
P. J. Stevens

Thiokol Corporation
G. J. Donovan

Systems Technology Inc.
R. H. Klein

United States Government

United States Department of Transportation
Dr. C. Brinegar
H. R. Schmidt
W. Steber

National Highway Traffic Safety Administration
W. A. Boehly
E. M. Chandler
F. A. Dilorenzo
H. J. Dugoff
V. J. Esposito
Dr. J. B. Gregory
A. E. Hirsch
A. J. Knievel
Dr. G. G. Mannella
D. R. Mela
L. R. Schneider
J. C. Shively
N. S. Stahler

United States Embassy London
T. C. Colwell

United States Senate
G. Kaganowich

United States House of Representatives
C. B. Curtis

United States Delegation to NATO
C. H. Sanders

United States Navy
Dr. C. L. Ewing

Volkswagen of America Inc.
C. Bahn
S. R. Miller

Contents

REPORT ON THE FIFTH INTERNATIONAL TECHNICAL CONFERENCE ON EXPERIMENTAL SAFETY VEHICLES

SECTION 1

CONFERENCE OPENING

Introduction By The Chairman

Dr. Gene G. Mannella, *Associate Administrator for Research and Development*
National Highway Traffic Safety Administration
United States Department of Transportation
and Chairman of the Fifth International Technical Conference on Experimental Safety Vehicles 3

The Right Honorable Fredrick Mulley, *Minister of Transport*
Department of Environment
Government of the United Kingdom 3

The Honorable Claude S. Brinegar, *Secretary of Transportation*
United States Department of Transportation 4

Gilbert A. Hunt, *President*
The Society of Motor Manufacturers and Traders, Ltd. 5

Dr. James B. Gregory, *Administrator*
National Highway Traffic Safety Administration
United States Department of Transportation 6

SECTION 2

STATUS REPORTS ON EXPERIMENTAL SAFETY VEHICLE DEVELOPMENT PROGRAMS

PART 1 **The United States**

Vincent J. Esposito, *Acting Director*
Office of Vehicle Safety Research
National Highway Traffic Safety Administration 11

Norman S. Stahler, *Senior Staff Member*
Structures Research Division
Office of Vehicle Safety Research
National Highway Traffic Safety Administration 15

PART 2 **European Experimental Vehicles Committee**

H. Taylor, *Head of Safety Department*
Transport and Road Research Laboratory
Department of the Environment 21

The Future for Car Safety in Europe
A Report of the European Experimental Vehicles Committee 24

PART 3 **France**

Michel Frybourg, *Director*
Institute of Transportation Research 55

Georges Boschetti, *Director*
Paris Research Center
Peugeot Automobiles 57

Philippe Ventre, *Engineer in Charge of Impact and Safety Tests*
The Renault State-Owned Works 60

Maurice Clavel, *General Delegate*
Anonymous Society of Citroen Automobiles 83

PART 4 **The Federal Republic of Germany**

Helmut Wagner, *Deputy Assistant Secretary*
Federal Ministry of Transport 103

Dr. Hans Scherenberg, *Member of Executive Board Corporate Chief Engineer*
Daimler-Benz AG 105

Edmund Hellriegel, *Chief Engineer*
Ford Motor Company 107

F. Lohr, *Chief Engineer*
Adam Opel AG 108

Dr. W. Lincke, *Head of Research Department*
Research and Development Center
Volkswagenwerke AG 120

Dr. Max Danner, *Research Scientist*
German Association of Liability
Accident and Motor Traffic Insurers (HUK-Verband) 133

PART 5 **The United Kingdom**

H. Taylor, *Head of Safety Department*
Transport and Road Research Laboratory
Department of the Environment 135

Charles Griffin, *Operational Director of Vehicle Engineering*
Austin Morris Group
British Leyland Motor Corporation Limited 136

Peter M. Finch, *Head*
Safety Test and Development Department
British Leyland Motor Corporation Limited 137

W. J. Daniels, *Chief Design/Development Engineer*
Austin Morris Vehicle Safety Research Project Section
British Leyland Motor Corporation Limited 178

M. Rodger, *Chief*
Research and Engineering Centre
Ford Motor Company Limited 213

PART 6 **Italy**

Dr. Augusto Sirignano, *Director-General*
Ministry of Transport and Civil Aviation 219

Vittorio Montanari, *Director of Applied Research*
Fiat Department of Motor Cars 223

PART 7 **Sweden**

Gustav Eckberg, *Head of Vehicle Department*
Swedish Road Traffic Safety Office 245

Sten Rundkvist, *Project Manager*
Swedish ESV Project 246

PART 8 **Japan**

Ichiro Noguchi, *Deputy Director-General*
Machinery and Information Industries Bureau
Ministry of International Trade and Industry
Presented by
Noriyosni Uno, *Director of Engineering Division*
Motor Vehicles Department, Road Transport Bureau
Ministry of Transport 251

The Test of Japanese Experimental Vehicles –
General Description
Yorshiro Okami
Japan Automobile Research Institute, Inc. (JARI) 253

The Test of Japanese Experimental Vehicles –
Summary of Toyota ESV Test
Yorshiro Okami
Japan Automobile Research Institute, Inc. (JARI) 275

The Test of Japanese Experimental Vehicles –
Summary of Nissan ESV Test
Yorshiro Okami
Japan Automobile Research Institute, Inc. (JARI) 301

SECTION 3

TECHNICAL SEMINARS

PART 1

Accident Analysis

Themes: **Data Collection; Use of Data for Car Safety; Examples of Particular Results**

Introduction

M. Halpern-Herla
ONSER 333

United States Overview
Donald F. Mela
Office of Statistics and Analysis
National Highway Traffic Safety Administration 335

European Overview
M. Halpern-Herla
ONSER 337

Japanese Overview — Traffic Accidents in Japan
Mr. Nagakuna
Traffic Bureau of the National Police Agency 338

United States Technical Papers

A Proposed New National System for Collecting Multipurpose Accident Data: SIR
James O'Day, *Head, Systems Analysis*
Highway Safety Research Institute
The University of Michigan 349

Empirical Relationships Between Car Size, Car Weight and Crash Injuries in Car-to-Car Crashes
Brian O'Neill
Insurance Institute for Highway Safety
Hans Joksch
The Center for the Environment and Man, Inc.
William Haddon, Jr.
Insurance Institute for Highway Safety 362

Factors Influencing the Performance of the Energy Absorbing Steering Column in Accidents
John W. Garrett and Donald L. Hendricks
Calspan Corporation 369

NHTSA's Evaluation of Air Cushion Restraint System Effectiveness (ACRS)
Donald F. Mela
National Highway Traffic Safety Administration 395

European Experimental Vehicles Committee Technical Papers

Synthesis of Statistical Data on Traffic Accidents in France, West Germany, Italy and United Kingdom
F. Hartemann and C. Tarriere
Association Peugeot-Renault 405

The Road Safety Checklist —
A Periodical Inquiry Regarding Traffic
J. C. Bluet
Organisme National de la Securite Routiere
ONSER 414

The Frequency of Corresponding Vehicle Damage in Crash Tests and Actual Accidents
Dr. M. Danner and K. Langwieder
German Association of Liability, Accident and Motor Traffic Insurers — (HUK-Verband) 421

Accident Investigation as an Aid to Planning the Future of Car Safety
Barbara E. Sabey and I. D. Neilson
Transport and Road Research Laboratory 427

Some Patterns and Causes of Injury in Car Occupants
E. Grattan and J. A. Hobbs
Transport and Road Research Laboratory
Department of the Environment 433

PART 2 Human Factors

Themes: Human Tolerance Levels to Injury; Use of Dummies and Similar Devices; Impact Testing and Accident Injuries

Introduction
B. Aldman
Department of Traffic Safety
Chalmers University of Technology
Goteborg, Sweden 441

United States Technical Papers
NHTSA Programs in Biomechanics
Arthur E. Hirsch, *Acting Head, Biomechanics*
Office of Vehicle Safety Research
National Highway Traffic Safety Administration 443

GM-ATD 502 Anthropomorphic Dummy — Development and Evaluation
J. A. Tennant, R. H. Jensen, and R. A. Potter
Advance Product Engineering
General Motors Engineering Staff 446

Relating Air Cushion Performance to Human Factors and Tolerance Levels
Louis C. Ludstrom, R. A. Wilson, and George R. Smith
Environmental Activities Staff
General Motors Corporation 463

European Experimental Vehicles Committee Technical Papers
Human Injury Tolerance Level Determination from Accident Data Using the OPAT Dummy
J. G. Wall and R. W. Lowne
Transport and Road Research Laboratory 501

Pedestrian Protection
C. Tarriére, G. Stcherbatcheff, P. Duclos, and A. Fayon
Laboratoire de Physiologie et de Biomécanique
Association Peugeot-Renault 508

Biomechanical Study of Side Impact Accidents
D. Cesari and M. Ramet
Organisme National de Securite Routiere 511

The Development of the OPAT Dummy
Peter Warner
David Ogle Limited 521

Performance Measurements on the OPAT Dummy
C. M. Haslegrave and M. A. Macaulay
The Motor Industry Research Association 526

Towards Pedestrian Safety
V. J. Jehu
Transport and Road Research Laboratory
Department of the Environment 533

A Review of Development of Passive Restraint Systems
Auto Restraint Systems, Ltd. 535

Reduced Laceration from a New Laminated Windshield
John Pickard, *Research and Development Manager*
Triplex Safety Glass Co., Ltd. 544

Japanese Technical Papers
Evaluation of Seat Belt System and Dummy Characteristics
Yoshio Serizawa, *Manager, Safety Research Department*
Nissan Motor Co., Inc. 553

PART 3 **Vehicle Structural Properties**

Themes: Design for Occupant Protection; Protecting Other Road Users; Energy Management

United States Overview
Edward M. Chandler, *Senior Staff Member*
Office of ESV Programs
National Highway Traffic Safety Administration 569

United States Technical Papers
RSV, Crash Hazards and Public Support
A. B. Kelley, *Senior Vice President*
Insurance Institute for Highway Safety 573

Front Energy Management Parametric Variation Study
William Rupp
Advanced Systems Laboratory
AMF Incorporated 602

Analysis of Test Results for AMF/Fiat ESV Head-on Collision
Sol Davis
Ultrasystems, Inc. 615

Big and Little Car Compatibility
Jerome M. Kossar
Office of Vehicle Safety Research
National Highway Traffic Safety Administration 620

Flexible Exterior Automotive Trim
Jonathan T. Lange
E. I. duPont de Nemours & Co.
Elastomer Chemicals Department 656

European Experimental Vehicles Committee Technical Papers

Compatibility on the Road
Ulrich Seiffert
Research and Development Center
Volkswagenwerke AG 665

Compatibility Between Vehicles in Frontal and Semi-Frontal Collisions
P. Ventre, *Head of Crashworthiness Department*
The Renault State-Owned Works 670

Relationship Between Vehicle Front-End Deformation and Efficiency of Safety Belts During Frontal Impact
J. Elsholz
Bayerische Motoren Werke AG 674

The Ford Automatic Safety Belt System
Edmund Hellriegel and Axel Rauthmann
Body Advanced Engineering
Ford Motor Company 682

Structural and Occupant Protection Systems of the Opel Safety Vehicle
E. S. Kiefer
Adam Opel AG 687

Some Considerations of Body Structure Crushability in Relation to Impact Speed
Dr. L. Rosti-Rossini
Alfa Romeo 692

A Research Study for an Energy Absorbing Sliding Seat
Gaetano Danese, *Director*
Rome Motor Vehicles Testing Center 699

Some Methods of Absorbing the Energy of Motor Vehicles and Their Occupants
P. Watson
Transport and Road Research Laboratory 703

Vehicle Compatibility in Car-to-Car Side Impacts and Pedestrian-to-Car Frontal Impacts
Peter M. Finch, *Manager, Vehicle Safety*
British Leyland Motor Corporation Limited 710

The Application of Computer Simulations in Vehicle Safety
W. C. Emmerson, *Manager, Engineering Development*
J. E. Fowler, *Project Leader*
British Leyland Motor Corporation Limited 712

The Role of Extrusion Devices in Energy Absorption for Safety
K. Deutsch, M. G. J. Elson, and R. W. Moore
Imperial Metal Industries Limited
Kynoch Works, Witton 721

The Dunlop Mk II Composite Energy Absorbing Bumper System
Dr. D. A. Newton, *Manager, Advanced Products Group*
Dunlop Limited 725

Energy Absorption by Variable Shear Strength Duplex Materials
R. S. Millman and M. J. Chappell
Wolfson Institute of Interfacial Technology
University of Nottingham 729

Reduced Laceration from a New Laminated Windshield
John Pickard, *Research and Development Manager*
Triplex Safety Glass Co., Ltd. 740

Occupant Protection in Frontal Impacts: A Static, Passive Restraint System
W. Egglestone and G. D. Suthurst
Ford Motor Company Limited 748

Occupant Protection in Rear Impact
D. Burland
Ford Motor Company Limited 760

Occupant Protection During Vehicle Rollover
K. Stone
Ford Motor Company Limited 769

A Practical Approach to the Protection of Motor Vehicles by the Absorption of Impact Energy
G. Persicke, F. I. Plant, and J. R. Child
Road Research Limited 785

Development and Manufacture of Automobile Bumpers
I. M. Thomson
Enersorb Limited 789

Japanese Technical Papers
Reviews on Accuracies and Results of ESV Collision Tests
Masahiko Sakimura
Sixth Research Dept.
Japan Automobile Research Institute, Inc. (JARI) 793

Characteristics of Body Energy Absorption and Restraint System
Kazuo Higuchi
Honda Motor Co., Ltd. 796

PART 4 ACCIDENT AVOIDANCE

Themes: Estimation of Effectiveness of Safety Measures; Engineering Systems; Interaction of Driver and Vehicle

Introduction
Dr. G. Pocci, *Ministry of Transport* 801

United States Technical Papers
United States Overview
Francis A. DiLorenzo
National Highway Traffic Safety Administration 803

Determination of Motor Vehicle Characteristics Affecting Driver Handling Performance
Gordon G. Hayes
Texas Transportation Institute
Texas A&M University 805

Driver/Vehicle Response Research
Richard H. Klein
Systems Technology, Inc. 810

European Experimental Vehicles Committee Technical Papers

The Relation Between Vehicle Handling and Accident Avoidance
Kurt Enke
Daimler-Benz AG 815

Measurement of Transient Slip Angles of Tires – A Contribution of Evaluating Safety-Relevant Driving Conditions
Dr. Rüdiger Weber, *Leiter der Reifenentwicklung*
Porsche AG 818

Safety and Motor Vehicle Equipment
Dr. Hansjörg Manger
Robert Bosch GMBH 823

Improving Directional Stability Under Braking
Detlef Banholzer
Audi NSU Auto-Union AG 829

Further Research on the Driver/Vehicle System Study of Braking in a Turn
F. Surace
Alfa Romeo 836

The Stop Lamp: Some Optimum Visibility Considerations
Paolo Soardo
Instituto Elettrotecnico
Nazionale Galileo Fennanis (IEN) 842

Safety Aspects of Denovo Run Flat Tires
T. Holmes and T. French
Tire Technical Division
Dunlop Limited 847

Factors Leading to Loss of Control – A Guide from Accident Investigations
G. Jones, *Manager, Safety Engineering*
British Leyland Motor Corporation Limited 856

Drivers' Limitations
H. A. J. Prentice
Transport and Road Research Laboratory 862

The Punctured Pneumatic Car Tire from the User's Viewpoint
M. A. Jacobson, *Chief Engineer*
The Automobile Association 868

Japanese Technical Papers

Characteristics of Japanese ESV Tires
Hideo Sakai
Third Research Dept.
Japan Automobile Research Institute, Inc. (JARI) 895

Steering and Handling Characteristics of a Vehicle When Fail-Safe Tire is Deflated
Shoichi Sano
Honda Motor Co., Ltd. 906

Closing Remarks

Dr. G. Pocci
Ministry of Transport 925

PART 5 OVERALL CAR REQUIREMENTS

Themes: Cost Benefit for Safety Measures; Interaction of Safety; Environmental and Energy Factors

United States Technical Papers

The Evaluation of Safety, Energy, and Environmental Factors in the Automotive Transportation System
William C. Steber, *Deputy Assistant Secretary*
Systems Development and Technology
U.S. Department of Transportation 929

An Overview of the U.S. Government Program to Evaluate Alternative Powerplants to the Conventional Internal Combustion Engines for Automobiles
John J. Brogan
Alternative Automotive Power Systems Division
Environmental Protection Agency 932

Research Safety Vehicle Crash Effectiveness Methodology
John J. Edwards
Ford Motor Company 937

European Experimental Vehicles Committee Technical Papers

Benefit/Cost Analysis as a Basis for Decisions in the Automotive Industry
Dr. Ernst Fiala
Volkswagenwerke AG 947

Cost-Effectiveness Study
N. Muhlrad
Organisme National de la Securite Routiere
ONSER 951

Study on Cost-Efficiency Ratio Based on "Basic Research Vehicle"
P. Ventre, *Head of Crashworthiness Department*
The Renault State-Owned Works 956

An Overview of TRRL Programs
Dr. John Fitchie, *Head of Department of Transport Systems*
TRRL 957

Japanese Technical Papers

A Feasibility Study on Small Safety Vehicles
Yoshio Serizawa
Nissan Motor Co., Ltd. 959

Cost-Benefit Analysis of Bumper Systems for Small Cars
Teiji Iida
Body Engineering Design Department
Toyota Motor Co., Ltd. 974

Discussion 979

PART 6 Application of Research Results to Production Cars and Future ESV Activities

Themes: Application of Research Results to Standards; Future Activities

United States Technical Papers
The Research Safety Vehicle Program and U.S. Motor Vehicle Safety Standards
Dr. James B. Gregory, *Administrator*
National Highway Traffic Safety Administration 985

European Experimental Vehicles Committee Technical Papers
Application to Regulations of Studies Made Pursuant to the ESV Programs
A. Osselet, *Chief Engineer*
Mines Department 987

Contribution of the German Motor Industry
Dr. G. Brenken, *Principal Manager*
Verband der Automobilindustrie e.V. 990

Translating the Results of the Experimental Safety Vehicle Program Into Production Cars
R. J. A. Sharp
Department of the Environment 991

Application of Results to Standards — Future Activities
Dr. Pocci
Ministry of Transport 993

Japanese Technical Paper
Application of Results to Standards — Future Activities of Japan
Noriyoshi Uno, *Director of Engineering Division*
Motor Vehicles Department
Road Transport Bureau 995

Discussion 997

SECTION 4

CONCLUSION

Closing Remarks

Alexander Silverleaf, *Director*
Transport and Road Research Laboratory
United Kingdom Department of the Environment 1001

Dr. Gene G. Mannella, *Associate Administrator for Research and Development*
National Highway Traffic Safety Administration
United States Department of Transportation 1002

SECTION 1

INTRODUCTORY STATEMENTS

CONFERENCE OPENING

DR. GENE G. MANNELLA, Chairman
United States

FREDRICK MULLEY
The United Kingdom

CLAUDE S. BRINEGAR
United States

GILBERT A. HUNT
The United Kingdom

DR. JAMES B. GREGORY
United States

SECTION 1

INTRODUCTORY STATEMENTS

DR. GENE G. MANNELLA, *Associate Administrator for Research and Development*
National Highway Traffic Safety Administration
U.S. Department of Transportation and Chairman of the Fifth International Technical Conference on Experimental Safety Vehicles

Welcome to the Fifth International Technical Conference on Experimental Safety Vehicles. My name is Gene Mannella and I am serving as chairman of this conference. I am sure you are all looking forward to a productive and stimulating exchange of ideas concerning the very important societal problems of traffic safety. I am equally confident that this conference will continue the established pattern of open and forthright dialogue that has been the distinguishing feature of past conferences.

We will now proceed to formally begin the conference proceedings and will call first upon representatives of the host country. It gives me great pleasure to introduce the first inaugural speaker, The Right Honorable Fred Mulley, MP, Minister of Transport, Department of Environment, United Kingdom.

The Right Honorable FREDRICK MULLEY
Minister of Transport
U.K. Department of Environment

It is my pleasure to welcome you on behalf of the United Kingdom Government to the Fifth International Technical Conference on Experimental Safety Vehicles.

At this Conference, organized jointly by the Society of Motor Manufacturers and Traders and the Transport and Road Research Laboratory, we look forward to hearing about the latest developments in car safety and to discussing the future direction of the program.

Road safety has for a long time been the subject of research and development in Europe. Nevertheless the initiative taken by the United States Government in promoting international collaboration in car safety has generated a renewed sense of purpose in the subject worldwide. There is now widespread cooperation between governments; and companies in a highly competitive industry are working together – and working with governments – in ways which would have been deemed impossible some years ago. As a result, a wealth of car safety knowledge has been made available and freely shared at these Safety Vehicle Conferences. I know that a great deal of work has been done to improve the safety of car occupants in crashes. I am, however, glad to learn that as the program has developed there has been increasing emphasis on providing compatibility in impact situations between vehicles of different sizes and types and on the protection of pedestrians who are struck by cars.

I believe that we have now reached an important stage in the program when we can begin to take steps to implement the knowledge gained; the aims of the car safety program will remain unfulfilled until the lessons learned are incorporated generally in production cars. Manufacturers are of course always introducing safety improvements on their own initiative and it would be very misleading to give the impression, as some have done, that as a class they must always be goaded and whipped into taking safety seriously. This is not so. Nevertheless, we are still to some extent in the situation where safety advances are featured in more expensive cars but not in those produced in very large volume. I fear that in this country, as I suspect in many others, the safety features of a car are not predominantly either in the sales propaganda or in the buyer's mind at the time of choice.

In the British view the primary objective of the ESV program has been much less to establish the theoretical limits of safety than to apply the results of intensive research to the mass of cars coming onto the road in the next few years.

I have not read all the papers submitted to this conference. In view of their volume that will not perhaps surprise you. Nor am I an automobile expert and I recognize that there are highly technical matters

involved here. Nevertheless, I would like to put it to you that one of your most important tasks, perhaps the most important, is to try to establish whether there is a consensus on the measures to be taken to improve significantly the safety of the cars to be produced in the near future, let us say in the early 1980s. This means envisaging standards which do not produce safety gains at the expense of much higher weight, cost or energy consumption – factors which are inescapably interlinked and which have assumed increasing importance in the light of recent events. It means making sure that in crashes the big heavy car or, indeed, the lorry do not have all the advantages, by designing the front of all cars in such a way that, at any rate in low speed accidents, their catastrophic effect on pedestrians is greatly reduced.

If there is agreement on these matters at a technical level – and I do not seek to prejudge it, for that is what you are here to talk about, then of course governments must come in and in full consultation with industry work out the appropriate legislation. This is vital so that in a competitive situation all manufacturers can work to the same rules. And it must be done at the international level for cars are made and sold worldwide.

It is too early yet to talk about having identical vehicle standards in every country and I recognize that local conditions, particularly on the question of air pollution, can require different solutions. Nevertheless, I believe that the major car producing regions of the world should be increasingly seeking ways of sharing their experience and their expertise in vehicle safety and applying it as harmoniously as possible.

These and other points have been very much in the minds of European governments since they formed the European Experimental Vehicles Committee almost four years ago; their report to be presented later today embodies their agreed views on these important subjects which mark a significant step forward.

It is my hope that this European response to the program when combined with American and Japanese views will assist us all to move from the research stage towards the goal of safer production cars which will make a real contribution to road safety worldwide. This conference provides an excellent opportunity to study the latest technical developments. It also enables you to explore the possibility of making important advances in the beneficial control of one of the most powerful and lethal tools yet invented by man.

In inviting you to rise to the challenge, I wish you a most successful conference and a pleasant stay in Britain.

The Honorable CLAUDE S. BRINEGAR
Secretary of Transportation
U.S. Department of Transportation

It is a great honor for me to address this Fifth International Technical Conference on Experimental Safety Vehicles. On behalf of the entire American delegation, I bid you all welcome. I'm confident that this will be a worthwhile and informative conference.

This series of conferences had its start in the common bond that we all share so strongly – the bond of international goodwill and cooperation. But we also share a common problem – one in which we in America are, sadly, the leaders – at least in total numbers. The problem is the thousands and thousands of deaths and injuries caused by the growing millions of automobiles. It is the desire and the hope for progress toward a workable solution to this most serious problem – the desire to save lives – that provides the strong incentive for this international cooperative effort. I know of few other international programs that are working on such a vital societal issue with such breadth and depth.

Much has been learned in the first five years of this cooperative effort.

We have learned that vehicle safety can be significantly improved over conventional private manufacturers' designs, *provided* safety is given a high priority and approached in a positive and an open-minded way. We have tried in this program to give it that priority.

We have learned that vehicle passenger restraint systems can be made effective and more acceptable.

We have learned that it is essential that vehicle safety be viewed not as an isolated element, but as an integral part of the vehicle's total system and design concept.

We have learned the importance of considering vehicle-to-vehicle relationships – what we call the aggressiveness problem – in determining crash survivability.

We have learned the importance of standardized testing and analytical procedures, and we have made good progress in developing these procedures.

We have learned that country-by-country differences in driving and road conditions can require different degrees of concentration on safety features. In some countries, for example, pedestrian safety can be a more serious problem than in other countries.

And, I believe, we have made progress toward identifying priority areas where future effort should now be concentrated, as well as in identifying reasonable upper limits of vehicle safety performance.

Each of the seven nations, has, in varying ways, contributed significantly toward this progress. Some

have developed fully integrated ESVs; some have concentrated on improving the safety aspects of essential subsystems, such as brakes, steering, and passenger restraint systems; and some have worked on testing and analytical procedures. This work, together with the discussions and technical papers presented at each conference, has helped the United States in its overall vehicle safety program, especially in rulemaking. This work has also helped us to re-focus the directions of our future work on experimental safety vehicles.

We well recognize that, as in most untried areas, we must learn as we go. In retrospect, some of our early ESV work was perhaps too narrowly focused and too rigidly conceived. We learned valuable knowledge from this work, but we also learned that we must avoid the impractical, that we must concentrate on that which *can* be converted from the test pit to the manufacturing line. It is certainly proper to set very high performance standards – personal safety can tolerate no less – but these standards must, in the final analysis, be attainable in the real world.

America's ESV program is now shifting in response to what we have learned thus far, and what we perceive to be the likely future operating conditions for our motor vehicles. I believe that it is particularly important that we shift our safety focus to the small, fuel-efficient vehicle. We recognize that here is an area where we in America can profit from your years of experience.

We have recently awarded five contracts for our next generation of ESVs – called, for administrative convenience, "research safety vehicles" or (RSVs). One of these contracts was awarded to Volkswagen. This program will concentrate, in its early phases, on overall vehicle specifications and on total systems engineering. The end product is directed toward test vehicles that are small and economical in fuel usage, not over 3,000 pounds in weight, reasonably nonpolluting, and attain their safety characteristics in ways that can be translated to the production lines of the 1980s without undue economic penalty.

While America's ESV emphasis is shifting, please let me assure you that our desire for international cooperation and our need for further progress in solving the vehicle safety problem continues as strong as ever. Vehicle deaths in America – though they have fallen significantly in recent months because of reduced speeds and reduced driving – are still at tragically high levels of nearly 50,000 per year. Recognizing the continued upward growth in motor vehicles everywhere in the world, I believe that America's safety problem of the 1970s, when viewed broadly, must be recognized as the not-too-distant edge of your safety problem. We invite your continued cooperation in our joint efforts to find solutions.

In particular, we invite you to continue to submit ESVs for our testing. These could be either complete ESVs or traditional vehicles with partial ESV subsystems. We will test them by our standardized procedures, and we will share our findings with all parties. We also invite your suggestions on new test and analytical procedures – especially cost/benefit methods of analysis – and on the determination of safety performance standards.

Areas that I especially hope to see new and innovative work in during the next few years include vehicle weight reduction, vehicle crash "aggressivity" reduction, passenger restraint system improvements, and knowledge on human tolerances to accident forces. I see these as prime areas for high payoff results.

Let me close by stressing this important point: We have, in a few years, made a good start on a complex, vital, and common problem. Progress has been slow, but with our continued joint efforts – with continued free exchanges of ideas and test data – I think we can achieve important, worthwhile results in the years ahead. The problem of vehicle safety is too important to treat otherwise.

Finally, I'd like to thank our host country, Great Britain, for its warm hospitality and its splendid exhibits. The TRRL sets a high standard for future conferences.

GILBERT A. HUNT, CBE
President
U.K. Society of Motor Manufacturers and Traders

It is a great pleasure and honor for me as President of the Society of Motor Manufacturers and Traders to greet you on this occasion on behalf of the motor industry in Britain. We welcome you most heartily and with the belief that the presentations, exhibitions, discussions and indeed disagreements which are to be brought forth over the next four days will have but one long-term conclusion – the reduction of the accident toll on the world's roads, and hence even greater freedom for all mankind to enjoy an ever-wider appreciation of the values of personal mobility through ownership and enjoyment of the products of our industries across the world.

We are especially indebted to the continued efforts of the United States Department of Transportation and to the governments and industries of all participating nations for making this week possible. Special thanks and congratulations are of course due to the United Kingdom Department of the

Environment through its Transport and Road Research Laboratory, whose officers have so ably shouldered the bulk of the organizational problems leading to this conference in London and the associated exhibitions and research track demonstrations at their center at Crowthorne.

This series of international conferences, of which this is the fifth, has been encouraging for many reasons. Probably most important is the noncompetitive linking as national delegations of representatives from both the motor makers and the motor legislators of all the world's major vehicle producing countries in a genuine endeavor to expand the boundaries of world knowledge and thinking on the mechanical aspects of world safety. We are not here to sell one product or idea as the best and definitive answer. Nobody is here to win votes from an electorate or to press some escapist ideology. From so many countries we are here to strive towards finding realistic, practical and economic answers to a problem which can be so shockingly personal to every citizen of every developed country – that of death and injury on the roads. At previous conferences many ideas have been put forward. Many theories have been propounded. Much hardware has been displayed. Here in Britain this week a further 15 months' development of these ideas, theories and hardware will be explained, discussed, displayed and in many cases demonstrated. Much of what was thought to be true five years ago has been found to be unreal or impractical. Much more is now known about the causes of accidents, the role of the vehicle in the accident, the functioning of the human body in an accident situation and, most importantly, there are already changes to be seen in every new model being produced which can be directly related to the work which this ESV program has encouraged and focused international attention upon. I have no doubt that internationally we will be far wiser again by the time this conference concludes on Friday.

No person here believes that the world's road toll can be eliminated solely by concentration on the "weapon" – the vehicle itself. But neither would anybody deny that there will always be room for improvements which will make vehicles easier to control and therefore to avoid accidents in the first place, and secondly improvements which can be made structurally and internally which will reduce the degree of injury to occupants – and to pedestrians – who may be involved in a road accident.

In a noncommercial, nonpolitical atmosphere such as we have here this week we must all be confident that new strides will be taken towards furtherance of all our knowledge on how best and most efficiently to meet the challenge of making motoring safer.

DR. JAMES B. GREGORY
Administrator
National Highway Traffic Safety Administration

May I also convey to you my personal welcome for traveling to London, a great distance for many of you, to participate in this Fifth International Conference on Experimental Safety Vehicles.

As you know, this is my first opportunity since becoming Administrator of the National Highway Traffic Safety Administration to attend one of these impressive technical gatherings. I am honored indeed to have the chance to meet personally with the government, industry, and scientific community, and the leaders of those communities as well, who have turned the concept of international cooperation for solving common technological problems from a promise into a reality. Indeed, in the very few months in which I've been in this assignment, I have begun to count many of you as close colleagues and friends.

I'm also most impressed by the continuing commitment of resources by the participating governments and their automotive industries. This dedication by the leading automotive producing nations of the world to improve automobile safety performance with the resulting potential for reduction in deaths and injuries on the world's highways has been truly outstanding. My sincere congratulations go to you all.

Perhaps most important of all, the ESV program has proven to most of us the necessity of ultimately addressing the automobile in our engineering work as a total system. For unless we look at the impact of proposed design solutions on overall vehicle performance, how can we properly make those difficult cost/performance trade-off decisions?

With this in mind, we have recently streamlined and integrated the motor vehicle research programs of the National Highway Traffic Safety Administration by combining into a single major office, entitled the Office of Vehicle Safety Research, the formerly separated offices which before addressed crashworthiness or passive safety, operating systems and experimental safety vehicles. This solidification of our vehicle research will encourage the input of important research findings from our systems and subsystems work into our total vehicle development projects. In turn, the more specialized single system research topics can take full advantage of our experience in the ESV program, and taken all in all, we will be able to apply the results of research, experimentation and demonstration into solid, sensible rulemaking. But experimental vehicle research, no matter how impressive the results, is only part of the picture. I am concerned, as we all are,

with the ultimate application, as I have said, of the important ESV and other research results to production cars.

In closing, I want to reaffirm my appreciation to the Government of the United Kingdom and to its automotive industry for so graciously hosting this Fifth ESV Conference. It is a pleasure and it's an honor for me to have this opportunity to participate with all of you in this most important program. Thank you very much.

SECTION 2

STATUS REPORTS ON EXPERIMENTAL SAFETY VEHICLE DEVELOPMENT PROGRAMS

PART ONE UNITED STATES

VINCENT J. ESPOSITO
NORMAN S. STAHLER

PART TWO EUROPEAN EXPERIMENTAL VEHICLES COMMITTEE

H. TAYLOR
E. E. V. C. – The Future for Car Safety in Europe

PART THREE FRANCE

MICHEL FRYBOURGH
GEORGES BOSCHETTI
PHILIPPE VENTRE
MAURICE CLAVEL

PART FOUR THE FEDERAL REPUBLIC OF GERMANY

HELMUT WAGNER
DR. HANS SCHERENBERG
EDMUND HELLRIEGEL
F. LOHR
DR. W. LINCKE
DR. MAX DANNER

PART FIVE THE UNITED KINGDOM

H. TAYLOR
CHARLES GRIFFIN
PETER M. FINCH
W. J. DANIELS
M. RODGER

PART SIX ITALY

DR. AUGUSTO SIRIGUANO

VITTORIO MONTANARI
Fiat Technical Presentation

PART SEVEN SWEDEN

GUSTAV ECKBERG

STEN RUNDKVIST

PART EIGHT JAPAN

ICHIRO NOGUCHI
Presented by Noriyosni Uno

YORSHIRO OKAMI
The Test of Japanese Experimental Vehicles (General Description)

YORSHIRO OKAMI
The Test of Japanese Experimental Vehicles (Summary of Toyota ESV Test)

YORSHIRO OKAMI
The Test of Japanese Experimental Vehicles (Summary of Nissan ESV Test)

SECTION 2

STATUS REPORTS ON EXPERIMENTAL SAFETY VEHICLE DEVELOPMENT PROGRAMS

PART ONE

UNITED STATES

VINCENT J. ESPOSITO
Director
Office of Vehicle Safety Research
National Highway Traffic Safety Administration

INTRODUCTION

Ladies and Gentlemen, it is my pleasure to present the initial portion of the United States ESV program status report. Following my presentation, Mr. Norman Stahler, who directs the ESV Testing and Evaluation Program, will report on the results of our most recent ESV testing.

Before I begin my formal remarks, may I take the opportunity to express our appreciation to the British Government for assuming the very demanding task of hosting this Fifth International ESV Conference. I would especially like to thank Mr. Taylor and Mr. Lister and their staff at TRRL and the people at SMMT for the excellence of the arrangements, which, as we all know, required a great deal of hard work. So to all of our British hosts, our sincere appreciation.

During this portion of the U.S. ESV status report, I would like to discuss two main topics. First, the lessons learned from our initial ESV family sedan development and our current efforts to conclude that project; and second, the status of our recently initiated Advanced State-of-the-Art Research Safety Vehicle or "RSV" project.

THE FAMILY SEDAN PROJECT

The ESV program began in 1969 with the United States' national effort involving the 4,000-pound full-size family sedan, a vehicle most typical of the U.S. highways at that time. Just two years later, in 1971, the International ESV Program became a reality under Memoranda of Understanding with the Federal Republic of Germany, France, Italy, Japan, Sweden, and the United Kingdom. The International ESV Program called for the United States to continue its work in the 4,000-pound family sedan weight class, while the other participating governments would sponsor extensive research leading, in most instances, to the development and fabrication of ESVs, primarily in lighter and smaller vehicle classes. All participants agreed to keep each other informed by the exchange of visits of engineering teams, open discussions at technical meetings, cooperative exchange and testing of vehicles, and publication and circulation of significant results.

These international agreements focused sharply the principal automobile producing nations of the world along with their engineering and manufacturing expertise on the difficult task of demonstrating the technology required to achieve a substantial advancement in the safety performance of automobiles on the world's highways. Certainly, at this point in time, we can say that the International ESV Program has made remarkable progress towards fulfilling its original objectives. However, as we all realize, much work remains to be done; and many important technological and safety policy questions remain yet unanswered. Indeed, as we have noted before, perhaps the most significant program accomplishment lies not in the problems resolved but in the problems more clearly defined or sharply delineated that still must be addressed.

I would like now to review, briefly, some preliminary program conclusions, as we see them, that can be drawn from the results of the initial ESV prototype developments.

Excessive Weight

Without exception, all preliminary and completely tested designs are indicating weight increases on the order of 20-30 percent, even with the use of high strength steels and some aluminum. Several obvious specification changes have now been identified that could lead to a decrease in the weight penalty without seriously compromising crashworthiness. Examples of such changes are in areas of rear seat restraints and driver visibility.

In addition, refinement of initial designs after testing could also result in weight reductions for some designs.

Notwithstanding the potential for some weight improvements, it is apparent that the existing ESV specifications, taken as a whole, result in designs of questionable near-term practicality, mainly because of excessive weight, but also because of the need for costly materials or because the designs require substantial changes in production methods. In considering problems such as excessive weight, however, we must remind ourselves that the initial goal of the ESV program was to establish the possible upper limits of safety performance. This goal, in spite of the demanding specifications in many areas, has generally been met; and the challenge now before us is to take the results of this experimental research program and find ways to apply them effectively to production cars.

Aggressiveness

A second problem that has emerged is the tendency to improve structural crashworthiness at the expense of unwanted aggressiveness in car-to-car collisions. The original U.S. ESV specifications included the requirement for lower aggressiveness (car-to-car) at various impact velocities, and several manufacturers developed unique hydraulic stroking mechanisms that demonstrated potential for high levels of crashworthiness combined with low aggressiveness. Other manufacturers decided in this initial phase to forego the lowering of aggressiveness in order to obtain high speed front-end crashworthiness while retaining less complex, less costly and more conventional frame design concepts. Two extremes have, therefore, demonstrated: (1) good crashworthiness and low aggressiveness in a design of questionable practicality, and (2) good crashworthiness but high aggressiveness in a more practical design.

Vehicle Handling and Other "Active" Safety Requirements

It appears that improvements in many areas such as braking, visibility and lighting can be made in most designs at the same time as marked structural changes are being made to obtain greater crashworthiness. While at the outset of the program, the ability to maintain adequate handling characteristics in combination with major structural changes was seriously questioned, it now appears that good handling can be achieved in such circumstances. The questions for most active safety areas now pertain more to, "What is the best and most cost effective specification?" rather than the feasibility of meeting the specification itself.

Areas Requiring Research

Certain critical technical areas were also highlighted by almost all participants as requiring additional research effort before the most cost effective vehicle design trade-offs can be accomplished:

- The question of how to provide for human tolerance to impact forces has been addressed again and again as a serious problem. Biomechanics as defined with respect to the ESV program means finding the tolerance limits for the human body under dynamic conditions of typical traffic accident situations, as well as the development of anthropomorphic research dummies and instrumentation. This vital area needs further research to increase our understanding of human tolerance in the violence of the crash environment.
- Better accident investigation data with respect to the frequency and severity of all types of accidents need to be developed. The cost benefit analyses used as a guide in determining the cost effectiveness of existing and proposed safety standards require the preparation of such a statistical base. As a corollary, it is necessary to develop a valid cost data base, to include not only the initial acquisition costs of proposed new safety performance improvements, but also the operating costs over the life of the vehicle. Any such analyses must, of course, eventually address the larger problem of cost benefit considerations within the total transportation system. Here broad questions concerning fuel consumption, air pollution, and total resource conservation must be considered.
- The topic of "accident avoidance" (active safety), particularly vehicle handling, has been identified as an area which is most difficult to quantify with respect to improved safety performance. Safety performance requirements based only on open-loop values derived from current production cars do not have high safety relevance. Requirements must be based on maneuvers that have been proven as critical to accidents with driver involvement.
- The problems inherent in providing crashworthy vehicle structures for protecting occupants at reasonable cost and without posing additional hazards to other users of the road are national in scope. Several countries have already made original and valuable contributions in this field. More needs to be done.
- Efficient restraint systems that are utilized by vehicle occupants are basic to providing vehicle crashworthiness from the occupant's standpoint.

No amount of structural effort can be very effective unless it is integrated with occupant protection systems. Promising concepts for such advanced occupant protection systems are in existence now as a result of developments in practically all participating countries.

- Many participants in the International ESV Program are expending large resources for updating and constructing test facilities, as well as conducting extensive vehicle testing. Although communications with respect to test results have been good, the same cannot be stated of the test methodology employed. It would be very beneficial to the overall ESV program to be able to employ more "common" denominators in the area of testing for the purposes of maximizing data correlation. Improved techniques for both accident avoidance and crash testing should be developed and agreed upon internationally.
- Automobile design which takes into account pedestrian safety involves very complex problems that must be addressed. Deaths and injuries as a result of accidents involving pedestrians are on the increase worldwide. The development of acceptable solutions to this problem compounds the design requirements for safer cars, and considerably more research is necessary. Because of the seriousness of this problem in Japan, the Japanese manufacturers have made impressive efforts in attempting to make their ESVs more forgiving of pedestrians. Several European manufacturers have also successfully addressed pedestrian safety in their ESV designs. As more and more automobiles populate the world's highways, more serious attention to pedestrian safety in automobile design is necessary. These requirements must be balanced with the other requirements of occupant protection and aggressiveness.

To summarize as concisely as possible our conclusions regarding the initial ESV research program, I would simply say that the high levels of safety performance demanded in the prototype specifications were indeed achievable, but the initial designs were not practical for near-term mass production.

Planning for the Future

The ESV program has been uniquely successful in demonstrating worldwide concern for improved highway safety. The commitment of talent and resources by the participating nations to ESV development work has been most impressive and the cooperation in the exchange of technical information unparalleled. The continuation of this international cooperation in finding the answers to the many questions raised by the initial ESV efforts is a most worthy challenge.

We have now essentially completed the initial development phase of the International ESV Program. All of the participating nations have or are in the process of fabricating and testing ESVs or are completing important subsystem developments and research studies. The international test phase of the program has begun with the exchange of the Fiat 1,500-pound, 2,000-pound, and 2,500-pound ESVs and Japan's Toyota and Nissan ESVs. Negotiations with other countries for ESV exchange are now in the final stages. Results of these tests will constitute a major input to the final optimization of the family sedan safety performance specifications and will materially assist in the structuring of follow-on ESV projects.

The Family Sedan Project in the United States is now being concluded by the preparation of a final report scheduled for issuance before the end of calendar year 1974. This report will summarize all ESV work accomplished by all participants to the extent that data have been made available to us and may include a proposed optimized safety performance specification for a full-size vehicle. In addition, the report will discuss research areas which have been identified by participants as requiring additional effort. Those research areas have already been briefly noted in the *Pilot Study Report on ESV* to the CCMS, and many will be discussed during the seminar sessions later this week as candidate possibilities for additional international cooperative research. The final ESV Family Sedan Report will be circulated worldwide so that all participants may comment on the substantive issues.

RSV

I would like to turn briefly now to my second topic, the Research Safety Vehicle program. In recognition of the continuing experimental and research needs concerning the motor vehicle as a total system, the United States has initiated an Advanced State-of-the-Art Research Safety Vehicle (RSV) project as the logical next step in its ESV program. This project takes special recognition of the projected transportation challenges of the 1980s, and is structured to address the problems of minimizing fuel consumption, controlling urban congestion and pollution, and maximizing resource conservation as well as safety. The initial proposed specifications for this project were distributed to all participants prior to the Kyoto conference in March 1973. Requests for proposals were forwarded to all governments and

manufacturers participating in the International ESV program. Contract awards for Phase 1, Concept Definition and Specification Development, were accomplished in January 1974.

In contrast to the original Family Sedan ESV, the approach to the development of final specifications for the RSV will depend substantially on the technically documented recommendations of the five Phase 1 contractors. In this respect, the program is attempting to establish firmly at the onset an agreed upon, realistic foundation for the subsequent design, fabrication and testing phases. The program will build on the engineering and research data base established by the initial ESV prototype developments and project these engineering solutions and concepts to the design of a 3,000-pound car for the 1980s.

The Memoranda of Understanding on ESV development and the exchange of technical information as a result of these memoranda certainly include the United States RSV Project. Specifically, copies of the RSV Statement of Work were included as Appendix 2 to the *U.S. ESV Program Quarterly Progress Report, Fourth Quarter 1973.* We expect RSV Phase 1 study results to be published in early 1975. Additional technical information will be forwarded as it becomes available. Because of the competitive nature of the RSV contracts, however, certain information cannot be released until the competition is completed. As in the initial ESV project, we fully intend to make all information generated during the RSV project available to all participants on as timely a basis as possible within the constraints of our contractual approach.

As the initial ESV prototype development phase draws to a conclusion, discussions are now in progress among the participating nations for meaningful follow-on projects. During this conference, these discussions have centered on the possibility of broadening international cooperation to include research areas identified by the initial phase of the ESV program as requiring additional research effort. Indeed, the conference agenda indicates the shift in emphasis from purely experimental vehicle development reports to discussion and cooperation in those related research areas so critical to us all.

On behalf of the National Highway Traffic Safety Administration and the Department of Transportation, I welcome this opportunity to continue our international ESV cooperative efforts and to extend this cooperation to include the necessary supporting research.

NORMAN S. STAHLER
Senior Staff Member
Structures Research Division
Office of Vehicle Safety Research
National Highway Traffic Safety Administration

It is indeed gratifying to be able to report on the status of ESV testing, especially since it is directly attributable to the cooperation of two participants in the international program, Japan and Italy. I would like to begin with a review of the significant milestones that have occurred since the meeting in Kyoto. Testing of the first generation U.S. ESVs was completed in mid-1973, and the reports of that testing have been distributed. Figure 1 identifies the vehicles that have been delivered to the Dynamic Science Facility this year, the broad categories of tests, and the schedule for testing those vehicles. The Nissan and Toyota ESVs that arrived in March have been subjected to braking and some handling tests, the results of which will be presented today. Accident

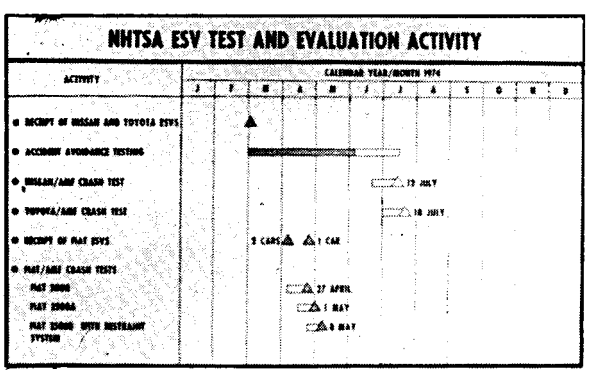


Figure 1

avoidance testing is continuing, and both vehicles should be completely tested by the end of July. We structured the program to allow concurrent accident avoidance testing and equal data presentation at this time for both Nissan and Toyota. Although there is some duplication with the testing of the Japanese Automobile Research Institute (JARI), the NHTSA has selectively incorporated different methodologies in an effort to improve its analytical tools; namely, the HSRI developed procedures for sinusoidal and trapezoidal steer tests to evaluate handling, and the drastic steer and brake test to determine overturn immunity. Also, rather than barrier crashing the Nissan and Toyota, front-to-front crash tests will be conducted with an AMF ESV, reduced to 4,000 pounds, at a closure velocity of 60 mph to better evaluate vehicle-to-vehicle compatibility relative to the traffic mix of large and small cars. The schedule calls for crash testing the Nissan on July 12 and the Toyota on July 18.

The three Fiats consisted of a 2.000-pound weight class vehicle and two in the 2,500-pound weight class, one of which included two front restraints. These vehicles were not designed for accident avoidance testing, but all three have been crashed; and the results will be discussed later.

Now, I would like to present the data that we have acquired with the Japanese ESVs beginning with the weights and measurements shown in Figure 2. It

WEIGHTS AND DIMENSIONS— NISSAN/TOYOTA ESV

	NISSAN	TOYOTA
• WIDTH	64 IN.	70.7 IN.
• HEIGHT	54 (54.3) IN.	54.5 IN.
• LENGTH	173.8 IN.	169.2 IN.
• WEIGHT	2744 LB	2924 LB
• WHEELBASE	100.5 IN.	90.1 IN.
• TREAD WIDTH		
• FRONT	53.8 IN.	59.0 IN.
• REAR	53.9 IN.	59.0 IN.
• PASSENGER CAPACITY	4	2
• CURB LENGTH/WHEELBASE	1.72	1.87

Figure 2

should be noted that the Japanese ESVs were developed on the basis of their own specifications which, in some areas, deviate from the original U.S. specifications. Seating of 95th percentile American males is marginal in both vehicles, but we should remember that in accordance with the Japanese specifications, the vehicles were actually designed for 150-pound passengers, which approximate the 30th percentile American male.

Let us proceed to the available accident avoidance test results, beginning with the Nissan braking performance, shown in Figure 3. Stopping distance on dry pavement was measured at 163 feet as compared to the Japanese design requirement of 164 feet.

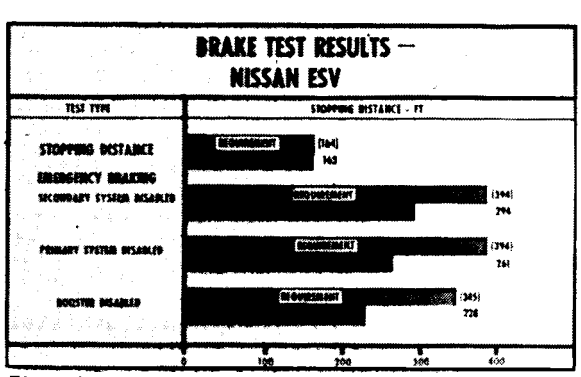


Figure 3

The Nissan braking system consists of front discs with booster and rear drums with anti-skid. Emergency braking tests were conducted with the front system disabled, the rear system disabled, and the booster disabled, with results as shown. In all cases, the measured performance was significantly better than the design requirements. Braking efficiency calculations have not been completed, and information required for computation of the idealized stopping distance with the actual test tires is being generated. This data will, of course, be presented in the final report on the Nissan vehicle.

Figure 4 presents pedal force gain results for the conditions of brake system normal, booster disabled, and front systems disabled. In all cases, the vehicle performed within the required pedal force gain envelopes. At the higher deceleration levels, the Nissan vehicle exceeded the 88-pound pedal force design goal for normal braking, cited in the Japanese specification.

Figure 5 summarizes the brake test results for the Toyota ESV. The Toyota has a four wheel disc, split system with booster and four wheel anti-skid. Again, results were obtained for stopping distance under normal and emergency braking. In each brake test category, the vehicle outperformed the specification requirement by a significant margin. The Toyota vehicle has demonstrated the best braking performance of any of the ESVs we have tested to date, stopping in a distance of 120 feet. This represents an average braking deceleration of approximately 1.0 g.

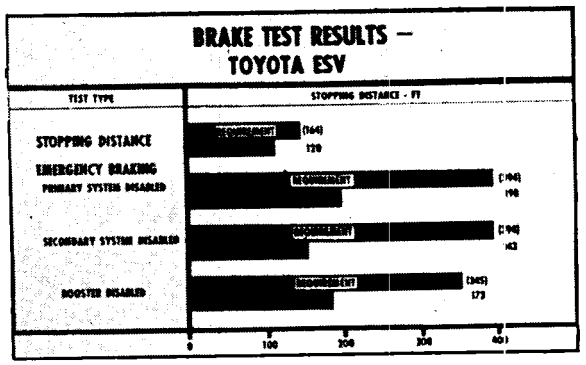


Figure 5

Pedal force gain results with the Toyota are presented in Figure 6. The vehicle performed within the required pedal force gain envelope, except during normal baking where pedal forces were somewhat lower than desired.

Now, to proceed with the handling test results beginning with the fixed control lateral acceleration for the Nissan, shown in Figure 7, the dotted line represents the minimum values in the Japanese ESV

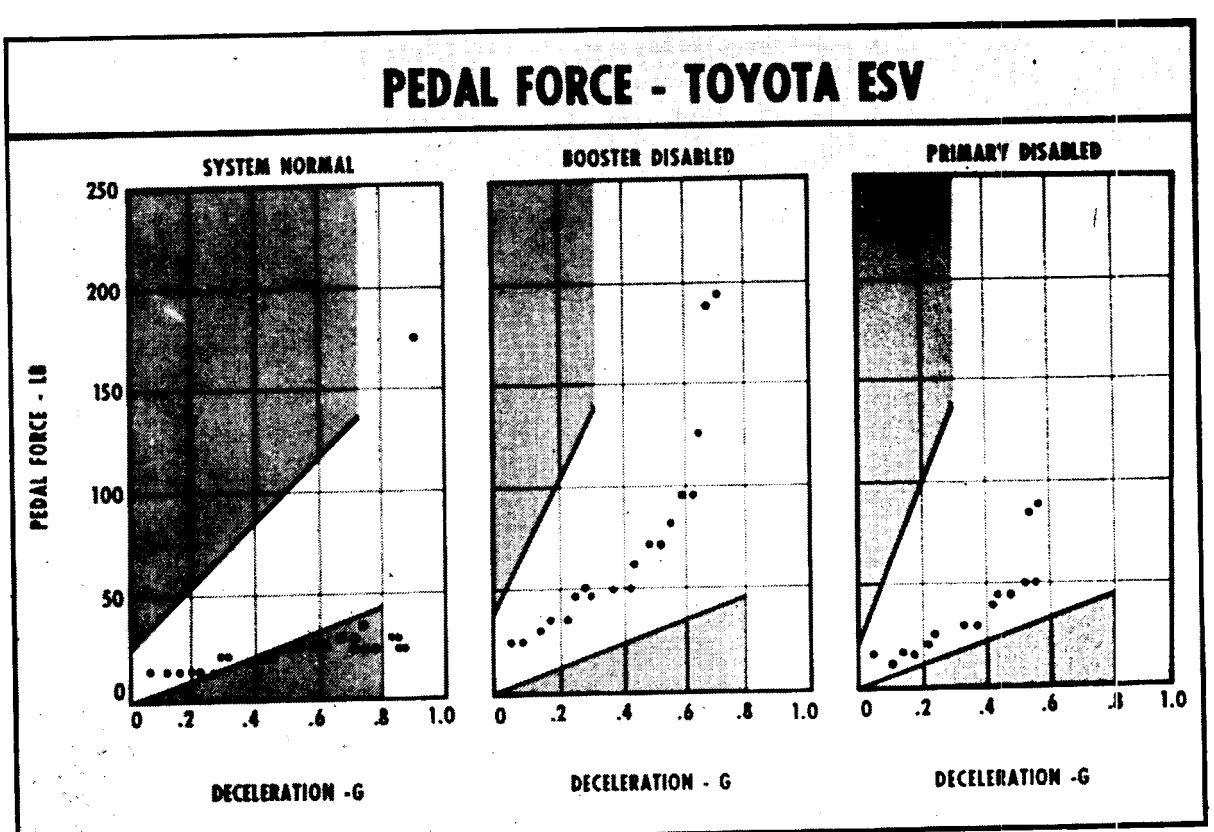


Figure 4

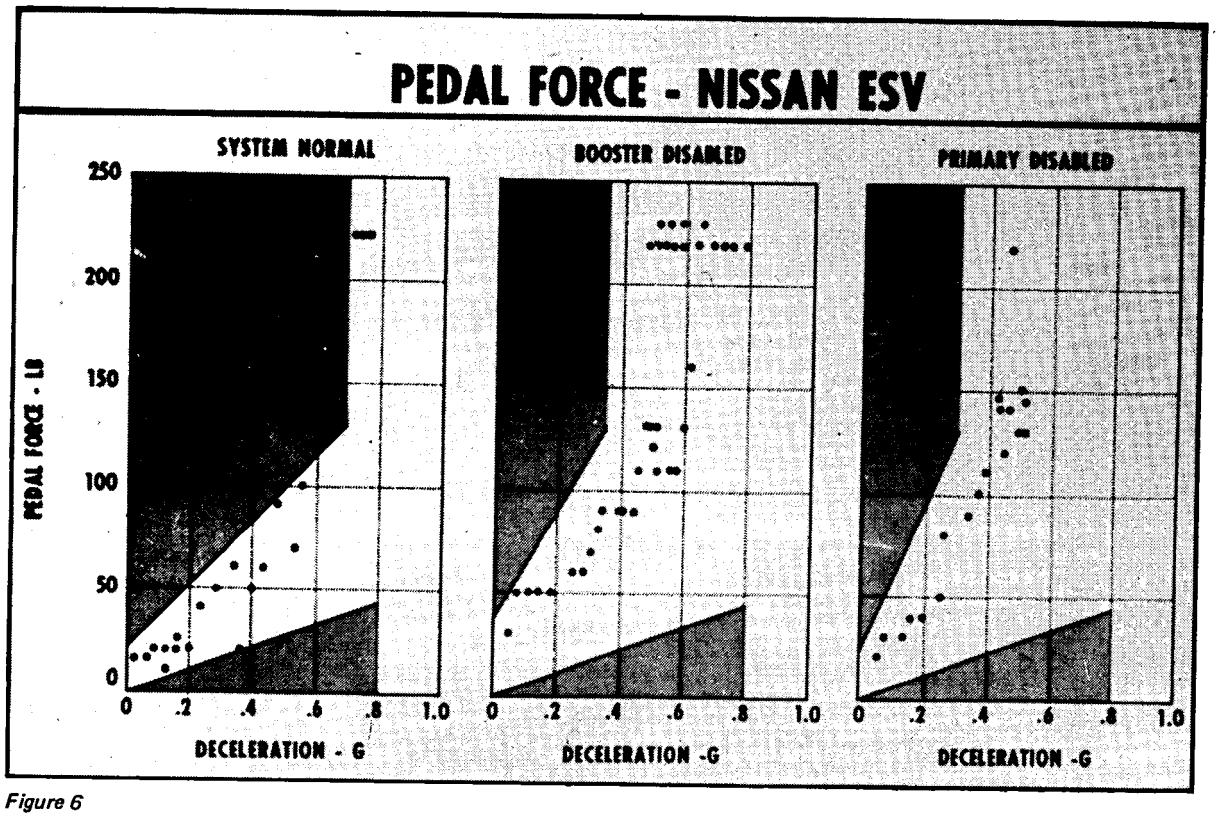


Figure 6

specifications. It can be seen that the Nissan more than satisfied the requirements of the specifications.

Figure 8 summarizes the results of the fixed control lateral acceleration test conducted with the Toyota. Quick examination of the results makes it evident that this vehicle also encountered no difficulty in bettering specification requirements.

The final tests involved the three Fiats that were crashed frontally with the AMF ESV. The first test was with a 2,000-pound class Fiat. Closure speed was 74.6 mph. Although this impact was a symmetric

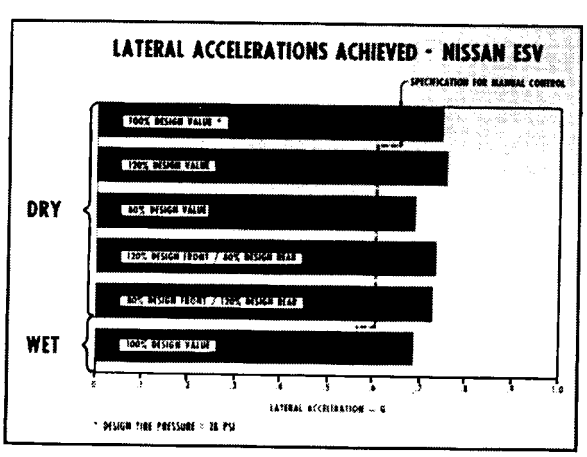


Figure 7

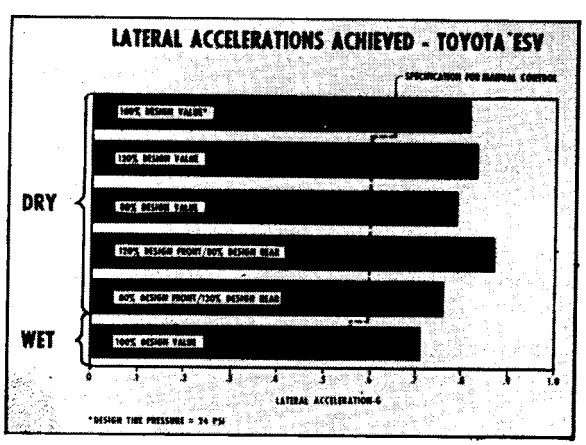


Figure 8

head-on collision, the AMF bumper rotated about 8°, indicating that asymmetrical loadings exist. This was partially due to the front wheel drive engine of the Fiat being located to the right of the vehicle centerline.

A summary of preliminary results is presented in Figure 9. The Fiat weighed 2,436 pounds (including instrumentation), and the AMF weighed 5,273 pounds. The Fiat velocity change was 56.5 mph which includes a 19.2 mph reverse velocity. The ratio of AMF/Fiat velocity change was almost inversely

SUMMARY OF TEST RESULTS– FIAT 2000/AMF ESV TEST

PARAMETER	FIAT 2000	AMF
TEST WEIGHT (LB)	2436	5273
VEHICLE SPEED (MPH)	–37.3	37.3
VELOCITY CHANGE (MPH)	56.5	24.6
DYNAMIC CRUSH (IN.)	24.4 @ 65 MSEC	21.IN./ 16.2IN
INTRUSION (IN.)	5.5IN./ 5.4IN	0
ACCELERATION (G)	–64.9 @ 46 MSEC	–26.2 @ 25 MSEC

Figure 9

proportional to the vehicle weight ratio, as we anticipated. The dynamic crush of the Fiat was 24.4 inches, while the AMF cylinder strokes were 16-22 inches. The maximum residual intrusions of five inches into the Fiat compartment were near the firewall centerline and should not present any real hazard to occupant safety.

Figure 10 shows the dynamic responses of the vehicles relative to deceleration and velocity histories for the occupant compartments. The peak vehicle decelerations were 26 g for the AMF and 65 g for the Fiat. The ratio of these peaks is approximately inversely proportional to the vehicle weight ratio.

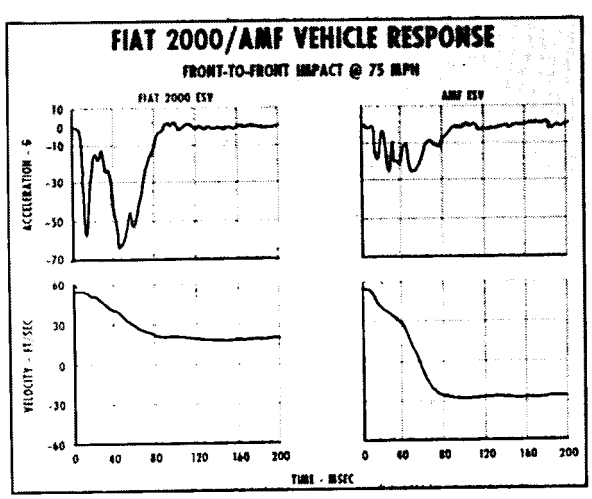


Figure 10

Another frontal crash test was conducted with a 2,500-pound class Fiat and the AMF ESV, again for the purpose of determining structural integrity and response. The AMF bumper stroked uniformly in this test, since this Fiat had a conventional front engine/rear wheel drive.

A summary of test results is presented in Figure 11. The test weight of the Fiat was 2,688 pounds, and that of the AMF, 5,273 pounds. The closure velocity was 75.6 mph and the Fiat velocity change was 56 mph. The dynamic crush was 24 inches while the AMF strokes were 19-20 inches. The maximum residual intrusion was less than five inches and occurred near the firewall centerline.

SUMMARY OF TEST RESULTS– FIAT 2500A/AMF ESV TEST

PARAMETER	FIAT 2500A	AMF
TEST WEIGHT (LB)	2688	5273
VEHICLE SPEED (MPH)	–37.8	37.8
VELOCITY CHANGE (MPH)	56.1	21.0
DYNAMIC CRUSH (IN.)	23.9 @ 48 MSEC	19.IN./ 18.8IN
INTRUSION (IN.)	2.6IN./ 4.5IN	0
ACCELERATION (G)	–64.5 @ 31 MSEC	–28.7 @ 24 MSEC

Figure 11

Dynamic responses of the vehicles are shown in Figure 12. They were very similar to those obtained in the first crash with peak vehicle decelerations of 29 g for the AMF and 65 g for the Fiat.

The third test was a frontal crash with the AMF ESV and a 2,500-pound class Fiat that contained two different load limiting front restraint systems in conjunction with two types of bucket seats. The steering wheel and column had been removed, and padding was employed across and below the dash. I should point out that the dummies were strapped in tightly, and the padding used in the upper portion of the seat back was not of head restraint quality. Since the restraint systems were of a very recent design, this was considered a development test and was con-

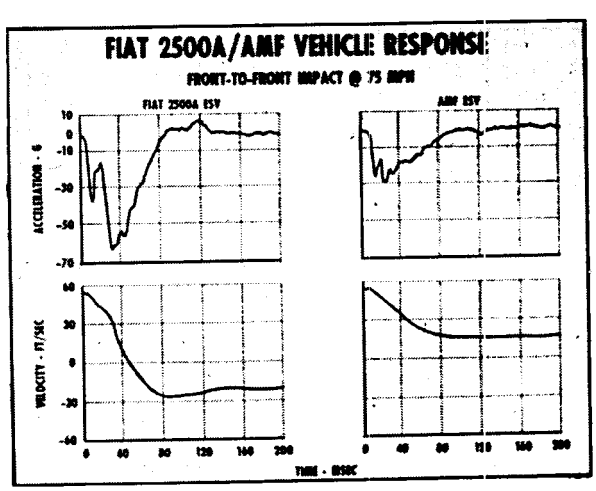


Figure 12

ducted at a reduced closure velocity of 49.3 mph. AMF weight remained at 5,273 pounds, and the test weight of the Fiat was 3,218 pounds, including the two dummies and instrumentation. The crush characteristics and dynamic response were at a lower level than those of the previously discussed test, so I will move on to the dummy data. Dummy head performance is summarized in Figure 13. The peak resultant

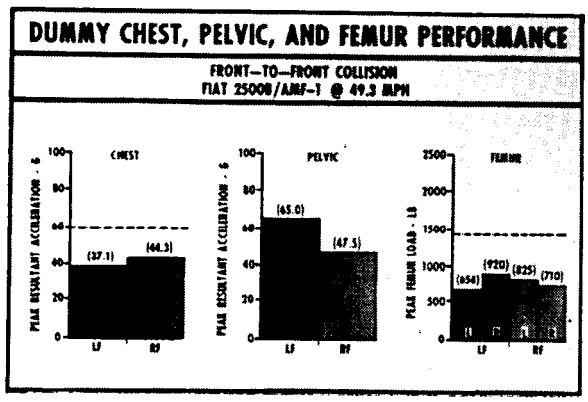


Figure 13

accelerations for both dummies during their forward motions were substantially less than the 1970 ESV performance criterion of 80 g. The severity indices during the 200 milliseconds following impact were also quite acceptable. Because of the vehicle pitching and the duration of the Fiat rebound, however, the dummies were still experiencing lower level accelerations well beyond 200 msc, which contributed to severity indices greater than 1,000 at 400 msc after impact. The left front dummy head also experienced high accelerations at 281 msc. This was a very short duration pulse, exceeding 80 g for only 3 msc, and resulted from the dummy's head impacting the unpadded B pillar during rebound. This rebound impact also increased the severity index, but the condition is not considered representative.

Chest accelerations are shown in Figure 14. The dummy chest performance is given in terms of peak resultant acceleration compared to an allowable of 60 g. Clearly, the chest performance for both dummies was well within that requirement. The peak pelvic accelerations and femur loads are also shown, and it is apparent that the femur loads were lower than the allowable 1,400-pound limit of 1970.

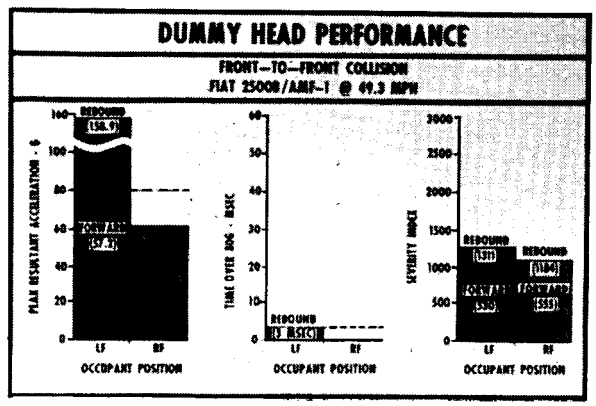


Figure 14

To summarize the Fiat tests, I believe it is appropriate to state that the vehicles performed very well and that fruitful knowledge has been gained by all parties involved.

In conclusion, we strongly encourage program participants to provide additional vehicles for cooperative testing in the United States. I would like to emphasize the word "cooperative," because we are more than willing to discuss, or negotiate, which tests are to be conducted, in order to avoid performing any that are counter to the desires of the developer. This is in consonance with the policy that we have followed in the past. We firmly believe that cooperation of this type can assist all of us in synthesizing the research and engineering results from each of the individual projects into significant and applicable knowledge.

Again, we wish to convey our sincere gratitude to those international participants who have furnished vehicles for us to test.

H. TAYLOR
Head of Safety Department
Transport and Road Research Laboratory
Department of the Environment and Chairman,
European Experimental Vehicles Committee

It is my privilege to speak on this occasion as Chairman of the European Experimental Vehicles Committee. I would like first of all to say something about the Committee before describing our most recent activities.

THE EUROPEAN EXPERIMENTAL VEHICLES COMMITTEE

The European Experimental Vehicles Committee was formed of government representatives in 1970 to coordinate the existing and proposed car safety technical activities of European participants in the international program; its scope was subsequently widened to include noise and exhaust pollution. Member countries are at present the Federal Republic of Germany, France, Italy, Sweden and the United Kingdom with the Netherlands and the EEC Commission represented as observers; most of the car designing countries of Western Europe are therefore represented. The EEVC has existed throughout the recent period of development of experimental safety vehicles consequent on the greatly increased public interest in providing better safety for car occupants. It maintains liaison between various European national research and development activities and provides a forum for clarifying views on the various technical options and on the response to the various international initiatives such as those stemming from the NATO Committee on the Challenges of Modern Society.

The EEVC has no executive or legislative function.

THE FUTURE FOR CAR SAFETY IN EUROPE

The work of EEVC has been considerably extended during the past year. Working groups have been set up to define on the basis of European experience, technical knowledge and facilities, a sufficiently common view for it to be able to present a soundly based assessment of the future needs for car safety in Europe. The working groups have benefitted from the participation of representatives of car manufacturing companies and some of their joint organizations.

I would like to turn now to some of the main findings in the EEVC report titled *The Future for Car Safety in Europe* prepared for presentation to this conference.

THE STUDY OF ACCIDENTS, THE BASIS FOR SAFETY DEVELOPMENTS

The collection of national road accident data has been continuing for about 65 years in some countries in Europe and the detailed study of accidents in one area has been in progress for some 20 years. So that today many basic analyses can be rapidly carried out which are vital to the planning of car safety and other road safety measures. It is clear that sufficient data exist on which to initiate car safety proposals though there is still room for improvement in collecting accident data. Much of the existing data are not published but are available to government officials and others concerned. Though ideally a high degree of uniformity and cooperation between accident investigations for car safety in Europe is desirable, a greater need may be to allocate support effort to make the best use of the data which are becoming available under existing arrangements. The need for uniformity or compatibility of definitions and procedures in accident studies is emphasized in the report.

Representing Accident Impacts by Test Procedures

Road safety is basically motivated by the public desire to save road users from injury. Damage to vehicles though of high economic importance is currently accorded a relatively low priority. Thus the damage and deceleration levels sustained by vehicles are primarily important insofar as they determine the injuries to their occupants. However, the link between them is by no means clear and it is considered that in the longer term the loadings on dummies and not vehicle characteristics only, must indicate the criteria for passing or failing an impact

test representative of frequent accident situations. In the short term there are several technical difficulties in doing this. Furthermore the rival claims of tests using complete dummies and those using simple forms representing single components of the human body must be considered. It was almost unanimously agreed that complete and representative dummies are necessary; simple forms often provide poor representations due to interactions between parts of the human body, some of which are brought to rest before others. For regulatory purposes it appears preferable to use a few impact tests making them as representative as possible of actual accidents.

A knowledge of human tolerance levels for each particular type of impact and consequent injury is essential for determining injury potential. The range of each human tolerance and how it varies with human size, sex, age and other differences between people has not always been fully recognized and allowed for. Of greater importance may be the need to calibrate the loadings on dummies to truly represent the tolerance levels of humans, which have themselves been estimated indirectly. The need is to record the level of accident impact resulting in some degree of injury in such a way that an impact test procedure can accurately reproduce the impact. Human tolerances have been estimated in the past from tests on human volunteers, animals, cadavers and by tests designed to match impacts resulting in known injuries in particular accidents. The last two procedures are probably the most useful, but whichever is used, decisions must be made in relation to the proportion of injuries which are to be avoided at a given impact level. This implies that enough data must be collected to discover the tolerances for various percentiles of the population. A review of the situation regarding the estimation of each tolerance level and the present estimates of each are noted in the report. There is a pressing need for a greatly increased amount of effort on determining human tolerances to the many different injuries occurring in vehicle accidents and the procedures should be chosen so that the results can bridge the gap between dummy test impact procedures and impacts in accidents on the road.

Test Procedures Which Should Lead to the Development of Safer Cars

Recent work in Europe and elsewhere has emphasized the need, not just for having test procedures to specify safety requirements for cars, but also for checking that cars complying with such requirements actually have better safety records than their predecessors. Better safety may not be achieved because either the tolerance or pass requirements are wrongly chosen or because the test is not really representative of accident situations. In the latter case, it is possible to be led seriously astray and for cars to be constructed with features that do not provide greater safety in practice, maybe at additional cost. Another need is to base requirements on good accident data, not only with regard to details but also for the frequency with which each situation occurs. Cost benefit considerations then show which tests have priority and for which accident situations only cheap modifications can be justified.

Although existing test procedures, such as the full frontal barrier impact, are the fully developed ones, this report suggests how they might be modified to take account of European accident studies, research into safety measures, and development such as in the ESV programs. These suggest that a substantial reduction in injuries may be expected if the proposals are implemented. However further assessment of the cost and benefits for a range of test severities for each of the tests suggested is needed before final proposals can be made. With this in mind the report makes the following comments on future test procedures and requirements for improving safety.

Accident Avoidance (Primary Safety). The necessity for studies in the field of primary safety or prevention appears less pressing than that for the other sectors of secondary safety and protection. Indeed most problems of primary safety have already been studied in detail and this has led to the establishment of strict specifications even on the international level. However, a list of topics, with an 'indication' of their degree of priority and practicability, has been established taking into account the fact that it is desirable and possible to progress in these fields.

Pedestrian Protection. Recent work in Europe has highlighted the possibility that the intractable problem of reducing pedestrian casualties may be partially solved by redesign of car fronts in profile and later on by preventing pedestrians struck by cars subsequently falling to the ground or striking the windshield or its frame. There is a clear need for further research and development in this field. Preliminary indications suggest that the front profile of a car should be low so that an adult pedestrian is struck below the knee by a bumper bar of reduced rigidity, which would reduce the severity of leg fracture and control his trajectory to some degree.

Occupant Protection (Secondary Safety). A great deal of effort of the EEVC was spent on car occupant protection needs. The report stresses the importance of the assumption that the wearing of seat belts by front seat occupants will become almost universal in Europe probably by an increase in compulsory

wearing. Little need was seen to provide protection for those choosing not to wear belts, although this may leave some car occupants unprotected. The general use of seat belts emphasizes the need to improve their convenience when putting them on and their comfort for those of well above or below average size. It also emphasizes the need for eliminating slackness by retractable belts or other means, and the great need for matching belts to cars' interior layouts so that they achieve their potential benefits, something that belts do not always do at present. Interlocks and passive belt systems were not fully discussed; the feeling being that the effects of compulsory wearing where introduced should first be assessed before requiring further equipment in the car.

All accident studies agree that the frontal impact is of greatest importance because of the high frequency of its occurrence. More limited evidence suggests that the large majority of injury producing frontal impacts (without belts in use) are at speeds equivalent to barrier impacts of less than 50 km/h. The test procedure to be selected is either a 60° or an offset frontal impact into a rigid barrier at 50 km/h with the measurements being recorded on two front seat dummies restrained by seat belts. Design studies are needed to find the cost penalty implicit in providing protection for occupants at different levels of impact.

The EEVC studied side impacts into cars and proposes that side impact tests should use as the striking object, a similar car to the one under test. The test, detailed in the report should also encourage designers to search for means of protecting the sides of the occupants and perhaps to have so little intrusion that in many car front to car side impacts the striking car may slide along the side of the car being struck and not lock into it.

This side impact test is particularly chosen to encourage the matching of the front and side structures of cars so that the fronts of cars are slightly less stiff than the sides and so do not easily intrude into them in these impacts. It is relatively easy to meet such side impact tests if the car front has a low bumper which strikes the side sills rather than the car doors and this design feature could also have some influence for providing protection for pedestrians struck by cars.

In Europe the frequency of rear impact leading to fatal or serious injury to car occupants is low. As a result the test suggested in the report for rear impact is intended to ensure that fire does not break out. It is also desirable that the strength of front seats should be considered. Fatalities to rear seat occupants can be reduced only by imposing a fairly severe rear impact test on the whole car and requiring structural integrity of the passenger compartment to be retained.

Though it may be hoped that the use of seat belts, the universal use of anti-burst door latches, and the now customary roof support strengths should be adequate to prevent most casualties in overturning incidents, a two rotation dynamic rolling test procedure is suggested. An alternative is to have a dynamic impact test on the roof front corner by pendulum or moving barrier. These suggestions reflect a lack of certainty that existing measures are adequate to prevent injury and some limited evidence supports this view; time will tell what is the actual situation.

THE FUTURE

The EEVC view is that these few carefully chosen car safety requirements should achieve worthwhile savings in casualties to car occupants in Europe. In the main these do not require sophisticated engineering designs for cars and should not be unacceptably costly nor impose much weight penalty. Car design teams need sufficient time to do their work of producing light and efficient designs. This latter feature is important for Europe in a situation of increasingly expensive fuel.

Europe is showing that it has the technical ability to study accidents, to make proposals and to carry them out in terms of new car layouts and equipment. With a more coordinated effort it will be possible to put these together to the advantage of the road user population in Europe and elsewhere.

The EEVC report provides important guidelines for improving car safety in the future, improvements which are not limited to countries participating in EEVC. These EEVC activities are not competitive with those of other organizations, who may benefit from the results, by helping to improve and harmonize, on a world level, technical regulations for the construction and equipment of vehicles. This work does not negate studies related to current technical regulations which have been carried out over a long period by existing organizations.

The report is by no means the last word on car safety; indeed one of its objectives is to highlight the work which still needs to be done. Nevertheless the committee hopes and believes that it will contribute to the improvement of safety standards on a worldwide basis.

THE FUTURE FOR CAR SAFETY IN EUROPE

A Report of the EEVC

FOREWORD BY THE CHAIRMAN

Cars have existed for only a relatively short time but during this period immense changes have taken place in the world, many of them as a direct result of the availability of powered road transport on a personal basis. With the many benefits have come some disbenefits; these have been largely attributed to cars which have attracted rapidly growing attention and public condemnation in recent years. The disbenefits are specifically the problems of road accidents, noise, exhaust pollution and traffic congestion; and, more recently, the rate of depletion of energy resources by road vehicles.

Research to improve the safety of vehicles has been carried out internationally for some time. Government administrators in collaboration with industry have started investigations to standardize regulations which differ from each other, and also to make recommendations for the highest possible level of safety, using available techniques, while remaining acceptable for production. This facilitates the sale of cars between nations, and at the same time improves their safety. Without underestimating this continuing work, some governments have initiated more advanced research, eventually to be used to strengthen regulations to give safer vehicles. Any gain in safety must be achieved within reasonable limits of cost, weight and dimensions. Furthermore, account must be taken of the special difficulties for small light vehicles and the problems of their compatibility on the road with larger, heavier vehicles.

In October 1970 a committee of European government representatives was formed which came to be known as the European Experimental Vehicles Committee (EEVC). The Committee maintains liaison between various European national research and development activities to increase safety and abate noise and pollution; it also provides a forum for clarifying views on the various technical options and on the response which should be made to the various international initiatives such as those stemming from the NATO Committee on the Challenges of Modern Society (CCMS). The EEVC has no executive or legislative function.

The report on car safety which follows is the result of initiatives sponsored by the EEVC which had the wholehearted cooperation of governments and in which representatives of the European car industry participated.

This report provides important guidelines for improving car safety in the future; improvements which are not limited to countries participating in EEVC. These EEVC activities are not competitive with those of other organizations, who may benefit from the results, by helping to improve and harmonize, on a world level, technical regulations for the construction and equipment of vehicles. This work does not negate studies related to current technical regulations which have been carried out over a long period by existing organizations.

The report is by no means the last word on car safety, indeed one of its objectives is to highlight the work which still needs to be done; nevertheless the Committee hopes and believes that it will contribute to the improvement of safety standards on a worldwide basis.

H. Taylor
Chairman, EEVC
June 1974

ROLE OF EEVC FOR CAR SAFETY IN EUROPE

The European Experimental Vehicles Committee has existed throughout the period of the development of experimental safety vehicles consequent on the greatly increased interest in 1970 in providing better safety for car occupants.

In spite of and further to the existing work, it was formed originally to coordinate the car safety technical activities of European participants in the international program; its scope was subsequently widened to include noise and pollution. Its members are mostly drawn from those working on research and development in this field for their respective governments but an observer from the EEC Commission attends meetings of the Committee for liaison purposes. Member countries are at present the Federal Republic of Germany, France, Italy, Sweden and the United Kingdom with the Netherlands and EEC Commission represented as observers; most of the car designing countries of Western Europe are therefore represented.

The work of EEVC has been extended during the past year. Working groups have been set up to define on the basis of European experience, technical knowledge and facilities, a sufficiently common view for it to be able to present a soundly based assessment of the future needs for car safety in Europe. The working groups have benefitted from the participation of representatives of several car manufacturing companies or some of their joint organizations. The conclusions represent the agreed position of EEVC and do not necessarily reflect the exact points of view of all participants. The remainder of the main text is a brief summary of the conclusions of EEVC, the working groups' more detailed reports being included as appendixes.

THE STUDY OF ACCIDENTS, THE BASIS FOR SAFETY DEVELOPMENTS

The collection of national road accident data has been continuing for about 65 years in some countries in Europe, with procedures being improved during this period, so that by today many basic analyses can be rapidly carried out, which are vital to the planning of car safety and other road safety measures. The study of accidents in one area in more detail by organizations other than the police has also been in progress for about 20 years, though procedures have now become systematic, with recently a better realization of the need for also collecting background information with which to interpret the results more fully.

The present position in Europe is discussed in Appendix 2, the report of EEVC Working Group 1. It is clear that sufficient suitable data exist on which to initiate car safety proposals although the report also makes recommendations for improvement in collecting accident data. The situation will improve when accident teams, which have only recently started work, have reached the reporting stage. Much of the existing data are not published but are available to government officials and others concerned. One problem has been the need to provide sufficient effort to analyze and publish studies once the actual investigations have been completed.

Though ideally a high degree of uniformity and cooperation between accident investigations for car safety in Europe is desirable, a greater need may be to allocate sufficient effort to make the best use of the data which are becoming available under existing arrangements. The recommendations of the Working Group 1 are given at the end of Appendix 2, the need for uniformity or compatibility of definitions and procedures in accident studies being a particularly valuable suggestion.

REPRESENTING ACCIDENT IMPACTS BY TEST PROCEDURES

Road safety is basically motivated by the public desire to save road users from injury. Damage to vehicles though of high economic importance is currently accorded a relatively low priority. Thus the damage and deceleration levels sustained by vehicles are primarily important insofar as they determine the injuries to their occupants. However, the link between them is by no means clear and it is considered that in the longer term the loadings on dummies and not vehicle characteristics only, must indicate the criteria for passing or failing an impact test representative of frequent accident situations. In the short term there are several technical difficulties in doing this. Furthermore the rival claims of tests using complete dummies and those using simple forms representing single components of the human body must be considered. It was almost unanimously agreed that complete and representative dummies are necessary; simple forms often provide poor representations due to interactions between parts of the human body, some of which are brought to rest before others. For regulatory purposes it appears preferable to use a few impact tests making them as representative as possible of actual accidents.

Test dummies should be strong, give repeatable results and be capable of calibration. They should be simplified rather than exact replicas of the human body with sufficient means of recording loads, deflections or decelerations appropriate to the test. Dummies stiffer and stronger than human beings are not ruled out if they can be used to verify the stiffness and energy absorbing properties of vehicle components struck by them. It may be that at some stage dummy design for regulatory purposes must be frozen so that a single design can be universally adopted for particular tests, though it would be preferable to be able to specify dummy performance requirements, rather than its design. In any case the development of both dummies and test procedures must continue and refinements incorporated in future requirements.

A knowledge of human tolerance levels for each particular type of impact and consequent injury is essential for determining injury potential. The range of each human tolerance and how it varies with human size, sex, age and other differences between people has not always been fully recognized and allowed for. Of greater importance may be the need to calibrate the loadings on dummies to truly represent the tolerance levels of humans, which have themselves been estimated indirectly. The need is to record the level of accident impact resulting in some

degree of injury in such a way that an impact test procedure can accurately reproduce the impact. Human tolerances have been estimated in the past from tests on human volunteers, animals, cadavers and by tests designed to match impacts resulting in known injuries in particular accidents. The last two procedures are probably the most useful, but whichever is used, decisions must be made in relation to the proportion of injuries which is to be avoided at a given impact level. This implies that enough data must be collected to discover the tolerances for various percentiles of the population. A review of the situation regarding the estimation of each tolerance level and the present estimates of each are noted in Appendix 4, the Working Group 3 report. There is a pressing need for a greatly increased amount of effort on determining human tolerances to the many different injuries occurring in vehicle accidents and the procedures should be chosen so that the results can bridge the gap between dummy test impact procedures and impacts in accidents on the road.

TEST PROCEDURES WHICH SHOULD LEAD TO THE DEVELOPMENT OF SAFER CARS

Recent work in Europe and elsewhere has emphasized the need, not just for having test procedures to specify safety requirements for cars, but also for checking that cars complying with such requirements actually have better safety records than their predecessors. Better safety may not be achieved because either the tolerance or pass requirements are wrongly chosen or because the test is not really representative of accident situations. In the latter case, it is possible to be led seriously astray and for cars to be constructed with features that do not provide greater safety in practice, maybe at additional cost. Another need is to base requirements on good accident data, not only with regard to details but also for the frequency with which each situation occurs. Cost benefit considerations then show which tests have priority and for which accident situations only cheap modifications can be justified. Procedures to ensure satisfactory progress are discussed in Appendix 2.

Although existing test procedures, such as the full frontal barrier impact, are the only fully developed ones, this report suggests how they might be modified to take account of European accident studies, research into safety measures, and development such as in the ESV programs. These suggest that a substantial reduction in injuries may be expected if these proposals are implemented. However further assessment of the cost and benefits for a range of test severities for each of the tests suggested is needed before final proposals can be made. With this in mind the EEVC would like to make the following comments on future test procedures and requirements for improving safety.

Accident Avoidance (Primary Safety). The necessity for studies in the field of primary safety or prevention appears less pressing than that for the other sectors of secondary safety and protection. Indeed most problems of primary safety have already been studied in detail and this has led to the establishment of strict specifications even on the international level. However, a list of topics, with an "indication" of their degree of priority and practicability, has been established by the Working Group 2 (Appendix 3), taking into account the fact that it is desirable and possible to progress in these fields.

Pedestrain Protection. Recent work in Europe has highlighted the possibility that the intractable problem of reducing pedestrian casualties may be partially solved by redesign of car fronts in profile and later on by preventing pedestrians struck by cars subsequently falling to the ground or striking the windshield or its frame. There is a clear need for further research and development in this field. Preliminary indications suggest that the front profile of a car should be low so that an adult pedestrian is struck below the knee by a bumper bar of reduced rigidity, which would reduce the probability of tibia and fibula fracture.

Occupant Protection (Secondary Safety). Most effort of the EEVC Working Group 2 described in Appendix 3 was spent on car occupant protection needs. The group stresses the importance of its assumption that the wearing of seat belts by front seat occupants will become almost universal in Europe probably by an increase in compulsory wearing. Little need was seen to provide protection for those choosing not to wear belts, though this may leave some car occupants unprotected. The general use of seat belts emphasizes the need to improve their convenience when putting them on and their comfort for those of well above or below average size. It also emphasizes the need for eliminating slackness by retractable belts or other means, and the great need for matching belts to car interior layouts so that they achieve their potential benefits, something that belts do not always do at present. Interlocks and passive belt systems were not fully discussed, the feeling being that the effects of compulsion should first be assessed before requiring further equipment in the car.

All accident studies agree that the frontal impact is of greatest importance because of the high frequency of its occurrence. More limited evidence suggests that

the large majority of injury producing frontal impacts (without belts in use) are at speeds equivalent to barrier impacts of less than 50 km/h. The test procedure to be selected is either a 60° or an offset frontal impact into a rigid barrier at 50 km/h with the measurements being recorded on two front seat dummies restrained by seat belts. Details are in Appendix 3. Design studies are needed to find the cost penalty implicit in providing protection for occupants at different levels of impact.

The EEVC Working Group 2 studied side impacts into cars and proposed that side impact tests should be with the striking object a similar car to the one under test. The test, detailed in Appendix 3, should also encourage designers to search for means of protecting the sides of the occupants and perhaps to have so little intrusion that in many car front to car side impacts the striking car may slide along the side of the car being struck and not lock into it.

This side impact test is particularly chosen to encourage the matching of the front and side structures of cars so that the fronts of cars are slightly less stiff than the sides and so do not easily intrude into them in these impacts. It is particularly easy to meet such side impact tests if the car front has a low bumper which strikes the side sills rather than the car doors and this design feature could also have some influence for providing protection for pedestrians struck by cars.

In Europe the frequency of rear impact leading to fatal or serious injury to car occupants is low. As a result the test suggested in Appendix 3 for rear impact is intended to ensure that fire does not break out. It is also desirable that the strength of front seats should be considered. Fatalities to rear seat occupants can be reduced only by imposing a fairly severe rear impact test on the whole car and requiring structural integrity of the passenger compartment to be retained.

Though it may be hoped that the use of seat belts, the universal use of antiburst door latches, and the now customary roof support strengths should be adequate to prevent most casualties in overturning incidents, a two rotation dynamic rolling test procedure is suggested in Appendix 3. An alternative is to have a dynamic impact test on the roof front corner by pendulum or moving barrier. These suggestions reflect a lack of certainty that existing measures are adequate to prevent injury and some limited evidence supports this view; time will tell what is the actual situation.

THE FUTURE OF CAR SAFETY IN EUROPE

The EEVC view is that these few carefully chosen car safety requirements should achieve worthwhile savings in casualties to car occupants in Europe. These mostly do not require sophisticated engineering designs for cars and should not be unacceptably costly or impose much weight penalty. Car design teams need sufficient time to do their work of producing light and efficient designs. This latter feature is important for Europe in a situation of increasingly expensive fuel.

Europe is showing that it has the technical ability to study accidents, to make proposals and to carry them out in terms of new car layouts and equipment. With a more coordinated effort it will be possible to put these together to the advantage of the road user population in Europe and elsewhere.

APPENDIX 1

THE WORKING GROUPS AND THEIR TASKS

The objective which the EEVC has set out to accomplish is to give recommendations based on research work for the definition of the characteristics that could be required of new production passenger cars which will be designed in the near future. The aim is to provide a more satisfactory level of safety for occupants, pedestrians and riders. This level should be established on the basis of a careful evaluation of the cost-benefit ratio and priority appropriate for each safety feature, derived from the best information available in Europe.

In order to accomplish the foregoing objective, three Working Groups (WG1, WG2 and WG3) were set up on an ad hoc basis within EEVC in the autumn of 1973 with the task of making quick assessments of present knowledge of the accident situation and the prospects for safer cars. The work was divided as follows:

WG1 WG1 was to review the sources of accident data available in Europe and to comment on how these should best be developed to further the aims of car safety. This would permit the definition and classification of accident problems in order of importance. Recommendations for the improvement of accident studies would also be made.

WG3 WG3 had the task of reviewing the technical problems involved in assessing occupant safety by impact test procedures. These studies were to include an assessment of currently available human injury tolerance limits, anthropomorphic dummies, and test techniques, together with recommendations for future research.

WG2 Based on the information discussed by WG1 and WG3, WG2 was to assess realistic safety requirements and to compare their priorities.

Apart from these groups, work is in progress to compare the various sets of economic assumptions used in European countries, as a basis for their cost benefit studies of safety measures.

APPENDIX 2

REPORT OF WG1 – A REVIEW OF DATA SOURCES FOR CAR SAFETY IMPROVEMENTS

AIM AND SCOPE OF THE REPORT

The European Experimental Vehicles Committee decided at its 5th meeting, held in Crowthorne in June 1973, to define the elements of a common position in the field of desirable safety improvements for new vehicles likely to appear on the European market in the early eighties. These recommendations, meant for national or international administrative authorities in charge of regulation and for automobile manufacturers should be based on the knowledge, as objective as possible, of the accident situation in Western Europe and on a comparison of the advantages and disadvantages of the intended measures.

Among the points to be examined to work out this common view, there is in particular the question of the sources of data on traffic safety. More precisely, the questions which can come up in this field are the following:

1. Do the different existing sources of data allow us to take up objectively and correctly the preparation or the decisions concerning the improvement of vehicle safety?
2. How is it possible to improve the existing information system in order to make it more efficient in respect of the aim of improving vehicle safety?

To answer these questions, it was decided to create a working group, called Working Group 1 (WG1), which got together twice in Paris, in November 1973 and in February 1974. (A list of participants in this group will be found in Supplement 1.) Besides representatives of national administrations, the presence of some representatives of the car industry appointed as national experts is to be noted. We also draw the attention to the fact that all the manufacturers who are members of the Group called the "CCMC" (Committee of Common Market Automobile Constructors) have given their viewpoint in an interview with the chairman of the working group. For material timing reasons, it was not possible to hear the representatives of the association of automobile constructors known under the name of the "BPICA," but there is scarcely any reason to think that they should disagree with the viewpoints given by their colleagues who have been consulted.

In these meetings, it appeared that there was a very broad consensus on how to deal with the problem of the sources of data on vehicle safety. The aim of this report is to state this common view. The report includes, besides a first introductory part, a chapter dealing with the present situation as regards the sources of data on safety and their present use, then a chapter dealing with progress still to be achieved in these fields. The last chapter summarizes the general recommendations of the group. It is here to be noted that, because of the wide variety of practice in European government organizations, some of the recommendations will mean the starting of some particular work in one country, but only the improving of an existing procedure in another. A series of Supplements describing briefly the present state of the information sources on safety, and a list of the principal study and research centers directly involved in these fields, is to be found at the end of the report.

REVIEW OF THE USE MADE OF ACCIDENT DATA IN EUROPE FOR THE IMPROVEMENT OF CAR SAFETY

The accident investigation situation in Europe is that national data from the police are generally available and that several detailed studies are in preparation, in progress or at the analysis stage. However, so far relatively few results have been

published, either because the results have been confidential to the organization carrying out the survey or because there has been insufficient staff effort available to complete the analyses and to report them formally. Thus the major purpose of this section is to state what data are available and how they are used. The availability is noted in the Supplements; the use that can be made is discussed in the main text.

Magnitude of Any Safety Problem

The first step is to assess how large is any safety problem being considered. National accident data supplied by the police may tell how many people are injured each year and to what severity, in a wide group of accidents including the problem under study. A detailed survey of accidents in one area may then show what fraction of these injuries are of the particular type and indeed whether the mechanics of the impact lead to the injuries in the way that was originally thought. Such a survey may supply an insufficient number of the particular type of accident and special arrangements may then have to be made to collect further evidence on which to design a suitable safety measure. An example will illustrate the procedure.

Suppose that a question is being asked about how front seat occupants can be protected from the high risks of being crushed by side impacts into their cars. National data might show that in one country some small percentage of all cars in which occupants were seriously injured were damaged at one side only, though in many other accidents frontal impact damage extended into the sides of the cars. A detailed crash injury survey of relatively few accidents might show that most of those injured had been sitting in the front, but that few of these had actually been injured by crushing, most having been hurt by direct impact with the car door or its surroundings when this had first been struck in the accident. From these details an estimate could be made of the annual total of those actually crushed in side impacts. But perhaps more significantly, the accident data would have shown that the wrong question may possibly have been asked.

In addition to estimating the magnitude of an accident problem for a recent year, it is sometimes important to estimate it on a similar basis for several earlier years so that the trend of the situation becomes clear. Accident problems may rapidly increase or diminish in importance at a quite different rate from the general traffic situation. For car safety situations, this usually arises from social or fashion reasons, for example increasing wealth has reduced the numbers of motorcycles in most countries so that the car/motorcycle impact problem has been diminishing, though the minimum for this particular trend may now have been passed. Another factor to affect trends is change in car design, for example the increasing proportion of front wheel braking for cars has reduced cars spinning due to rear wheels becoming locked and so side impacts in single car accidents seem to be decreasing as new models gradually replace old ones on the road.

The extent to which magnitudes of safety problems can be assessed in the European countries can be seen from the information in the Supplements. Broad estimates can be made in several countries based on police national accident data, while detailed estimates can be made in two or three countries. These enable particular car accident problems to be studied in some detail, and many such studies have been carried out. Doubts are sometimes expressed about the accuracy of police and other data and though it is true that some might be more accurate than others, it is also true that sufficiently accurate estimates can be made for many situations by specialist staff familiar with any problems or shortcomings of the data which they are using. A particular problem is that of some accidents not being reported at all; these largely fall into particular categories such as single vehicle accidents. Work in Sweden and elsewhere has highlighted the problem, enabling estimates to be corrected for appropriate situations.

Identification of High Risk Situations

Though it may be sufficient to know the magnitude of a safety problem, it is often useful to identify the background situation in which accidents occur. Are the accidents mostly in urban or rural areas, by day or during the hours of darkness, on motorway or other classes of road? National and more detailed studies give information about this, though the detailed studies are usually biased by being situated in one particular area, though allowances for this can often be made. Then safety problems differ, some occur frequently with a low risk of injury, while others occur infrequently but nearly always lead to injury. Studies of accidents without injuries as well as those with injuries are needed to show up the difference. Human attitudes to safety measures may be different in high and low risk of injury situations and difference is of importance when estimating the effectiveness of safety measures.

Accidents should be studied in relation to their background so that not just numbers of accidents, but data about accident rates will be available. Typical rates are accidents per hundred thousand

population, per hundred thousand kilometers driven, with different rates for distances driven on roads of different class or speed limit, by day or by night, or for journeys to or from work, or for social purposes during evenings or weekends. The possibilities are many, the value of them, however, much depends on the car safety investigation being undertaken. For example the study of loss of control of cars is as much a study of the drivers compared with the whole population of drivers and for these studies accident rate comparisons with good background data are invaluable.

The situation in Europe is that there are countries in which the attendant circumstances of accidents are recorded in some detail. The feature which is least often noted is the estimate of speed of impact because this is not easily estimated, even by experts. However it would be of great value to analysts of accident data. Sources of background data are noted in the Supplements but they are relatively few and often not matched precisely to corresponding accident data. The situation is best for national human and vehicle populations, road traffic and lengths of road available. Background data relating to the road layout situation, traffic speed distributions, driver and pedestrian behavior is just starting to be collected for one or two detailed accident surveys and these are particularly needed for primary safety studies.

The need for good background data to enable assessment of accident rates to be made, is shown by the overall comparison of accident rates between the various European countries. Most rates have a range of at least two to one between largest and smallest. No explanation of these differences has been conclusively demonstrated, though it seems likely to be attributable to differences in driving behavior. More detailed studies of background situations in which accidents arise, seem to be required; the present results show that no obvious differences in road usage account for the differences in accident rates.

Prediction and Ranking of Safety Measures

One task is to predict the effectiveness of a safety measure in saving accidents; for this two estimates are needed. First, a prediction is needed of the magnitude of the accident situation for which the safety measure might be of use, and second, an estimate of what fraction of these a particular form or specification of safety measure might save. The fractional contribution multiplied by the number of accidents which the measure could affect then gives the effectiveness, which can be compared or ranked with other safety measures. The estimation of the magnitude or number of accidents has been discussed earlier; the estimation of fractional contribution is now discussed.

Estimates of the fraction of accidents saved by a safety measure are made in several ways:

- Relying on expert assessment of accidents in which the measure was not available,
- A careful comparison of a few accidents in which the safety measure was present with a much larger number in which it was not,
- Large scale trials of cars with the safety measure in everyday use, or
- The final comparison of when the safety measure is in general use with the previous situation.

The estimation of the effectiveness of a safety measure is necessarily the work of an expert, familiar with both the results of detailed accident investigations and the effect that a safety device or impact specification might be expected to have. If one or two examples of a safety measure exist, they can be tested to improve the basis of the estimates. Having a firm idea in mind of the effect of the safety device, the investigation will consider each of a whole set of accidents in a study in turn and estimate for each, what probability the proposed safety measure might have had on the outcome of the accident if it had been present. A difficult aspect of this work arises if it is possible that in a minority of cases the measure might make an accident situation worse, because in such circumstances there may be no reported accidents on which to assess this effect. If the measure can be specified with, for example, different test impact speeds, it is important to repeat the estimates of effectiveness for several speeds; the final estimate of performance is then shown to vary from zero for a test impact speed such that all existing cars meet the requirement, up to some value for a test speed higher than the speeds at which cars actually have their impacts.

If several examples of a safety measure exist they can be tested on the roads to check that they actually work and do not develop defects or prove to be unusable. If the numbers are larger, their use in service can be checked to look out for any involved in accidents which would give the first direct evidence whether safety protection has been provided. This is the "clinical" approach: comparing the evidence for one accident with that of another matched accident in which all conditions were as similar as possible, except that in one the safety measure was not present.

Occasionally it may happen that many cars are in service with the safety device fitted to them, either by chance design or because a manufacturer has adopted a measure before it is legally required. This situation is similar to that of the retrospective review

of a safety measure, which is discussed in the next section.

Whichever procedures are used to estimate the effectiveness of a safety measure, it is inevitable that the estimates are based on incomplete knowledge and in fact reasonable proof may be enough, as scientific proof is usually impossible.

The ranking of a set of suggested safety measures is another problem. Each has to be considered with respect to engineering feasibility for production and in use, cost to produce on a large scale, effectiveness in saving accidents or injury and acceptance by the public whether accepted voluntarily, after propaganda, or when required legally. From these considerations a degree of justification for each device may be agreed and also the best specification (such as of impact speed) and the earliest possible date of introduction may become apparent.

From this evidence a ranking policy can be proposed, the ranking being in chronological order of introduction of each safety measure. Some would be rejected and others left to await satisfactory engineering development if thought to be desirable in other respects. The question of interaction is most important for ranking because the introduction of one device before another may reduce the possible effectiveness of the later one. In general terms secondary safety measures protect the same set of accidents as those prevented by primary safety measures and so there is interaction between them. Different primary measures less often interact with each other and similarly for secondary measures. The effect of interaction, which applies to the whole road safety policy (road, road user and vehicle), is that measures introduced early on assume great importance. Interaction explains why the effects (in terms of casualties) of several measures is not usually the sum of the effects of each measure if it was taken alone. Exceptionally, two measures put together may save more injuries than the two separately (e.g., wearing belt and some redesign of car interiors or lighting speed), but this does not alter the general situation that interaction usually reduces total benefits. It is to be noted that, besides interaction, some other reasons explain why the effectiveness of the whole set of possible measures is never total: there are some specific accidents and injuries which cannot be saved in the present state of technique as we are not able to design any suitable measure which would apply to them (e.g., the pedestrian problem), and besides, no economically possible measure is ever a hundred per cent efficient.

The situation in Europe is that several countries have been predicting and ranking safety measures for some years on the basis of barely sufficient accident data, but sufficient progress has been made to justify the effort. With results of further accident studies becoming available as indicated in the Supplements, the accuracy of the estimates is likely to improve. Because this situation has been reached, it has been possible for the EEVC to set up the three Working Groups to review the overall situation in Europe.

Evaluation of Existing Safety Measures

The final stage in evaluating a safety measure is to check its efficiency in reducing accidents or injuries when in service. If introduced on a large scale voluntarily before legislation, this may be done then to ensure that legislation for universal adoption is correctly based, otherwise it must lag several years behind legislation until many cars are on the road and have had sufficient accidents. Statistical studies of the number of cars to be fitted with a safety device show that large numbers are needed for most accident situations for definite conclusions to be reached about the effectiveness of a safety measure. Thus these trials can be very expensive and difficult to manage. A careful matching of accidents in trial and control groups of cars can ease the difficulty of analysis.

The situation in Europe is that only a few final evaluations have been satisfactorily carried out. No entirely satisfactory comparison between toughened and the recent laminated glass has been completed because the different glasses are used in different countries. On the other hand it does appear that some forms of energy absorbing steering column have not been providing much protection. There have been small trial comparisons between antilock braked, load sensing braked and ordinary braked articulated vehicles; these have shown the expected results in most respects but have not been conclusive.

Methods for Assessing Human Performance in Accidents, Human Attitudes to Safety Measures and Human Tolerance of Injury

One reason why slow progress has been made with primary safety measures, relating to the relationship between the driver and his car (his comfort, control loads and movements, his perception of the driving task from the driving seat, and of warnings by instruments), is that it has proved difficult to measure what the driver does and what sensations influence him at the actual time of an accident involving him. A little progress has been made by careful interviewing of drivers after accidents in detailed accident studies. More has been done to measure what a driver does when driving normally but this may differ from his reactions to accident situations.

It can be misleading to rely on interview methods for assessing driver attitudes to safety measures, such as the wearing of safety belts or their response to warnings of car troubles or hazards ahead. Again, it seems best to rely on evidence of behavior at actual accidents, as collected in some current investigations.

Working Group 3 is concerned with stating the situation regarding the estimation of human tolerance to injury. What is needed for progress in car safety design is injury threshold levels which can be used in test impacts for design purposes. All that need be said in this section is that useful progress can be made by using the actual accidents to find how people are injured and by then comparing these impacts with similar test impacts in the laboratory, using the same model of car in similar impact conditions. The speed of impacts to produce similar damage on the parts of the car structure hit by the occupants can then be found and from this the test speed for the injury threshold. Existing detailed crash injury studies are adequate for this work when the deformations of car components struck by the casualties are measured.

THE FUTURE NEEDS FOR DATA SOURCES ON CAR SAFETY IMPROVEMENTS

Introduction

Earlier parts of this report have reviewed the existing state of data sources on road accidents in Europe. The aim of this section is to outline the future needs as seen by the members of WG1.

It was thought that realistic planning of research should consider the problems of today and the next ten years. To consider the future beyond the next decade was thought to be too speculative. On the other hand many aspects of vehicle design require a considerable lapse of time before they can be incorporated into vehicles, and then distributed throughout the vehicle population in significant numbers. Good accident research must allow for this, and also it should predict where possible specific problems will develop in the future, as a result of changes in the system taking place today.

General Aims of Future Research

The collection and analysis of data on actual road accidents should be integral parts of the wider problem of traffic safety and associated measures which aim to reduce death, injury and property damage. Field accident research must therefore be closely linked to government and industry, and thought of as a necessary prerequisite to any design or legislative change.

In essence, accident studies, as far as the vehicle is concerned, should provide answers to two types of question.

First, specific mechanisms of injury and the nature of particular collisions need to be understood. Although this work can be conducted to a degree in the laboratory, with the use of volunteers or analogues of one type or another (animals, cadavers, mathematical models or dummies), the final test of any design feature is its actual influence on a real person in a real collision. The roads of Europe each day provide a laboratory in which designs are perforce being tested, and this information should be evaluated as scientifically as possible.

Second, accident studies should provide knowledge on the frequency of various events. Only with information of this type is it possible to identify the relative importance of specific problems, establish priorities for remedial measures, and predict the likely benefits of certain courses of action. The laboratory cannot provide this type of data.

In recent years as accident research has become more scientific, the great importance of good experimental design has become apparent. This has produced the need for more carefully structured studies to solve the problems of representative sampling, illustrating the great need for compatible control data, for general background data on vehicle ownership and use, and for particular attention to be paid to the study of accidents which produce no injury. In planning accident research in the future it is important to resist short term unstructured projects which attempt to examine too many uncontrolled variables, and fail to produce significant results because of inadequate experimental planning. Although there are many urgent problems requiring immediate answers, it is important for the users of the results of accident studies to realize that future work may contradict the present conventional wisdom, and a continuous refining and monitoring program is necessary if optimum legislation and vehicle design for crash protection are to evolve satisfactorily.

Specific Aims of Future Research

This section outlines certain aims towards which future research should be directed. These aims are common to all motorized countries, and additional benefits to knowledge could be achieved if, without compromising specific national priorities which undoubtedly vary, compatibility and uniformity of

data collection and analysis procedures could be introduced internationally.

The Magnitude of the Problem

It is fundamental that each country should have a national system for collecting some information on all injury producing accidents occurring within its borders. This information is necessary to evaluate the place of road traffic injuries in comparison to other causes of death and morbidity which are prevalent. Only then can a reasonable policy for the allocation of national resources between the competing demands on public and private expenditure be developed. Reports obtained through the police are, in most countries, the best basic source of data. However, checks are required to evaluate what proportions of accidents of differing types are underreported by the police system.

The Working Group recommends the adoption of a basic accident recording system of relatively simple format. More emphasis should be given to vehicle parameters, and all the information collected should be of a simple, factual nature, with no room for value judgements. This latter type of information, for example the descriptions of driving behavior in such terms as excessive speed or absence of training, as explanations for the occurrence of a collision, have been shown to be of little value, and indeed may be harmful. It should always be remembered that these nationwide data are in most countries supplied by policemen, who are performing this task as a subsidiary one to their main activities.

The advantages of uniformity of definitions between nations are important for these basic data. While most countries for example now use the 30 day standard definition for a traffic death, injuries of lesser degrees are defined very differently from country to country. In the long term it will be essential for the development of common European policies, to have common agreed definitions of the basic characteristics of accidents.

Beyond the collection of basic data on a national scale, the Working Group recommends the development of two further levels of field accident study. The second level would produce more comprehensive data than the total national system, but would be restricted to, for example, 20 percent of all injury producing accidents. This sample would be carefully structured to ensure that it is representative of the relevant variables. These second level studies would be of limited duration to provide representative data on specific problems. The particular data collection procedures would depend on the problems being studied. In France, for example, a project is underway in which every fifteenth accident will be photographed by the police, thus providing detailed information on vehicle deformation. Similar sample studies are underway in Australia and Sweden, where procedures have been established so that all injured car occupants who were wearing seat belts are evaluated. In the Federal Republic of Germany the HUK organization provides detailed descriptions of the collision circumstances and associated injuries based on a data source of some 30,000 accidents obtained from insurance files.

Of particular importance for Europe at the present time is a need to resolve the conflicts which exist between the protection requirements for different classes of road user and different collision types. The Working Group recommends that the following studies be initiated at an early stage:

- A study of the conflicting requirements between pedestrian protection, car occupant protection and reduction of vehicle repair costs.
- A study of the frequency of various collision types and their speeds in relation to proposed impact tests. Information is already available which demonstrates that the conventional 50 kph barrier test is representative of only a minority of collisions. The appropriate test conditions for front, front corner, side and rear impact collisions need to be justified before their adoption as legal requirements.

The third level of field accident study recommended is the specialist in-depth approach, where relatively small samples of accidents (a few hundred per year) are investigated in considerable detail. This type of research is necessary for an understanding of specific mechanisms of injury. While the small sample sizes may preclude these studies from having strict statistical validity, they can nevertheless be related to the larger population of accidents by careful use of scaling procedures related to injury severity and collision severity, such as the equivalent test speed concept.

The Working Group therefore suggests that the development of this three level approach to field accident data should form the basis of future work in each country with as much compatibility of definitions as can practically be achieved. The accident investigation studies initiated by CCMS-NATO provide an excellent start, and such cooperative projects should be developed further.

Detection of Specific High Risk Situations

In addition to the basic accident data collection projects discussed above, it is considered vital for parallel sources of data to be developed for the detection of specific high risk situations.

Here the need for control data becomes apparent. For questions concerned with primary safety for example, it is necessary to know the nature of the traffic mix, the relative proportions of different makes and models of vehicles in use, their speeds and their respective annual mileages. Similarly other problems require knowledge on occupancy, age, sex, presence of children, seating positions, use of seat belts, use of headlights and many other factors.

In the past the acquisition of these types of control data have largely been neglected, with the result that the significance of much accident data was lost. The Working Group recommends that careful experimental design techniques be used more widely, so that the value of relatively expensive accident data is enhanced by the collection of appropriate controls.

To this end there is much useful information which can be obtained by integrating information from a variety of sources such as licensing authorities, motor trade associations, insurance companies, vehicle fleet operators and the like.

Priorities of Proposed Countermeasures

The aims of improved vehicle design and vehicle safety legislation are to reduce the frequency and severity of accidents and their attendant injuries. These apparently simple aims may in reality not be as easily discernible as at first sight appears. This is because of a great amount of interaction between all the contributing factors which come together to create traffic injuries. Many examples of this interaction exist: for instance, head restraints reduce the frequency of injury to the neck, they can also in some cars obstruct rearward vision; the front structures of cars can be optimized for maximum occupant protection in the standard barrier 80 kph collision, but if they are, then in head to side collisions the bullet car causes excessive intrusion into the side of the target car.

Primary safety countermeasures in particular require more analysis. The Volvo studies on the likely benefits of antilock braking are examples of how a priori assessments of countermeasures can be made.

The evaluation of any countermeasures, either for primary or secondary safety, is a three stage process. First, its effectiveness is assessed at the laboratory level. Second, the frequency of the particular events which would be modified by the countermeasure are estimated. Third, if the countermeasure appears to be effective thus far, an experimental trial can be designed, the aim of which would be to measure the success of the device in quantitative terms under controlled circumstances.

Implicit in the assessment of countermeasures is the value to be assigned to an injury. The Working Group is convinced of the importance of cost estimates as a necessary prerequisite to cost-effectiveness studies. Although necessarily approximate and indeed notional for some items, cost benefit assessments are considered essential for establishing priorities of countermeasures. It should be remembered that most cost benefit studies are comparative, in the sense that the consequences of one possible course of action are being compared with those of another. Because of this fact, the results from such studies need only to rank choices, so that absolute values of costs do not have to be precise. More research is required on the costs of accidents and injuries however, because average figures for all types of injuries are inadequate for the assessment of countermeasures which influence certain specific types of injury.

The Working Group recommends that all suggested regulations should, where possible, be examined from a cost-effective point of view, before their implementation. Also, when appropriate, research programs should be similarly scrutinized, so that priorities for future research can be developed more rationally than at present. For example, recent work on the frequency of overtaking has shown that to concentrate a large effort on that particular aspect of driving behavior is not likely to produce great benefits, because it is associated with a relatively small number of accidents.

Only by cost-effective analysis, based on the known frequency and severity of various events can the conflicts in vehicle design between front, side, rear and pedestrian protection be resolved at optimal levels. Therefore, attempts should be made wherever possible to predict effects on the population of collisions before the introduction of changes.

Evaluation of Design Changes and Regulations

Once a change has been introduced, it is essential for follow-up studies to be undertaken to quantify the effectiveness of the change. The experience of all previous accident, and indeed all epidemiological, research has shown that the reality is often very different from the conception. Common sense has that curious property of being more correct retrospectively than prospectively.

Wherever possible, the introduction of a change should be preceded by an evaluation of the problem, the first half of a before and after study. Only then can a sound evaluation of the success of the measure be made.

As consequence of this fact, the introduction of several measures at the same time should be resisted, even though there may be administrative attractions

for such a course. If this is not possible then experimental trials should be initiated before decisions are made.

One aspect of effectiveness studies which will become more important in the future is the examination of noninjury accidents. This type of collision presents special difficulties if it is to be systematically evaluated, but this research will become more important in the future. For some specific questions it is already essential, the evaluation of the effectiveness of seat belts for example.

Human Tolerance to Impact and Performance Criteria

The Working Group agreed that one of the most important and immediate problems at the present time is to develop realistic injury criteria for car design. The establishment of performance standards is to be favored over design standards, but a necessary prerequisite to such standards is a knowledge of the appropriate values (in engineering terms) which can be assigned to the various parts of the body, as tolerance levels to impacts.

More work is also required on the severity of injury for which a tolerance level is specified. It is unrealistic, for example, to attempt to prevent all injuries from the minor to the fatal in all accident circumstances. Certain levels of injury must be accepted in some situations. Presumably the avoidance of death is the primary requirement, but the consequence of such a criterion is the acceptance of moderate levels of injury in less severe collision circumstances.

In-depth field accident studies can provide injury tolerance data. Laboratory studies can also contribute knowledge in this area, particularly work on cadavers. It is important that these two types of research are coordinated as closely as possible to ensure that the laboratory conditions realistically reflect actual circumstances on the roads. This type of work is paricularly important in the light of recent findings, which have demonstrated that dummies at their present state of development are far from being realistic simulations of the human frame. Of vital importance at the present time is the correct definition of head and chest injury criteria.

Field accident studies however are absolutely fundamental to the development of correct injury criteria, and the Working Group recommends that the objectives of the in-depth studies should be coordinated, internationally where possible, in an effort to produce better criteria than those proposed at present. These studies, by their nature are expensive, and therefore their value should be maximized by good exchange of knowledge at the technical level.

A particular type of research to be encouraged is the reproduction of specific collisions in the laboratory. With the actual injuries to an individual established from the field, the accuracy of the proposed tolerance levels and their measurement can be evaluated in an experimental collision.

An important part of the human tolerance question about which there is almost total ignorance at the present time is how human tolerance values vary throughout the population at risk because of age, sex and exposure differences. This is an important area of inquiry which needs immediate study. First indications are that for some major body areas, the chest for example, human tolerance values may well vary by a factor of four between the fifth and the ninety-fifth percentile levels. The consequences of this fact on the single values currently suggested for injury criteria requires evaluation. The in-depth field accident studies discussed above should be directed at this question.

RECOMMENDATIONS

As a conclusion of this report, the members of the Working Group 1 make the following recommendations:

1. Generally speaking, the study of vehicle safety should be placed in the overall framework of accident studies and use all the available data collected by public authorities, insurance companies, car manufacturers, research organizations, etc. These data should be available without any restriction and made compatible as far as possible. Field accident research must be closely linked to government and industry and thought of as a preliminary for most legislative changes.
2. In order to know the importance of the different safety problems, a data recording system on road accidents with casualties should be available at three levels:
   - At the national level, information collected by the police should state objective facts, simple to write down, and include the essential data on vehicles.
   - To this first level, an intermediate level should be added, to provide representative data dealing in a more detailed way with specific problems considered as having priority.
   - The third level of data collection is an in-depth investigation by multidisciplinary teams of samples of accidents which are necessarily small and not always representative.

- The definitions and procedures used at these three levels should be made as uniform, or at least as compatible as possible. For this purpose, it is suggested that the EEVC should have a minimum list of the data to be collected and of the corresponding definitions made to send to the competent administrative authorities for adoption.

3. In order to detect abnormally dangerous situations, it is necessary to complete the collection of accident data by data measuring risk exposure (e.g., distances covered by different categories of drivers and vehicles), so making it possible to compare groups of casualties with control groups. For this purpose, the classical techniques of experimental design for data collection should be used systematically.
4. The proposals of measures for vehicle safety amelioration should be examined, as often as possible, from the point of view of their probable cost and efficiency. When a potential measure seems to be expensive, efforts appropriate to the importance of the measure, should be made in order to estimate its efficiency by means of experiments before its definite adoption. The orientation of research programs should also take into account, as often as possible, the probable advantages and disadvantages of the measures which are to be examined from a technical point of view.
5. The decisions taken in the field of vehicle standards should be assessed a posteriori. But because of the practical difficulties which are often raised in this type of assessment, it is recommended that the decisions should be taken step by step, in order to facilitate the experimental study of the efficiency of the different intended measures.
6. The accident data collected should make it possible to study the tolerance of the human body and the vehicle performance criteria.
   - As far as the tolerance of the human body is concerned, priority should be given to the studies of head and thorax injuries which take into account the variability of data on the population exposed to risk. As for vehicle performance, priority should be given to studies of the definition of crash test conditions representative of the real situation.
   - Also, a study of the conflicting requirements between pedestrian protection, car occupant protection and the reduction of vehicle repair costs should be initiated.
   - These studies should be based on a combination of the observations collected in real accidents and in accidents reconstituted under controlled conditions.
7. Although there are difficulties in investigating primary safety, this field should not be neglected in future accident research.
8. The cooperation and coordination of these particularly difficult studies between the technical research centers should be widely encouraged internationally.

SUPPLEMENT 1

LIST OF PARTICIPANTS IN WORKING GROUP 1

Federal Republic of Germany:

Fellerer	VDA/BMW
Friedel	BAST
Langwieder	HUK-Verb
Kreuzberg	Opel

France:

Bluet	ONSER
Halpern-Herla Chairman	ONSER-IRT
Ledru	SETRA
Muhlrad	ONSER

Italy:

Strampelli	Ministry of Transport
Gastaldi	Fiat

Sweden:

Bohlin	Volvo

United Kingdom:

Mackay	University of Birmingham
Neilson	TRRL
Waters	DOE

SUPPLEMENT 2

ROAD ACCIDENT INVESTIGATIONS IN THE FEDERAL REPUBLIC OF GERMANY AND BERLIN (WEST)

Road accident research in Germany is characterized by a great number of individual activities and investigations. A meaningful coordination and rational plans for the promotion of

accident research are the most important tasks of the Department of Accident Research of the Federal Road Research Institute, conceived as central accident research agency.

In the following the sources of the most important road accident investigations are laid down in the form agreed upon at the Second Meeting of EEVC WG 1. The objectives of these investigations are distinguished into the groups mentioned below:

1. Scope and severity of road safety problems.
2. Identification of dangerous conditions of situations.
3. Investigation into the problem: accidents versus motor vehicles.
4. Effects of accidents on human beings.
5. Effectiveness: priority listing of countermeasures.

A detailed list of references can be obtained from the Bundesanstalt fuer Strassenwesen (Federal Road Research Institute), Bruehler Strasse 1, 5 Koeln 51, Federal Republic of Germany.

Accident Data

1. Basic Statistics

Organizations – (a) Federal Bureau of Statistics; (b) State Bureau of Statistics; (c) Local Authorities Statistics (e.g., city or county police); (d) Federal Motor Vehicle Department (KBA).

Aims – 1, 2.

Numbers, dates and areas – Recorded are: the number of accidents; the number of persons involved; the number of fatalities and injuries; and the number of accident causes.

The Federal Motor Vehicle Department (KBA) publishes annually statistical data on the degree of motorization, private motor vehicle ownership, commercial vehicles, trailers, registration of vehicles and certificates of title, transfer of title, and deregistration of motor vehicles and trailers.

2. On-the-Spot Surveys of Accidents

Organizations – (a) leading car manufacturers; (b) Federal Road Research Institute (BAST).

Aims – 3.

Numbers, dates and areas – 200-400 in-depth studies of cars including detailed technical and medical analyses. A part of these investigations is included in the NATO/CCMS pilot study: "Accident Investigations."

3. Detailed Analysis of Causes of Accidents

Organization – Motor Insurers Bureau (Verband der Haftpflicht-, Unfall-und Kraftverkehrsversicherer, HUK-Verband).

Aims – 1, 2, 3.

Numbers, dates and areas – In 1970-71, 63,084 files on serious injury traffic accidents were evaluated. The objective of this study was the establishment of a general data bank to enable the identification of the most critical accident causes and the involved traffic participants.

Organization – Federal Road Research Institute (BAST).

Aims – 1, 3.

Numbers, dates and areas – Due to the unfavorable development of traffic accidents in 1970, a general study of the relation between traffic accidents and highway traffic was made. The study was mainly based on statistical data and publications from the Federal Bureau of Statistics, the State Bureau of Statistics, the Federal Motor Vehicle Department, the Federal Ministry of Transport as well as on other relevant publications.

Organization – State and community police.

Aims – 2 and improvement of geometric design of roads.

Numbers, dates and areas – A special card system on a local basis detects critical accident types and gives information on the frequency of typical collisions, run off the road accidents or roadside obstacle accidents.

4. Crash Injuries Studies

Organization – HUK Association (as mentioned above).

Aims – 3, 4.

Numbers, dates and areas – The study covers 30,000 accidents with passenger injuries. A further study is in progress this year, covering 100,000 accidents. These studies will provide conclusions on the risk of suffering a certain injury depending on the car model and the seating position in the car.

Sources of Background Data for Accident Studies

5. Vehicle Examination Studies

Organization – Motor Vehicle Inspection Department (TUV).

Aims, numbers, dates and areas – In the Federal Republic of Germany, all motor vehicles are subject to a roughly biennial inspection, in particular in respect to proper function of essential technical safety elements such as brakes, lighting equipment, etc., at any one of the Official Motor Vehicle Inspection Departments where experts and inspectors are

in charge of this task. Statistical records are kept on all defects detected. The statistics have no bearing on traffic accidents.

Organization – Motor Vehicle Institute of the Technical Allianz Center.

Aims, numbers, dates and areas – Special vehicle examinations are made in respect to structure, primary safety and driver behavior.

Organization – All car manufacturers.

Aims, numbers, dates and areas – To improve the primary and secondary safety of standard car models and newly developed car models investigations, comprehensive studies and tests are made.

Organization – German Motor Vehicle Trust Institution (DAT), and the German Driver Control Association (DEKRA).

Aims, numbers, dates and areas – Experts, reports give conclusions as to technical defects and car damage.

6. Other Data

- In the Federal Republic of Germany, a great number of individual research activities are made, in particular in the fields of: human factors; cost benefit analysis; road improvement; speed regulation; research into the technical features of motor vehicles; and traffic medicine and traffic psychology.
- In the Federal Republic of Germany, no official data on the consequences of accidents in monetary terms are available (e.g., costs of a fatality). Measures to improve traffic safety, currently contemplated by the Federal Government, are published in the Traffic Safety Program. The section "Motor Vehicle Design and Equipment" deals in particular with the following points: technical measures to prevent accidents; technical measures to mitigate the consequences resulting from accidents; head rests; laminated safety glass and others; safer occupant compartments; and experimental safety vehicle.

SUPPLEMENT 3

ROAD ACCIDENT INVESTIGATIONS IN FRANCE

Accident Data

1. National Road Accident Report Procedure

Organization – Data provided by police for SETRA (Service d'Etudes Techniques des Routes et Autoroutes).

Aims – To provide help for decision making and to be used by administration and research organizations.

Numbers, dates and areas – About 250,000 injury producing accidents and the corresponding 400,000 casualties, every year. These cover practically all the injury producing accidents occurring in France.

Details – The form which is filed by the police includes only simple information, about the place when the accident occurred, the road design, the vehicle type, the state of the occupants. The basic data are published every year (e.g. Accidents corporels de la circulation routiere – 1973 – SETRA).

2. Insurance Companies' Files

Organization – Accident files collected by the general association of the Insurance Companies (AGSAA); these companies insure about 80 percent of all vehicles.

Aims – Each accident reported to an insurance company is recorded in order to establish liabilities and evaluate costs.

Numbers, dates and areas – All accidents being reported to an insurance company belonging to AGSAA, whether these accidents produce injury or not. About 5,000,000 vehicles involved in accidents are recorded each year. A report is published every year.

Details – Though orientated towards the specific purpose of insurance companies, these statistics can be used by research organizations to estimate costs of accidents or relations between injury producing accidents and purely material accidents, or to compare the estimates of injury producing accidents with those which are provided by police. Most accidents with only damage to the liable car are not reported.

3. National Accident Investigation

Organization – Data provided by the police and gathered by ONSER.

Aims – To provide an intermediate level of statistics between the road accident report procedure and the detailed accident investigations.

Numbers, dates and areas – The data are collected on a representative sample of accidents, including a fifteenth of all accidents occurring in France (about 16,000 of them). The collection is only starting now.

Details – The source of data is not the accident form filed by the police, but the regular accident declaration report. The results of data

analysis will be more quickly available than those of the national file.

4. Bidisciplinary Accident Investigations

Organization – Investigations carried out by ONSER (Laboratory of Lyon-Bron) in an area around Lyon, by the Renault-Peugeot Association in an area near Paris, and at the Salon-de-Provence hospital. Citroen will also participate in collaboration with ONSER in Lyon.

Aims – To provide precise data on vehicle deformations and on crash injuries, to research organizations and to the car manufacturers.

Numbers, dates and areas – The study in the Lyon area is more than four years old and about 600 cars involved in accidents have been fully investigated; the results are being analysed at ONSER. The data from the Paris Region study (about 1,400 vehicles) and the Provence study are now being coded.

Details – The investigations have been carried out by three teams of physicians and car engineers. The results will be used to study specific problems, such as the relation between injuries and car deformations, the effectiveness of some safety devices already in service, or the study of priority of benefits. The sample of accidents being necessarily biased, some of the data will have to be balanced.

5. Specific Investigations

Organization – Investigation carried out at ONSER with the help of police or administration.

Aims – To provide the necessary data for some studies or specific aspects of the safety problem.

Numbers, dates and areas – The number of accidents investigated vary according to the problem concerned. These investigations have been carried for the last five years; for example: Vehitest (1971), the alcohol rate of the drivers (1969-1970), the relations between pedestrian behavior and accident risks (1971), real accidents on safety-guides (1973), various kinds of windshields (1973).

Details – The specific studies concerned here were undertaken to evaluate a priori the effectiveness of some measures. For example, the aim of Vehitest was to estimate the importance of technical defects of the vehicles as a factor of accidents and to provide necessary data for a cost benefit analysis of different systems of technical car inspections, 1,600 accidents were investigated; the cars involved were inspected by engineers and the accident declaration reports were provided by the police.

Sources of Background Data for Accident Studies

1. Daily Traffic Data

Organization – Data collected and analyzed by SETRA.

Aims – To provide statistics on the average daily intensity of traffic on national roads and motorways.

Numbers, dates and areas – The traffic is observed daily in 900 permanent spots of the national road network. Night observations are also carried out for periods of three months on sections of the main traffic network. The results are published every year by SETRA. The traffic mixture has been analyzed in 1970, with data provided by 25 of the 900 pilot spots, and such an analysis is to be repeated every fifth year.

Details – The observations are based on the notion of homogeneity, the criteria for homogeneity of a section of road being the density of population along the section, the width and the number of traffic lanes, the intensity of traffic, the administrative department.

2. Safety "Dashboard" Study

Organization – The investigation is carried out by ONSER.

Aims – To provide data on traffic out of towns of more than five thousand inhabitants and also on traffic in towns (traffic intensity and mixtures, average speed, use of safety belts), and on the relations between the driver and the vehicle.

Numbers, dates and areas – The investigation on traffic out of towns has been going on for two years and has taken place in 20 departments. During the first year of investigation more than 25,000 vehicles have been observed. The two other types of investigations are just beginning. They will take place in the same 20 departments as the first one.

Details – The 20 departments have been chosen to constitute a representative sample, on the basis of their geographical situation and of their petrol consumption. The observation spots are distributed on the various categories of roads in proportion to the length of the corresponding networks. Results are published every month

by ONSER (e.g., Tableau de Bord mensuel de Sécurité Routiere – Décembre 1973).

3. Sample Survey on the Road Transport of Goods

Organization – Survey carried out by INSEE (National Institute of Statistics) and the Department of Transports.

Aims – To provide data on the traffic of heavy vehicles, such as number of lorries in service, annual distance driven by an average lorry, etc.

Numbers, dates and areas – This survey is done every year. One survey is carried on for a fortnight, and about 50,000 vehicles are observed in a week.

SUPPLEMENT 4

ROAD ACCIDENT INVESTIGATIONS IN ITALY

Accident Data

1. National Road Accident Report Procedure

Organization – Data is provided by Road Police, City Police and Military Police (Italian name: Carabinieri) mainly for justice need. Brief data transferred in standard form sent to "Istituto Centrale di Statistica" (ISTAT-Rome). All data stored on computer with tabulating, listing and mapping programs.

Aims – To be published into two books put up on free sale. First book with 41 tables concerning type of load, vehicle, time, day, month, driver's age, type of crash, death, injury (driver, passenger and pedestrian), etc. Second book reports number of accidents, deaths and injuries for each kilometer of Italian highway and motorway.

Number, dates and area – About 200,000 injury producing accidents and 130,000 accidents with damage once a year, for the past four years or more, covering the whole of Italy.

2. National Driver's Behavior Report Procedure

Organization – Summary report is sent by police to local office of Motorizzazione Civile (UPMCTC) and to Prefects granting driving license for every accident producing death or serious injury (more than 40 days before recovery).

Aims – To provide technical data for driver's license suspension and user factor contributing stored on computer.

Number, dates and area – About 16,000 injury producing accidents a year for the past four years or more, covering the whole of Italy.

3. Car Manufacturers' Accident Studies

Organization – Data is provided by branch to Fiat and Alfa Romeo.

Aims – To collect vehicle damage and injury data in detail for vehicle design purposes.

Number, dates and area – 3,000 accidents in two years – Torino (Fiat); 100 accidents in one year – Milano (Alfa Romeo). Data ready but not stored on computer.

4. Highway Statistic, Traffic

Organization – Azienda Nazionale Autonoma delle Strade (ANAS-Roma) provides pointed check of traffic survey during some hours, day (holiday or not), month.

Aims – To compile each year Censimento Circolazione (Traffic Statistic). 1970 is the latest.

Number, dates and area – Traffic data is measured at about 5,000 points in particular sites all over Italy. Data listed with ten types of vehicles.

5. Registration and Traffic Vehicles

Organization – Associazione Nazionale Fra Industrie Automobilistiche (ANFIA-Torino) compiles each year "Automobile in Cifre" (Vehicle statistic). 1973 is the latest.

Aims – Book put on free sale, concerning production traffic (from tax payment), registration.

Number, dates and area – Whole Italy, every year.

SUPPLEMENT 5

ROAD ACCIDENT INVESTIGATIONS IN SWEDEN

Traffic accident investigations on various levels are carried out in Sweden by government authorities and institutions and by private companies – the car manufacturers and insurance companies.

This list of data sources has been produced by B. R. Nilsson of AB Volvo, Car Division, Göteborg, Sweden.

Accident Data

1. National Road Accident Report Procedure

Organization – Swedish Road Safety Office (SRSO). The SRSO has a team for certain analysis of traffic accidents.

The study started in conjunction with the change to right hand traffic in September 1967. NTSA reports every day on fatal traffic accidents. The main information sources for reports are the 119 police forces and the Swedish Central News Agency.

Aims – The study is mainly a day-by-day follow-up study of the accident frequency situation for policy and traffic regulation decision making.

Criteria – Fatal road traffic accident.

Number, dates and area – About 1,200 fatal accidents a year for the past 6 years covering Sweden.

Details – Data are about time, place, type of road, classes of vehicle and numbers of dead and injured in each accident.

2. General Accident Studies

Organization – National Central Bureau of Statistics (NCBS).

Aims – Official statistical data on traffic accidents. Basic data on site of situation and circumstances in which accidents occur.

Criteria – Road traffic accident involving personal injury.

Numbers, dates and area – About 28,000 injury accidents a year. The accident studies started about 50 years ago. Accidents involving injuries to a person are reported by the 119 police forces to the NCBS.

Details – For the purpose a special form is used, with about 60 questions about accident circumstances, classes of vehicle, environmental factors and injury data.

Storage – All data are stored on computer.

3. Special Studies

Organization – National Swedish Road and Traffic Research Institute (NSRTRI).

Generally – The NSRTRI makes various special studies, such as traffic accidents with heavy vehicle combinations, influence of speed on road accidents, effect of stud tires, etc. These studies are primarily based on the statistical data available but also involve interview questionnaires for people involved in accidents. Some studies, e.g., the stud tire investigation in 1972-73, are accomplished in cooperation with car manufacturers and insurance companies.

4. Car Manufacturer Accident Investigation

Organization – AB Volvo. The in-depth/multidisciplinary investigations are carried out by two teams – one team for car accidents and one team for truck accidents, in cooperation with police and emergency alarm centers and hospitals. The statistical investigations are made in close cooperation with the Volvo 5-year guarantee organization, Volvia Insurance Company, Volvo dealers and hospitals.

Aims – To create relevant data for various automotive safety questions as: vehicle design, human tolerance factors, product safety and liability, crash test program and specification.

Numbers, dates and area – GO-teams: About 100 fatal and injury accidents a year of each team. The area covered is roughly within 1-hour drive from Göthenburg.

Criteria – Traffic accident involving Volvo 140 and 164, causing occupant injury or any truck accident.

Numbers, dates and area – Statistical investigation: 5,000 to 6,000 of the most severe accidents with Volvo cars a year. This survey was started in October 1973 and is supposed to run continuously.

Criteria – Traffic accident involving Volvo 140 and 164, causing major property damage.

Details – Statistical investigation. The data collection includes an interview questionnaire mailed to the driver involved.

Generally – In addition to the basic investigation activities mentioned above, Volvo has various temporary projects related to traffic accident research, e.g., skid accident investigations, analysis of all the fatal car accidents in Sweden in 1973, human tolerance data derived from road accidents.

Organization – Saab-Scania.

Generally – Saab-Scania has a team within its development department for certain traffic accident research. The data collection and analysis is retrospective in its nature and is limited to the most severe Saab vehicles in accidents in terms of vehicle damage. The research work has temporarily involved a go-to-the-scene activity. A statistical accident survey has also been made by Saab.

5. Insurance Companies Accident Studies

Organization – Folksam Insurance Company and others.

Generally – Since 1969 the Folksam Insurance Company has had various accident studies going on. The activities concerned are partly located at the main office in Stockholm and partly at an experimental bodyshop in Växjö. The main

purpose of the Folksam studies is to reach conclusions from field accidents reported to the company concerning the injury producing mechanisms of the interior design of the vehicle. The insurance company Skandia has recently carried out an investigation related to the child-in-road traffic, especially the child of preschool age. Quite recently all the traffic insurance companies in Sweden have agreed to organize a road traffic safety committee, which is thought to have traffic accident investigations on its program. The nature of the investigations has not been announced yet.

6. Sources of Background Data for Accident Studies

Organization – Swedish Vehicle Inspection Company, owned by the Swedish State, the car insurance companies, Automobile Clubs, and the Automotive Association.

Aims – Annual traffic safety inspection of all motor vehicles – cars, trucks, trailers and motorcycles – with an age of three years and older.

Numbers, dates and areas – About 3 million vehicle inspections a year for the past nine years (Sweden).

For information write to the Swedish Road Safety Office at Statens Trafiksäkerhetsverk, Box 508, 126 05 HÄGERSTEN, Sweden.

SUPPLEMENT 6

ROAD ACCIDENT INVESTIGATIONS IN THE UNITED KINGDOM

Road accident data and background information are collected in several different ways in the United Kingdom. The main sources of data are listed below. Information is available from those sources marked with an asterisk. Abstracts of reports describing some of these accident data, or the uses to which they are put, can be selected and sent out by the United Kingdom, IRRD Centre, which is at TRRL, Crowthorne, Berks, UK.

Accident Data

1. National Road Accident Report Procedure (Stats 19)

Organization – Data provided by police for Department of Environment (DOE) and its Transport and Road Research Laboratory (TRRL*).

Aims – To provide basic data for policy decision making and a data bank for use by government, research, local authorities and industry.

Numbers, dates and area – Quarter of million injury accidents a year for the past 15 years or more covering England, Scotland and Wales (most accidents except for some single vehicle, pedalcycle, motorcycle and car accidents).

Details – About 50 questions about casualties, vehicles involved and attendant circumstances, most with between two and twelve multichoice answers. Accuracy based on booklet of definitions (Stats 20). Compilation by headquarters staff at the 63 police forces, with checking at DOE/TRRL. All data stored on computer with tabulating, listing and mapping retrieval programs. Data are about time, place, type of site, classes of vehicle and numbers of injured in each accident. Basic data published annually e.g. 'Road Accidents 1971' (HMSO*).

2. On-the-Spot Survey of Accidents

Organization – TRRL*.

Aims – To find how and why accidents occur and to identify road, vehicle and road user factors contributing, using survey for policy making and as data bank for government, research and industry.

Numbers, dates and areas – 2,000 (mostly injury) accidents by four years up to Spring 1974 in east of Berkshire, England.

Details – Many details but not a full in-depth study and no injury data. Vehicle and road scene information with interviews of those involved. Some background data collected. Filing of paper records with about 200 items of information stored on computer for each accident for indexing and tabulation purposes. Used as a basis for primary safety assessments. Special studies of loss of control, car color, brake and tire condition and driver behavior.

3. Crash Injury Study

Organization – TRRL*.

Aims – To find how car occupants and others are injured, for policy making and as a data bank for government and research into incidence and human tolerance of injury.

Numbers, dates and area – Main study is of 1,000 serious or fatal injury accidents since 1965 in Thames Valley area.

Details – In-depth study of car occupant injury causation. Many accident details recorded as

well as medical investigations into injuries. Separate studies of how pedestrians, motorcyclists and others are injured and of performance of windshields and steering columns in accidents. Filing of paper records with computer storage for indexing and tabulation of over 50 accident and 150 injury items per casualty. Used as a basis for car secondary safety assessment but does not record severe impact accidents without serious injury, low in urban accidents.

4. Crash Injury and General Accident Study

Organization – Birmingham University, Department of Transportation and Environment Planning* in conjunction with the Birmingham Accident Hospital.

Aims – Data bank for studying accident and injury causation for use by government, research and industry.

Numbers, dates and area – 2,000 vehicles and occupants in injury accidents since 1965, many in urban Birmingham and outskirts.

Details – Detailed studies, particularly for injury investigations, but also for primary safety with various special objectives at different times. Continuing with injury and crash performance of current model cars and pedestrian injury in relation to car exterior design. Much of work planned to suit particular car safety inquiries.

5. Additional Police Investigations Into Accidents

Organization – Almost all of 63 police forces.

Aims – To supply further accident data to assist local authorities responsible for roads. Also special studies, some to assist TRRL, others to place the determination of responsibility of accidents on a more scientific basis.

Numbers, dates and area – Most police forces help local authorities, also occasional studies when need arises.

Details – Most record road data, additional to those for Stats 19 and store it all on local computers. Special studies include: (1) all fatal accidents; cars into rears of heavy goods vehicles; (2) extra motorway accident data; (3) metropolitan police study of effect of car color at intersections; (4) interpretation of skid marks.

6. Car Manufacturers, Accident Studies

Organization – Ford Motor Company and others.

Aims – To improve design of cars for safety.

Numbers, dates and area – Usually close to car manufacturers' sites, various numbers and dates.

Details – Results are not generally made available, but are usually to study secondary safety performance of recent car models or to study reported defects.

Sources of Background Data for Accident Studies

7. Vehicle Examination Studies

Organization – Vehicle Engineering and Inspection Divisions, DOE.

Aims – To investigate possible vehicle defects in service and in accident to provide data for need for manufacturers' recall campaigns.

Numbers, dates and areas – All areas. About 500 vehicles a year found to have defects, mostly goods and public service vehicles.

8. Highway Statistics, Traffic and Other Data

Organization – Department of Environment (HMSO*).

Aims – To publish data annually (e.g., "Highway Statistics 1971") to inform those needing data on vehicle registrations, traffic, road mileage etc.

Numbers, dates and areas – Traffic data is measured at 200 points at all times and 1300 points in 1960, 1966, 1973 at random sites over Great Britain.

Details – Further details of national data in: "National Travel Survey", "National Income and Expenditure", "Private Motoring in England and Wales", "Family Expenditure Survey", "Characteristics of drivers obtained from large scale inquiries (TRRL, LR 389)."

9. Special Counts of Background Data

Organization – TRRL and others.

Aims – Various aims to help understanding of accident data.

Numbers, dates and areas – Varied, but usually as small as possible in particular areas and not repeated regularly.

Details – Examples include fitting and wearing of seat belts (repeated regularly) and condition and pressure of tires.

10. Local Authority Data

Organization – Most local authorities.

Aims – Data collected for planning purposes in local area.

Numbers, dates, areas and details – Most areas have large origin – destination studies for road planning. Also measurements of road surface slipperiness, traffic counts and accident studies for monitoring road aspects of safety.

APPENDIX 3

REPORT OF WG2 – THE ORDER OF PRIORITY AND MAJOR REQUIREMENTS FOR SAFER CARS FOR THE NEAR FUTURE

PROGRAM'S MAIN GUIDING PRINCIPLES

After considering the information discussed by Working Groups 1 and 3, this group was to proceed with an analysis of the various problems to lead to the definition of the corresponding safety requirements and their order of priority. Finally proposals for possible future action were to be made to the main committee.

The actual speeds and other detailed suggestions made by Working Group 2 in this report for possible impact tests and other procedures are preliminary indications rather than final statements of an EEVC point of view. Time was not available for WG2 to estimate costs and benefits to be expected from a range of measures, such as speeds of impact for a test procedure, so that optimum conditions could not necessarily be selected.

The WG2 task program has developed along three main guiding principles and with the following priorites:

- Car internal and external design features for occupant protection.
- Car external design features for protection of other exposed road users.
- Primary or preventive safety design features.

CAR OCCUPANT PROTECTION

The car occupant safety characteristics must be established as a function of the following two requirements:

- Reduction of direct impact and consequential severity of injury in the various accident modes.
- Elimination of indirect dangers ensuing from such accident events (fire, impossibility of timely aid, etc.)

The above two basic requirements should be met by specifying suitable performances for standard impact tests conducted on cars with restrained or suitably protected dummies. The performances to be required could be as follows:

1. Compliance with biomechanical tolerance limits.
2. No bursting open of doors during impact.
3. Possibility, after collision, of opening at least one door without tools.
4. Possibility, after collision, of removing the complete dummies.
5. No fuel spillage or fire.

Restraint Systems

Among the presently known restraint systems, the seat belts (3-point type, in particular) are certainly the most effective and simple in providing a reasonable direct protection of car occupants in the majority of road accidents.

It is desirable to have future regulations which make it mandatory to install and wear seat belts in European countries. In view of this, utmost R&D efforts should be devoted to seat belts in order to improve their present features and performance. Ameliorations should be concentrated on the following aspects:

- Installation on car
- Dimensional and strength specifications of the different components
- Location relative to occupants
- Occupant comfort
- Manual fastening
- Automatic adjustment and locking
- Dissipation of occupant kinetic energy through absorbing devices
- Starter inhibition or some other interlock when belts are unfastened (possibly)
- Warning systems for when belts are unfastened.

The rational solution of the different problems associated with the use of seat belts will require the close coordination of all the effort spent in this field.

Further development of passive restraint systems should be investigated.

Test Methods for Impact Simulation

The discussion of the answers given by the various National Delegations to the questionnaire prepared

by WG2 has led to a common attitude on the four main impact modes intended to verify the occupant protection performance.

For each of said impact modes, at this time, the alternative of different test methods was indicated: the final choice will be made once the comparative test results and accident analysis data will both be available.

The comparative tests on current production cars should highlight the severity level of each impact mode being investigated from the standpoint of damages to the car and possible consequences on the occupants.

In this connection, cooperation by European Car Manufacturers will be requested.

1. Frontal Impact Test

Test Procedure: To be selected between the following two tests A and B, both are considered to be practical modifications of the existing head-on test. They are likely to lead to further reductions in injury according to predictions based on existing accident studies.

A. *Impact against barrier* angled at 60° to vehicle main axis.

B. *Offset impact against barrier* with radiused edge (15 cm. radius). The impact must involve half of the vehicle front (provisional agreement). As a rule, the impact half must be the steering wheel side but the test can be repeated on the opposite side, when found advisable.

Test Velocity: 50 kmh

Test Conditions: Vehicle in running order. Two dummies (50th percentile, male) in the front outboard seating positions. Restraint systems in the normal position and conditions specified to enable them to act on the dummies.

Requirements to Be Met: As specified under "CAR OCCUPANT PROTECTION", items 1 to 5 inclusive.

2. Side Impact Test

Test Procedure: Apart from improving the protection available for occupants of cars struck in the side, these tests should encourage compatibility between the fronts of vehicles and the sides of cars which they strike. At present the test may be selected from A and B, but these may be further developed by substituting for the striking vehicle an impactor with a standardized front, representative of future European car frontal structures.

A. *Stationary vehicle* struck on its side by the front end of an identical vehicle. The velocity vector of the striking vehicle must make an angle of 75° to the main axis of the struck vehicle. The main vertical plane of the striking vehicle must pass through the driver's seating position H point.

B. *Moving vehicle* struck on its side by the front end of an identical vehicle. The main axes of the two vehicles must be set at 90°. The relative velocity vector of the striking vehicle must make an angle of 75° to the main axis of the struck vehicle. The main vertical plane of the striking vehicle must pass, at the instant the impact begins, through the driver's seating position H point.

Test Velocity: 40 kmh (relative velocity of striking vehicle to struck vehicle).

Test Conditions: Two dummies (50th percentile, male) in the seating positions adjacent to the struck side. Restraint systems in the normal position and conditions specified to enable them to act on the dummies.

Requirements to Be Met: As specified under "CAR OCCUPANT PROTECTION," items 1 to 4 inclusive.

3. Rollover Test

Test Procedure: To be selected between the following two:

A. *Rollover test* with two full rotations.

Test Velocity: 50 kmh (initial speed).

Test Conditions: Vehicle in running order. Windows closed. Two dummies, (50th percentile, male) in the front outboard seating positions. Restraint systems in the normal position and conditions specified to enable them to act on the dummies.

Requirements to Be Met: As specified under "CAR OCCUPANT PROTECTION," items 2 through 5 inclusive. Additionally, no ejection (even partial) of dummies and absence of excessive deformations (collapse) of roof.

B. *Dynamic impact test* on roof front corner by pendulum or moving barrier having a mass corresponding to 60 percent the curb weight of the test vehicle.

Test Velocity: 10 kmh.

Test Conditions: Vehicle body fast on ground. No dummy on board.

Requirements to Be Met: Absence of excessive roof deformations (collapse).

NOTE: The test could be run statically by applying to the roof front corner a preestablished load by means of a rigid flat plate. Complementary static tests could be carried out to verify the capacity of the door locks to prevent accidental door opening under

loading from inside and outside the passenger compartment.

4. Rear Impact Test

Test Procedure:

A. *Stationary test vehicle* struck from rear along the longitudinal axis by a moving barrier or pendulum of 1,100 kg.

Test Velocity: 35 kmh.

Test Conditions: Empty vehicle, in running order, unbraked and in neutral.

Requirements to Be Met: As specified under "CAR OCCUPANT PROTECTION," items 2 through 5 inclusive.

Compatibility

It is clear that the problem of compatibility must be viewed within reasonable limits and that the possibility of compatibility should therefore be ruled out in the event of collisions between vehicles quite dissimilar as regards mass, size, shape and structural characteristics (e.g., cars and trucks).

The objective of compatibility should therefore be confined to cars and, presumably, to a limited range of these.

For an exact definition of the limits of said range, the following data should first of all be analyzed:

- Characteristics of cars on the road in Europe (weights, size, mechanical layout, etc.).
- Mass ratios in the various car accident modes.

The final compatibility performance will almost certainly amount to meeting requirements 1, 2, 3, 4 and 5 listed under "CAR OCCUPANT PROTECTION," in front, side and rear impact tests.

The main problem will indeed be to define a representative impactor. Taken to the extreme, this could be reduced to a single structure simulating the front end of a car whose shape, size, mass and stiffness (local and overall) are representative of those of all cars pertaining to the range considered.

Another criterion could be that of testing using a standard obstacle (deformable barrier, large framed sheet metal restrained at either side, etc.) on which to measure intrusion depth, space, piercing, etc.

PROTECTION OF PEDESTRIANS AND EXPOSED RIDERS

The problem of the protection of exposed road users is second only to car occupant protection.

However, potential solutions are not very encouraging and even the more optimistic proposals are somewhat lacking in terms of effectiveness. According to the present knowledge, there are only a few possibilities of improving the safety features of cars for the protection of pedestrians at collision speeds above 10 kmh.

The accidents covered by this area of safety can be classified according to the topic of investigation as follows: (a) pedestrian and (b) pedalcyclist and motorcyclist.

Pedestrian Protection

The most important of the various types of accidents involving a pedestrian consists of three phases as follows:

1. Pedestrian is hit at leg level by the outermost part of car front end.
2. Pedestrian hits hood and can be hurled onto windshield.
3. Pedestrian falls on road.

At low speed, impact severity and risk of fatality grow rapidly in phases 1 to 3, whereas at medium and high speeds phases 1 and 2 may already cause death.

Safety requirement investigation will be carried out in the above phase sequence in order to:

- Assess the effect of shape, size, stiffness and location of car front end protrusions on risk of fatality at initial impact.
- Evaluate the effect of shape, size, and stiffness of hood and windshield on risk of fatality at second impact.
- Examine the potential of pedestrian restraint systems designed to prevent third impact.

Pedalcyclist and Motorcyclist Protection

Though no laboratory test information is available on simulated accidents with pedal- and motorcyclists, it can be assumed that the sequence of events differs from that of accidents with pedestrians mainly at initial impact, when, in most cases, only the car and cycle come into contact with one another, involving the front, side or rear of the car. As a consequence, second impact can involve areas other than the hood or windshield.

Some of the safety requirements for pedestrian protection may well apply also to pedal- and motorcyclist protection, at least for straight-ahead impact against car front or rear end.

A definition of specific requirements is unlikely.

ORDER OF PRIORITY OF SECONDARY (OR PROTECTIVE) SAFETY MEASURES

The following numerical code is used for priority and practicability ratings:

Priority: 1 = Maximum; 2 = Medium; 3 = Minimum.

Practicability: 1 = Available; 2 = Foreseeable; 3 = Doubtful.

Priority is an overall assessment indicating the need for work to be carried out, whether this be further investigation or final development of test procedures.

Practicability is the engineering practicability for producing cars with the safety measure of the performance suggested.

	Priority	*Practicability*
Seat belt improvements to belts and cars to increase performance, convenience and comfort related to their use and standardization of buckles.	1	1
Investigations to improve protection for pedestrians when struck by cars.	1	3
Frontal impact measures for restrained occupants.	1	1
Side impact measures.	2	2
Rollover measures (prevent door opening and roof collapse).	3	2
Rear impact measures.	3	1
Fire prevention.	3	1
Release of occupants whether injured or uninjured.	3	1

PRIMARY SAFETY

The need for new or improved primary or preventative safety requirements seems to be much less urgent than that for secondary or protective safety for car occupants and other road users.

In fact, many primary safety improvements have been introduced in the past, and at present detailed accident investigations are showing to what extent various safety measures may actually contribute to safety. The following notes summarize tentative conclusions of this work.

	Priority	*Practicability*
BRAKES		
Antilocking systems (good potential but need assessment and further development for reliability).	2	2
TIRES		
Low pressure and deflation warning.	2	2
Safety tires.	3	1
DRIVING AIDS		
Warning or driver control devices for unexpected hazards, driver fitness and car speed are all potentially useful, but need development and trials.	2	2
Ergonomics of driving task (comfort and optimization of controls and layout).	3	1
HANDLING		
Research needed to study car behavior, drivers and their interrelationships.	2	3
LIGHTING AND VISIBILITY		
Conspicuity of cars by warning and signalling lights and other means, need reassessment. Driver's view at night and in adverse conditions also needs reassessment.	2	1

SUPPLEMENT 1

LIST OF PARTICIPANTS IN WORKING GROUP 2

Numbers in parentheses indicate the first and second meetings respectively.

Federal Republic of Germany:

Friedel	BAST
Kleinsteuber	Vd-TUEV
Lincke (1)	Volkswagenwerke AG
Seiffert (2)	VDA

France:

Osselet, A.	Ministère de l'Equipment
H-Herla	ONSER
Bluet	ONSER

Italy:

Pocci, G. – Chairman	Ministry of Transport
Danese, G.	Director CPA
Schinaia, C. (1)	CPA
Lomonaco, C. (1)	CPA
Strampelli, R. (2)	Ministry of Transport
Palieri, A. (2)	CPA
Tibiletti, C.	ANFIA
Franchini, E. (2)	ANFIA

United Kingdom:

Sharp, R.	DOE
Baxter, W.L.	DOE
Lister, R.D.	TRRL
Neilson, I.D.	TRRL

Finch, P. (1)	British Leyland	**At-large:**	
Knight, J. (1)	SMMT	Tarriere, M. (2)	CCMC
Hollings, J. (2)	SMMT	Puleo (2)	CCMC

APPENDIX 4

REPORT OF WG3 – HUMAN TOLERANCE LEVELS AND OCCUPANT PROTECTION EVALUATION TECHNIQUES

PURPOSE AND SCOPE OF THE WORKING GROUP

At the fifth meeting of the European Experimental Vehicles Committee at Crowthorne in June 1973, it was agreed that a common view should be established on a number of subjects relating to safety features for production cars coming onto the European market in the early 1980s. The objectives of these features would be to provide greatly improved safety at an economical cost.

Working Group 3 was set up by the Committee to consider one group of subjects, namely human tolerance and test techniques for assessment of car safety features. Two meetings of this working group were held at the Transport and Road Research Laboratory at Crowthorne. The membership of the group is given in Supplement I and consisted of experts from national administrations, some branches of the motor industry and testing and research establishments. In addition other interested parties were given the opportunity to comment on the various questions under consideration. CCMC did so but BPICA, because of the short time available, was not able to do so.

The objectives of the group were: first, to consider the maximum loadings (human tolerance) to which road users can safely be subjected in road accidents, and to state any doubts which exist about the loadings for particular types of impact, together with proposals for experimental work or other investigations necessary to determine what levels should be chosen; and second, to consider test techniques and dummies for verifying that human tolerance loadings are not exceeded in car test impacts.

The group felt that its report should be in realistic terms and proposals should be applicable to mass production vehicles.

The report which follows consists of four sections dealing with Human Tolerance Levels, Test Techniques, Test Devices and Recommendations.

HUMAN TOLERANCE LEVELS

If tolerance levels are specified their use must be justified by accident data, they must be representative of the population of vehicle occupants and the form in which they are specified must be suitable for evaluation in impact tests.

Many tolerance levels for various areas of the human body are quoted in the literature but relatively few of these are universally accepted. The collection and refinement of such data is of prime importance if a great deal of time and resources is not to be expended in designing safety features for cars to inaccurate specifications, such features as a result not being fully effective.

Nevertheless, a list of human tolerance levels is given in Supplement II as the best guide for designing safer road vehicles that the group could present at this point in time.

Each of the tolerance levels listed cannot be specified in isolation, the total injury spectrum must be reviewed. For example, in extremely severe accidents, skeletal fracture in one part of the body may be acceptable if it precludes fatal organ decelerations in another.

The actual level of loading which it is appropriate to specify depends on how it is to be used. If, for instance, the design of the restraint system is such that the loading applied to the body is dependent on severity of the accident, as is the case with standard seat belts, then a high tolerance level which could be expected to injure say 25 percent of the people involved in relatively rare severe accidents, might be justified to improve the protection for the remaining 75 percent at high speeds and of the whole population at lower speeds. On the other hand if the loading is applied in all accidents, as is the case with pretensioned seat belt systems at present under investigation, then an injury rate of 25 percent in minor accidents would not be acceptable and a lower tolerance level must be specified.

Research is needed to provide information on the statistical spread of human tolerance levels in the

population and how these levels vary with age and sex. In addition, there is need for a summation technique to be developed to enable the more serious effects of multiple injury to be assessed.

Four methods of determining human tolerance levels have been used in the past, namely, tests of human volunteers, cadaver tests, interpretation of accident data and tests on animals.

Although each method has some advantages, scaling from tests on animals and extrapolation from human volunteer tests produce results of doubtful value. In order to approximate to tests on the live human being, tests reproducing road accidents are being performed using cadavers. There are problems of dispersion between individual cadavers, of obtaining cadavers representing the younger age range and of obtaining sufficient results for statistical validity. In addition there is the problem of knowing how measurements made on cadavers can be related to measurements made on dummies. Some research is being undertaken in which the performance of belted cadavers is being directly correlated with that of anthropomorphic dummies. In other investigations accident data is being interpreted using dummies and impact devices in simulations of the accidents, to obtain tolerance levels directly applicable to these dummies and impact devices, and relating to the type of people who are being injured in accidents.

The fact that human tolerance levels to deceleration or force, determined in tests with cadavers cannot be directly applied to tests with dummies, because of the difference in performance between dummies and cadavers, is extremely important and needs to be borne in mind when drawing up proposals for development objectives or legislation.

The human impact tolerance levels specified in Supplement II should be adjusted if comparative dummy tests under the same test conditions as those in which the tolerance levels have been determined using cadavers, produce output readings different from those obtained from the cadavers.

TEST TECHNIQUES FOR THE ASSESSMENT OF CAR SAFETY FEATURES

The selection of the impact conditions for which it would be most appropriate to assess car safety features was the assignment of a separate working group, but Working Group 3 was asked to review the technical aspects of the currently available test techniques and Supplement III of this report gives detailed comments on the various techniques. For technical reasons and economy in testing, relatively few key tests should be required, mainly on complete vehicles. These tests should be backed by test data from laboratory rig tests, etc., and, in some instances, substitution of such data in the event of failure of one or two instrumentation channels in the key tests rather than requiring a complete retest, might provide valid information.

It may not be possible to design a vehicle to provide optimum protection for both restrained and unrestrained occupants and it is probably impossible to provide effective restraint at the present time for all sizes of occupant. For example, pregnant women or very large or small people may not be able to use restraint systems. So, although every effort should be made to ensure that as many people as possible use the restraint systems provided and priority should be given to providing protection for restrained occupants, some protection should be afforded for unrestrained occupants, care being taken to ensure that the provision of such protection does not excessively impair the protection of restrained occupants. If additional requirements are proposed for unrestrained occupants, then either the same tests as proposed for restrained occupants but at lower impact severities, or simpler tests using dummies in body shells mounted on rigs or tests using test-forms to impact critical components as required in ECE and U.S. regulations would be sufficient.

There are some arguments for limiting the speed of impact testing into blocks and paying greater attention to increasing the compatability between different vehicles in impacts. Other authorities think that car to car compatability is not a high priority problem and that more accident data are needed to define the problem more closely. If, as a result of safety research, cars with stiffer front ends are developed, compatibility might become even more important in the future.

The performance of the vehicles in the tests listed in Supplement III could be assessed using test dummies together with the human tolerance criteria listed in Supplement II, provided these values are adjusted to allow for the difference in performance between test dummies and living human being (where possible by correlation tests with accident data and cadaver tests) and provided that continued research effort is put into confirming or revising the criteria.

TEST DEVICES

Test dummies for occupant protection assessment should be sufficiently humanlike in performance, be strong, give repeatable results, be capable of calibration, be standardized and manufactured to comply with performance requirements. They should be calibrated before and after testing to ensure compliance with these requirements.

Some opinion has it that reproducability of results is of paramount importance and can only be attained by specifying in detail the design of the dummies and even, perhaps, the source of supply of the dummies and of spare parts. On the other hand, some consider it too early a stage in dummy development to freeze the design of dummies and that a performance standard is more appropriate calibrating the tolerance levels measured with any particular design of dummy against established cadaver and accident data.

All dummies currently available commercially have limitations when assessed against the requirements for an ideal test dummy. These limitations included, nonhumanlike performance, excessive frangibility and spurious readings. Problems have arisen with the Hybrid II dummy, which was originally only intended for evaluating air bags, in that dummies that have been purchased after calibration by the manufacturer, have been found not to comply with the required specification.

Greater simplicity would be an advantage in test dummies. Some of the points in the OPAT dummy and the HSRI dummy might be an advance in this respect. The TRRL side impact dummy is designed with more emphasis on load measuring capabilities than other current dummies. The design philosophy on which it is based was to build a dummy structure more rigid than the components it was intended to subject to impact so that a check is made on the stiffness and energy absorbing properties of the items impacted which can be correlated with data from real world accidents.

The ONSER frangible dummy in which an attempt is made to reproduce the strengths of the human bones is an interesting development as a research device but it is not intended to be used as a compliance testing dummy.

RECOMMENDATIONS

The following recommendations can be drawn from the report:

Tolerance readings in terms of accelerations or forces determined from tests on cadavers should not be applied directly to readings obtained from tests using dummies.

There is a need for a series of comparison calibration tests to determine what readings from tests using any particular design of dummy correspond to tolerances which have been determined using cadavers or accident data.

Data from effective, efficient continuing accident studies is needed to identify the important areas of the body for which tolerance levels should be assessed in impact tests and accurate human tolerance data is needed for these areas to enable safer cars to be developed as rapidly, efficiently and economically as possible.

More information is needed on the statistical spread of tolerances within the population and variations with age and sex.

A technique is needed for evaluating tolerances to multiple injuries.

The following areas have high priority for human tolerance research:

- Brain and cervical spine injury.
- Tolerance levels in side impacts.
- Thoracic and visceral injury tolerance.
- Confirmation or revision of tolerance levels for other areas.

The number and complexity of tests to which vehicles are subjected need to be restricted on economic and test reliability grounds. For the same reason substitution of data from appropriate rig tests might provide valid information, in full scale compliance impact tests if some instrumentation channels develop faults during the test.

There is a need for tests to assess the protection afforded to both restrained and unrestrained occupants but the protection of unrestrained occupants should not be allowed to unduly compromise the protection of restrained occupants.

Test dummies should be strong, repeatable, standardized, capable of calibration, manufactured to comply with performance requirements and sufficiently humanlike in performance.

Of the different frontal impact tests available the one most representative of the major injury producing accidents should be selected by the comparison of test photographs with accident photographs and statistics.

There is need to develop a compatability assessment test for vehicles, particularly in front to side impacts.

Design requirements and performance tests for pedestrian protection assessment need further research and development.

Because both sides of the passenger compartment are stressed in a side impact test it will probably not be practical to carry out such a test on a vehicle which has undergone a partial frontal barrier impact test.

SUPPLEMENT I

LIST OF PARTICIPANTS IN WORKING GROUP 3

Federal Republic of Germany:

Guelich, H.	BAST
Reidelbach, W.	Daimler-Benz

France:

Cesari, D.	ONSER
Chapon, A.	ONSER
le Guen, H.	UTAC
Leroy, J.	ONSER

Italy:

Franchini, E.	Fiat
Schinaia, G.	Centro Prove Autoveicoli

United Kingdom:

Lowne, R. – Secretary	TRRL
Macaulay, M.	MIRA
Wall, J. – Chairman	TRRL
Waller, J.	Vauxhall Motors Ltd.

The CCMC viewpoint was given by Dr. Tarriere (France). Professor B. Aldman (Sweden) and Mr. Bussemaker (Netherlands) were invited to comment on the draft report of the Working Group before it was finalized.

SUPPLEMENT II

A REVIEW OF TOLERANCE LEVELS FOR VARIOUS PARTS OF THE HUMAN BODY

The following list of human tolerance levels is the best guide for designing safer road vehicles that the group could present at this point in time.

Some values are omitted from this list either because insufficient data is available or it is thought that the problems of measurement at the present time or rarity of accidents involving such injuries, justify their omission. Some of the values quoted are not yet generally accepted and require further research to confirm or revise them.

The ONSER report, "Revue des connaissances apport des researches actuelles en matiere de tolerance humaine a l'impact," June 1973, was circulated to the group and used to help draw up this list together with a bibliography of relevant German biomechanical publications prepared by the German delegation as part of an evaluation of German and international publications on this topic being carried out by the German Federal Road Research Institute.

Head/Brain

Separate tolerance levels for the brain to linear and rotational acceleration are desirable as well as levels for forces applied to the head specified in terms of location direction, rate of onset, duration, maximum value, average value, pulse shape and force distribution. At present, no tolerance level is sufficiently well established to put forward. Criteria of 80 g for 3 milliseconds and a "head injury criteria" of 1,000 are currently used in many European countries but only because these values measured on dummies are required by American legislation. Because of the nature of the brain, some means of assessing pulse shape is required but, because the mechanism of brain injury is not yet understood, it is not possible to say what the appropriate assessment should be.

A tolerance of the brain to rotational acceleration of 1,800 rad/sec^2 has been proposed by Ommaya of the U.S. but this has not gained general acceptance. The measurement of angular acceleration on dummy heads would present problems of instrumentation and repeatability.

In the absence of established tolerance levels for the brain a requirement that restrained dummy occupants should not experience head contact in frontal impact tests would be extremely restrictive on the design of restraint systems.

There is urgent need for further research into tolerance levels for the brain but the problem of determining such levels is not likely to be easily or readily solved.

Face

Tolerance levels for the bones of the face are better established than for the brain. This is probably because bone, being a somewhat brittle material, is more sensitive to peak force than to shape of the force-time curve. The published literature indicates that the Zygoma is the area with the lowest load tolerance and this could be taken as the tolerance limit for the entire facial area. Schneider and Nahum, "Impact Studies of Facial Bones and Skull," 16th Stapp Conference (1972), quote a force of 890 N applied on an area of 6.45 square cms. Measurement of the tolerance on a dummy would not be simple and it might be necessary to devise subsidiary rig tests to evaluate the facial impact area.

While laceration can be detected using a double layer of moist chamois leather over a layer of foam plastic it is not easy to quantify and it might be preferable to try to design out contact with lacerative parts of the vehicle rather than specify a tolerance to laceration.

Neck

Because of the low tolerance of the front of the neck to direct load and the difficulty of measuring such loads, it might be decided to specify no direct frontal neck contact. Gadd, Culver and Nahum, "A Study of Responses and Tolerances of the Neck," 15th Stapp Conference (1971), indicate tolerance levels of 80° in rearward hyperextension and 60° in

sideways flexure. The Mertz data, "Strength and Response of the Human Neck," 15th Stapp Conference (1971), indicate a rapid rise in torque and moment as the hyperextension approaches 80° and in view of this it may be advisable to work to a somewhat lower figure. Rate of neck rotation and combination of head rotation and neck flexure might, as a result of future research, prove to be important parameters in determining neck injury.

Clavicle

In future vehicles using seat belt restraints the important loading on the shoulder would be the load on the clavicle and this could be expressed in terms of seat belt webbing load and angle of the seat belt. In side impacts the transverse loading on the shoulder would need to be specified.

TRRL experimental interpretation of accident data indicates a tolerance level of 8 kN expressed terms of seat belt shoulder strap tension for the clavicle using the OPAT dummy (the belt making an angle of 35° with the torso line of the dummy) and a level of 6 kN for end loading on a clavicle in a side impact measured with TRRL side impact dummy.

Thorax

The true cause of injury to the internal organs of the chest is not yet fully understood but in frontal impacts loading on the rib cage, deflection of the rib cage and deceleration of the chest will probably provide sufficient data to specify tolerance levels and in side impacts loading on the ribs and deceleration of the chest will probably suffice. Which of these parameters is important will need to be decided and may depend on the type of test dummy being used. Cadaver-dummy correlation tests carried out with seat belt restraints indicate that a tolerance level of 60 g for 3 milliseconds for frontal impacts measured in the Sierra dummy is too high, resulting in flail chest injuries in the cadavers.

Kroell, Schneider and Nahum, "Impact Tolerance and Response of the Human Thorax," 15th Stapp Conference (1971), quoted data indicating a tolerable deflection of 60 mm measured on cadavers. Some evidence indicated that this was too high and that 50 mm would be a more appropriate figure. Nahum quoted a figure of 50 mm for a single rib and loads varying between 580 and 850 N. TRRL tests with the OPAT dummy to correlate with injuries in road accidents indicate a critical deflection of 45 mm measured on this dummy. If such tolerance levels are adequately determined and specified there will be no need to specify steering column collapse loads.

Accident correlation tests using the TRRL side impact dummy indicate a tolerance level of 1 kN measured at each of the four "rib" stations on the dummy.

Abdomen

Because of the known low tolerance of the intraabdominal organs, seat belts should not be permitted to ride up over the iliac crests of suitable dummies during the loading cycle in a frontal impact test. To assess this some sort of reference line might be established on the dummy.

Pelvis

Injury to the pelvis apart from the hip joint is thought to be unlikely in frontal impacts if the tolerance levels for other parts of the body are respected but in side impacts the loading in a lateral direction needs to be specified.

TRRL accident damage correlation experiments have indicated a tolerance level of 5 kN for the sum of the transverse loads measured at the iliac crest and hip joint with the side impact dummy.

Knee Thigh-Hip

The axial force along the thigh to cause fracture dislocation of the hip, the force to fracture the femur and to produce patella fractures are all somewhat similar and from Patrick's data appear to be of the order of 6 kN in cadaver tests although TRRL correlations of accident damage and injury with measurements using a kneeform impactor indicated a lower tolerance level in the region of 4 kN measured with this device for hip joint injuries in actual accidents. Other data quote higher values and the U.S. regulations specify a maximum level of 7.65 kN.

To avoid penetration injuries of the knee joint any force applied to the knee should be well distributed.

Lower Leg

The bending load in terms of load and point of application is important both in the case of pedestrians and vehicle occupants. No published data is available but Young in unpublished data quoted a tolerance level of 4.5 kN for the tibia.

Whole Body Tolerance to Acceleration

This might be of importance in extremely severe rear impacts where efficient head restraints prevented neck injury.

Fire

Rather than consider tolerance levels to fire, which is not practical, vehicles should be designed to guard against the risk of fire following impacts.

SUPPLEMENT III

TEST TECHNIQUES

The group reviewed the technical aspects of the various test techniques listed below which are available for assessing occupant protection in vehicles during different types of impact. A selection of suitable tests could be made from this list depending on the recommendations of the other working groups.

Frontal Impacts

Three test methods can be used for assessing occupant protection in frontal impacts, full head-on impact, partial frontal impact and angled frontal impact into massive blocks. The test most representative of frequent injury producing accidents should be selected and to enable this to be done, accident statistics and photographs should be compared with photographs of vehicles from the three different tests.

If a partial frontal impact has been carried out on a vehicle it will probably not be practical to utilize the undamaged side of the vehicle in a side impact test as both sides of the passenger compartment are stressed in such a test.

Full Head-on Impact Perpendicularly into a Massive Block. This is the conventional test for frontal impact protection.

Partial Frontal Impact into a Massive Block. This test is more severe in the penetration damage inflicted on the vehicle structure than the full frontal impact but the deceleration levels on the occupant compartment can be somewhat lower. If the corners of the impact block are suitably rounded, this test might be used to encourage car design to guard against interpenetration and structural interlocking of opposing vehicles in partial frontal impacts.

Angled Frontal Impact into a Massive Block. At suitable impact speeds and angles of impact this test can induce unrestrained dummy head contact with items (such as the windshield pillar) which are known to cause injury to real life accidents. However, this problem could also be assessed with laboratory rig tests and this would be preferable in order to reduce the number of full scale impact tests required by legislation and improve the repeatability of the tests.

Side Impacts and Compatibility

A new approach is needed to the question of structural compatability assessment and side impacts are probably the next major area of research likely to produce high returns in injury reduction, particularly as compulsory seat belt wearing would substantially reduce frontal impact injuries. There is a need for a test to ensure that deformation of doors in side impact does not lead to door opening by the release of tension door latch mechanisms. A static crush test would be suitable for this. Three possible test techniques for compatability assessment are outlined below but there is a need for further research to investigate their practicability.

1. Assessment of compatability from force deflection records obtained from frontal car to barrier impacts and side impacts with a mobile barrier.

   This technique is not yet sufficiently developed to enable it to be adopted but it is currently being investigated. It would enable the occupant protection afforded by interior padding to be assessed.

2. Frontal barrier impact into a barrier of specified stiffness to ensure that the front structure of cars is not excessively strong and side impact by a mobile barrier with the same stiffness to ensure that the side structure is adequately strong.

   Such a test method might present problems in reproducing the required stiffnesses of the barriers and would need to be carried out at a speed different from that using a rigid barrier to maintain an equivalent impact severity. Such tests might be carried out using dummies in both front seat positions and the angle of impact in the car might be from 15° forward of the perpendicular to its longitudinal axis.

3. Static side crush test possibly coupled with requirements that the front structure be not excessively strong (evaluated from frontal impact tests).

   This method has the advantage of simplicity and reproducability but it would require additional rig testing to evaluate occupant protection devices such as padding. Such rig tests are already developed.

Rollover Evaluation

A dynamic 720° car rollover test to assess whether doors will come open or restrained occupants be ejected has been suggested but current rollover tests are difficult to perform and show poor repeatability. A series of laboratory tests which are already developed to assess structural integrity by static loading on the roof and door latch performance by exerting a force on the release area could be substituted. Additional tests of interior padding using impact forms might also be required. Again these tests are already available.

Rear End Impacts

Such tests are probably not justified on cost and injury producing grounds but a rigid mobile barrier impact such as that specified by ECE or an equivalent rig test could be used to assess fire risk due to fuel leakage following rear impacts.

The rear impact protection provided for occupants could be assessed using existing regulations for head restraints (modified to ensure that head restraints are not positioned too low by the user) without the need for a full scale dynamic rear impact test.

Pedestrian Impact

No suitable full scale test is yet available and at present, design requirements on height of bumpers, length of hoods and external protrusion are appropriate. The bumpers should be positioned to ensure that initial contact is below knee level and impact form tests might be used to assess their propensity to produce tibia fractures. External protrusions could be assessed under existing regulations.

MICHEL FRYBOURG
Director
Institute of Transportation Research

I would first like to review the previous conferences held over the last three and a half years: The first conference was held in Paris in January 1971; Stuttgart, in October of that year, afforded an opportunity for assessing the scope of work undertaken; the meeting in Washington in June 1972, highlighted renewal of land transportation; the meeting at Kyoto, in March 1973, permitted an appraisal of the first results and the London meeting will take stock of a substantial program, which is anticipated to have a major impact on motor car manufacturers worldwide.

An important initial observation concerns the scientific and technical backing which the various countries are now in agreement in giving to regulation.

The regulations, in some countries, were drafted by civil servants with a legal background; in others by engineers but no country could claim to have a scientific decision-preparation agency providing the necessary back-up for a regulatory program requiring time-consuming and costly preliminary studies; reliance was placed on know-how deriving from an interprofessional private/public-sector consensus.

However, the deployment of technical resources was not without consequences. Some countries believed that regulation was the stimulant of research and that research would be encouraged by fixing, in principle, the performance standards to be attained at a credible level. It belonged to the government to say what had to be done and to industry to find the ways and means to do it. The goal was technological progress utilizing mass production techniques within acceptable economical limits in order to maintain a dynamic pace. One word won wide acceptance as characterizing this attitude: *challenge.* A man's reach must exceed his grasp, not to reach the moon which concerns only a few, but to attain goals which satisfy the greatest number.

The question remains, however, as to what criteria will be adopted as the basis for performance standards. At what level must the challenge come? Fixing one's goals too high is wasted effort with poorly foreseeable economic consequences, especially for medium-sized firms which cannot cover their mistakes by "exporting the cost overruns." Setting goals too low would encourage complaisance guaranteed by public authority.

One can even go farther and state that the optimum level for challenge culminating in the most intense research effort probably does not coincide with maximum effectiveness for the community which is measured in human life saved or pollution reduced. France has always demonstrated extreme reserve in the face of such a challenge and has endeavored to devote maximum effort to assessment studies in order to fix performance standards at the proper level. By requiring, for example, levels of occupant protection much higher than reasonably necessary, one can indirectly preclude very economical and effective expedients such as seat belts.

The reports which follow will illustrate and confirm the two basic conclusions which it seems to us to be indispensable to draw from past efforts.

The first conclusion is that henceforth future motorcar regulations will be based on a substantial program of scientific investigations, designed for the community, not for the mobilization of opinion in favor of spectacular expedients determined in advance. International cooperation in the conduct of such scientific studies will be intense because it will not culminate in patents and provide the best guarantee of the desired internationalization of the regulations. Science is a stranger to boundaries and tariff protectionism.

The second conclusion lies in the importance of assessment studies. It is good to make probative models calculated to verify the validity of certain concepts; it is even better to work on performance criteria which will permit enhancement of our knowledge as to the goals to be attained and consequently as to what the test vehicles will specifically have to prove. In a word, now that scientists have been mobilized behind this basic question of passenger protection, account must actually be taken of what they have found rather

than following a regulatory road basically laid out before the study program is started.

Effective international cooperation in the vehicle safety research field can be meaningful only if the participants consider in future regulations the results obtained on the basis of such cooperation. Futhermore, in order to facilitate cooperation between the United States and the various European countries bound to that country by bilateral agreements on experimental vehicles, it would be highly desirable for a dialogue to begin between the responsible American authority, the NHTSA, and the EEVC, the agency within which the common European vehicle safety viewpoints are defined.

The assessment studies carried out in France relate in particular to the establishment of a photograph library of controlled collisions. The purpose of the library is to permit grading of actual collisions on the basis of an impact violence index calibrated by comparison of a sampling of controlled collisions and simulated accidents reconstructing real accidents. We will thus learn the statistical distribution of the energy dissipated during accidents in order to formulate realistic passenger protection performance standards.

Since 1971, the French Government has allocated 9,790,000 francs of credits to contracts totalling 14,700,000 francs devoted to the Thematic Vehicle Safety Program. These contracts deal with the following subjects classified by major topics:

- Structural improvement for passenger protection
- Interior and exterior fittings
- Improvement of restraining devices
- Braking
- Speed limits
- Black boxes
- Photograph library

The first three topics will be developed in the manufacturers' reports. The fourth topic concerns the design of a braking corrector for lorries and the initiation of a controlled experiment as to the effectiveness of antilocking braking systems. The fifth, concerning speed limits, relates to:

- The feasibility of a governor device
- The development of passive systems for automatic vehicle speed limitation by roadway control
- Black boxes refers to devices which record the speed and deceleration parameters at the time of a collision. Black boxes equipping a suitable sampling of vehicles will provide confirmation and supplementary information furnished by the photograph library.

These preliminary indications which will be developed in the reports of the French delegation are good illustrations of the foregoing remarks; regulation must proceed concurrently with studies and research, not precede them. It is not enough to find technological means calculated to meet predetermined objectives but at least as much to fix the level of those objectives in order to reach the sought-after goal – improved safety.

Although the division of labor has demonstrated its efficiency in production, it has never given satisfactory results in the conceptual field. This is precisely why we are all meeting in London, civil servants and industrialists, public sector and private sector, competitors and collaborators. Vehicle regulations in the 1980s will be based on the results of the research in progress and will be international not only in application but also in conception.

GEORGES BOSCHETTI
Director, Paris Research Center
Peugeot Automobiles

Since the Kyoto conference, Peugeot Automobile Research works on safety have been conducted as follows:

- Research carried out within the framework of "Actions Thématiques Programmées" (Subsystems Joint Research Program) of the French Government Research Program and jointly developed with the Régie Renault.
- Independent research work performed within the framework of the Peugeot-Renault Association and in particular work carried out by the Laboratory of Physiology and Biomechanics led by Doctor Tarriere.
- Research work performed by "Automobiles Peugeot" alone, which concerns some additional analysis, particularly synthesis research materialized by our Crowthorne presentation.
- Research work carried out within the framework of CCMC, which, through direct contact between the best European specialists, defines the best means available to really improve safety within the shortest delay.

All of these projects are complementary thus, in order to make the study easier, I shall group our results according to the various big technical problems.

FIRST IMPACT TESTS

Frontal Impact

Within the framework of the "Actions Thématiques Françaises" our purpose was to keep a space large enough to allow correct operation of restraint means in real impact conditions and at speeds as high as possible.

As a basis, we used the investigation of the association on the statistical study on frequency of more or less dissymetrical impacts and the detailed study of special accident cases which we reproduced in impact tests with the same cars.

We developed also personal research on comparisons of vehicle-against-vehicle impacts in simulated accidents, shifted on 1/4 or 1/2 on vehicle width and on 90°, 85°, 60° and 45° fixed barrier impacts.

This study showed the worthiness of the 60° impact whose representativeness, in real accidents, was later confirmed. We could reach the aim contemplated at research level as regards structure up to 65 km/h under these impact conditions. There remains now to check whether these results are really applicable at benefit/cost level.

Our firm is particularly interested in this 60° impact which seems to be the most representative. It is indeed a dissymetrical impact, as in 70% of real accidents, and is easier to reproduce than a shifted impact. Although it is more severe for the structure than the 90° frontal impact, we think it better to choose a more difficult test which is more representative of real accidents than to keep a less worthy and less difficult test and try to make it more severe through an arbitrary and useless increase of the test speed.

When we state we are in favor of a more difficult but more representative test, it is understood that this new test will be fully substituted for the older one and not added to the former one as some rulemakers are seemingly tempted to do. It is useful to make this point clear.

Also within the framework of the "Actions Thématiques Françaises" we are taking part in the creation of a national photo library of road accidents.

Lateral Impact

Within the framework of the governmental program too, we tried to reduce intrusion on the impacted vehicle side. This step obviously confirmed the problems of aggressiveness in lateral collision between vehicles.

We could obtain intrusions that were less than 10 cms in lateral collisions over 50 km/h at the cost of substantial reinforcements, some of them being used to cross-brace both sides of the vehicle at seat level. The idea is to let the clearances be absorbed first until the big side reinforcements come into contact to transmit the force through the seat structure.

Presently, we go on with our work through adding to the structure improvements those which may be obtained by an inner protection of compartment dissipating energy.

As for the frontal impact, the spirit of synthesis between structure improvement and inner protection directs our current works, taking as criteria the biomechanical data on occupant protection.

Rear Impact

We have pursued our own research works on the various possible locations of gasoline tanks; between rear wheels, under the rear seat, above the rear suspension taking up part of luggage compartment.

Our opinion is that there should not be drawn too strict conclusions based on hasty experience. The main point is to take advantage of the protection provided by rear wheels in case of rear-end crushing, which avoids the use of substantially heavy and costly reinforcements. It seems that accident investigations show that the benefit/cost ratio of rear impact protection is very low.

Impacts and Structure

We are studying the problem of compatibility within the CCMC. It is on the behalf of the CCMC that Dr. Seiffert, will make a presentation on this subject at the seminar on vehicle structure.

OCCUPANT PROTECTION

Within the framework of "Actions Thématiques Françaises," we have entered upon several studies:

- First of all, the development of a *protection system for children* from six to eight months old to 4 years old.

  Following our preliminary tests, our purpose was to develop specialized restraint means ensuring for children a protection equivalent to that provided for adults at 50 km/h impact speed.
- We also study a *low-speed passive protection system,* namely energy-absorbing dashboard materials, protection against occupant compartment inner walls as well as flexible nonlacerating safety windshields (in cooperation with Saint-Gobain).

  Our firm attaches much importance to low speed passive protection. Indeed urban collisions are very frequent and in most cases correspond to a severity level not exceeding the 30 km/h reference impact test against an extremely heavy and rigid wall. On the other hand, it is acknowledged that belt constraint is not so well accepted in town and that any research allowing to make up for this gap must be pursued.

  In addition to this French official program, our Association Laboratory has proposed more sophisticated protection criteria and, more particularly, a protection criterion of internal organs in order to avoid the consequences of submarining.

  These results will be presented, on behalf of the CCMC, at the seminars. Mr. Hartemann will deal with accident investigations, whereas Dr. Tarriere will bring out the relationship between human tolerances to impact and protection measures.

PEDESTRIAN PROTECTION

Within the framework of the French program, we conduct a statistical analysis of pedestrian accidents and a simulation of those accidents on current production vehicles. We try to bring out the main parameters to be retained for the design of future vehicles.

PRIMARY SAFETY

We currently conduct studies on:

- rear lamp visibility, which provides information on speed levels adopted by other road users in order to avoid "chain" impacts on motorways
- wheel anti-locking devices, while limiting our ambitions in order to avoid developing sophisticated and fragile systems for the sake of it within the CCMC, we participate in the critical study of road-handling tests which could be used as a basis for the establishment of possible standards and for the development of a "black box" giving information on maneuvers carried out by the driver prior to the accident Doctor Enke will discuss the problem during the seminar on behalf of the CCMC.

Taking into account all these works, as well as the comparison of our results with those achieved by our counterparts, we note that from one ESV Conference to the other, substantial progress has been made in the field of scientific knowledge. We are now in a position to choose better orientations and we should therefore be optimistic.

However, in conclusion, I would like to raise two concerns:

- We already had the opportunity to say that, though ESV conferences are playing an unquestionable role in research and at international levels, ESV vehicle exhibitions partially serve advertising purposes and by no means show the same efficiency as the technical presentations about impact mechanics, accident investigations, inner protection, aggressiveness and compatibility studies, which are the only ones to make automobile design progress.

Nowadays, advertisements on safety encroaches upon motor shows. ESV exhibitions within the framework of international conferences might also become invaded by advertising and thus pass unnoticed. There may be some doubt about the future of such exhibitions.

On the other hand, we are very much in favor of large scientific international conferences of this type with the participation of the United States, Japan, Europe and other important communities.

- My second concern is that technical progress might lead to exaggerated confidence in science, technology and regulation to the prejudice of man, especially man as a driver, both so weak and powerful. The current trend may be too much in favor of protecting man at all costs, in spite of himself.

Secondary safety will probably soon reach its limits as occurred before with primary safety. It was formerly thought that primary safety was the main concern. Its limits have been defined.

Secondary safety, afterwards, showed its usefulness efficiency. However, it should not be

considered as the most important solution even if a minimun safety level associated with a minimum aggressiveness level has been achieved.

Accidents result from dangerous driving configurations, which are consequent upon either driving aggressiveness or wrong appreciation of the risk which the driver may run.

The third course still little explored may be the study of driving aggressiveness and the tolerable limit of risks. This might well be the most rewarding.

PHILIPPE VENTRE
Engineer in Charge of Impact and Safety Tests
The Renault State-Owned Works

INTRODUCTION

To considerably increase the protection afforded occupants by motor vehicles, two different methods have been applied.

In the U.S.A., with the vitality and strength which are typical of this nation, they wanted to act without delay. Based on accidentology studies, a statement of work was drawn up fixing the characteristic values of protection which had to be attained and the speeds corresponding to the principal accident configurations (Experimental Safety Vehicles programme).

Taking account of the speed limits which were then generalized on roads and motorways throughout the U.S.A., the realization of this programme was destined to eliminate the problem of road casualties.

In Europe, a more prudent method was used, less ambitious to start with, calling largely on analysis. Noting that the development of what actually happens during an accident was little or not known, and that no quantified data were available, it was concluded that it would be illogical to try and propose solutions to a problem that had not yet been clearly stated.

The method, therefore, consisted in defining the problems and then progressively resolving them. This is the principle of E.S.S.S. (1) which has resulted, in France, in the forming of the government programme on A.T.P. (Action Thématique Programmée) (Programmed Thematic Action) which is jointly financed by the government and constructors, and in which RENAULT and its associates PEUGEOT have greatly participated.

LOGIC OF THE METHOD

To bring the task that had been undertaken, to a successful conclusion a rational procedure and logic was necessary. This was introduced in 1970 in a way which, while having been much less spectacular at the beginning than the E.S.V. programme, has attained knowledge that whilst being elaborate is still perfectible.

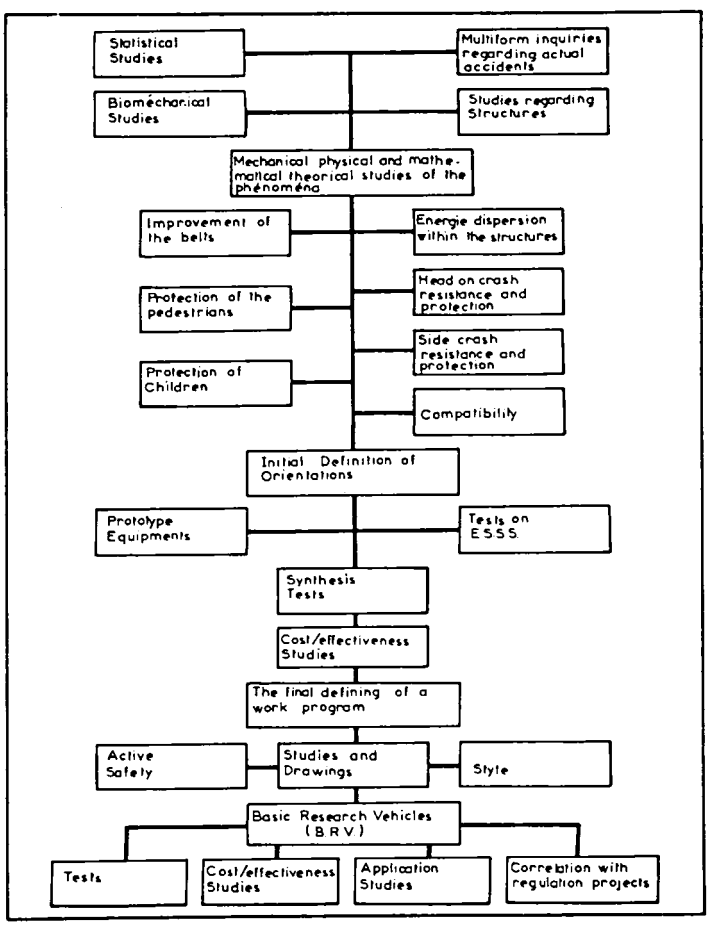


Figure 1.

(1) Experimental Safety Sub Systems.

CONSTITUTING AND ANALYSING STATISTICS

In 1971 in France, it was possible to dispose of important statistical data which allowed priority studies to be started. Since then, further studies as well as many comparisons with a great quantity of American work, have been conducted by the constructors themselves in Europe.

THOROUGH ANALYSING, CASE BY CASE, OF REAL ACCIDENTS

At the beginning of 1970, the PEUGEOT-RENAULT Association set up a multi-purpose survey bringing together teams of police, doctors and technicians, to draw up a file for each accident, showing the circumstances and the consequences for the vehicle occupants.

This survey necessitated analysing methods being drawn up for certain parameters, like those, for example, allowing the violence of the collision for the occupants to be determined as exactly as possible, based on the circumstances of the impact and the state of the vehicles.

Since that time, nearly 2,000 accidents have been analysed in detail, thus producing a precise and real basis, allowing the problem of protecting motor vehicle occupants to be correctly defined.

The most fundamental conclusions obtained up to now are as follows:

- one's attention should be given to occupants and not to the vehicle (the state of the latter after the accident serves only as a technical basis for analysing the impact);
- the notion of "survival space", which is purely mathematical as it is linked with the vehicle and the passengers, is not efficently involved in explaining the consequences of accidents on occupants; and so this notion will in no way whatsoever be used hereafter;
- overall means of evaluating should be available allowing the quality of protection with relation to the limits of bio-mechanical tolerances to be characterized.

These conclusions have culminated in the study of "protection criteria", measured on occupants or, for obvious reasons, on anthropomorphic dummies.

It is certain that the conclusions arrived at have no other value than that for the sample being studied. In spite of the amount of work that has been done and continues to be done, the question of the representativeness of the sample still remains unanswered. It is certain that, in this field, development on a national scale, or even in groups of countries, would give even more valid results.

TRIGGERING AND INTENSIFYING RESEARCH WORK

Stating the problem with general statistics and thorough investigations will not suffice to resolve it on the technical plane. It must be translated into "mechanical" data for the vehicle and "bio-mechanical" data for the occupants. Several types of research work were thus simultaneously engaged in.

Bio-mechanical research work

Protect the occupants, this means succeeding in keeping the dynamic stresses that the occupants undergo during a collision within the limit where serious injury occurs. The bio-mechanical problem is thus to define the "tolerances" of the human body under the different stresses it could meet with in road accidents. These tolerances should then be expressed in terms that can be utilized by the engineers charged with developing means of protection.

These terms are, generally speaking, forces, accelerations, pressures.

The bio-mechanical research work can only be carried out by doctors or with their help in special laboratories in such a way that data that can be used results from the co-operation between doctors-technicians. The physiologic and bio-mechanics laboratory of the PEUGEOT-RENAULT Association has been charged with following up the work that has been carried out on this subject throughout the world, to favour the development of work that has not yet been started, and to carry out certain tests itself.

Fruitful international co-operation has gradually been established in this field, and namely within the Committee of Common Market Constructors (C.C.M.C.).

The bio-mechanical research work should not only specify the mechanical limits of the human body but should also be done in such a way as to allow representative dummies to be made.

The representativeness of the dummies is a problem that has not yet been resolved, but this state of affairs should not prevent the right opinion from being taken, even if the measuring equipment is not well adapted.

Recent progress in bio-mechanics has allowed the differences between human reality and dummies to be better appreciated. This knowledge has caused some of the conclusions obtained solely with dummies to evolve.

This progress has thus changed the protection priorities, in particular for occupants who are belted in. The difficult areas are, in order: the thorax, the pelvis then the head.

Technological research work

The behaviour of a unit as complex as a motor vehicle in cases of collision as varied as real accidents, is very difficult to be simply quantified.

Furthermore, a certain quantity of essential data was lacking, for example, knowledge on the behaviour of material at high speeds of deformation.

Parallel research work has been carried out over several years now on:

- The behaviour and variations of material mechanical characteristics at high speeds of deformation (fig. 2);
- The behaviour of sub-assemblies comprising the material that has already been studied under rapid and varied deformation (fig. 3).

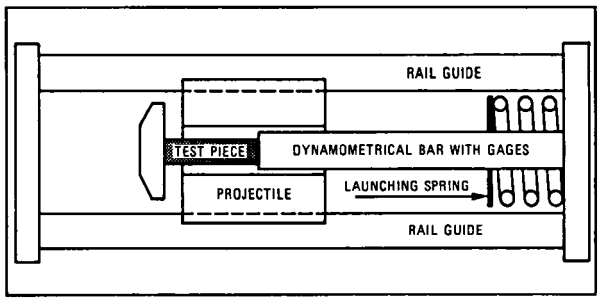


Figure 2. Testing machine for studying the behavior of materials at high speeds of deformation.

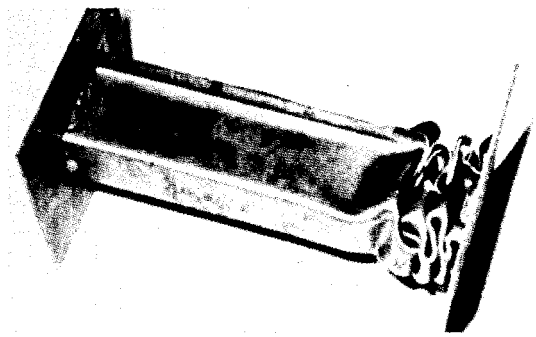


Figure 3. Dynamic compression on elementary structure.

- The behaviour of complete structures, even complete vehicles, in representative configurations of real accidents (fig. 4).

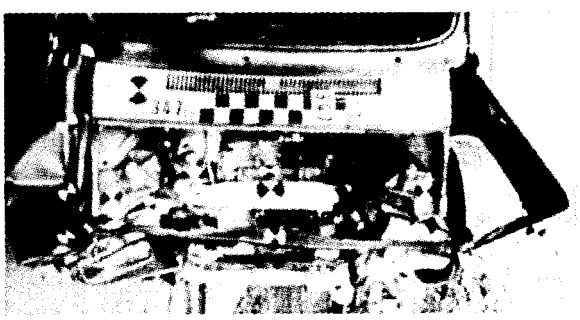


Figure 4. Dynamic crushing test for Renault 12 without engine (50 km/h).

- Defining means for protecting occupants.

This work has called for much time and many resources.

Special research work

Analysing what happens in reality has shown that problems exist that have either been forgotten or underestimated. To quote just two examples, we have carried out thorough research work on the problem of compatibility of vehicles, dimensions, masses and different architectures between themselves (fig. 5), as well as research work on those parameters that determine the seriousness of injuries in pedestrian-vehicle collisions.

Figure 5. Collision Renault 5 against big vehicle.

A great part of the research work is carried out under contract from the French government within the framework of the French Programmed Thematic Actions.

STUDYING SOLUTIONS

The preceding research work has allowed the present situation to be "photographed" and quantified. These data have allowed the solutions to the different problems to be specified, as much on the theoretical plane with mathematical modelizations as on the practical plane with real tests.

But each solution has been studied in great detail. Most of the time, several solutions have been outlined and tested for each case and compared from the point of view of performances and industrial results such as weight, price and possibilities of realization.

Once again, the analytical method prevailed and the studies and tests were carried out on sub-assemblies and not on complete vehicles.

This authorizes a progression that is more complete, simpler and les sonerous than making a complete vehicle at the research stage.

COST/EFFICIENCY RELATIONSHIP

As work has progressed, an extra notion has gained importance: the relationship between the extra expenses for vehicles and efficiency for the community.

This question is not purely and merely a material one; but the increase in price and the making heavy of vehicles presents considerable economic problems which are made worse by the shortage of energy and raw materials. There is therefore an equilibrium to be found and consequently methods for determining this equilibrium.

And so we have tried to specify a method for calculating the cost/efficiency relationship of the protection methods that are being studied.

This evaluation, which is still approximate, of the cost/efficiency relationship is carried out on the basis of fragmental data, and in no relationship with those that the governments should have at their disposal. It is certain that the final choice does not rest with constructors, but with Administration; but what is more, they should have the resources to do this at their disposal.

WHY A BASIC RESEARCH VEHICLE FOR SAFETY?

After three years of intensive work on sub-assemblies, it has been possible to distinguish two types:

- those which it is possible to study in great detail whilst keeping the principle of the method intact because interference with the complete vehicle is slight; this is the case for frontal collisions, rear collisions, rolling over, safety belts, etc.;
- those that cannot be studied in great detail without gradually affecting the vehicle as a whole because they interfere greatly with the latter. This is namely the case for the side collision, which affects the design of the body side, the floor, the doors, the seats, interior fittings, etc. This leads imperceptibly to a complete passenger compartment to which it is interesting to adjoin the "front collison", "rear collision" etc. sub-assembly.

There are many reasons for passing from the sub-assembly analytical stage to the complete vehicle synthesis stage.

1. It is not possible to advance in great detail on certain sub-assemblies that are too integrated in the vehicle without completely designing the latter.

2. The compatibility of the different sub-assemblies between themselves must, in any case, be ascertained.

3. A synthetic basis for reflection is necessary for the problems as a whole concerning safety and the rest of the usual parameters of a vehicle:amenities (aspect, comfort), performance, price. On this score, it is a basis for studying the feasability of the safety vehicle.

4. A *formulated* basis for studying cost-efficiency is necessary.

To make a valid reckoning, not only the cost of the solutions must be studied, but also the cost of their interference and the costs involved.

It is not therefore a "dream-car", and it is strictly dependent on the aims; the latter, furthermore, can be modified during the study should the cost-efficiency relationship not be good or certain problems have been underestimated. However, if the experimental vehicle allows as to go beyond what is demanded by future regulations based on cost-efficiency, in all cases, it is the technological solution and its cost which will set a study of advantages in motion. Then, perhaps, certain points will have to be taken up again.

5. At some time or other, the knowledge that has been acquired will have to be put in concrete form, for on certain points, continuing abstract research work will only mean marking time. But the sum goes over and above that for sub-assemblies, for, besides checking the compatibility of sub-assemblies and the validity of synthesis, a material basis is necessary *for reflecting upon technical improvements* with a view to lowering the price and weight or transposing the solutions to types of vehicle other than the model taken as a basis.

6. At a given moment, it is necessary to inform, on the outside:

a) Administrations, for secondary safety will depend strictly upon the way in which the economic and technical work is translated into a regular language which is forcibly limited. Furthermore, problems like the compatibility of vehicles between themselves can only be resolved by harmonizing international regulations:

b) all those who are concerned with feasability, the weight and the price of all the performances that are proposed. Intellectual speculation needs, furthermore, to be put into concrete form at a given time to prove its veracity;

c) the public, and in this case, it can be multi-purpose:

- for our own personal information by starting talks (but it must be stated that it will always be impossible to precisely define the exact limits of protection and the cost of safety will have to be separated from that depending on the degree of finish of the vehicle;
- for educational use, because it puts in a concrete form the quantity of work that is being, or has been carried out, work which would not otherwise be known about, for the public has at its disposal no information allowing it to judge the merits of the expenses foisted up on it.

PHILOSOPHY OF SYNTHESIS OR WHICH SYNTHESIS?

CHOOSING A BASIS

Synthesizing safety sub-assemblies should lead to a vehicle comprising, naturally, typical characteristics of the RENAULT range: Saloon with tail-gate, front-wheel drive, 4/5 seats, extra-long suspension, 4 independant wheels.

Furthermore, so as to limit studies, and because these vehicle sub-assemblies are not directly connected with secondary safety, standard mechanical units should, as far as possible, be used.

CHOOSING THE DEFINITION

One of the aims of this synthesis study is to see whether the "secondary safety" parameter can be introduced, at the same time the vehicle keeping all that makes it attractive at the present time.

To do this, the vehicle must be comfortable and spacious; it must be neither dreary nor coercive. But the solutions that are retained must also be feasible with the technology that is known or can be foreseen in the near future; the weight and price must remain within acceptable limits for the category of vehicle: secondary safety must remain saleable.

As probably most of the main constructors have done, we have studied proximity radar, heavy duty, hydraulic bumpers (fig. 6) etc. But all these sub-assemblies, once

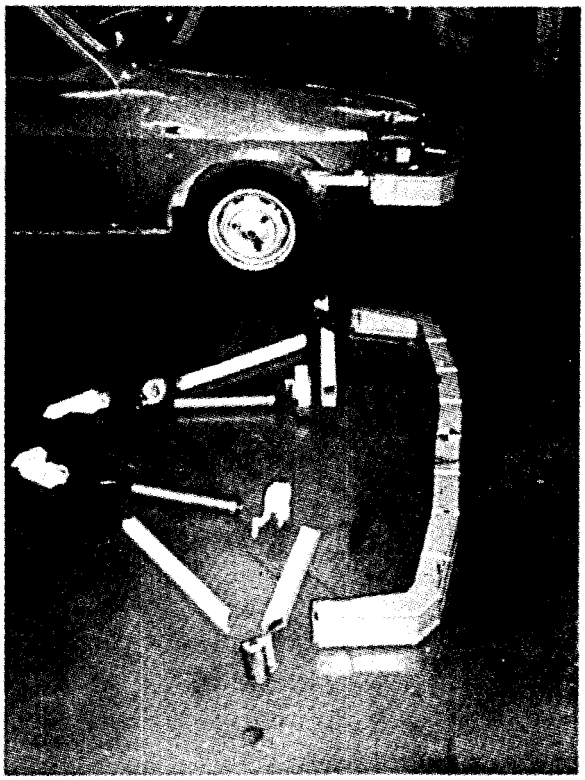


Figure 6. Hydraulic bumper.

they hab been studied, made and tested, proved to be so costly and of so uncertain an efficiency that it was quite obvious that they were not good solutions in the order of priorities.

In the field of occupant protection, it proved obvious to take only restrained passengers into consideration.

As far as the restraining device is concerned, statistics, multipurpose investigations and cost-efficiency studies, even though they are roughly worked out, have shown that the safety belt is the best means that is presently known.

The French government's decision to make the wearing of the belt compulsory dispenses with fitting coercion devices, at least, for the front seats.

The air bag is still in the hypothetical field and it is very difficult to evaluate its potential efficiency in all configurations of collisions (lateral impact for example).

CHOOSING TECHNICAL SOLUTIONS

The different work that has jointly been carried out has led to two types of basic data:

- a list of representative accidents and priorities, and a breakdown of the types according to the severeness of the accidents;
- technological solutions and an evaluation of their costs as a function of their performances.

Specifying an analysing and classifying method for collision severness has allowed results to be obtained whose validity depends on that for the accident samples on which the study is carried out. These results seem to approach those that may be published elsewhere for Europe.

However, to put our work into concrete form using technical solutions which must necessarily satisfy performance requirements, the levels were fixed arbitrarily, following many studies and tests, by choosing the as high as possible performance value on which the costs increase extremely quickly due to the necessity of using technologies which are more and more sophisticated, whereas the efficiency increases only very slowly, as the percentage of occupants concerned constantly decreases.

But naturally, the final choice on the performance levels to be called for can only be decided at the nation level; it is even very much to be wished for that this will done in groups of nations.

TECHNOLOGIE OF SOLUTIONS AND RESULTS

The preceding reflections and work, taken as a whole, have led to the results included in this synthesis. Their details are given in order of decreasing statistical importance according to the types of collision.

FRONTAL COLLISION

Recall concerning statistics and accidentology

It is always difficult to sum up a complex situation in a few figures. Furthermore, all the data are not always available simultaneously. In this particular case, of the statistical data for France as a whole, we know in detail those for 1969. But, it is now rather far off; we know the total number of people involved in accidents in 1972 as well as the number killed, but we do not know how they are classed as a function of the type of collision and obstacle. However, to use the most recent data, we have supposed that this breakdown is not different in 1972 from that in 1969.

The first data that we have at our disposal for checking this hypothesis show that the actual death rate for 1972 is rather lower than this estimation. In the case of frontal collisions, the situation can be summed up in three tables, knowing that they represent 58 % of the vehicles involved in accidents.

It can be seen therefore that the most severe type of obstacle is the fixed and rigid one; the most frequent, another car; and the most representative, the collision at 60°. Fully analysing the accidents case by case has confirmed this finding, and the typical accident can be represented by this case drawn from our investigation (fig. 7):

Choice of performances

In spite of the vastness of the work that has been carried out and the knowledge already gained, knowledge on

BREAKDOWN OF PEOPLE INVOLVED IN FRONTAL COLLISIONS (FRANCE 1972)

Category	Number	% of the category as a whole all types of accident together	% of those involved as a whole in this type of collision
Killed	4 762	55.2	1.7
Seriously injured	32 406	59.6	11.6
Slightly injured	88 604	57.7	31.8
Uninjured ...	152 721	57.7	54.9

BREAKDOWN OF FRONTAL ACCIDENTS AS A FUNCTION OF THE OBSTACLE

Type of obstacle	% of cases	% of people killed
Car	68.8	29.1
Utility vehicle	16.9	25.6
Fixed and rigid obstacle	12.6	39.6
Others	1.7	5.7

BREAKDOWN PER TYPE OF COLLISION

Type of collision	all accidents	serious accidents
Apportioned	22 %	25 %
60° (to LH or RH) .	36 %	57 %
Very much offset...	35 %	13 %
Narrow obstacle on same centre-line	7 %	5 %

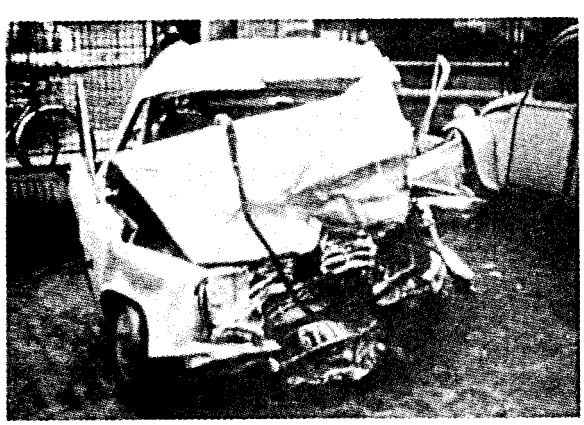


Figure 7. Typical frontal collision (taken from accident investigation).

phenomena concerning accidentology as a whole is far from being totally clear.

The considerable number of different cases of accidents which are due to the different types of collisions and the different areas collided with, the differences in behaviour of a same vehicle in these many cases, and the variations in individual tolerances in all these, make it difficult, if not impossible, to precisely quantify the problem.

However, the ratings and research work based on average hypotheses allow us to advance along logical though somewhat imperfect lines.

As far as possible, empiricism gives way to rigour.

We have studied the breakdown of collision violence for the occupants.

The classification of the violence of these collisions, which we published, by the way, at the last international conference on safety vehicles at Kyoto, shows us that it is very important te determine, with precision the speeds for which we must provide occupant protection.

Indeed, the cost of solutions varies very much with these speeds at the same time affecting fewer and fewer occupants.

It is by succesive progression of the different research work that the most efficient solutions have been able to be approached. This method differs from that consisting in working with a "a priori" statement of work.

When taking stock at a given moment, it is possible to translate into "performance level" the stage that has been reached, without that meaning however that it is a technical limit or an economic aim that is to be wished for.

TYPES OF TEST AND TEST SPEEDS

Taking account of statistics and accident investigations, the front Sub-Assembly was designed to protect the occupants in the following conditions:

- frontal, orthogonal collision with fixed barrier at 65 km/hr., on same centre-line as the latter. Two dummies belted into front seats. This test is not representative of a great number of real cases. However, present legislation has made this, over the past ten years, "the" criterium of evaluation at 50 km/hr. (fig 8).

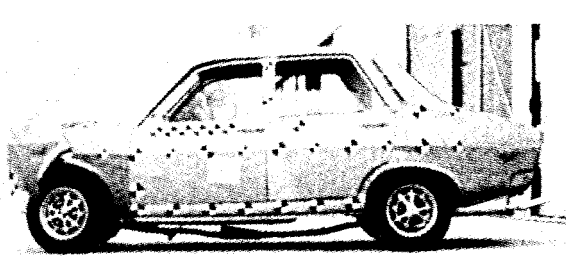


Figure 8. Experimental collision with Renault 12; frontal crash 50 km/h.

Consequently, to evaluate the progress achieved with a valid basis for comparison, this type of test has been maintained.

- frontal collision, on same centre-line as barrier, with fixed barrier inclined at 60° on trajectory centre-line, at 65 km/hr., with two dummies belted into front seats (fig. 9).

This test would seem to us to be the best compromise for covering the maxium of frontal tests (asymmetrical) and for establishing an average between the collisions between

vehicles (the more numerous) and with fixed obstacles (the more severe).

- collision with fixed, narrow obstacle (post of diameter 305 mm.) at 50 km/hr.; two dummies belted into front seats (fig. 10).

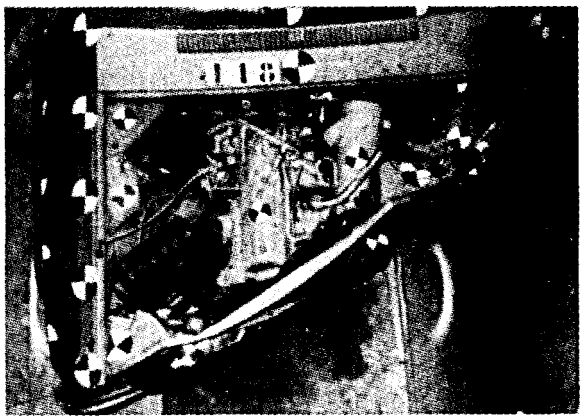


Figure 9. Experimental collision with Renault 12; 30° 65 km/h.

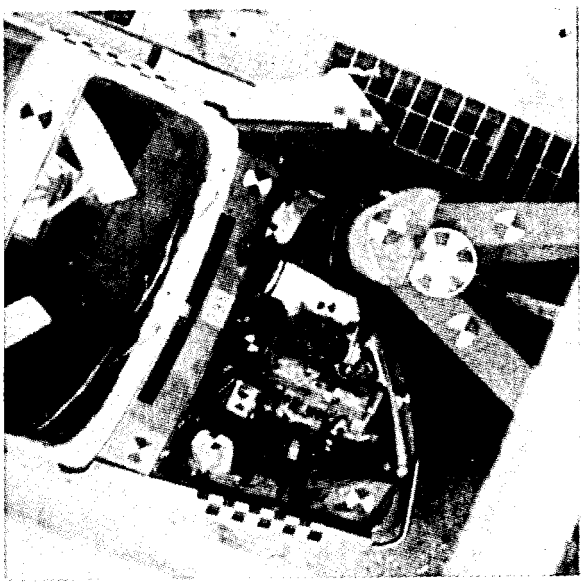


Figure 10. Experimental collision with Renault 12; offset pole crash — 50 km/h.

The point of impact must be somewhere over the width of the vehicle. This test is very severe structure and, depending on the point of impact, on the restraining device.

Two remarks must be made:

— the problem of collisions with heavy goods vehicles is treated elsewhere, as the solution cannot be found with light vehicles;

— no test has been specified at low speed with unrestrained dummies; at too low a speed, this test is not selective and at high speed it is too severe. It is not realistic, for the actual trajectory of the occupants, unrestrained, depends mainly on the collision circumstances, and the result then becomes a matter of chance: either the head comes into contact with the windscreen, which can be tolerable, or the head comes into contact with the uprights or cross-member, which may be fatal. Furthermore, it is in the speed zone for which the belt is totally effective.

CRITERIA FOR EVALUATING

The sole aim of all this work as a whole is to see that occupant protection is ensured. Consequently, the evaluation of the result can only be done by seeing that human tolerances to the mechanical stresses of a collision have not been gone beyond.

This supposes that the tolerances are known, which is unfortunately far from being the case, and that they can all be measured, which has not been done either.

Nonetheless, as certain values could be translated into physical parameters that could be measured, it has been possible to make progress using anthropomorphic dummies.

The aim of the research work is then to obtain lower and lower measured values, showing lower and lower stress levels.

We have reasoned only for occupants restrained with three point belts.

The data provided by accident investigation and bio-mechanical research work tend to show that the principal protection to be provided is that for the thorax much more so than for the head.

The Medical research work that has already been carried out in the world on the limits of thorax strength indicates that the human tolerances in crushing stresses of the rib cage are relatively low.

There is a major difficulty in reproducing the behaviour of the human body in this area with dummies; however, an analogy can be had by expressing these maximum for values as maximum deceleration measured on the dummy.

60 g maximum value is today accepted as the limit not to be gone beyond. Our own work has led us to think that, in the future, it will be preferable to lower this figure.

As far as head protection is concerned, accident investigations are unanimous in showing that for people who are belted in and whose heads have hit nothing, no serious injury appears, even in cases of very violent accidents.

Furthermore, the bio-mechanical work having been carried out mainly for cranium impacts with infinitely rigid surfaces (block of metal or concrete), the limits that have been determined are not applicable to impacts that can take place with the interior walls of an automobile.

Consequently, we consider that on this point, bio-mechanics has much progress to make, and it not being possible to fix acceleration values having a real bio-mechanical meaning, we are trying to obtain levels as low as possible.

Reasoning for belted in occupants, we are obliged to be anxious about protecting the internal abdominal organs.

So our physiological and bio-mechanics laboratory has developed a criterion for protection called:

- Sub-marining criterion.

It consists mainly for us to see that the sub-abdominal part of the belt never passes over the iliac peaks (strong part of pelvis) during all the collision phase.

Finally, to ensure that the bone chain comprising the knee, the femur and the pelvis is protected, we observe the compressive stresses which are conveyed in the femur in the case of contact with the hard part of the front of the passenger compartment.

It is unanimously admitted at the present time that the maximum stresses can go up to 750 kg per femur.

This reasoning remains valid for all types of collision as a whole.

Technologies

After having tried several solutions (some of which have already been presented), some technologies were chosen for their weight-performance-price relationship.

STRUCTURE (fig. 11)

The solution that has been adopted is as follows: two parallel, lower side members and two diverging, upper members connect the front of the lower members to the top of the front pillars. The side valance and an enclosing sheet form these box-section, side members, which dip to the front. This arrangement allows the compressive stresses to be transferred to the top and bottom of the structure, the body sides to be used in compression, and the side member-caisson to be made from thin sheet but of large section.

The upper and lower side members are fixed to a large, front, lower cross member acting as structural cross-member, anti-agressiveness cross-member, distributor of offset stresses in the members, and shield support cross-member.

This structure leads to the lower side-members being wide apart and consequently to wide tracks.

The front unit is connected to the passenger compartment through a very strong frame formed from both the front pillars and the lower, windscreen cross-member. Without the sections being very large, they are of ample dimensions and, theres is no problem of transition.

The assemblies have been specially designed to eliminate bending moments, unfolding, tearing. This has led to complex assemblies being made whose feasability and cost (difficulty of making certain units) are not yet known.

The frame comprising the front pillars and the bay cross-member is partly closed at the top part by hollow sections connecting the front structure lower side members to the valances.

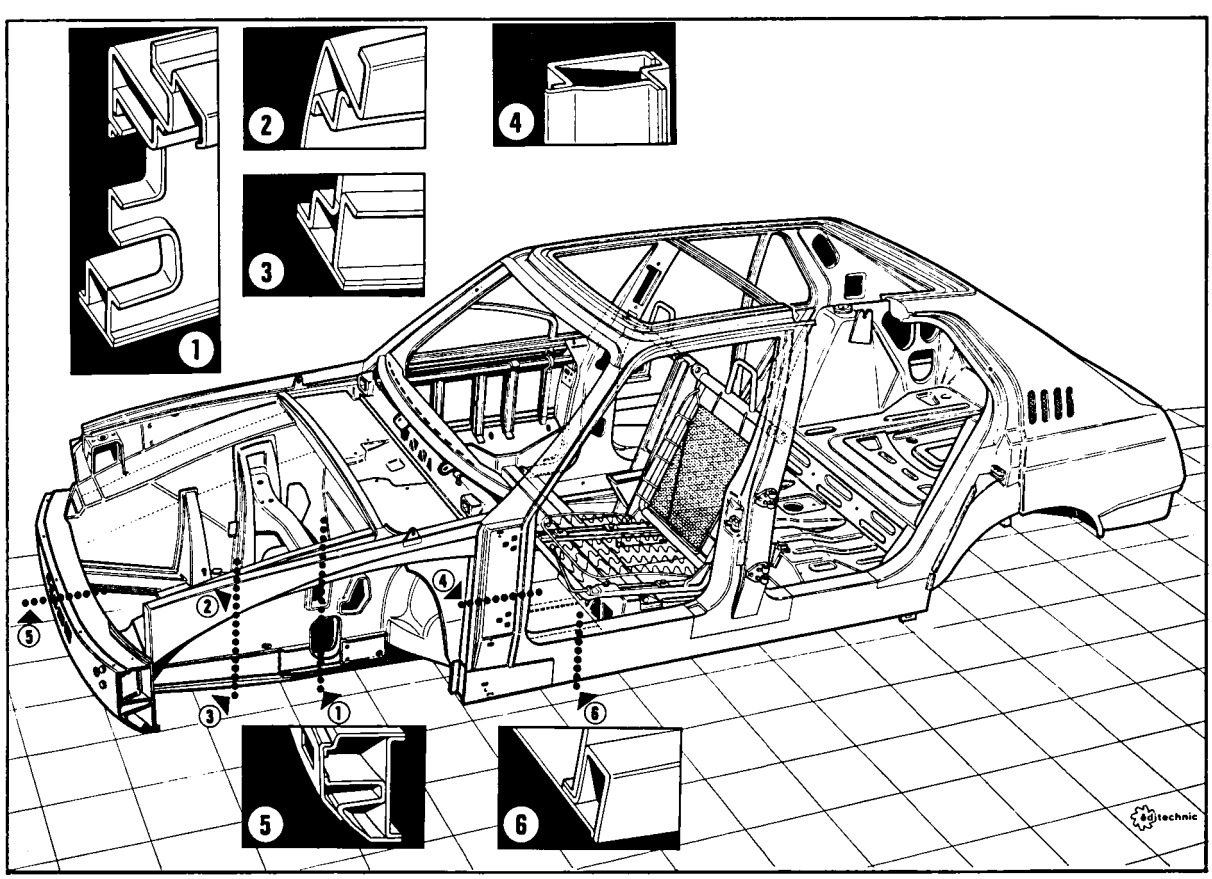


Figure 11. Structure of vehicle; frontal crash.

Besides their classical structural function, these sections hollow take part in passenger compartment strength by preventing, for example, the wheels from intruding in off-set collisions.

The floor is made up of a sheet with a tunnel down the centre having, small-radius stamping to ensure stability when being compressed; the two side-members from the front unit, and the two side valances.

The solution can be considered as being satisfactory, but it is very demanding on the front axle.

RESTRAINING DEVICES

Even before the French government took the decision to make the wearing of seat belts compulsory outside built-up areas, we had chosen this means of protection whose cost remains low and whose technology is well-known and can be perfected by ourselves.

The future of the air-bag is still hypothetical and furthermore, it is very difficult to evaluate its efficiency in all configurations of collisions.

On the other hand, after hesitating a long time between active and passive safety belts, the government decision simplified matters. The belts are active, i.e. the occupant belts himself in; but, taking the present technological evolution into account, they are of the retractor type.

The quantity of work that has been carried out within the framework of advanced studies on "restraining devices" and ATP, has resulted in a sophisticated seat belt being designed which, besides the webbing and the energy absorbers being slightly different from the standard production ones, is fitted mainly with a pre-loader which is operated when the collision occurs and whose purpose is to tighten the belt on the occupant within a few milli-seconds, thus taking up any play and increasing the restraining strength very quickly to levels sufficient enough to ensure correct coupling between the occupant and the vehicle (fig. 12).

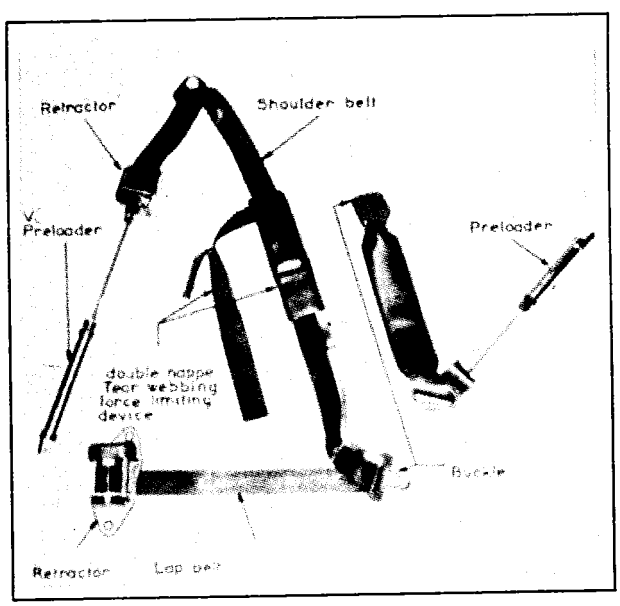


Figure 12. Safety belts with preloaders.

This belt has been developed on dummies, but its specification remains to be improved upon and namely the loading of the absorbers and the width of the webbing in the thorax area.

The rear seats are equipped with 3 points belts for 3 people. They are ordinary belts (cost-efficency relationship), but it would be possible to fit preloaders if necessary. For children's protection, we have turned to optional systems: children's seat types or cushions placed against the front-seat backs.

EXTRA INTERIOR FITTINGS

On the dash-board (fig. 13)

The main controls are grouped together in a functional unit in front of the driver; the extra controls will be on a console. Facing the passenger, and because the restraining device will not necessitate extra means, the dash board is relieved as far as possible to maintain the impression of room. This choice is deliberately different from the "padded" dash boards which are meant to furnish passive protection. The dash board is fitted with no luxuries or superflous accessoires. On the other hand, it should give the impression of safety from its aspect.

Figure 13. BRV dashboard (vehicle being assembled).

Steering column

The design of the steering column depends mainly, as far as secondary safety is concerned, on the restraining device specified for the occupant. It cannot, in any case, be identical when protecting an occupant in free flight, checked with an air-bag or restrained with a belt.

In the first case (free flight), it seems that a column that moves upwards in a primary collision is not a bad thing; on the contrary, it affords considerable protection because of the stresses, which are quite low but not zero, that the occupant undergoes when bending it back to the dash board and the windscreen, and therefore over a great distance.

On the other hand, this undesirable movement, which cannot be tolerated in the case of an air-bag, should be and remain in a perfectly specified position when the occupant hits it and is checked by it.

It this case, the column should be capable of extra absorption by crushing under load, which presents several

technological problems for all those cases of collision, and they are the majority, in which the centrelines are offset.

In the case of the occupant restrained with a belt, the problem is again different. As with the air-bag, the column must be in a precise position with relation to the occupant and therefore no, or very little, undesirable movement will be tolerated during the first phase of the collision.

On the other hand, it is no longer the thorax-head masses together which come into contact, but only the head which, at the end of the movement, strikes the steering wheel. It is, therefore, no longer necessary to require the column to absorb energy for a mass of approximately fourty kilos., but the steering wheel to check from 5 to 10 kg. The technological solutions are therefore different.

In the case of this synthetic vehicle, the column, which is perfectly connected to the windscreen, lower cross-member, is practically insensitive to the primary collision, remains relatively stiff, and the steering wheel has a rim that is less stiff than the present rims and has a great deal of central padding; the latter, which is restraining from the aspect and instrument visibility point of view, could be replaced with a solution presently at the research level, i.e. the mini air-bag or inflatable cover.

Test results

The tests are carried out in two parts :

1. CHECKING STRUCTURAL BEHAVIOUR (fig. 14-15)

Because the orthogonal frontal collision with a fixed barrier is presently the only international reference, and in spite of its lack of representativity, only the results of this type of test will be given: for a test at 65 km/hr. (65.7 exactly) the results are :

- *dynamic crushing;*

60 cm; this stopping distance is a little short, for, in fact, 65 cm at least had been aimed for.

Due to this, the violence of the collision is a little too much for the occupants, and it must be optimized; this will be done by calibrating the members. Sufficient crushing distance remains available at the front of the passenger compartment.

- *dynamic steering wheel recoil;* 20 *mm*
- *deceleration law:*

Approaching the required trapezoidal law, it presents nonetheless a favourable, initial "early impulse" point for the occupants but is a little high for compatibility with other vehicles.

2. CHECKING THE BEHAVIOUR OF RESTRAINING DEVICES (fig. 16)

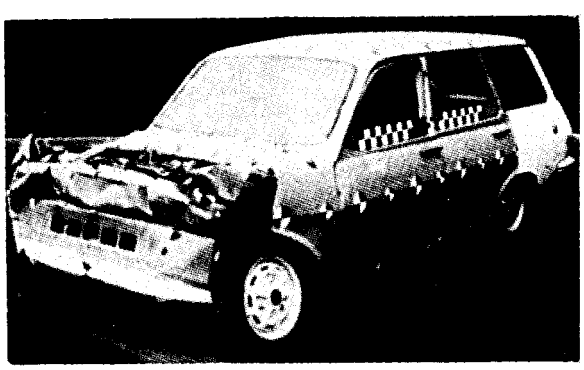


Figure 14. Dummy vehicle (Renault 12 based) in front collision 65 km/h.

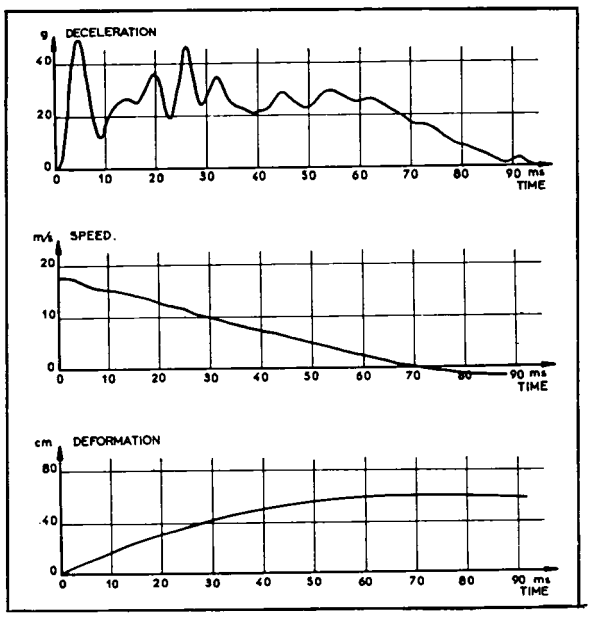


Figure 15. Test results – vehicle.

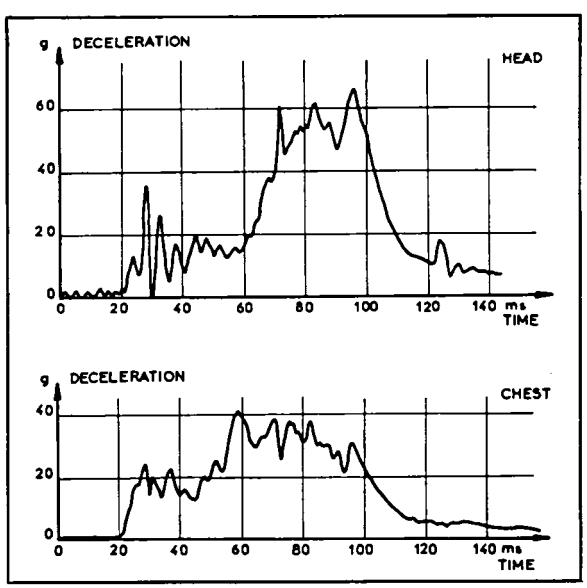


Figure 16. Test results – occupant.

The tests are carried out on a trolley representing the test-vehicle passenger compartment, including possible dynamic deformation, and in particular, steering wheel recoil; the trolley is subject to a deceleration law identical to that for the vehicle.

Protection criteria	Results
Head	60 g
Thorax	38 g
Pelvis	No submarining
Femur	250 kg

Cost/efficiency

To make a choice between the different technical solutions that are possible, or to grade them, the cost-efficiency ratio must be calculated.

Although evaluating the cost of solutions presents no particular problem, the difficulty is great when evaluating their efficiency, especially if overall and subjective estimations are to be avoided.

The method that has been adopted is the following:

1. Characterization of French accidentology as a function of the parameters that are decisive for occupant protection :

- obstacle collided with
- area of vehicle collided with
- type and mass of vehicle
- relationship between the severeness of the collision and seriousness of the injuries,

with the seriousness of the injuries being taken as reference.

2. Institution of a sample of accidents that are representative of the previously described situation, with investigation data. The parameters for each collision are therefore all known.

3. Analysing, case by case, file by file, the influence of the solutions that are envisaged, either taken separately or in combinations. This work is carried out by a multi-field team.

4. It is possible to estimate from the results thus obtained, and with the sample being considered as representative, the overall influence, on the country level, that certain dispositions could have, when applied to occupant safety in vehicles. This method, which is a little complex because it is very analytical, has the advantage of studying accidentology per vehicle family and thus evaluating the efficiency of the solutions being envisaged as a function of these families, and then extrapolating at the total car-population level.

GENERAL REMARKS

The car population taken into account is supposed to be entirely made up of equipped vehicles. With 10 % renewal every year, this implies total renewal in 10 years. Another hypothesis analysing the influence of progressively introducing a device would not alter the final order.

- The study on efficiency took into account, in frontal collisions taken as a whole, only those collisions with OFR (fixed and rigid obstacle) and with other light vehicles. Collisions with heavy goods vehicles have been left aside because they are special.
- All the measures that have been envisaged increase the number of uninjured, but the latter do not appear in the tables.

PROTECTING ARRANGEMENTS BEING ENVISAGED

- Two, three-point belts without retractors, fitted for front seats of vehicle with unmodified structure;
- Two, three-point belts without retractors, fitted for front seats + lap belts for rear seats, vehicle with unmodified structure;
- Two three-point belts with retractors, fitted for front seats of vehicle with unmodified structure;
- Two, three-point belts without retractors, fitted in a vehicle with a modified structure for different collisions with a fixed and rigid obstacle at 50 km/hr. + lap belts for rear seats + special steering column + padded steering wheel + laminated windscreen;
- The same arrangements as above in a modified structure for various collisions at 65 km/hr., with belt pre-tightening device:
- Air bags installed in a vehicle that has been modified as above.

ANALYSING EFFICIENCY

The table hereunder gives the percentages of people who have been saved by the solution(s) envisaged for the three categories of people involved whilst taking into account possible transfers from one category to another (in both directions) and for a 100 % effective application.

Protecting arrangements	Deaths avoided	Seriously injured saved	Slightly injured saved
	%	%	%
2 front seat belts present structure	49.4	34.1	38.1
Front and rear belts present structure	56.4	42.5	36.4
Front and rear belts + strengthened structure 50 km/hr.	71.1	73.2	31
Front and rear belts + strengthened structure 65 km/hr.	80	84	20
Front belts + preloaders + normal rear belts + structure 65 km/hr.	84	84	40

CALCULATING THE COST

The cost of the solutions is known for a certain number of vehicles.

By extrapolating the total car population and taking an average value, it is possible to obtain the overall cost for this car population which is estimated at 14 million vehicles.

The calculations are carried out in francs at a constant value as the economists can then involve all compound cost hypotheses.

RESULTS

They can be put in form table giving the overall cost over 10 years as a function of a percentage evaluation of victims spared in frontal collisions alone and for the French car population.

A comparison with the percentages of avoided deaths per most optimistic and most pessimistic classes of speeds has furnished a relative error over all deaths of approximately 10 %.

Equipment	Use rate	Cost*	Deaths avoided	Seriously injured avoided
	%	$\overline{M}$F**	%	%
3-points belts ..	0.4	1.68	20	14
	0.6	1.68	30	20
Retractor belts.	0.6	3.92	30	20
	0.8	3.92	40	27
50 km/hr. structure + retractor belts	0.6	8.1	42	44
	0.8	8.1	56	58
	1	8.1	71	73
65 km/hr. structure + preloader belts	0.8	13.4	67	67
	1	13.4	84	84
65 km/hr. structure + Air bags	1	18.5	84	84

* Total cost for equiping car population calculated in francs 1974. Price for clients - tax included.
** Milliards of F:

SIDE COLLISION

Recall concerning statistics and accidentology

The hypotheses being the same as for the frontal collision, and knowing that it represents 27.2 % of the vehicles involved in accidents the situation for the side collision is presented as follows :

BREAKDOWN OF PEOPLE INVOLVED IN ACCIDENTS

Category	Number	% of the category as a whole all types of accident together	% of those involved as a whole in side collisions
Killed	2.692	31.2	2
Seriously injured	14.192	26.1	10.9
Slightly injured	42.536	27.7	32.6
uninjured ...	70.935	26.8	54.4

BREAKDOWN AS A FUNCTION OF THE OBSTACLE

Type of obstacle	% of cases (sample France)	% of people killed (Peugeot-Renault sample)
Car	70.5	66
Utility vehicle	15.5	17
Fixed, rigid obstacle	9	12
Others	5	5

BREAKDOWN PER TYPE OF COLLISION

Collision area	All accidents	Serious accidents
	%	%
Outside passenger compartment area ..	45	20
In passenger compartment area	55	80

The most frequent obstacle and the most severe is therefore another car colliding with the passenger compartment area.

Analysing the accidents has confirmed this point, and the typical accident can be represented by this case taken from our investigation (fig. 17).

Figure 17. Typical side collision (taken from real accident investigation).

Choice of performances

TYPES OF TEST AND TEST SPEEDS

The side collision sub-assembly was designed to protect the occupants under the following conditions :

Side collision by another vehicle whose centre-line is in line with the front occupants' hips, at an angle varying from 90° to 45° to the front.

As agressive vehicles will exist for a long time yet within the world automobile population, the striking vehicle is not another safety vehicle with reduced agressivity.

Basic research work has been carried out on solutions allowing collisions with present-day production vehicle to be withstood.

The choice of impact speed depends on the striking vehicle's mass; the aim is to protect the occupants with an impact speed of 50 km/hr. for a striking vehicle with a mass equal to that of the vehicle collided with.

The collisions can be simulated with either the vehicle collided with being stationary, or with both vehicles in movement so as to reproduce the exact kinematics of the occupants.

Protection performance with a fixed, narrow, rigid obstacle will be considered as a consequence of what happens in the car/car collision. In this last case, as a first approximation, the protection speed limit will be about 30 km /h.

CRITERIA FOR EVALUATING

The reflections for the frontal collision are even more valid for the side collision.

If the biomechanical data for the first case are still fragmentary, for the second case they are even less known.

Indeed the classification of the protection priorities that has been drawn up for the frontal collision (thorax, internal organs, head) has no meaning for the side collision.

The nature and the seriousness of the injuries will depend, for the main part, on the nature of the obstacle and the behaviour of the side structure of the vehicle, and on the fitting out of the inside surfaces.

However, analysing accidents and the little bio-mechanical work that has been carried out in this field have shown that the thorax tolerance in a side impact is rather lower than in the front collision, that pelvis protection can this time be expressed in terms of maximum acceleration, which seems furthermore rather low, and finally that the tolerance of the skull in side accidents as well as that for the brain in side accelerations is also lower than in frontal collisions.

Furthermore, although accident investigation has shown that there is apparently no problem for occupants who are belted in for important headbending movements on an anteroposterior centreline, the tolerance of side bending of the head would seem low and we have taken as a basis not to go over a head-trunk angle of 45°.

Technologies

STRUCTURE (fig. 18)

The severe requirements for the side collision have led to a rather unusual body side structure being designed.

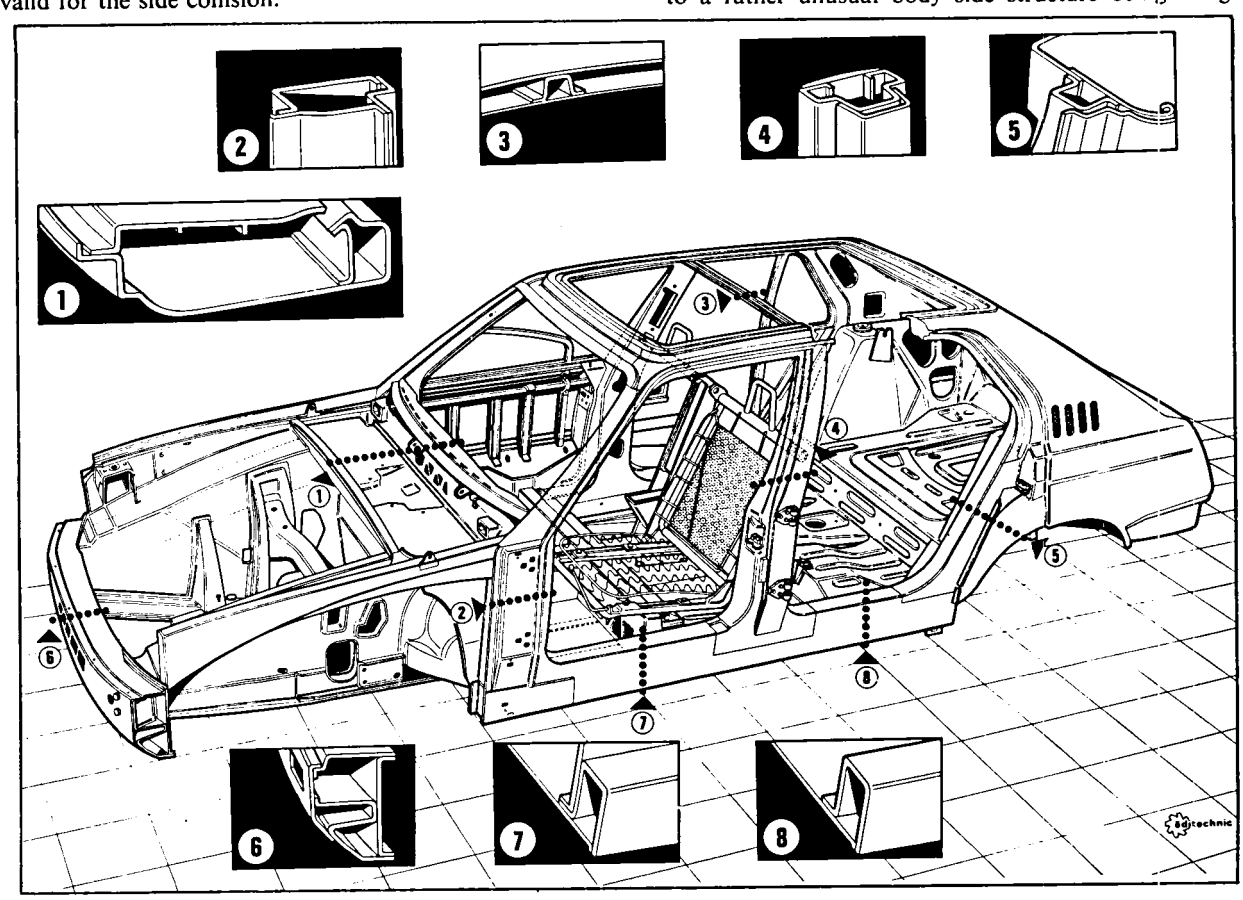


Figure 18. Structure of vehicle; lateral impact.

The sections of the windscreen uprights and the rear pillars are unorthodox, the former for reasons of strength and maximum visibility (contradictory requirements) the latter for transition reasons with the upper side section and the valance.

The middle pillar is greatly tilted backwards so as to limit interference with the front passengers' heads and allow the front shoulder belt anchor point to be correctly positionned.

The design of the front pillar sections and upper side sections is special because the main worry was to have an excellent bearing surface between the pillar and the door (longitudinal compression strength and possibility of opening the doors after a collision), easy transition of sections and no sharp edges inside the passenger compartment, which leads to a different sealing strip being stuck on the body than the "snappon" one now used on the Renault 12.

The doors cover and bear on the valances thus actively taking part in penetration strength.

They are an essential element of the body-side strength and are made at the present of many units, and it is probable that they will have to form the subject of optimizing work as far as their weight and cost are concerned.

The "door with window frame" solution has been kept.

At the top part, the hope of having only one beam acting simultaneously as a compression beam for the frontal collision thus being positioned rather on the inside of the vehicle so as to better line up with the front pillar, and as a bending beam between hinges and locks for side collision and rather on the outside of the vehicle could not be put into practice.

So, for the moment, both functions are distinct and require two elements, the window passing between the two.

The beam technologie has not been finally decided upon and the choice will be made between ordinary sheet, alloy steel sheet of the Soldur or Corten type, and aluminium section. It is the best weight-cost-efficiency compromise which will determine the choice.

For the B.R.V., the solution that has been retained is the use of commercial steel tube with an elastic limit of 60-69 kg/mm^2 and an O.D. of 36 mm.

At the bottom part and so as to withstand the stiff units of the striking vehicles, the door is in fact a deep caisson made of thick sheet that is capable of dissipating energy.

In this area, it is the outside wall of the caisson which is the skin sheet; the join will be covered with the light accident protection fillet or a finishing strip.

On the floor, the important innovation which was necessitated by the problem of side protection, is the cross-member under the front seat. It is made up of two levels: the first one between the valance and the top part of tunnel, of trapezoidal section made from 1.5 mm sheet, perforated to make it lighter and to allow heating air to circulate to the rear wells; the second one from valance to valance, continuous above the tunnel, of closed section made from 2.5 mm sheet.

A small extra cross member is placed at the rear to ensure transversal strength in the rear passengers' area.

INTERIOR FITTING

The technologies that have been retained for the floor, the body side and the doors are not sufficient to provide the necessary strength for satisfying the required aims in the side collision. Using the seats as a strut between both body sides is indispensable for general strength and, in particular, for the window sill.

Two solutions have remained in balance over a long period of time; the first one consisted in keeping the present day seat characteristics: considerable longitudinal adjustment and seat-back adjustment. The problem then consists in the seats always acting as a strut between themselves and the middle pillars.

The advantages are that the driver's controls are in no way modified and that the vehicle keeps its usual functionality;

The disadvantages are that it leads to seats of uncommon thickness taking up too much room in the passenger compartment and mainly having a too prohibitive weight and price.

However, the solution is viable and tests carried out on a dummy-vehicle have shown that the aims have been attained.

The second solution which has finally been retained is that with fixed seats acting as a strut between the middle pillars (fig. 19).

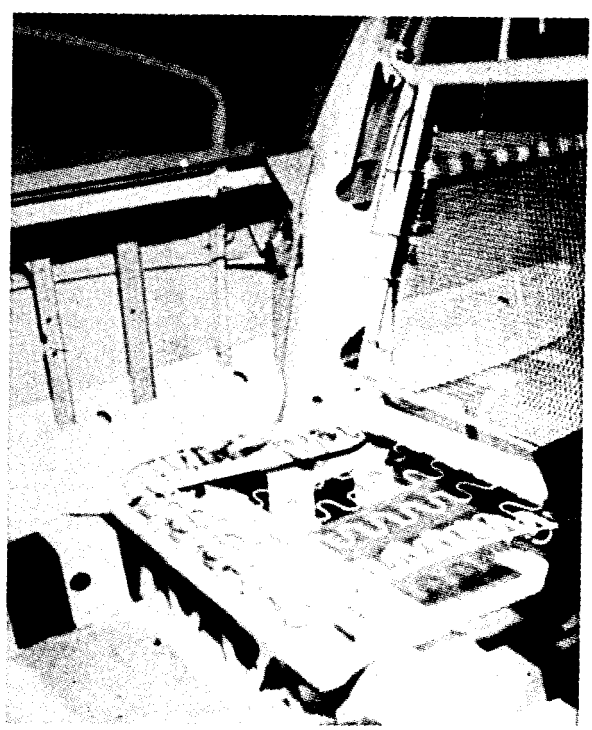


Figure 19. Front seat structure.

The technological realisation leads to the seats being hang on the stay-tube which is connected to the pillars at the window sill level and to leaving them with slight longitudinal adjustment coupled with seat-back reclining.

This solution has the advantage of leading to seats that are simple and, therefore, their price differs but little from standard seat prices. It also has the advantage of positioning the front passengers practically always in the same place, which makes the task of protecting them easier.

On the other hand, it has the disadvantages of:

- modifying the known and appreciated automobile functionality;
- necessitating extra adjustment concerning the driver's controls (for the moment, there is only the pedal assembly concerned and it is using the vehicle that will tell us if this should be extended to the steering column);
- taking up a little passenger compartment length thus providing sufficient room for tall rear passengers;
- reducing the impression of roominess inside the passenger compartment.

The solution will allow a report to be drawn up which is supported with the weight-price-efficiency-utilisation parameters.

The fixed seat solution in a research vehicle, will allow the problems of side collisions and front occupant protection in rear end collisions and rear occupant protection in front end collisions to be resolved with relatively light seats, but they cannot be adapted to existing structures.

The rear seats are similar to those for the Renault 16, but the upper seat-back tube is locked in the normal position to provide staying of the rear part of the passenger compartment for the side collision and also protection for luggage.

PROTECTING OCCUPANTS

Windows

This is one of the most advanced technological points of the vehicle. Supported by the progress presently being made by glass-makers on laminated glass, we have banked on safety windows having no glass on the passenger compartment side.

The choice is more subjective for the windscreen which, theoretically, the restraining devices should not allow to be collided with, than for the side windows which are the only "padding" that we count on for protecting the head by limiting the head-trunk side angle and the head side deceleration.

In this spirit and because the need for this protection is permanent, we have been led to making fixed that part of the side window situated in an area concerning the head positions from the 5th. to the 95th. occupant percentile. The part of the window that winds down is therefore reduced.

For the moment, in the rear doors, the window is totally fixed for simplicity's sake, but the solution for the front door could be adapted.

These windows which are very difficult to transpierce, are efficient protection against ejecting even though it be partial, but they raise the problem of getting to or getting out occupants in the case all the doors are blocked following multiple collisions; the tests will tell if the back window can remain of tempered glass.

Padding (fig. 20-21)

In two places:

- at the window sill level to limit thorax deceleration;

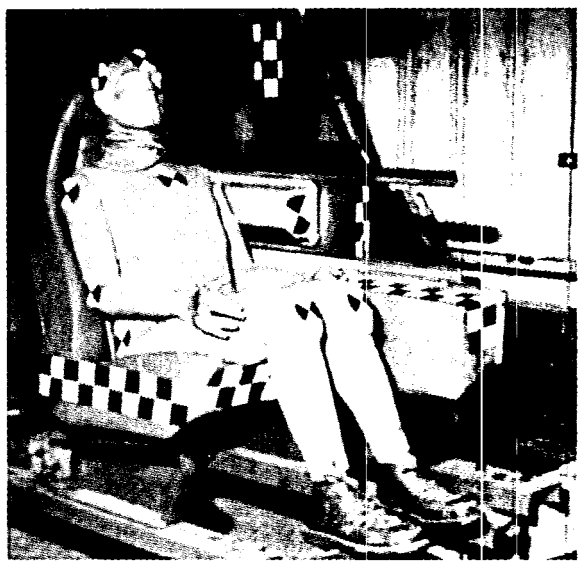


Figure 20. Test for door padding definition.

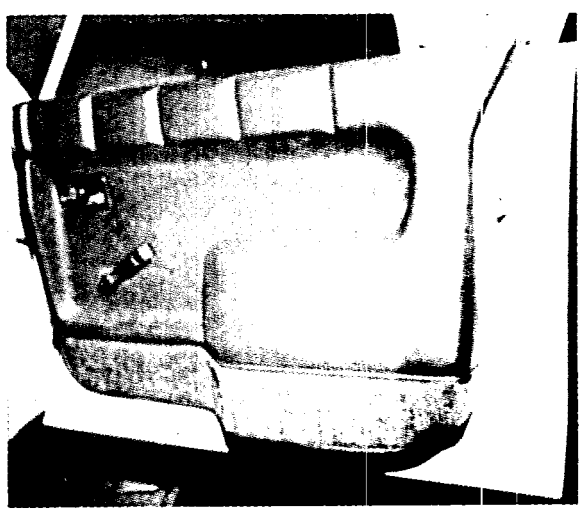


Figure 21. BRV front door panel.

- at the hip level to protect the pelvis.

This padding has been developed through many calculations studying the influence of wall displacement (intrusion) and tests to adjust stiffness; it is made from semi-compact foam.

Test results (fig. 22-23-24)

A synthetic test was carried out with a vehicle whose side-collision areas were modified to correspond with the definition of the research vehicle.

The front part of the striking vehicle represented the average present day agressiveness of vehicles in the French car-population.

Protection criteria	Results
Head	40 g
Thorax	29 g
Pelvis	50 g
Head-trunk angle	15°

The collision was carried out in a manner such that the centreline of the striking vehicle was in line with the centreline of the driver's hip with both vehicles being perpendicular to each other.

For an average vehicle acceleration of 20-25 g, the results are as follows:

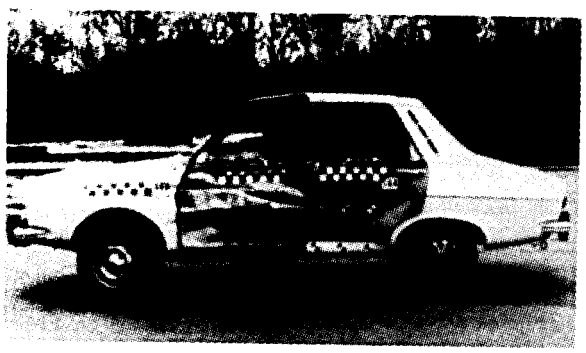


Figure 22. Dummy vehicle (Renault 12 based) side collision – striking Renault 16 – 50 km/h.

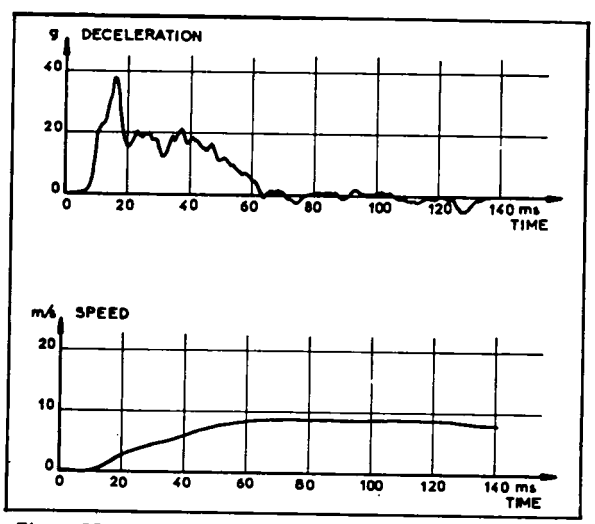


Figure 23. Test results; measured on opposite side of impact.

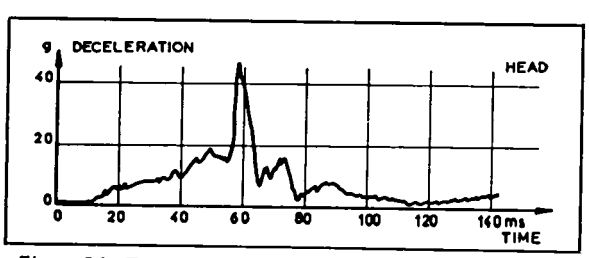


Figure 24. Test results; occupant.

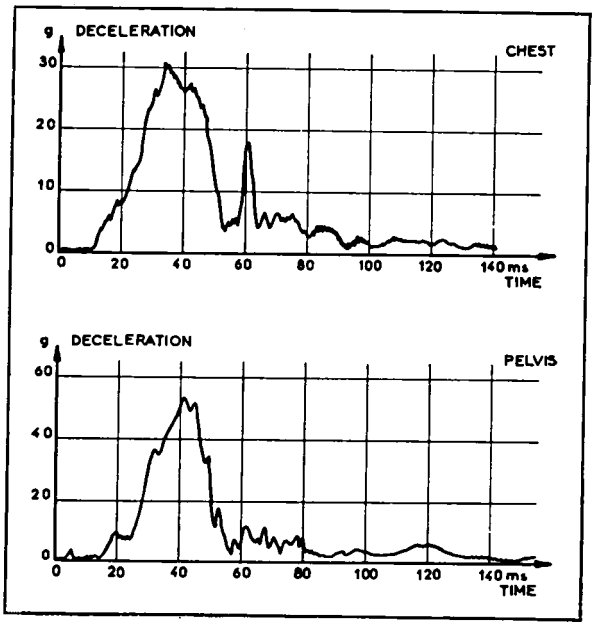


Figure 24. (Continued)

ROLL-OVER

Recall of statistics and accidentology

This represents 7 % of vehicles involved in accidents. The breakdown on people involved is as follows:

Category	Number	% of the category as a whole all types of accident together	% of those involved as a whole in this type of collision
Killed	975	11.3	3.1
Seriously injured	5 927	10.9	18.7
Slightly injured	11 516	7.5	36.4
Uninjured ..	13 234	5	41.8

Two significant numbers are to be noted in the Peugeot-Renault sample:

- 40 % of those ejected are killed;
- 1.1 % of those not ejected are killed.

A typical, average accident can be represented by this case (fig. 25).

In conclusion, contrary to what was believed for a long time, the criterion to be taken into consideration is non-ejection.

There seems to be no direct relationship between crushing of the roof and the state of the occupants. An American enquiry has shown that a relationship between the state of the vehicle and occupants starts when roof is crushed in more than 600 mm.

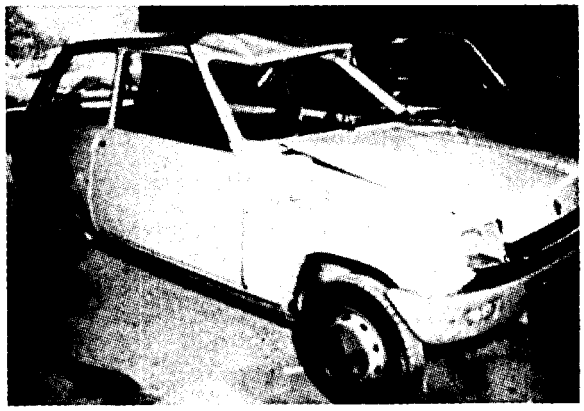


Figure 25. Typical roll over accident (taken from real accident investigation).

However, in a research vehicle, it seemed interesting to place the accent on structural strength in roll-overs.

Choosing performances

TEST TYPES AND TEST SPEEDS

Dynamic roll-over of at least 720° (2 turns) which means a vehicle speed of approximately 50 km/hr.

The vehicle is inclined, free, on a trolley catapulted at 50 km/hr and abruptly stopped.

Inquiries having shown that roll-overs along a longitudinal centre-line were not the most severe, the vehicle is placed in such a way that its longitudinal centre-line forms an angle of 45° with the trolley centre line, and the rear higher than the front (fig. 26).

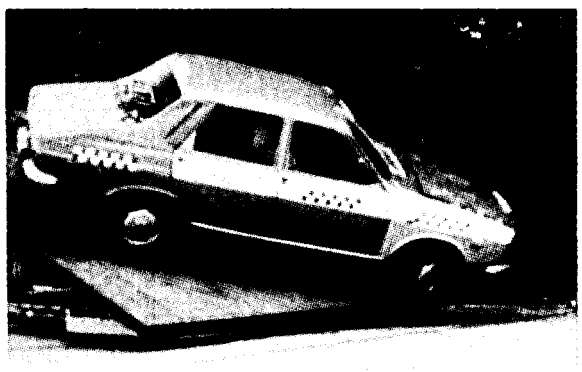


Figure 26. Dynamic roll over; test procedure.

This severe test has the disadvantage of being difficult to reproduce.

CRITERIA FOR EVALUATING

- No ejection, even partial, of dummies belted into front seats;
- doors must not open during test;
- At least one door must open after test;
- Dummies must be able to be removed without difficult.

Technology (fig. 27)

The arrangements made for the front and side collisions constitute a solid basis.

The remainder was carried out using a double roof and a cross-member between the middle pillars.

The double roof, besides its strength function ,acts as padding and avoids any head contact with agressive metal protrusions.

The efficiency of the whole appears to be better than what would be neccessary and sufficient.

Test results

The tests carried out show that for a dynamic roll-over at 50 km/hr. with two complete rotations of the vehicle, the arrangements that have been made fully satisfy the aims.

REAR COLLISION

Recall of statistics and accidentology

Represents 7.9 % of vehicles involved in accidents.

BREAKDOWN ON PEOPLE INVOLVED

Category	Number	% of the category as a whole all types of vehicle together	% of those involved as a whole in rear collisions
Killed	198	2.3	0.48
Seriously injured	1 848	3.4	4.5
Slightly injured	10,903	7.1	26.8
Uninjured ..	27,791	10.5	68.2

BREAKDOWN AS A FUNCTION OF THE OBSTACLE

Type of obstacle	% of cases	% of killed
Car	79.5	Too few to have a valid breakdown
Utility vehicle	15.5	
Fixed and rigid .... obstacle	3.4	
Others	1.6	

The typical accident is represented by this case (fig. 28):

The conclusion is that the statistical weight of this type of accident is very small: consequently, it would seem that no particular attention should be given to it.

However, as it is a question of finding solutions so as to analyse constraints and costs, a test on collision between vehicles will be carried out.

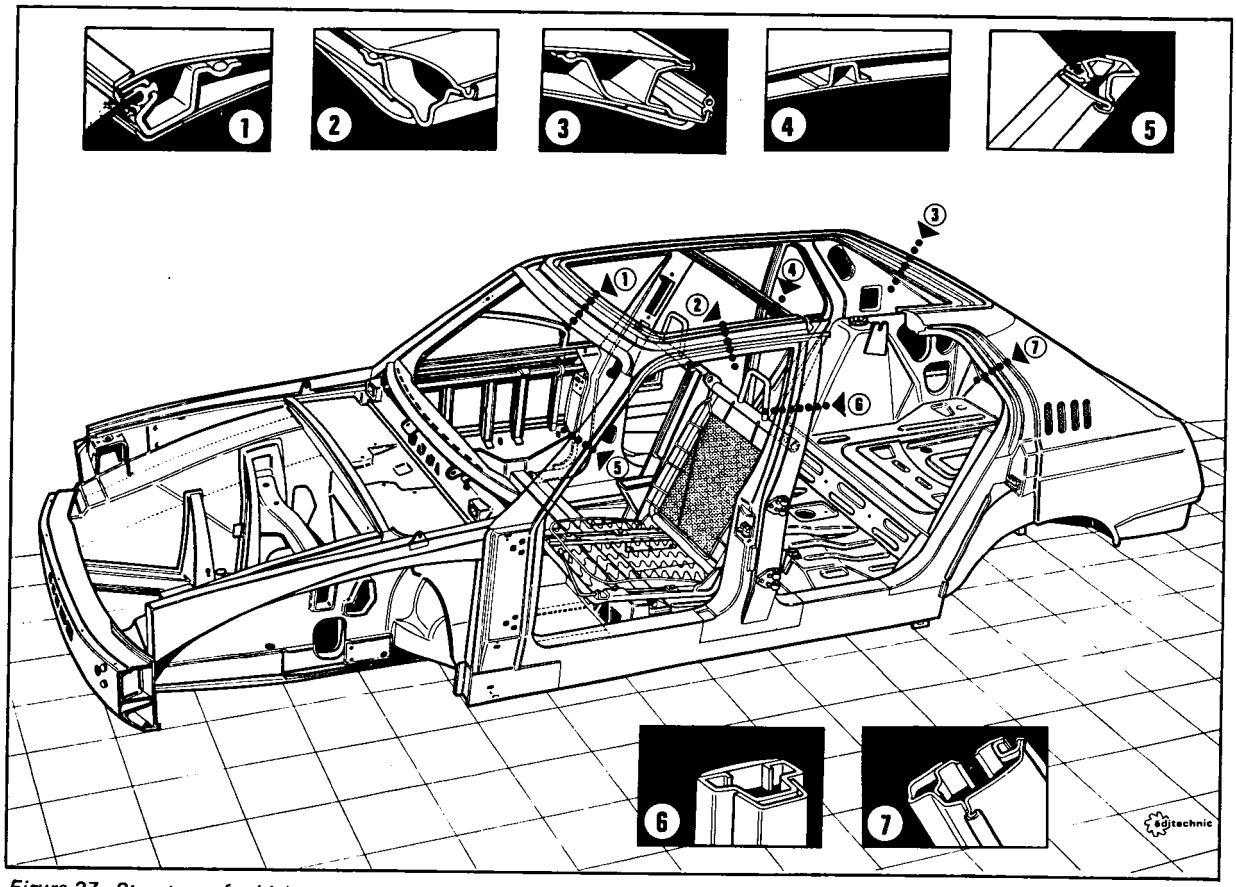


Figure 27. Structure of vehicle; roll over.

Figure 28. Typical rear collision accident (taken from real accident investigation).

Choosing performances

TYPES OF TEST AND TEST SPEEDS

Rear collisions happening mainly with another car, this is the type of collision that has been retained; the choice of speed (60 km/hr.) was made with a view to obtaining a good performance, more to check the advantages of making a technological choice than as a function of a rational cost/efficiency relationship.

A test with an indeformable, mobile barrier is more reproducible and sufficient to check on the protection afforded the occupants by the vehicle for this type of accident.

CRITERIA FOR EVALUATING

- doors not opening during the collision;
- doors able to be opened after the collision (at least one);
- no petrol leakage.

Technologies

STRUCTURE (fig. 29)

Choosing a 5-door saloon, type Renault 16, leads to very strong body sides.

The rear pillar, which is built into the body side in the form of a very strong, hollow section, plays an efficient role in the passenger compartment strength.

The petrol tank, which is positioned under the rear seats floor, is not accessible for the collision violences that we know of today.

But, taking into account the fact that accidentology does not state that one must go this far, and the constraints that this solution imposes on the vehicle, to obtain a cost/efficiency ratio it would seem preferable to adopt other solutions for avoiding petrol leakage.

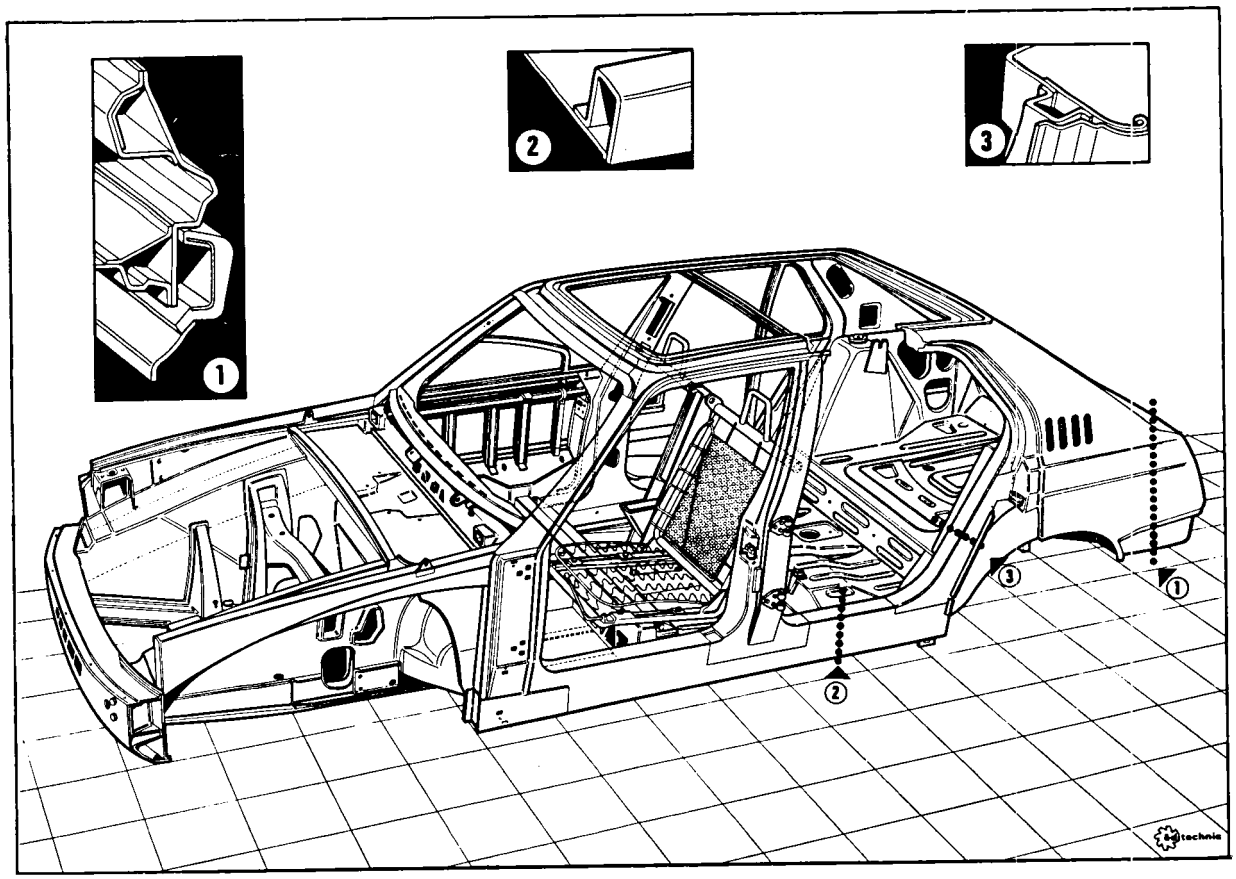


Figure 29. Structure of vehicle; rear impact.

A cross member for the re-inforced rear apron has been specified to support the rear shield, which is of identical design to the front one.

INTERIOR FITTING

Fixed head rest for front seat and removable head rest for rear seat.

Test results (fig. 30)

The specified aims have been reached:
- no petrol leakage;
- doors remained closed during the collision;
- at least one door able to be opened after the collision.

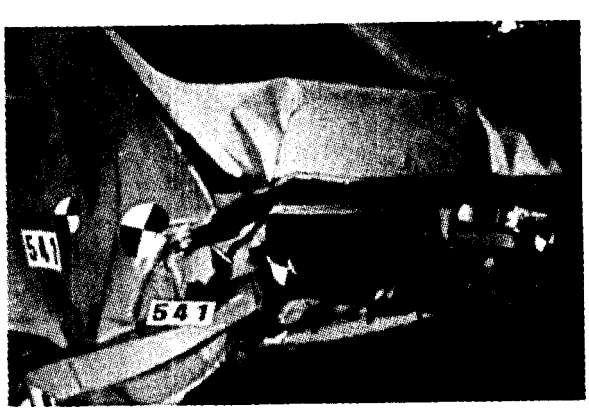


Figure 30. Dummy vehicle (Renault 16 based) striking Renault 12 – 60 km/h.

PEDESTRIAN PROTECTION

Recall of statistics and accidentology

Vehicle/pedestrian collisions represent 7 % of all those involved and concern 20 % of those killed.

This represents approximately 3,200 people per annum (in 1972).

Studying cases of accidents has shown that a vehicle/pedestrian collision can be broken down into three phases:
- vehicle contacts lower part of body;
- head-vehicle contacts;
- contact body + head-ground.

Systematic analysis of the determining parameters has been undertaken within the A.T.P. government contracts.

Choosing performances

TYPE OF TEST AND TEST SPEED

Research work, as much biomechanical, theoretical with mathematic models, as practical in the framework of a vast test campagne, has resulted in relatively clear conclusions being formulated but which remain, however, to be ratified.

Contacts with the front end of the vehicle should be situated firstly quite low so as to be below the knee joint of small people and then above the knee of tall people on each side of a less injuring area (fig. 31).

All these areas should also integrate the children's protection parameter.

The first contacts should have regard for the average rupture tolerances for the lower members and pelvis for adults, the thorax and the head for children.

The areas that may then be contacted with the adult head are to be equipped; but these areas are not independant of the technological choices made for designing the front surface of the vehicle, for they determine the cinematics of the pedestrian during the collision.

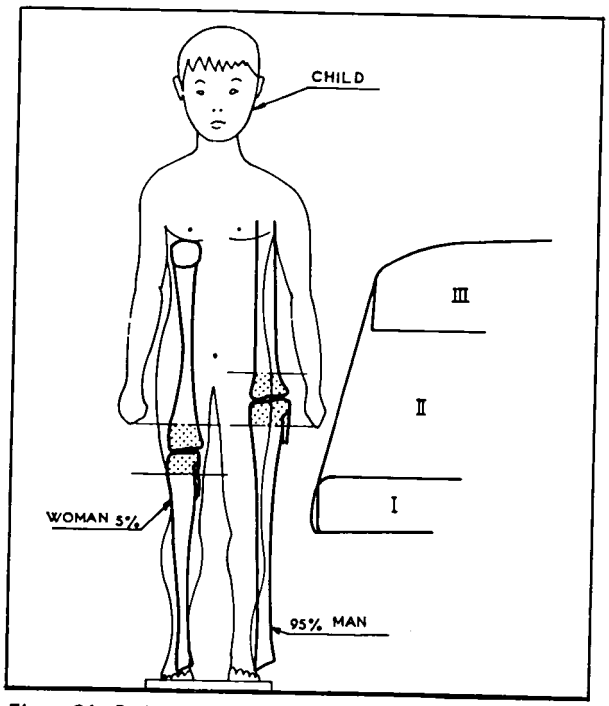


Figure 31. Pedestrian protection.

The aim is to lower as far as can be done the physical parameter values measured on pedestrian dummies during a collision at 32 km/hr., with the vehicle, which is braking, striking the side of the dummy which is placed upright on the ground on the trajectory.

CRITERIA FOR EVALUATING

The method adopted for the occupants is kept for the pedestrians. However, particular parameters will have to be specified for leg protection and the dummies used (which are presently identical to those for the occupants) and will have to be adapted to these types of tests.

Technology

Research work in this field is very much in the foreground. Many solutions are being studied whose industrial feasability has not yet been proved, or whose cost is not yet satisfactory; it is therefore too early to come to a conclusion.

RENAULT's research vehicle has been equipped on the vehicle front cross-member with a covering of polyurethane foam, thus ensuring lower protection (fig. 32).

Protection at average height is ensured by the bonnet front being made of plastic which, used to support the headlights and constitute the front part of the front wings, becomes a veritable, indeformable front section.

This unit is therefore particularly connected with styling, and it is in this field that numerous technologies are still in competition.

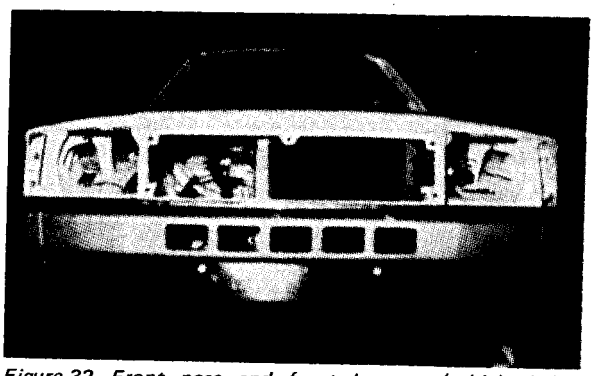


Figure 32. Front nose and front bumper (vehicle being assembled).

Test results

The many tests that have been carried out have allowed an opinion to be formed on the values that may be attained and which correspond with acceptable bio-mechanic limits.

- force developed on the leg in the order of 7,500 N;
- force developed on the pelvis in the order of 15,000 N;
- head deceleration limited to 80 g/3 in s.

COMPATIBILITY

Taking its mechanical and physical meaning in a wider sense, compatibility may be considered as the aptitude of two vehicles to ensure survival, and even the absence of injury, for all the occupants involved in a collision.

In the word "compatibility", it is understood that, in the case of a collision, the units brought together *all* take part in dissipating energy.

Whatever the nature of the obstacle, the result of the collision for the occupants of the vehicle under consideration will always depend on two groups of connected parameters:

a) circumstances of the collision: speed variation, total deformation distance and law of deceleration (levels and time):

b) the behaviour of the structure whether modifying or not the occupant environment and the behaviour of the restraining devices.

The reasoning is, for the moment, limited to front and side collisions.

Physical definition

Given two theoretical vehicles M1 and M2 at speeds V1 and V2; mutual collision will bring about for both of

them a speed variation of Δ V1 and Δ V2, with a stopping distance 11 and 12.

Suppose that vehicle V1 is the more deformable.

A limit exists for which Δ V1 maxi. associated with *11 and 12* maxi. gives an average γ 1 maxi. such that the passenger compartment of vehicle V1 not being deformed by more than 11 maxi., the restraining device installed will allow the protecting criteria to be satisfied.

The maximum closing speed VR = V1 + V2 which corresponds with this total value is the maximum compatibility speed for the two vehicles in question.

Definition of incompatibility

This is the case of a collision in which, the collision circumstances being within the limits of compatibility, occupant protection is not ensured.

This definition is obviously only valid in as much as the vehicle is in good condition and the restraining devices are operational.

In practice, there are two sorts of reasons for occupant protection not being ensured for a collision speed lower than the maximum speed:

- going beyond 11 thus modifying the environment of the restraining device for average γ 1 lower than average γ 1 maxi;
- going beyond average γ 1 maxi. for 11 lower than 11 maxi.

This reasoning is obviously identical for vehicle V2.

Aims

Making vehicles compatible is therefore going to consist in specifying limits for the three following variables:

a) speed variation (Δ V), which is going to depend on the vehicle weight ratio and the statistical risk of collision for a given weight ratio;

b) the deceleration level at which this speed variation took place and whose limit will depend on the behaviour of the restraining device;

c) the available deformation length of vehicles which depends on the design of the structure, the type of vehicle, and the vehicle dimensions.

To satisfy the problems brought up in questions *b* and *c*, the work that has been carried out for protecting in different cases of collision has provided answers.

Specifying the maximum speed variation has not yet been entirely finished.

A thorough analysis of road reality has provided the first element of an answer, and calculating with a computer has allowed the probability that the mass ratio of two vehicles in collision is lower than a given value, to be estimated using the distribution of vehicle masses in a given population as a basis.

It can also be seen that it is not necessary to require performances of extreme classes of vehicles in the car population that are as high as the difference in masses could lead one to believe in the first instance.

Measuring method

The aim being to define a new minimum compatibility and not an "aggressiveness index", present research work, which is far from being finished, is taking the direction of a test with a deformable, mobile barrier.

After many other unfruitful tentatives, this test seemed a good means for taking account of the problems of relationship between Δ V, γ and structural resistance, contrary to the orthogonal collision with a fixed and rigid obstacle which makes γ at a given Δ V more severe.

Technology

Analysing the structure of the front part has already given the main part of the direction taken in matters of compatibility:

- a strong, lower, front cross-member distributing the stresses to the structure as a whole in frontal collisions, and acting mainly on the sub-structure of vehicles that are collided with from the side;
- upper member dipping down to the front end so as not to convey too much stress to the upper level;
- limiting crushing forces on structure.

Results

It is presently only possible to proceed by comparison.

With relation to a vehicle presently being produced and with average aggressiveness:

- the lower side stresses are increased by 50%;
- the upper side stresses are decreased by 30%;
- the weight effects due to the engine on vehicle centre-line are decreased by 60%.

The tendencies are therefore good, but it is not yet possible to say whether the result is satisfactory.

LIGHT COLLISION PROTECTION

Although it is not a safety requirement, limited protection imperatives have been decided upon for the vehicle in light collisions:

- for the front and rear, no damage up to 8 km/hr relative speed with different types of existing vehicles.
- for the sides, protection for urban "contacts"; in this field, it is practically impossible to define exact aims, and efficiency will be the consequence of those technological solutions that have been retained.

This concern is not new and some models in the RENAULT range supply answers to the front and rear problems.

But although many, more effective technical solutions exist or are on the point of culminating, their high cost and the problem of the cost-efficiency relationship is highlighted.

Studies that have been carried out in different countries have not yet furnished clear answers to this question.

On the research vehicle, protection is provided by polyurethane foam padding built into the front and rear structure cross-members and the body sides.

Studies are being continued, for the cost and the weight of these solutions have proved to be high.

STYLING

For the esthetic elaboration of this vehicle, we have endeavoured to state the organic elements of safety, without concealing or amplifying them, by considering them not as extra constraints for styling design, but as a new source of inspiration.

We wished to show that peripheric protection–pronounced step on the surround–the pronounced set of the windows for the side protection of heads, the significant front overhang with built-in soft nose, large glazed surfaces, signalling and lighting devices that are rationally laid out, combined with facility of access to the passenger compartment and the rear boot which, furthermore, characterizes our models, detracts in no way from the equilibrium of the lines and shapes.

The smooth expanses which present no dangerous protuberances, are sprayed with nacreous yellow paint which has been specially designed for visibility at night and in fog by constantly supplying sharp contrast with the ground.

On the inside, we have shown that safety and room are not antinomic.

An anti-glare dash-board, which is greatly cut away on the passenger side, and covered with a matt and dark material, padded door panels and steering wheel, anatomical seats whose upper back is entirely built in with the structural cross-member between the middle pillars, make the passenger compartment particularly attractive and comfortable. It makes one "feel safe".

CONCLUSIONS

Although analysis and research work is far from being finished, what has already been done supplies several conclusions.

The analytical method has proved itself and it is important to persevere in this direction.

Technical collaboration in research work, whether public or private, national or international, has allowed knowledge to be furthered in an important and non-selective way.

Priority orders have now been drawn up thus giving a direction to research efforts.

The most critical problem remains the industrial developing of occupant restraining devices which are valid for and usable by all automobile users; this means intensifying biomechanical research work to define the aims and technological research work for finding rational solutions.

The "controlled" strength of the front and rear parts of structures can now pass to the industrial stage, in so far as the regulations do not require performances that are needlessly excessive and are not closely related to road reality.

It is the problem of side collision protection which will remain the most difficult to resolve, and which will lead to the heaviest, most costly and constraining solutions for the automobile, depending on the performances required.

Progress remains to be made in the field of compatibility of vehicles between themselves.

The difficulty is characterizing it and specifying a means for evaluating it.

It is not unreasonable to hope for an improvement of pedestrian protection up to collision speeds which are not absurd for urban traffic.

However, the two latter points, compatibility and protection of pedestrians, which means redesigning the structure and the architecture of vehicles, will not be able to find a short-term industrial solution.

Finally, the importance of the economic weight of decisions modifying the automobile makes thorough, systematic and international studies indispensable.

Making a complete research vehicle will allow the weight and cost of a relatively high but homogeneous level of safety to be taken stock of on the different points, depending on their importance.

The analysis which could be made in terms of cost-efficiency as the accidentology and bio-mechanical studies prove themselves and the corresponding technologies will be deeply delved into, may be able to serve as a basis for choosing solutions to be adopted in the cars of tomorrow to make them safer, whilst losing nothing of their qualities and leaving them at a price that is accessible to everybody.

BIBLIOGRAPHY

1. Distribution and gravity of collisions as a function of the damaged part of the vehicle and the obstacle hit, Claude BERLIOZ, ONSER, Report on the Second International Technical Conference on E.S.V. 1971.
2. Work Program of the Peugeot-Renault Partnership, Philippe VENTRE, RENAULT, Report on the Second International Technical Conference on E.S.V. 1971.
3. First conclusions from analysis of highway crashes, Dr. Claude TARRIERE, Laboratory of Physiology and Biokinetics of Peugeot-Renault Association, Report on the Second International Technical Conference on E.S.V. 1971.
4. Enquête médicale et technique sur les accidents de la route, F. HARTMANN, Association Peugeot-Renault S.I.A. nº 12, 1972.
5. De la réalité des blessures aux critères pris en compte dans les tests de sécurité, Dr. Claude TARRIERE, Association Peugeot-Renault S.I.A. nº 12, 1972.
6. Efficiency of the 3 point belt in real accidents, Dr Claude TARRIERE, Association Peugeot-Renault, Report on the fourth International Technical Conference on E.S.V. 1973.
7. Proposition d'une méthode d'analyse et de classification des sévérités de collision en accidents reels, P. VENTRE et J. PROVENSAL, RENAULT, International Conference on the Biokinetics of Impacts, Amsterdam 1973.
8. Rating accident severities of occupants, Claude PROST-DAME, RENAULT, Report on the Fourth International Conference on E.S.V. 1973.
9. Méchanisme du choc, Théorie du deuxième choc, Philippe VENTRE, RENAULT, S.I.A. nº 12, 1972.
10. Comparison of the aggressiveness of different vehicles and the safety they afford, Claude BERLIOZ, ONSER, Report on the Third International Technical Conference on E.S.V. 1972.
11. Homogeneous Safety amid Heterogeneous car population, P. VENTRE, RENAULT, Report on the Third International Technical Conference on E.S.V. 1972.
12. Proposals for test evaluation of compatibility between very different passenger cars, P. VENTRE, J.C. RULLIER, Report on the fourth International Conference on E.S.V. 1973.
13. French developments in original deployable systems with reference to safety belts, TARRIERE, MAURON, VENTRE, Association Peugeot-Renault, 2nd International Conference on Passive Restraints 1972, S.A.E. 720410.
14. Simulation bio-mecanique des membres inférieurs, G. STCHERBATCHEFF, A. FAYON, C. TARRIERE, Association Peugeot-Renault, International Conference on the Biokinetics of Impacts, Amsterdam 1973.
15. Proposal for a Protection Criterion as regards abdominal internal organs, C. TARRIERE, A.A.A.M. Conference, Oklahoma 1973.
16. Les moyens d'essais de collision à la Regie Nationale des Usines Renault, P. VENTRE, S.I.A. nº 5, mai 1974.

MAURICE CLAVEL
General Delegate
Anonymous Society of Citroën Automobiles

Introduction to CITROËN report

Since it participates actively and steadily to International Conferences on Experimental Safety Vehicles, CITROËN Corporation has never failed to support, in a context of research of optimum safety, the survival of the small vehicle (sub-compact), basis of mass motorization and of democratic diffusion of motor car.

For this purpose and since 1971, our Company has been regularly reporting, in WASHINGTON and in KYOTO, the progression of its test and research program, carried out with the agreement and support of the French Government, according to the program it established three years ago. This program, entitled "Experimental Safety Sub-System" (E.S.S.S.), is based, - I remind of it, - on the study of sub-assemblies and partial structures of a vehicle of a maximum weight of 1,500 lbs, and of the restraint systems which may be fitted to it.

After having investigated the first part of the problem brought up by the structure of such a vehicle (sub-compact), on the occasion of the 4 th. International E.S.V. Conference in KYOTO, in the month of March 1973, CITROËN has laid down the general outline of its basic research work and the rational solutions it attained, and which can be required from such a vehicle, as regards frontal and side impact performance.

During the same time, CITROËN determined limits to this performance, together with configurations, and alterations to structure which could be kept with a view of optimum cost/efficiency.

This investigation was carried out in co-operation with Cornell Laboratory and with the use of test devices and biokinetic and statistical data currently available.

Of course, data and issues are still uncompleted, but the Research Departments of European manufacturers meeting in the Committee of Common Market Automobile Constructors have undertaken to study them thoroughly and systematically.

At the end of this study, and particularly as regards the orthogonal impact against a fixed barrier of infinite rigidity, probably poorly representative of actual accidents, but much more severe for the occupants, we have concluded, reporting it in KYOTO, that for a small sub-compact vehicle of maximum weight of 1,500 pounds, with 2 embarked dummies, it was risky to require an impact speed above 60 km/h without going beyond the absolute limits of energy absorbing capabilities of a vehicle of inevitably shorter length (approx. : 3,60 m).

So, we feel to-day some concern in front of the latest proposal of amendment to N.H.T.S.A. Standard n°208, which resumes a priori performances put forward, 4 years ago, by the Federal Administration into its initial E.S.V. programme concerning a 4,000 lbs vehicle, and speaks again of a 50 M/h test against a fixed barrier.

Such a scheme would condemn European type cars, that means compact and sub-compact vehicles according to the American terminology.

Therefore, we hope that the Federal Administration, will come back to the conclusions drawn by CITROËN, as well as by other European and Japanese manufacturers, as regards optimum energy absorption in the event of an impact, for a vehicle of sub-compact type, at the very moment when the severe gas shortage suffered by the Western world, unquestionably confirms the interest we have always taken in the small car, which is from now on booming in the U.S. home market.

As regards us, this investigation concerning small vehicle structure being completed, the topic and first results of the second stage of our research work, regarding the Restraint Systems, will be set forth to you by means of a film, during the 10 minutes we have left.

RESTRAINT SYSTEMS IMPROVEMENT

(Pelvic Restraint Device)

INTRODUCTION

– PRINCIPLES OF OUR RESEARCH WORK AS REGARDS OCCUPANT PROTECTION DURING COLLISIONS AND VEHICLE ACCIDENTS.

Schematically, we may admit that during impacts the human occupant of a motor car undergoes harmful effects of several classes :

– Physio-pathological effects due to decelerations

- which may be different from one individual to another,
- vascular (hydraulic) and nervous answers appear only after some time, and it seems that their effects will be all the less detrimental as they last a shorter time, and will happen at the very beginning of the impact.

– Impacts, and occupant stoppage and restraint loads

- **The unrestrained occupant** will accomplish his "deceleration" against the walls and elements of the passenger compartiment.
  These facts have been the first to be studied in BIOKINETICS.
  The deceleration is then performed in an ultra-short time, and on an extremely small distance.

- **The restrained occupant** will accomplish his deceleration against the restraint system, which implies, according to the design of the latter, more or less considerable stresses that we ought to reduce to a minimum.

– Movements resulting from vehicle stopping during impact

- These shiftings, concerning the whole of the body relatively to the passenger compartment, and also the various segments and organs of the human body, in relation to one another.
- Making allowance for these facts, CITROËN research work has been directed up to 1973 towards :
  - Improvements in structure designs allowing :
    - to obtain a better dispersion of energy,
    - a modulation of the deceleration curve,
    - a control of vehicle warpings,
    - to maintain, after impact, a residual space sufficient to allow the restraint system develop its action, according to anticipations, and with the highest efficiency.

We have reported these investigations and their results during the E.S.V. (Experimental Safety Vehicle) International Technical Conferences of **Washington** in 1972, and of **Kyoto** in 1973.

– Development and achievement of a "UNITY MAN/SEAT/RESTRAINT/STRUCTURE" which is more particularly the purpose of this report.

Safety does not seem as attainable without injury to the occupant, by improvement of only one element of the system, but by an harmonious and complementary grouping of the possibilities offered by every one of its components.

It seemed to us essential to :

– Select bearing points on the human body :
  * to maintain vulnerable segments with maximum efficiency,
  * by use of bearing areas providing maximum strength.

– to distribute and "modulate" loads and stresses according to bearing areas anato-physiological possibilities, and according to their own properties.

LIMITS OF THE STUDY.

– The studied systems are intended to equip a vehicle of approx. 1,500 pounds (i.e. 750-800 kg), unladen, for this type of vehicle :

  – represents the wide-spread "EUROPEAN" vehicle, the importance of which has been thoroughly established on the economical ground by the late gas shortage,

  – sets particulary critical problems owing to :
    * its restricted weight,
    * passenger compartment and over-all dimensions.

  – allows to apply the solutions which have been acknowledged satisfactory to heavier and more important vehicles.

1 – OCCUPANTS :

We consider that the sizes and weights of the users of these restraint systems have, as lower limits, those of the 5 th Percentile adult female, and, as upper limits, those of the 95 th Percentile adult male, corresponding to mini and maxi sizes of the grown-up population.

By priority we have chosen to turn our attention to the driver's protection.

2 – VEHICLE :

The systems studied being intended for a vehicle of 750-800 kg, the peculiar characteristics of this type of vehicle will be taken into consideration :

– available space to insure occupant's restraint and protection,
– occupant's sitting in normal driving position,

- value and conditions of vehicle crushing,
- law of deceleration belonging to the vehicle itself.

3 – TESTING SPEEDS AND CONDITIONS :

The conditions will be those of a vehicle impacting a fixed and rigid obstacle normal to its trajectory.

The protection of vehicle occupants using the studied restraint systems will be ensured up to a speed of 60 km/h.

When the studies will be achieved, checking will be carried out at a speed of 70 km/h.

4 – PROTECTION CRITERIA EMPLOYED :

The efficiency of the restraint systems will be appreciated relatively to protection criteria, measurements being done using anthropomorphic test devices.

- **Head**
  At present, no valid criteria.
- **Thorax**
  Acceleration shall not exceed 60 g's, except for cumulative duration ≤ 3 ms.
- **Abdomen**
  No upward movement of the pelvic restraint element above iliac crests.
- **Pelvis**
  The load applied to pelvis through the pelvic element will be at most equal to 1,800 daN (~ 1840 kgf).
- **Femur**
  The compressive stress measured in each femur will be at most equal to 670 daN (~ 700 kgf).

5 – ECONOMIC INCIDENCE.

Despite the need of new designs, the cost of envisaged restraint devices must remain consistent with the price of a "Small Vehicle".

ACHIEVEMENT OF THE UNITY MAN/SEAT/RESTRAINT/STRUCTURE.

1 – GENERAL :

Every one of the elements of this unity must be improved according to its peculiar imperatives, but always keeping in mind the general context of the whole.

2 – THE HUMAN BODY :

- must be "Nidied" at buttocks level, without compression of femoro-popliteal vessels of the back part of the thighs,
- must be "Propped" in an anatomical manner, as high as possible at the level of the trunk, especially at the level of the dorsal upper region, and of lumbo-sacral hinge, particularly vulnerable areas.
- must not submarine by shifting forward and downward beneath the pelvic element of the restraint system.
  And this, for the various weights, sizes, and volumes of occupants.

3 – THE SEAT.

During long its design was mainly intended to improve-**comfort;** new concept, without too much impairing it, must be directed relatively to SAFETY FACTOR.

The seat is a paramount element :

- it must permit to place the occupant in a **predetermined** position allowing for the efficiency of the restraint system,
- the subject must remain in this position and, to do so, must feel "comfortable" only in it.

  More stiffness in the seat, alterations of upholstery will be needed, but they must not impair comfort in normal use.

4 – THE RESTRAINT SYSTEM.

- So called 3-point belts have constituted a definite improvement and have revealed an unquestionable relative efficiency, perfectly evidenced by actual accident investigations.
- Good in their principles, it is necessary to effect betterments and improvements in their achievement, in order to increase their efficiency, as well as to decrease the eventuality of sequels, injurious to the user.

- It is absolutely necessary to take into account very accurately :
  - the possibilities and high tolerances of some areas of the body, as the pelvis;
  - the imperatives and sensitiveness of some areas, as the thorax

  and to redifine, according to these principles :
  - the bearing areas, on the human body, of the restraint system components,
  - the characteristics and geometry of the various components of the system

  relatively to :
  - human anatomical data (contours, strength),
  - physiological data of the living subject,
  - loads to be anticipated and sensitiveness of the envisaged bearing areas,
  - findings established during the enquiries made on actual accidents, and of the experimental reproduction of same types of accidents,
  - results from studies carried out according to known data.
    - on mathematical models
    - on anthropomorphic test devices
    - on improved test devices, with a view to achieve a better representativity of the possibilities of physiological "Balance" of the human body of a living subject.

When it has been possible to define and achieve a restraint system meeting these new imperatives of safety and efficiency, it becomes appropriate, according to the possibilities and

without impairing the efficiency, to promote the use of the device by simplifying its positioning, reducing uneasiness, and by enabling ready adjustment, whatever the sizes and morphologies of the users, men or women.

DEFINITION OF THORACIC AND PELVIC RESTRAINT SYSTEM.

The characteristics of the two constituents will be studied successively.

1 – DEFINITION OF PELVIC RESTRAINT.

1.1 – Biokinetic considerations.

- Let us resume that a lap belt alone is rightly considered as poorly efficient, and this all the more as the position departs from the thighs and becomes more distinctly abdominal. An abdominal upward movement is prone to entail severe injuries to abdominal organs, this upward movement being still favoured by the submarining of the subject, at the beginning of the occupant's drift resulting from impact.

- The pelvis constitutes a noteworthy anatomical restraint possibility, if **bearing** is taken **and remains below the upper and anterior iliac crest.**
- On the physiological ground, the pelvis proves particularly strong if the direction of load is correctly applied. Moreover, the anatomical elements are favourable to a bearing on the upper part of the thighs, in order to insure "iliac locking during impact".

Figure 1

Figure 2

- Our research work is aimed at determining an angle of seat base/back, promoting the achievement of an angle thigh/pelvis allowing the positioning of the pelvic restraint according to an optimum direction for adjustment and keying below the upper and anterior iliac-crest, which protrusion is more or less pronounced according to the morphology of the flabby parts of the subject.
  This optimum direction must be determined by means of accurate studies, so that it can be achieved for all the subjects.

- Correctly positioned, this pelvic restraint tightly applied, is **perfectly tolerated without discomfort.**

  Let us point out that a true pelvic restraint must be able to provide the restraint of the greater part of the human trunk, and all the more alleviate the load applied to the thorax level.

1.2 – Experimental definition.

- On a small sampling of subjects, taking into account the above-mentioned considerations, we position the pelvic restraint. The selection of these subjects allows us to define the limits into which the pelvis belt must be positioned. At first, we shall verify how varies the direction of the element and its position with regard to the iliac crests, then, we will try to find which must be, with regard to a given seat, the mean direction and the position of the anchorages providing the best compromise for the population we have taken into consideration.

  The subject having taken his place, in driving position, bare hands on the steering wheel, and feet on the pedals, heels resting on the floor, we locate the iliac crests. The pelvic restraint is then correctly adjusted, measurements and photographs are taken.

  The resulting negatives are superimposed by squaring the seat marks. We notice little scattering in the positions of iliac crests (x = 35 mm; z = 35 mm), this scattering is still more reduced if we consider the upper point of the pelvic restraint.

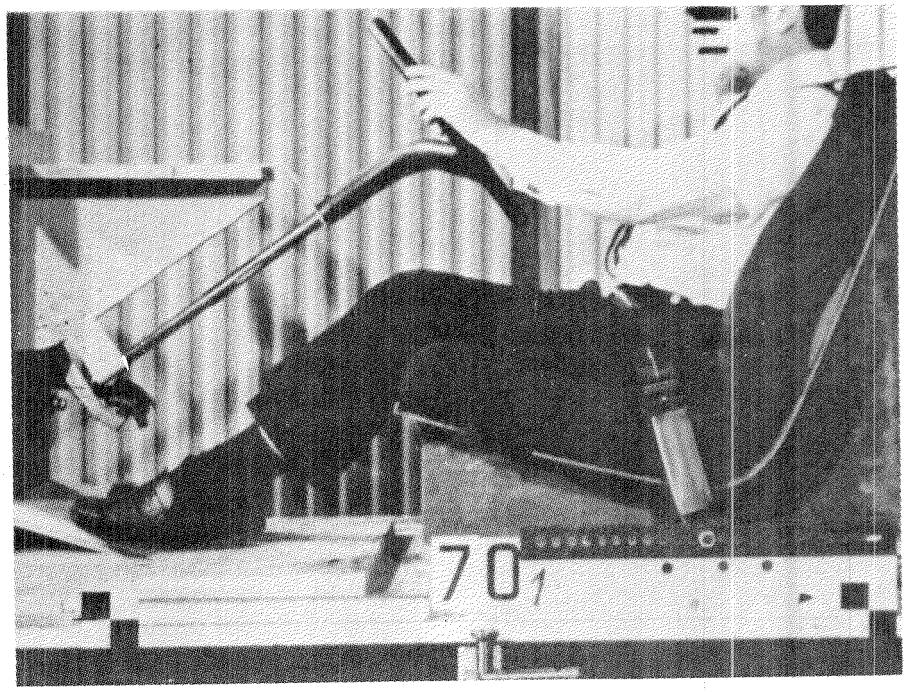


Figure 3

The last part of this work of definition consists in verifying that the previously defined position of the anchorage points, enables a right positioning of the pelvic restraint on all the subjects whose size and weight are included within the limits established by ourselves.

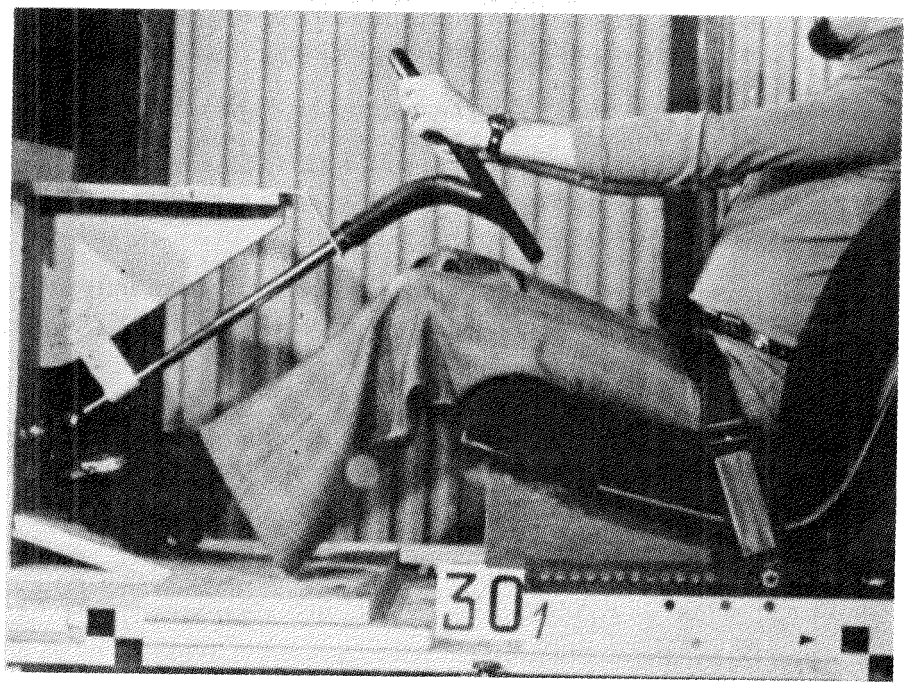


Figure 4

A right positioning of the pelvic element implies that its fixed points are able to follow seat adjustment.

2 – DEFINITION OF THE THORACIC RESTRAINT.

2.1 – Biokinetic considerations.

– If the pelvic restraint ensures the support of the lower part of the subject (pelvis and lower part of the trunk), as well as of the lower limbs, linked with it by the two coxal-femoral articulations (hips), the upper part of the trunk, neck, and head, must be restrained by means of an other device.

"3 - point" belt current arrangement consists of a strap describing a laying out in the form of a shoulder-belt **of very varying patterns,** according to vehicles and this, mostly in sympathy with the position of anchorages and, especially, of the upper point.

Scapular belt (collar-bones and shoulder-blades), and thorax, are elements allowing to apply a restraint system on condition, however, of taking into account their anatomo-physiological possibilities of strength and tolerance.

The damping possibilities to the action of loads generated by the restraint action are neither the alone fact of the resistance of an independant "costal grill", nor of the further action of muscles, but of a balancing hydro-pneumatic thoraco-abdominal system used by man, in normal conditions, for the trunk and spinal column statics, in pure vertical statics and during efforts.

Loads correctly applied on the thorax, on a good surface and during a sufficiently long time, will be perfectly tolerated by the living subject, whereas the test device will be inert, and the cadaver, unbalanced, will suffer more or less severe injuries.

It is therefore advisable to determine an anatomo-physiological laying out, making the best use of thorax damping capabilities, and to make the bearing areas and the value of the applied load consistent with the LIVING subject tolerance.

In actual accident, thoracic "flaps" (simultaneous double fracture of several adjoining ribs) are mainly the fact of impacts against hard elements of small areas. Likewise, severe intrathoracic organic injuries are the fact of **PRE-EXISTING** pathological injuries at the level of these organs. Thoracic devices could be different according to the speeds to be "covered", and the type of vehicle.

A good thorax restraint must provide an improved restraint for head and neck.

2.2 – Experimental definition.

– The position of the upper anchorage point must result from a compromise between the possibilities of securing to the structure the thoracic restraint element, and of the positions of bearings on the body of various subjects.

The position we have adopted for the point takes into account manufacturing imperatives of a sub-compact 4-door sedan, the anchorage point being located on B-Post.

A correct applying of the webbing on the lower part of the thorax could be obtained by bringing together the thoracic restraint lower fixed point and the pelvic restraint internal anchorage point.

TESTS

1 – PURPOSE

– **We have deemed, for the reasons previously treated that it was important to avoid submarining and the upward movement of the pelvic element into the flabby parts of abdomen** (also see Annex).

A series of manipulations on models lead us to redefine the position of the lower anchorage points of restraint elements.

The position of these points being in connection with the travelling of the seat, we have reached a good positioning of the pelvic element, whatever the size of the subject.

– **Our dynamical tests have been carried out in order to verify that this new webbing geometry, in conjunction with a rectified seat, really enabled to suppress submarining.**

These tests also supply our Computing Department with a number of useful coefficients for a better development of its mathematical models of restraint systems.

The optimization of belts, or the grouping of other restraint devices will be carried out with the help of mathematical simulations.

2 – TEST CONDITIONS.

– On a 4-wheel sledge, fitted with tires, is placed a part of vehicle passenger compartment.

This part of vehicle compartment enables to verify the possible contacts of the test device with vehicle inside, and the securing of belt and seat.

The test device is placed at the driver's position.

The sledge is propelled at a speed between 48 km/h and 50 km/h.

It crushes a collection of calibrated tubes in order to reproduce the law of deceleration of a vehicle in frontal orthogonal impact against a rigid barrier.

We will successively make use of a 50 th percentile male test device (SIERRA 292.1050) and a 5 th percentile female test device (ALDERSON VIP 5 F).

The 5 th percentile female device corresponds to the user most liable, a priori, to submarining when the position of the anchorage points is invariable on the car body.

On test devices, deceleration measurements are taken in the thorax.

In some tests the loads in belt strands were measured by means of sensors placed between the anchorage points and the webbing fixing shackles.

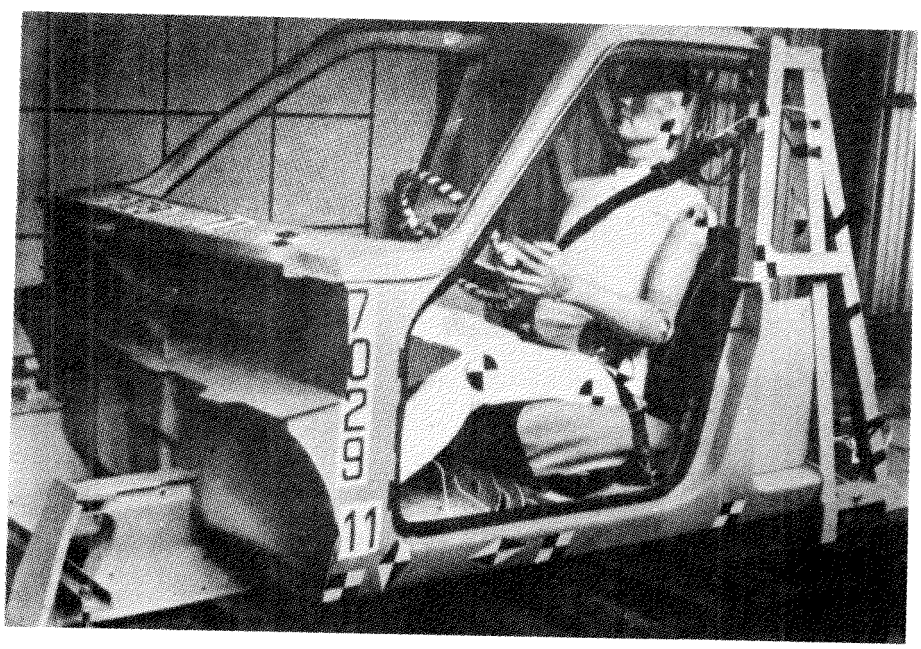


Figure 5

The absence of upward movement of the pelvic restraint above the iliac crests is verified by means of opto-electronical devices fixed on the iliac crests of test device pelvis.

The possible passage of the pelvic element in front of a photodiode releases these devices. The ambient lighting provides for the necessary source of luminous energy.

The test devices are set on actual seats which have been the subject of special investigation. Owing to the first importance of the share of the seat in the restraint of its occupant, this seat will be changed at every test.

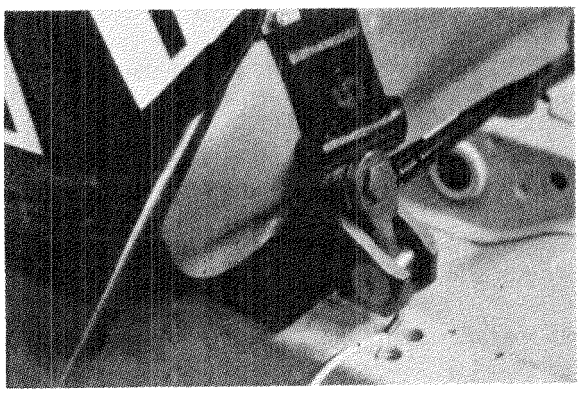


Figure 6

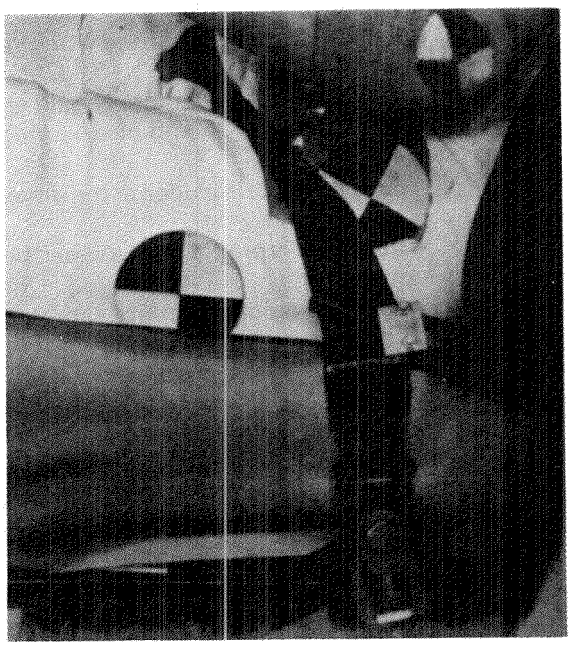


Figure 7

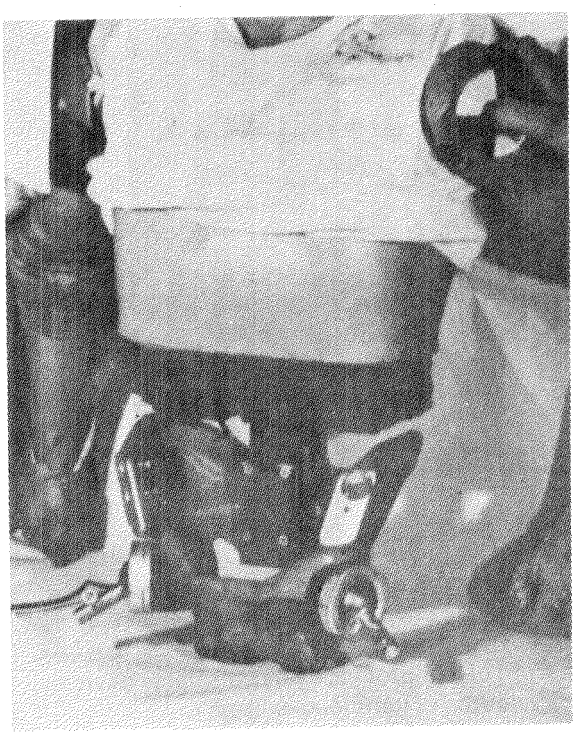


Figure 8

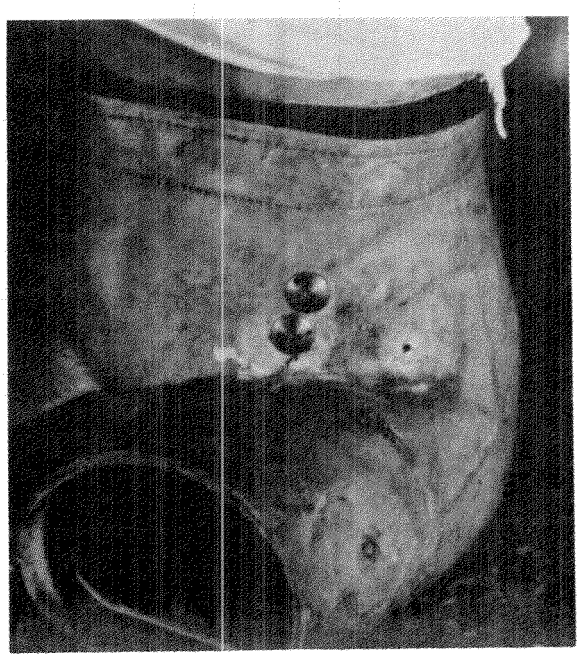


Figure 9

The test devices are restrained by standart type belts, with 3 anchorage points, the webbing having a width of 50 mm and an elongation of 12 % under 1,000 daN. These belts admit only one adjustment, the locking latch being free on the webbing constituting the pelvic and thoracic restraints.

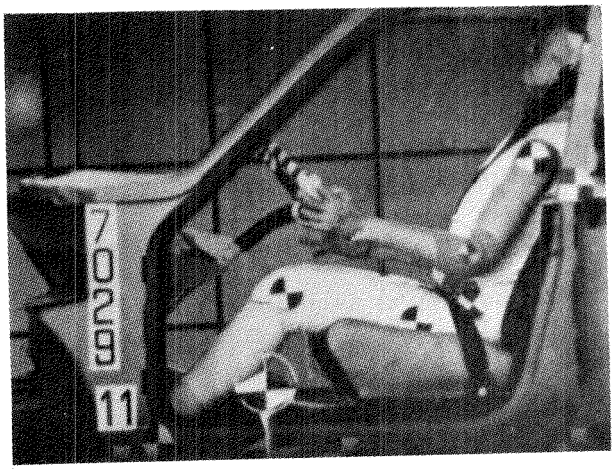
The tests are carried out according to the following configurations :

1°) **50 th. percentile male test device, belt lower anchorages fastened to the seat.**
2°) **50 th. percentile male test device, belt lower anchorages fixed to the vehicle body.**
3°) **5 th. percentile female test device, belt lower anchorages fastened to the seat.**
4°) **5 th. percentile female test device, belt lower anchorages fixed to the vehicle body.**

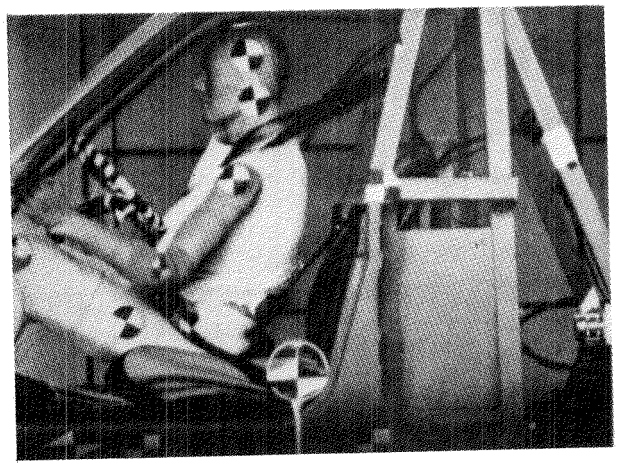
DYNAMIC TESTS ON CATAPULT WITH 50th. PERCENTILE MALE "SIERRA" TEST DEVICE

	TEST N° 1	TEST N° 2
Position of belt anchorages	**Seat**	**Car Body**
Initial speed (V_o)	51.3 km/h	50.3 km/h
Sledge stopping distance	52.5 cm	51 cm
Sledge mean deceleration	19.6 g	19.3 g
Maximum acceleration in the thorax ($\ddot{x}$)..	50 g	58 g
Corresponding time	t_o+57 ms	t_o+64 ms
Maximum load in the plane of the pelvic belt	1,715 daN	1,680 daN
Maximum displacement of hip point ...	22 cm	20 cm
Belt upward movement above iliac crests.	**NO**	**YES**

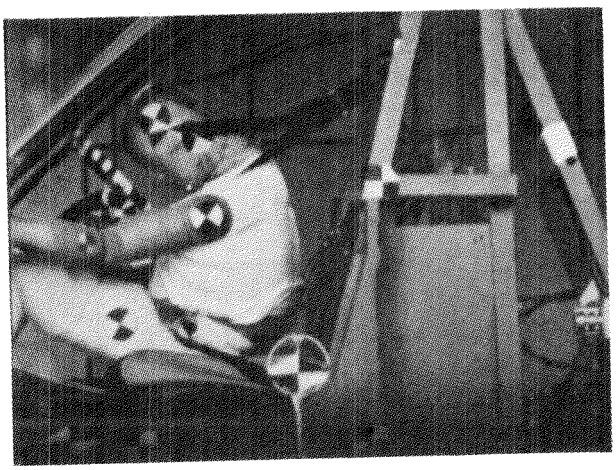
TEST N°1



t_o

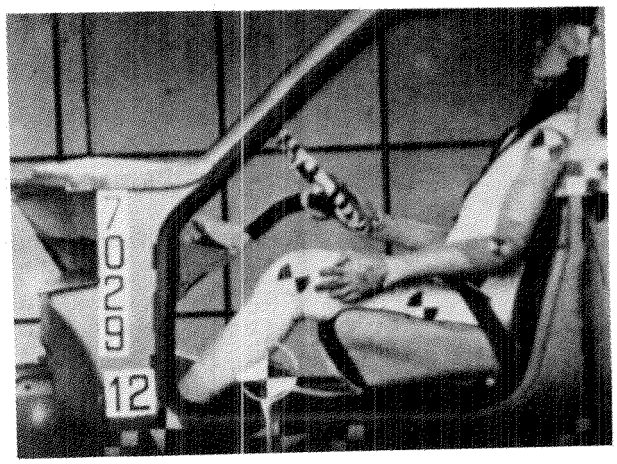


t_o + 60 ms

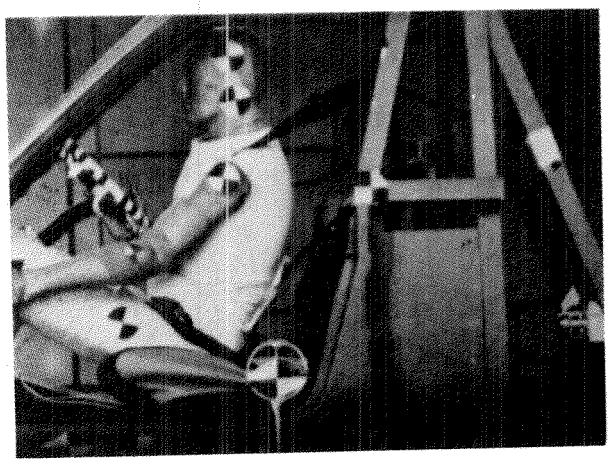


t_o + 80 ms

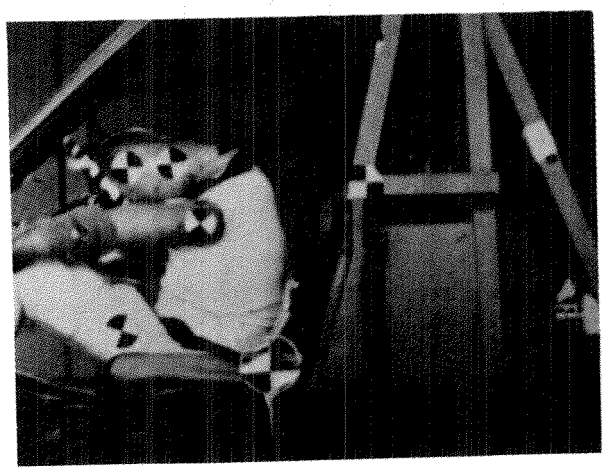
TEST N°2



t_o



t_o + 60 ms

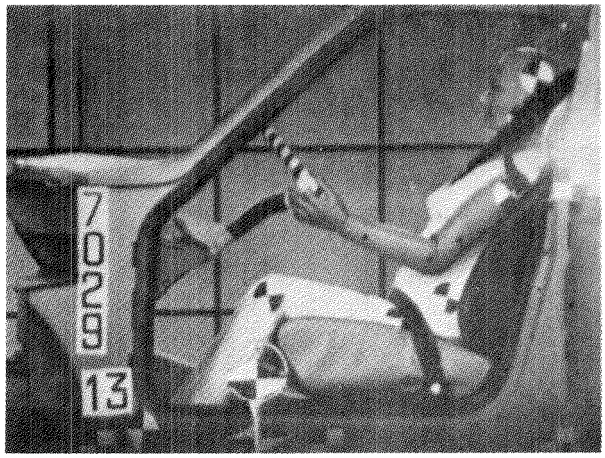


t_o + 90 ms

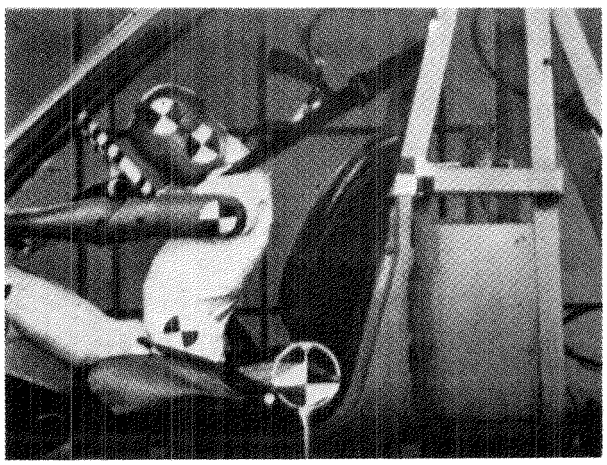
DYNAMIC TESTS ON CATAPULT WITH 5th. PERCENTILE FEMALE "ALDERSON" TEST DEVICE

	TEST N° 3	TEST N° 4
Position of belt anchorages	**Seat**	**Car body**
Initial speed (V_o)	50 km/h	52.2 km/h
Sledge stopping distance	47 cm	52.5 cm
Sledge mean deceleration	20.9 g	21.2 g
Maximum acceleration in the thorax ($\ddot{x}$)	45 g	46 g
Corresponding time	to+69 ms	to+75 ms
Maximum displacement of hip point ...	15 m	22 cm
Belt upward movement above iliac crests	**NO**	**YES**

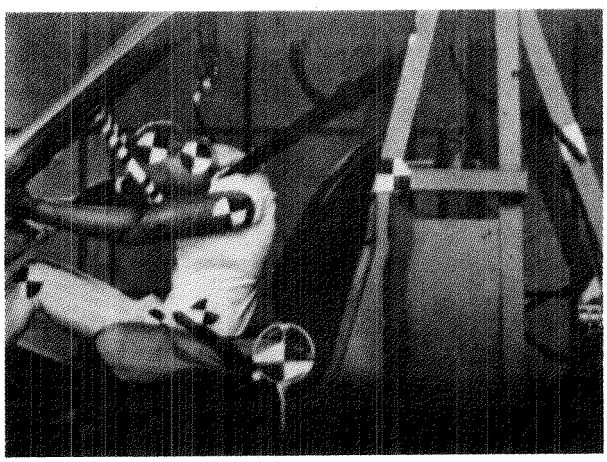
TEST N° 3



t_o

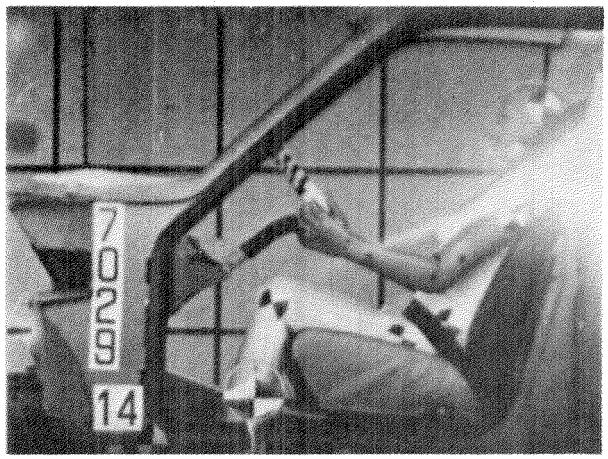


t_o + 70 ms

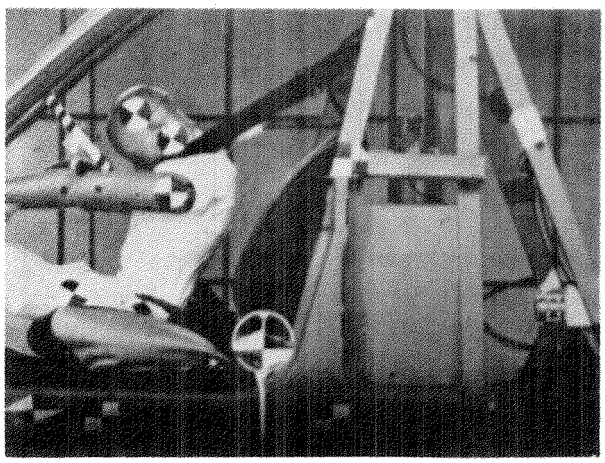


t_o + 90 ms

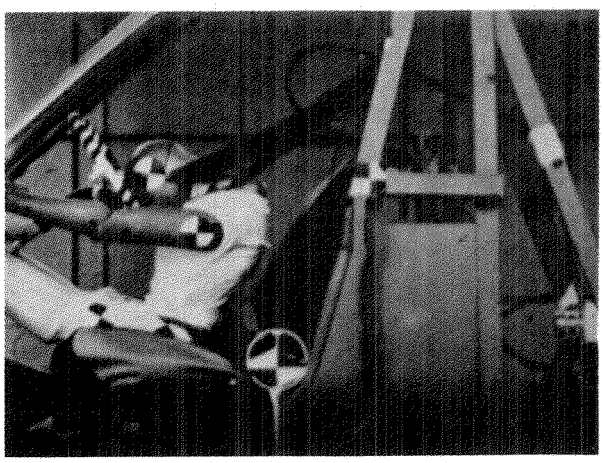
TEST N° 4



t_o



t_o + 70 ms



t_o + 90ms

CONCLUSION.

- Our tests reveal that submarining, particularly very important for a 5 th percentile adult female, could be suppressed thanks to a pratically vertical position of the restraint pelvic element.

  In order to maintain this direction of belt and achieve a good applying below the iliac crests, whatever may be the size of the subject, we have found that the position of anchorages, relatively to **hip point,** should be **constant,** therefore that the anchorage points should follow the seat adjustments.

  Fastening the safety belt to the seat is a solution meeting these conditions.

  The progress of our research programme provides for taking up anew the improvements of thoracic restraint, work that we have submitted at the 4 th. International Technical Conference, March 1973, and for combining the two following improvements :
  - pelvic restraint, and
  - thoracic restraint.

  Nevertheless, we are expecting valid protection criteria for the head, and means to check these criteria.

  We are trying to derive the maximum possibilities from these two restraint systems, in order to provide an actual protection to users in the event of impacts similar to those of a small vehicle in frontal orthogonal impact, at 60 km/h, against a rigid and undeformable barrier.

- In this programme, experimental phases (dynamic tests) and mathematical simulation are closely related. This method enables us to reach the aim more rapidly and in a less expensive manner.

ANNEX

COMPARISON BETWEEN VARIOUS FRONTAL IMPACTS UNDERGONE BY "GS" CITROËN VEHICLES.

Our research work relative to the improvement of the pelvic restraint has been due to our first findings in actual accidents.

Indeed, we have mainly observed on victims, restrained by 3-point belts :

- **on the thorax :** ecchymosis without severity at the level of belt bearing,
- **on the abdomen :** lack of urine emissions of variable durations which may show a bruise of the bladder, this one being situated immediately above the pelvis.

In these accidents, thoracic stresses were very low.

Now, similar tests, theoretically equivalent, have provided values which would evidence considerable thoracic injuries according to present estimates.

The annexed tables sum up the work of reproduction of an accident, "GS" CITROËN sedan against a "FORD CAPRI", of which we possess the following elements :

- reference of impact,
- vehicle masses, speeds,
- characteristics of impact,
- description of occupants and restraint systems.

Our Company attaches peculiar importance to these reproductions of actual accidents for which the reports are very comprehensive.

These reproductions will reveal the probable over-estimate of the actual violence of crashes caused by impact experimentations against a rigid and fixed obstacle.

For the reproductions of the accident GS against CAPRI, we have made use of a GS instead of CAPRI, the curbweights, unloaded, of these two vehicles being similar (950 kg).

The relative speed of the two vehicles is approximately 100 km/h.

In the two following tables we enter the results of :

- Accident GS against CAPRI
- Test No 1 GS vehicle against GS with great offset
- Test No 2 GS vehicle against GS with small offset
- Test No 3 GS against rigid barrier at 60° (speed 50 km/h)
- Test No 4 GS against rigid barrier at 90° (speed 51,4 km/h).

In the accident, the driver (alone on board) was restrained by a 3-point belt (standart equipment). In the tests 1-4, we have placed two 50th percentile test devices on the two front seats. These test devices were restrained by 3-point belts.

Table n° 1

– Measurements, recorded after tests, of vehicle typical and permanent sets.

Dim. (mm)	Struck Vehic.	Test No 1		Test No 2		Test n° 3	Test n° 4
		Vehic. N° 1	Vehic. N° 2	Vehic. N° 1	Vehic. N° 2		
A	– 500	– 565	– 565	– 815	– 820	– 485	– 370
B	– 60	– 65	– 55	– 65	– 55	– 125	– 365
C	– 250	– 345	– 275	– 350	– 350	– 245	– 210
D	+ 55	+ 45	+ 35	+ 55	+ 60	+ 20	– 163
E	– 72	– 97	– 72	– 185	– 202	– 104	– 127

E = rearward movement of dashboard at driver's level.

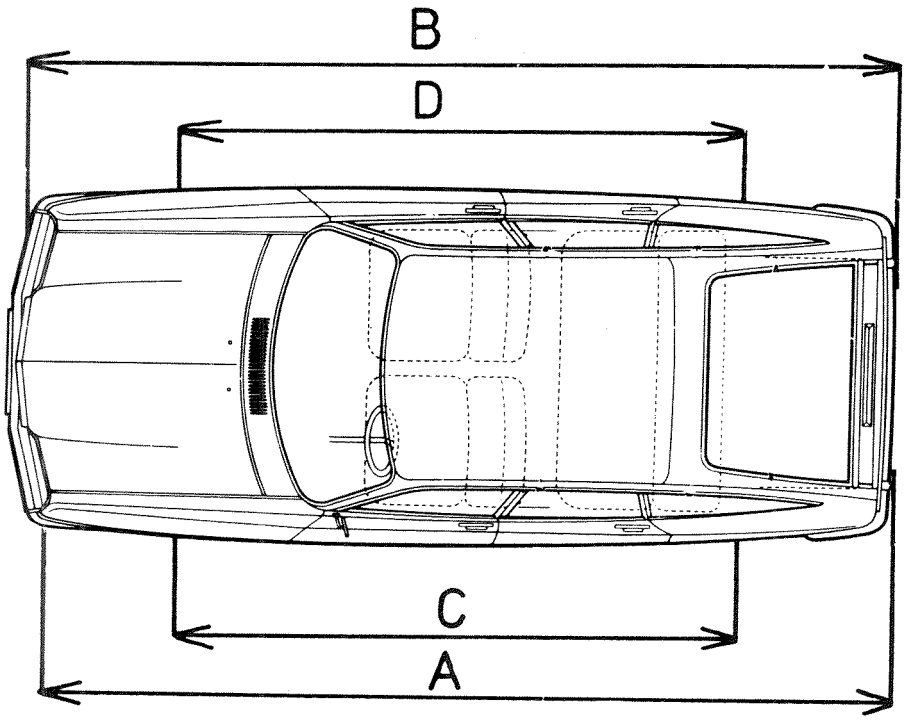


Table N° 2

– Maximum accelerations recorded on the vehicles and embarked test devices during testing, as well as measurements of loads on femur and belts.

	Test n° 1		Test n° 2		Test n° 3		Test n° 4	
	Driver	Passeng.	Driver	Passeng.	Driver	Passeng.	Driver	Passeng.
$\ddot{x}$ Car Body .. L.H.	30		42		48		80	
$\ddot{x}$ Car Body .. R.H.	18		30		50		78	
$\ddot{x}$ Thorax	30	26	40	37	62	40	82	75
$\ddot{x}$ Head	30	26	40	-	-	45	160	–
Femur ..... G	<100	<100	150	200	-	-	160	720
Femur ..... D	<200	<100	850	100	-	-	840	400
F (3-point belt, upper point) ..	425	560	700	720	700	900	800	1000
F (outside straps)	390	355	480	750	550	725	600	725

$\ddot{x}$ = expressed in multiples of g (9.81 $m.s^{-2}$)

F = in decaNewtons

– = no results of measurements

An examination of these tables brings out considerable differences between the results of tests N°1 and N°2 (accident reproductions) and those of tests N°3 and N°4 (impacts against fixed barriers).

These deviations are very important for the values of accelerations measured on the car body, and on the anthropormorphic test devices.

NOTE :

- Without willing to draw a premature conclusion, we think that :
  1°) Even with the undeformable barrier at 60°, the level of protection criteria recorded during the experimentation is **abnormally high.**

  2°) Consequently, and to better represent the reality proceeding from accident investigations, we should :
    - either envisage a **movable deformable barrier,**
    - or decrease the experimentation impact speed against rigid and fixed barrier.

ACTION :

- Multiply the reproductions of actual accidents according to the C.C.M.C. programme.

- Try to find a ratio = $\dfrac{\text{Test speed}}{\text{Reproduction relative speed}}$

- Undertake the study of the characteristics of
  - deformability
  - displacement

    of a movable and deformable barrier.

HELMUT WAGNER
Deputy Assistant Secretary
Federal Ministry of Transport

I have the honor to give, on behalf of the government of the Federal Republic of Germany, a short report on the work achieved in my country in connection with the ESV problems.

On the 23rd of November 1973, the Federal Government published a program on road traffic safety which has been partly based on an evaluation of the previous ESV conferences.

At the same time the efforts concerning technical measures for a reduction of the noise caused by road traffic were increased. Accordingly, the situation is as follows:

- The Federal Government is in favor of further improving the safety of the passenger compartments of motor vehicles. Uniform provisions are to prescribe that the steering column and the steering wheel are to be designed in such a manner that in case of an impact the kinetic energy is reduced in the least harmful way possible. The endeavors made in respect of improving the safety glass are to be supported.
- Moreover, it is to be ensured that the head restraints offered on the market and used in the vehicles can guarantee an adequate protection, by excluding any danger for other occupants of the vehicle or impairing the sight of the driver to the back.
- It is intended to introduce the compulsory utilization of safety belts by the occupants of motor cars effective 1976; this obligation will apply to front seats in passenger cars and vans. In preparation of this regulation, a provision took effect on the 1st of January 1974, according to which the equipment of the front seats of these vehicles with type-approved safety belts and the equipment of the other seats with safety belt anchorages has been made compulsory.
- For the prevention of accidents the Federal Government intends to promote the development of the braking systems, the improvement of the visibility of the driver and provisions for the distance covered by the headlight beams.
- In order to control the noise caused by road traffic the maximum permissible noise level of the vehicles is to be reduced.
- New motor vehicle requirements will be decided by the Federal Government in close cooperation with the international organizations concerned and, above all, with the European communities.
- The Federal Government welcomes the common activities carried out in the ESV conferences. In its view, however, the results of the tests carried out so far have shown that, in the case of a speed of impact of 80 km/h, the goal to protect the occupants of a motor vehicle from serious injuries in the case of a head-on collision can only be attained by means of expenditures which in all probability will meet with strong opposition in Europe. The government of the Federal Republic of Germany therefore suggests we consider a modification of the test program.
- The Federal Government made arrangements for large-scale tests to investigate speed-related problems in road traffic. Part of these tests will be completed by the end of 1975, and the rest will be terminated on the 30th of September 1977.

The studies carried out by the automobile industry of my country resulted in the following:

- The planning and initiation of an economically justifiable development of the safety technology would require further intensive basic research.
- Focal points of future research activities must be seen in cost benefit analyses and in a systematic evaluation of accident statistics.
- In the studies of component research, priority is to be given to bumpers and passive restraint systems.
- Further studies will include approaches to the solution of tire problems.
- The information systems for drivers must be improved.
- Future international safety regulations for motor vehicles should be prepared on the basis of sound and profound technical-scientific and medical findings, in joint cooperation at supranational level.
- Special specifications for a new Experimental Safety Vehicle including the experience gained from the present state of the ESV development,

component research and series production can only be prepared if a satisfactory solution to the problems specified above has been found. A German manufacturer has already started to work on these specification under an NHTSA contract.

Repeated tests carried out on an advanced ESV (speeds of impact over 50 km/h on a rigid barrier) have confirmed that it is not possible to obtain essential protective effects for the occupants of a vehicle without considerably increasing the weight and fuel consumption of the vehicle.

The Association of Liability, Accident and Motor Traffic Insurers has continued its research on accident causes. Its compilation of data concerning accidents involving passenger cars is the largest throughout Europe.

Details on the studies of the automobile industry and of the Association of Liability, Accident and Motor Traffic Insurers will be given in the status reports following, as well as in the seminars.

DR. HANS SCHERENBERG
Corporate Chief Engineer
Member of Executive Board
Daimler-Benz AG

I am pleased to have the opportunity to speak to you for the fourth time on the subject of the Experimental Safety Vehicle program of Daimler-Benz AG.

At previous conferences we have presented three fully functional prototype vehicles, given detailed histories of their respective developments, and data obtained in numerous test series.

The first two vehicles, the ESF 05 and the ESF 13 were designed according to the original specifications established by the U.S. Department of Transportation. With a weight penalty of 50% over the series production base vehicle, the Mercedes-Benz 250, we managed to meet all essential requirements on active as well as passive safety, in particular those requiring occupant protection at a speed of 50 mph in a frontal collision against a rigid barrier.

Like many others, we believe the 50 mph test condition to be unrealistic. Therefore, in the second phase of our ESV project we aimed at the goal of researching occupant protection in the speed range between the 50 mph ESV requirement and the present standard impact speed of 30 mph.

Our intent was to attain sufficient occupant protection at 40 mph impact speed with the application of 3-point safety belts of modern European design. Result of this endeavor was the ESF 22 with a weight penalty of 20% over the base vehicle, a Mercedes-Benz sedan of the S-class. Again, this result was unacceptable for use in production.

In order to define the design consequences of a head-on collision at 40 mph even further we have developed our ESF 24 which we will present at this conference as our final step in this program. In this research vehicle we designed the body structure and the restraint systems solely to meet the conditions of a 40 mph frontal barrier test. The safety performance in the other types of collision – rear impact, side impact, and rollover – correspond to that of production S-class vehicles. The front structure, including the bumper system, had to be extended by 15 cm. From the experience with this research vehicle we would expect a weight penalty of approximately 150-200 kp or 8-11% of the base weight. This weight increase may appear relatively low, but again it is not acceptable, particularly in view of the present concern over energy and raw material supply.

As a result of our experience I believe it is possible and necessary to give a final evaluation of the Experimental Safety Vehicle Program.

We have previously voiced reservations, both in Washington and Kyoto, about the original target of this program and repeat this today. We do admit that vehicles like the ESF 22 and ESF 24, which differ from the original specifications of the Department of Transportation, come closer to realistic solutions of vehicle safety. Also, three years of extensive work have well demonstrated the capabilities of automotive engineering. Many improvements of details were obtained, and we also believe it is meaningful to actually demonstrate such major effort at least once and discuss it on an international level. This program has certainly increased the sum of our knowledge and traffic safety will undoubtedly profit from it.

But, on the other hand, even more valuable is the realization that we still lack important basic knowledge and that a period of intensive basic research should now follow in order to assure that further advancement in safety technology is firmly based on a realistic and economic foundation.

It is our feeling that the Experimental Safety Vehicle program has served its purpose and can be regarded as completed.

Meanwhile, the U.S. Department of Transportation has embarked on a new research safety vehicle program. According to the four-phase timetable a proven concept of a practicable safety passenger car should evolve by 1977/1978, assuming that vehicles of this type will be producible in large numbers during the Eighties and will meet all requisites for traffic and transportation systems of that decade.

For the first time the program calls for cost consciousness in production and economy in vehicle operation as well as recognition of the sociological conditions of the Eighties. The RSV program does not start by listing rather speculative requirements for a pure safety vehicle which are not scientifically founded. Instead it first contains the basis task to define and substantiate the requirements on automotive technology, always bearing in mind the necessary compromise between mobility, safety and environment.

Daimler-Benz A.G. will not be actively engaged in the phase of program definition and performance specification. We do not believe that the prediction of traffic and sociological conditions for the Eighties in the U.S. is the one and only important yardstick, because Europe has also become a part of the world with heavy motorization.

We, therefore, intend first to study future conditions in Europe and in other parts of the world so as to enable establishment of a basis for a worldwide synthesis at a later time. Simultaneously, we shall continue with our traditional, consistent

development of safety elements for our production vehicles. Daimler-Benz AG is and will remain interested in the research safety vehicle program, and might actively cooperate in the later phases. Actually, our work following the ESF 13 until the ESF 24 of today now appears to be in close relationship to the RSV program. We could rely directly on this work as soon as the program targets are clearly established at the end of the definition phase.

At this time these goals have yet to be defined and approved. We are, of course, all the more surprised that the National Highway Traffic Safety Administration opened Docket 74-15 and proposes in Notice 1 a considerable increase of the requirements on occupant protection of passenger cars produced after 1980 to a level of 45 or even 50 mph frontal impact speed against a rigid barrier.

This is proposed:

- Although the rigid barrier impact has been determined to be an unrealistic accident simulation
- Although the necessity for such high impact speeds so far has not been demonstrated but rather has been statistically contradicted
- Although the increased enforcement of speed limits and recommended speeds will further reduce the statistically relevant collision speeds
- Although technical feasibility is not ascertained, even at weight penalties of up to 50%
- Although such a requirement would prevent the necessary compromise between safety, environmental protection, and energy demand

Once again we can only repeat our recommendation to analyze and to apply the available results not only of research, but also of practical experiences, for example by utilizing the protection performance of modern 3-point safety belts through mandatory belt usage. Further important insights should be collected before proposing requirements with such far reaching implications.

Daimler-Benz AG welcomes the beginning dialogue between the U.S. authorities and the European Experimental Vehicle Commission (EEVC) and recommends further mutual exchange on the safety programs in Europe and the U.S. The guiding concept of the EEVC is a practically safe vehicle intended to ascertain reasonable and convenient solutions permitting application in standard production. The EEVC program puts research, which was recognized to be necessary, into a foremost position. On the grounds of thorough statistical analysis of accidents in the European countries, which has already begun, this program will define adequate test procedures. A preliminary list of guidelines for the requirements of vehicle safety is based upon data available today and refrains from exaggerated requirements which are economically unjustified.

We believe that the necessary synthesis of requirements can be based upon this program because, in general terms, safety on the roads in Europe is much the same as that in other parts of the world. For this reason we have never fully understood why safety standards in the U.S. should be different from those in France, Sweden, or the Federal Republic of Germany, for example. We sincerely hope that this unsatisfactory condition will terminate in the near future through a coordinated international effort of governmental authorities and industry resulting in uniform legislation.

EDMUND HELLRIEGEL
Chief Engineer
Ford-Werke Aktiengesellschaft

For many years, the motor vehicle industry has developed and constructed motor vehicles that fulfill the requirements of the safety laws. We have reached a certain degree of perfection in this matter. Therefore, we can predict the behavior of the vehicle to such an extent that adjustment operations can be kept within a reasonable limit.

In our efforts to meet the statutory requirements, we have developed so-called "in-house standards" which we must also satisfy. These standards refine and complement the statutory requirements; for example, the behavior of the steering wheel in the case of frontal impact. The statutory requirement states that the steering wheel must not move to the rear more than 5 inches. The "in-house standards" require 4 inches, since we must take production allowances into account. Thus, while we exceed the statutory requirements in this case, we must also be conscious of the fact that we thereby increase the delay factors that affect the vehicle passengers. We aim for 25-30 g over a maximum of 5 ms. Similar complementary in-house standards are set up for all the statutory requirements.

It is our goal above all to avoid accidents and, insofar as accidents are unavoidable, to reduce the consequences. This would be our goal even if there were no statutory regulations. In the course of our efforts, we have also set ourselves a further limitation: that the dimensions and weight of motor vehicles should not be increased.

Within the framework of the safety program, we are attempting to increase the vehicle impact velocity, in the case of frontal collision, from 30 to 50 mph. Experiments are being performed in order to determine on a scientific level whether this increase is correct or not.

These tasks are subordinate to the goal of protecting human life and reasonably altering existing statutory requirements, insofar as this is necessary.

One other limit value that has been retained unaltered since the origin of safety regulations, also seems worthy of examination. This is the behavior of the steering column in the passenger compartment in the case of frontal collision.

A decisive factor for the survival of the vehicle passengers is that of sufficient survival room, and that of keeping the delay factors within reasonable limits.

It can be assumed with certainty that a relatively young individual with modern safety devices can survive a frontal collision at about 50 kmh if his safety belt is buckled. We also know that the same chances of survival are less for an older person, because the loads effected by the belt are too great. These loads, because of bone brittleness, can lead to internal injuries that are fatal in most cases.

I, therefore, would consider it wise to check the changes in positions of the steering column in frontal collision tests, to increase the distance as much as possible. In this way delay factors can be kept within limits which would afford older persons a genuine chance for survival, without the necessity of increasing the external dimensions of the vehicle.

OPEL'S SAFETY DEVELOPMENT PROGRESS REPORT

F. LOHR
Adam Opel AG

INTRODUCTION

Automotive safety has always been inherent in Opel's vehicle development. For example, in 1935 we first introduced the integrated steel body which already showed energy absorbing characteristics which are typical for modern automobiles, Figure 1.

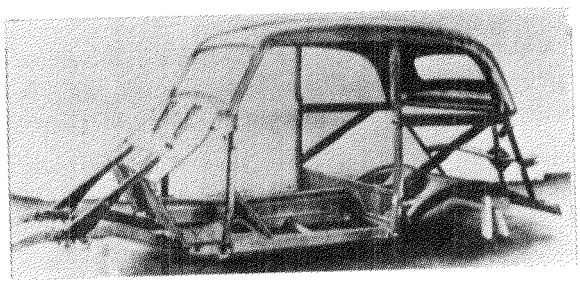


Figure 1

In March of 1971 Opel began to participate in the development of an ESV according to the specifications of the Association of the German Automobile Manufacturers (VAA), Figure 2. During our first development phase we learned very soon that any vehicle which could fulfill these specifications would be unrealistic in an economic sense, and would call for an excessively long and heavy design. Opel's next development phase, started in June of 1971, defined its goal to be the development of an economical, low weight, safety vehicle using high volume production related components, methods and materials.

The impact specifications were based on the findings of our extensive accident research program which had demonstrated that a barrier impact velocity of 40 mph covered the majority of all actual traffic accidents, Figure 3. Higher impact velocities would result in unrealistic concepts since the recent material and energy shortage made significant weight increases even more unacceptable.

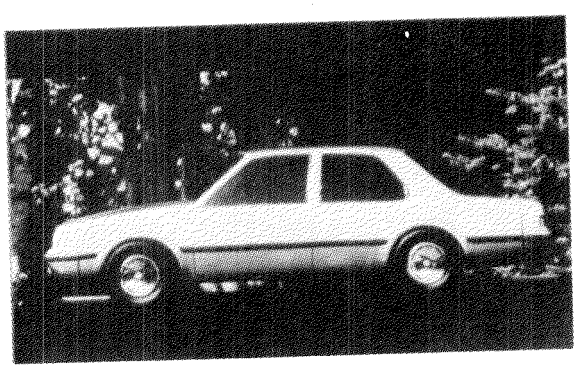


Figure 2

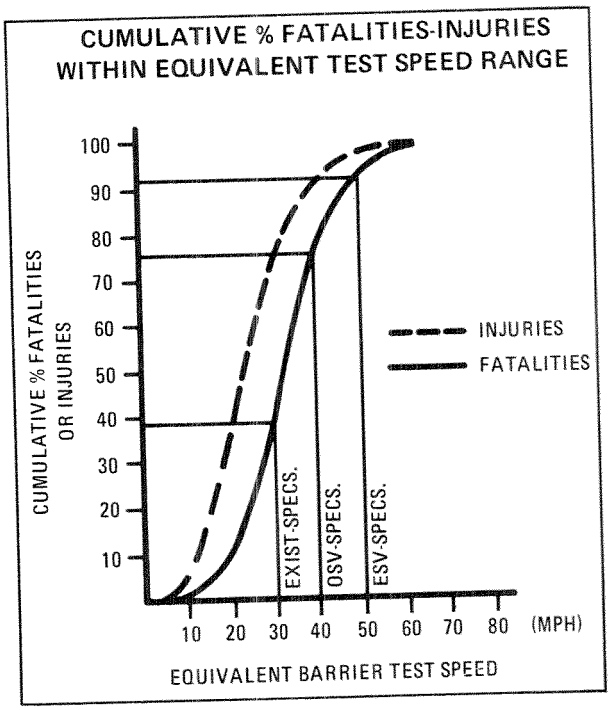


Figure 3

The 1971 development effort reflected the independence and originality of Opel's research with its goal that of producing the Opel Safety Vehicle–OSV. Major findings of this research phase went into the design of the Kadett unveiled in August 1973. With the aid of computers, maximum collision safety parameters for 30 mph front impacts were also obtained for this Opel model and the body was designed accordingly.

Thanks to barrier crash tests and special investigations on a crusher facility which provided impact data for the computer safety program, the new body's impact behavior was checked during initial development phases and systematically brought up to the high engineering standards targeted.

The production model of the Kadett was a realistic choice upon which to base Opel's safety vehicle–the OSV 40–for another important reason: the maximum of good handling characteristics for improving safety by helping to avoid accidents.

Opel production vehicles comply with performance criteria for vehicle handling properties which represent a summary of the proposed safety vehicle requirements, Figure 4. An extensive handling research program on front suspensions, and rigid, independent and De Dion rear axles resulted in production chassis with good handling characteristics;

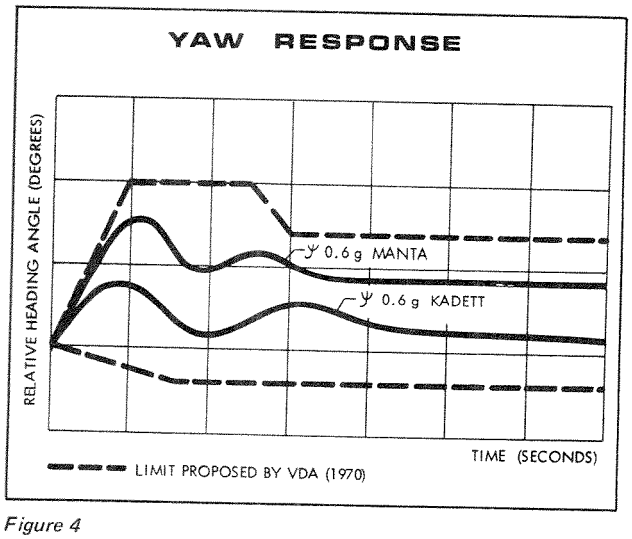


Figure 4

development of a special chassis for the OSV 40 was not necessary. We concentrated, instead, on improving the occupant protection features and, as a base, efficient structural components with controlled crush behavior were developed. Although we tailored the structure for an Opel Kadett, the concept could be applied to any modern passenger car size, Figure 5.

Figure 5

On the occasion of the 5th ESV Conference in London, Opel is displaying the OSV 40, an experimental safety vehicle designed for 40 mph barrier impact speeds and one based on a 4-door Kadett Sedan with 1.2 liter engine, Figure 6.

Figure 6

GENERAL VEHICLE SPECIFICATIONS FOR OSV 40

The general vehicle specifications for OSV 40 reveal these data, Figure 7.

- OSV 40, four door sedan.
- Overall length 4318 mm.
- Overall width 1580 mm.
- Overall height 1370 mm.
- Front tread 1330 mm.
- Rear tread 1301 mm.
- Weight of "body-in-white" 300 kg.
- Curb weight 965 kg.

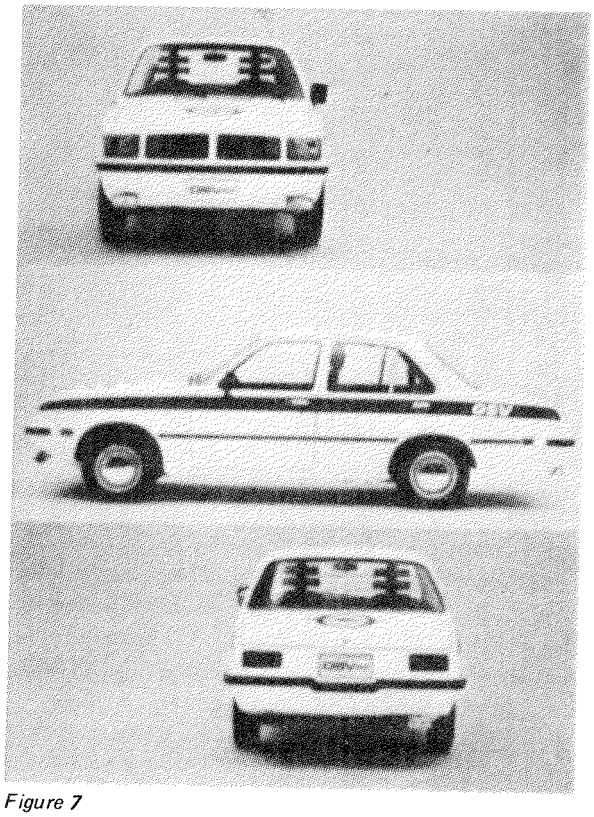


Figure 7

CHASSIS

The essential chassis components were adopted unaltered from the production model Kadett.

Front Wheel Suspension

The wheels are independently suspended on double wish bones and mounted in maintenance free rubber bushings, Figure 8. The pivoting axles of the wish bones are articulated to help absorb the brake torque. The steering knuckles are mounted on the wish bones via two maintenance free ball-and-socket joints. The front axle cross member, which is tilted five degrees to the rear, is joined to the body at four separate points. The coil springs are installed between the lower control arm and the front cross member. The dual-action hydraulic telescopic shock absorber is mounted close to the wheel on the upper control arm for optimum wheel control and comfort. The stabilizer bar controls the roll tendency of the body. Additionally, it assures the vehicle's tendency towards a slight and well-controlled understeer under all load conditions.

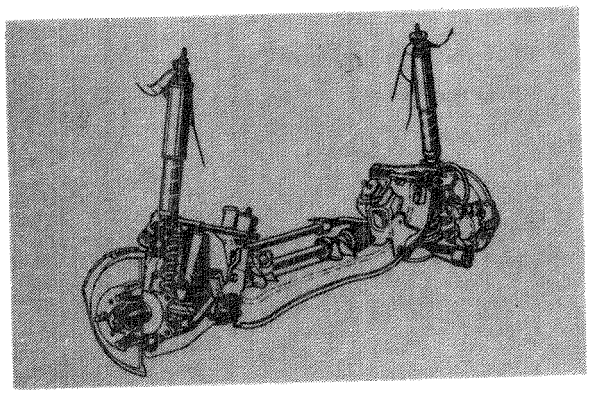


Figure 8

Rear Wheel Suspension

The rear axle is a short torque tube design, Figure 9. The differential housing is extended to the front by a tube which is suspended to the body by elastic rubber mounts. The torque tube transmits acceleration and braking torsion of the axle. The axle is controlled by two longitudinal swing arms and a Panhard rod for lateral control. The progressive coil springs are affixed directly in front of the axle, and the dual-action hydraulic telescopic shock absorbers are set vertically and far outboard for maximum ride comfort. The stabilizer bar is linked at its end with the axle via independent supports and its back is fastened to the body structure by means of two rubber bushings.

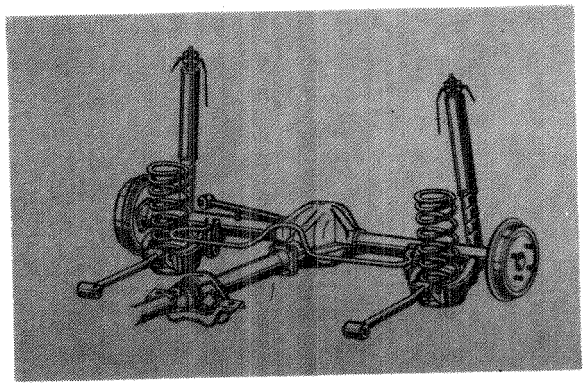


Figure 9

Brakes

The car is equipped with a dual-circuit braking system with booster, front disc brakes and rear drum brakes, Figure 10.

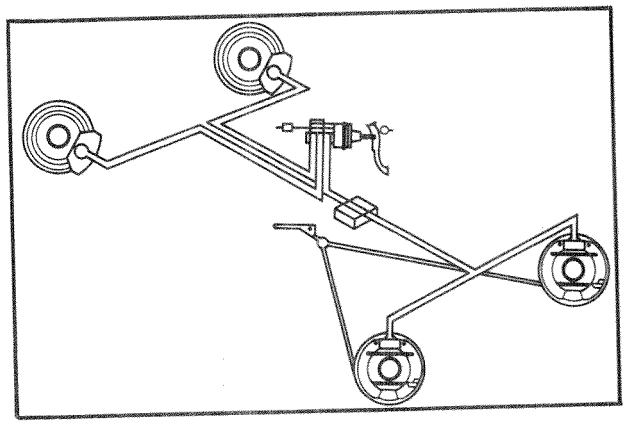


Figure 10

Steering

A service free rack and pinion type steering system is installed in the vehicle. Overall gear ratio is 18.8:1.

Wheels and Tires

Steel belted radials, 155 SR 13, are mounted on 5 J x 13 steel rims.

ENGINE

The engine is of a four cylinder in-line design, 1196 cm^3 displacement, generating 44 kw (65 hp) at 5600 rpm; torque 70 ft/lbs at 3400 rpm; compression ratio 9.2; fuel 98 roz.

BODY

The body structure of the OSV 40 is based on that of the Kadett. All energy-absorbing structural elements are filled with polyurethane hard foam, Figure 11.

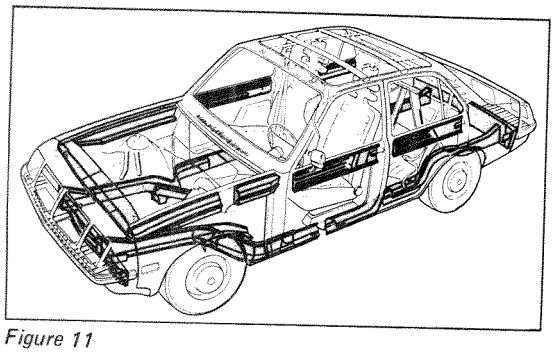


Figure 11

The foam front end structure consists essentially of two upper and two lower foam filled side rails whose foam filling is adapted to the specific profile cross section and material gauge. These profiles crush in a controlled manner at impact speeds above 5 mph. The upper side rails in combination with the lower ones reduce the vehicle's aggressiveness and provide effective protection when hitting a truck. The lower side rails are supported towards the tunnel in the area of the torque box in order to increase the load resistance of this area. The structure described can withstand the impact of an 5 mph bumper test without deformation. Deformation of the front end occurs at higher impact speeds and therewith the energy absorption of the body structure. The transmission mounting is equipped with a shearing mechanism in order to achieve easiest possible deflection of the drive cluster. In this event the energy absorption remains restricted to the controlled deformation of the front end.

The passenger compartment features foam-filled rocker panels to increase compression resistance. High level foam-filled door beams also increase load resistance and additionally offer protection on side impacts. The door beams are set high in order to impede the usual "override" on an impacting vehicle.

All pillars are reinforced in order to hold roof disfiguration to a minimum in event of rollover. Here the laminated safety windshield, bonded directly to the sheet metal, and the roof mounted front seat backs also contribute to roof stability in event of rollover. These features make a rollbar superfluous.

The rear body structure is reinforced with enlarged foam-filled side rails to achieve sufficient stability in 5 mph bumper tests. Further reinforcements were not required. Crash tests have shown that the relatively crushable rear end structure wards off an impacting vehicle with its great deformation capability.

As in the production Kadett, the fuel tank is installed in a safety zone above the rear axle. It is protected from damage by a sheet metal wall between it and the luggage compartment. A fire proof wall between the tank and the passenger compartment is designed to protect the occupants. The fuel tank is of light weight high-molecular polyethylene of great pliancy.

All interior sheet metal surfaces with which occupants could come into contact are padded with a semi-hard polyurethane foam with an average 20 mm thickness. Its effectiveness in reducing deceleration peaks and lacerations has been substantiated in actual tests.

In response to the 40 mph test speed set by Opel, the proven safety steering system of the Kadett was augmented by addition of an extra collapsible portion in the lower steering spindle to prevent intrusion of the steering wheel at the increased impact velocity, Figures 12 and 13. The upper collapsible spindle

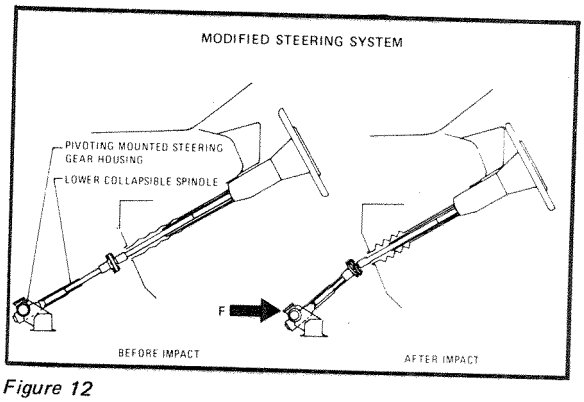


Figure 12

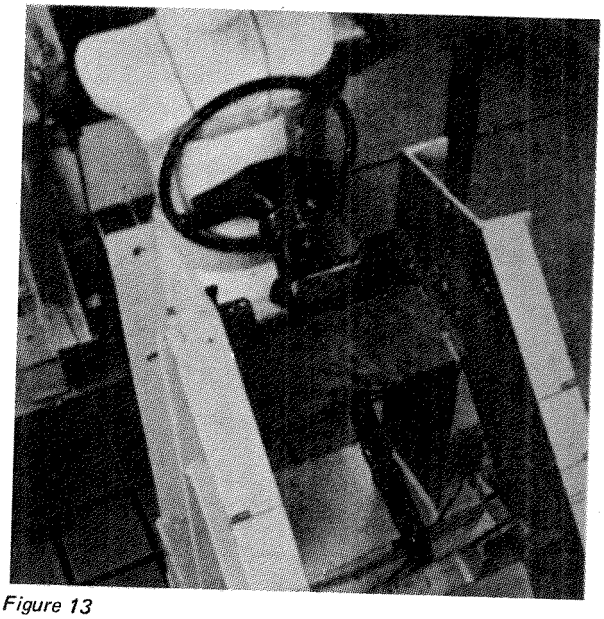


Figure 13

works in conjunction with the energy absorbing steering column and the shear-off support as an energy absorbing system in event the unbelted driver is thrust against the steering wheel. The steering wheel itself does not generate any pressure peaks to the thorax, but distributes contact pressure as evenly as possible.

The passenger compartment was modified to adjust the occupant protection to the required impact velocity. New front seats have been developed with energy absorbing roof mounted seat backs, with lateral shoulder support and with an optimized seat shape which minimizes "submarining" and ensures maximum comfort. The lateral support serves to prevent collision of driver and co-driver in event of side impact. Front seat headrests are integrated with the seat structure and designed to have minimum effect on the driver's rear vision. In addition, the high back rest prevents extensive roof deformation in rollover tests, Figure 14.

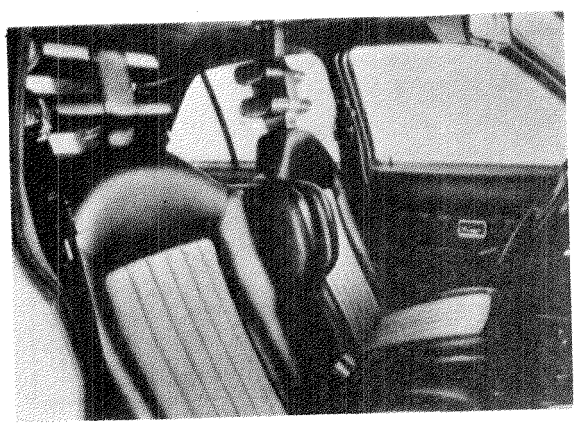


Figure 14

In rear-end collisions, on the other hand, the driver (or co-driver) is pressed with the multiple of his body weight against the backrest, causing extremely high load on its tilting mechanism, Figure 15.

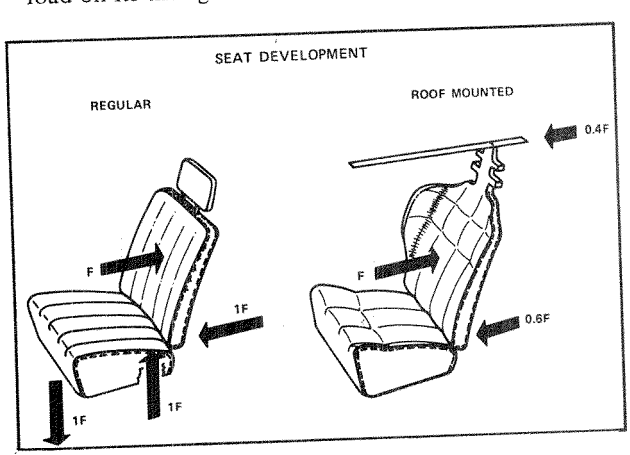


Figure 15

For this reason, load on front seat backrests of the OSV 40 was relieved by seat belt anchorages on the roof. This also securely prevents tipping of the seat. The energy absorbing stretching of the roof belt assures cushioning of the backrest in event of rear-end impact. The adjustability of seats and backrests is not affected by the roof mounted belts, since the guiding system blocks automatically only in event of impacts, Figure 16.

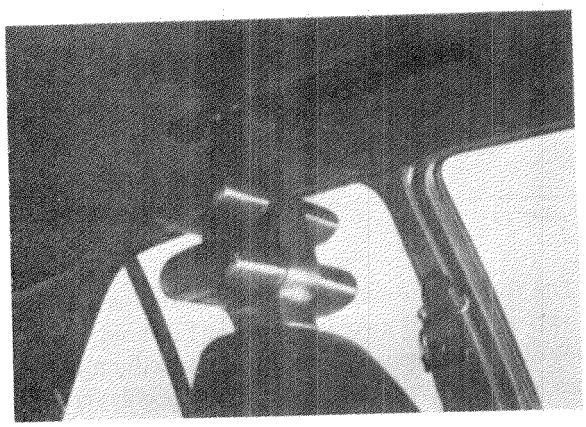


Figure 16

The OSV front seat concept has been extensively tested in a series of sled tests.

A new headrest concept of a shutter type design has been installed to obtain sufficient head support for rear passengers in rear impacts, Figure 17. The vertical belts in such a design minimize obstructions to the driver's rear field of vision.

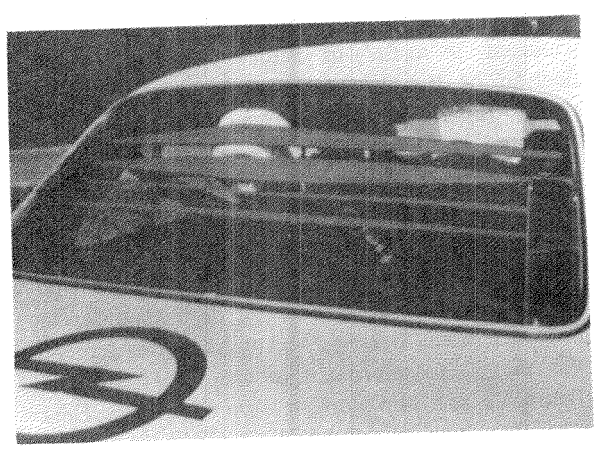


Figure 17

The vehicle is equipped with three-point safety belts for all occupants. Automatic belts with impact sensitive belt tighteners were selected for the front seats. This belt tension system compensates belt slack, thus permitting full utilization of the occupant's forward motion for energy absorption.

After extensive laboratory tests, a polyurethane foam was selected as padding material to reduce the risk of occupant injuries, Figure 18.

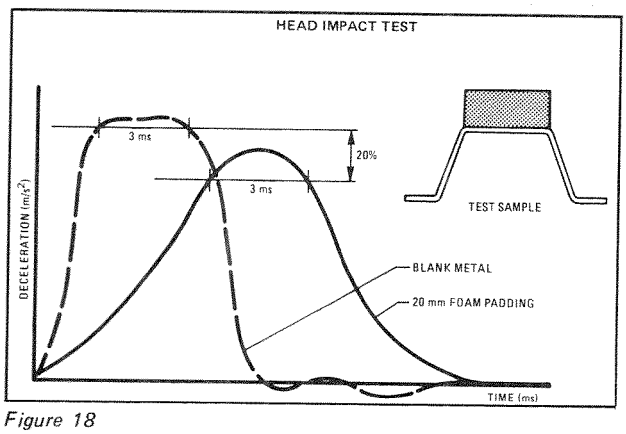


Figure 18

OPERATIONAL SAFETY

All controls are located within easy reach of the driver, even when he has his belt fastened.

From what is generally known about the interactions between driver and vehicle, several components were selected which help to prevent dangerous situations. The failure warning system controls 11 safety related functions of the vehicle and requires only *two* indicator lights, Figure 19. This system fulfills two requirements: first, the number of instruments is kept to a minimum to avoid reducing the driver's concentration capability; second, all safety related functions of the vehicle are monitored for prompt spotting of system failures.

The two indicator lights report signs of wear or operational defects to the driver in a manner which alerts him to the extent of reduced operating safety by different type signals. The repair shop, then, can directly localize any source of trouble via an electronic diagnostic system. This system can be linked with the warning system via a central socket and will indicate specific failures by the lighting of a bulb.

Good road illumination and discernability of the vehicle even under unfavorable lighting conditions are a key contribution to accident prevention. The OSV 40 is equipped with halogen headlights, fog lamps, turn signals and side markers. The rear of the vehicle is outfitted with tail lights (similar to production-type, turn signals), brake lights, back up lights and a fog tail lamp, Figure 20. The side markers assure that the vehicle is easily seen by other drivers.

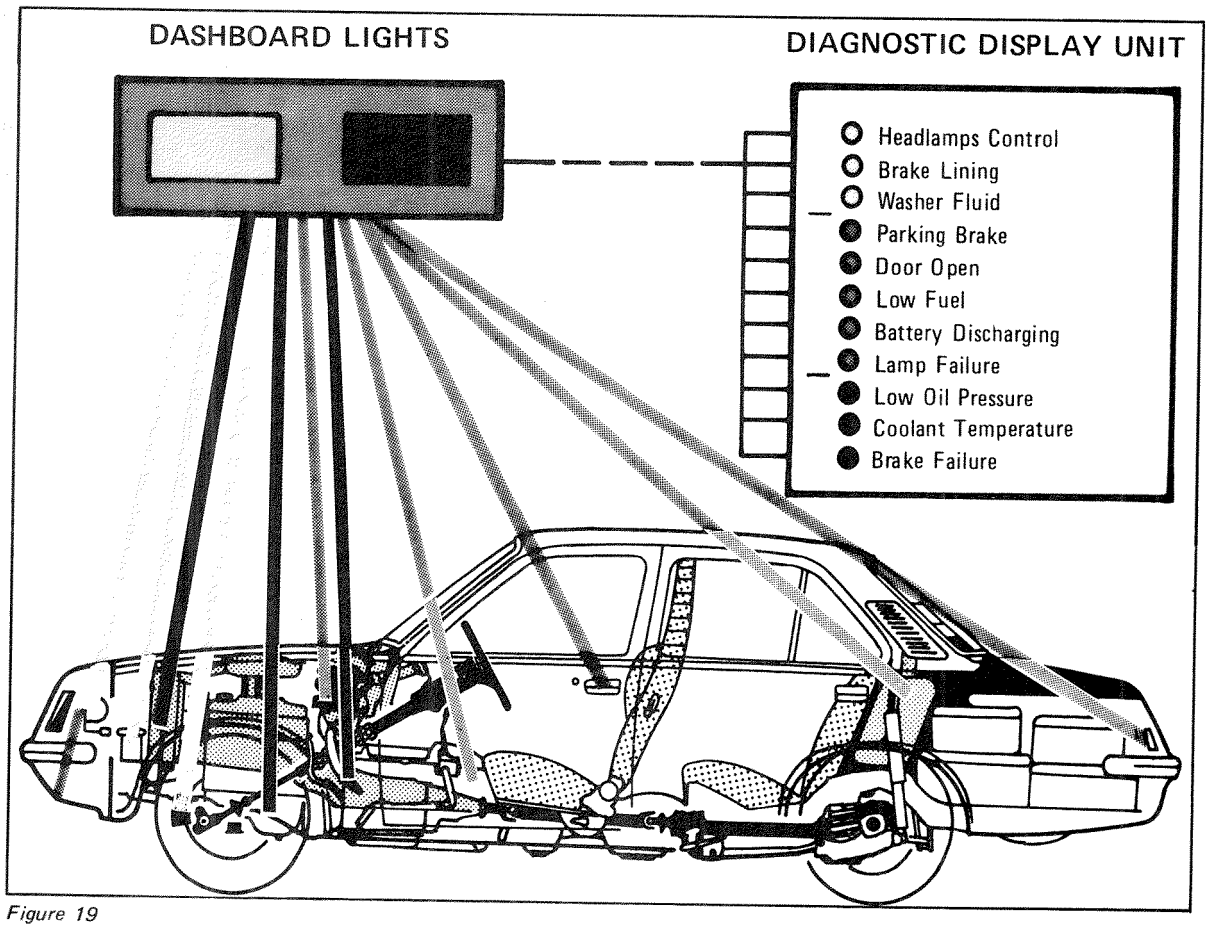


Figure 19

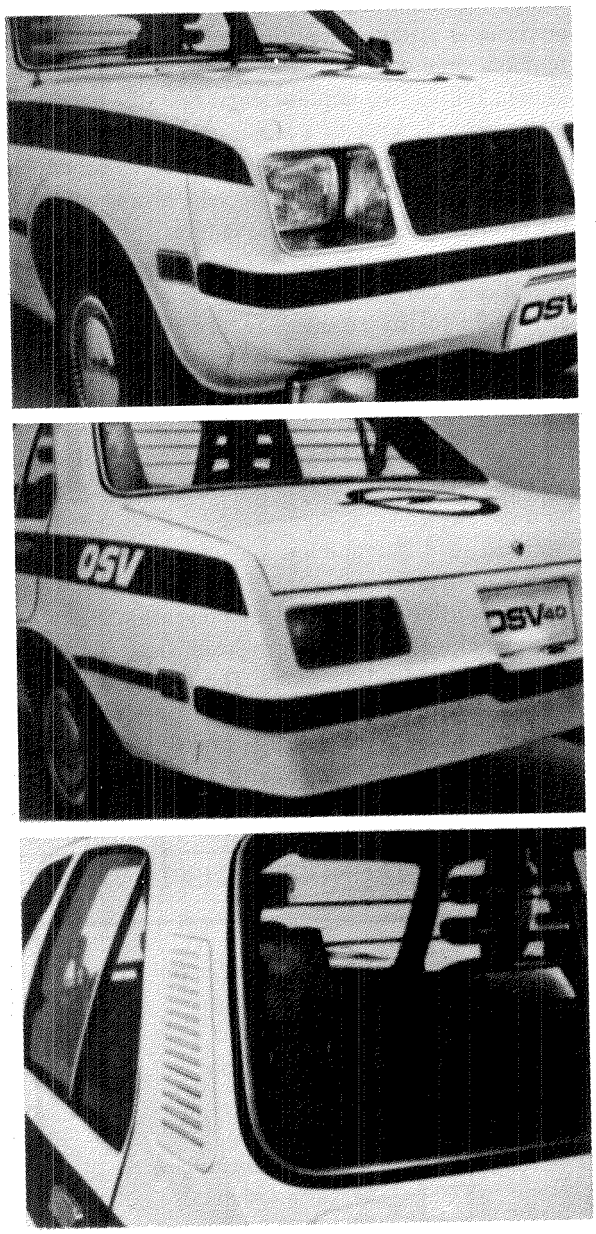


Figure 20

Two additional high-level warning lights behind the backlight alert following vehicles to hazardous situations with both warning flashers and "emergency braking" lights. Thanks to their high-set positions, they are visible from a great distance–even through the windshields and rear windows of other vehicles–and are also protected from damage in event of rear end collisions.

The headlamp position is adjusted to various vehicle load conditions by a headlamp aiming device or levelling system, Figure 21. A headlamp cleaning system in combination with halogen headlamps and headlamp aiming device provide improved visibility without annoying the oncoming traffic.

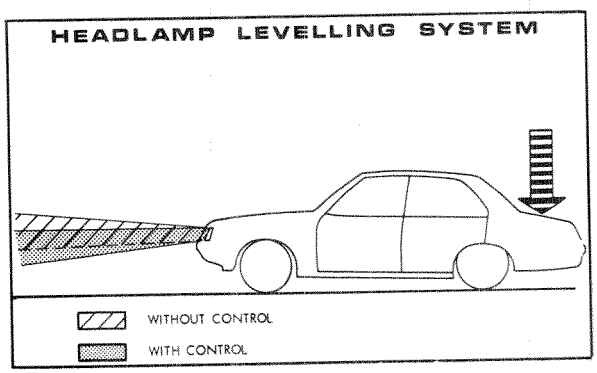


Figure 21

The large, heatable rear window combined with the broad rear-view mirror provides a good field of vision even under poor weather conditions. A new outside mirror with two separated fields of view eliminates the blind spot which is inherent in most present mirror systems, Figure 22. Field evaluation of this concept will have to be performed.

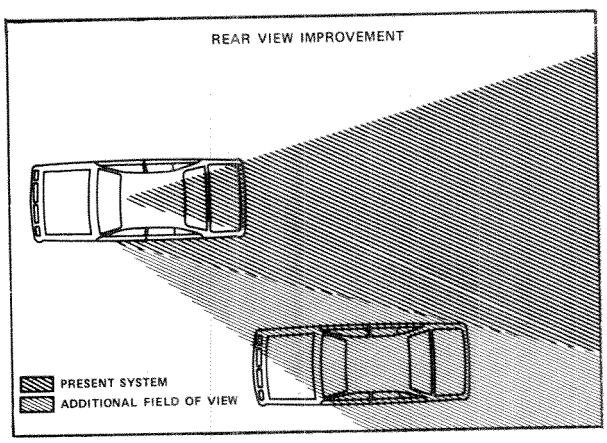


Figure 22

An automatic transmission and a FM radio for traffic news round out the equipment.

The installation of the described components and the imposition of the requirement of a front impact velocity of 40 mph result in a weight increase of the body-in-white of 60 kg as compared with the Kadett. The curb weight of the OSV-40 is increased by 120 kg, Figure 23.

PEDESTRIAN SAFETY

Improved pedestrian protection is enhanced in very low speed impacts with polyurethane front and rear fascia supported by plastic honeycomb structures, Figure 24. The severity of local injuries of impacted pedestrians is reduced by the pliancy of these fascia. Their smooth rounded contours will cause an impacted pedestrian to roll on the vehicle's hood, thus substantially diminishing the risk of injuries.

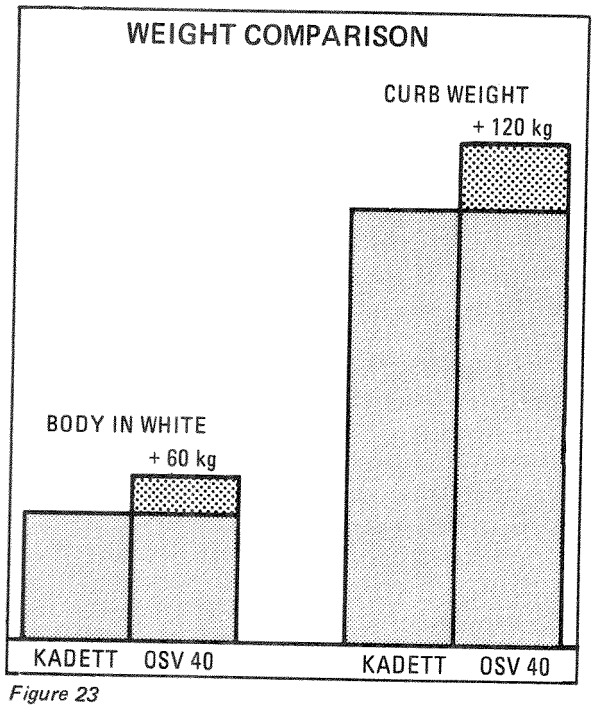


Figure 23

Like the production Kadett, the OSV 40 is outfitted with smooth exterior contours with no protruding parts, rounded door handles and an easy-break-off exterior mirror to generally reduce the injury risk of impacted pedestrians.

In addition to providing pedestrian safety, the polyurethane parts with their reinforced elastic honeycomb structure provide an alternative to the presently used aluminum bumper with hydraulic shock absorbers. There is an advantage to these pliant front and rear sections in that repair costs for collisions above 5 mph are lower than with hydraulic systems.

OCCUPANT PROTECTION

General test conditions provided that all test vehicles were driveable prior to the tests. The fuel tank was filled with non-inflammable liquid for safety reasons. All vehicles were loaded with four 50 percent Hybrid II dummies. The vehicle weight including cameras and test equipment was equivalent to fully loaded. The front seats were fixed in their next to last position.

To reduce the test evaluation to significant data, forces and decelerations only were measured in the endangered body areas of the dummies.

Requirements provided that the injury criteria of the occupants should not exceed criteria given below. Either the occupants should be able to leave the vehicle after the test or they should be removed without use of tools. Dummies should not be damaged by structural parts. No liquid should be

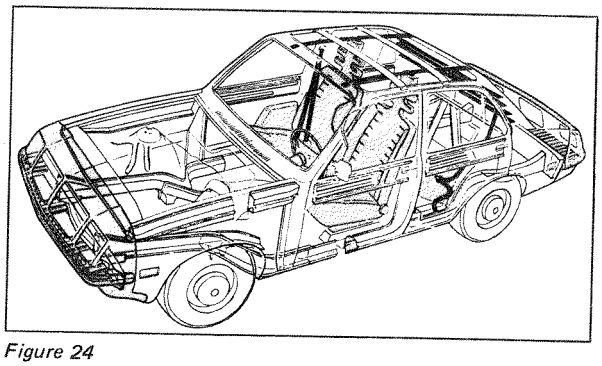


Figure 24

spilled from the fuel tank or the fuel lines during or after the tests. In addition, all doors should remain closed during the rollover test and no part of the dummies should extend past the vehicle contour.

OPEL OCCUPANT INJURY CRITERIA[1]

Head

The resultant head acceleration should not exceed 800 m/s^2 for any continuous period of more than 3 ms. The Head Injury Criteria (HIC) should not exceed 1000.

$$HIC = \left[\frac{1}{t_2 - t_7} \int_{t_7}^{t_2} a\,dt\right]^{2,5} (t_2 - t_7)$$

Chest

The resultant chest acceleration should not exceed 600 m/s^2 for any continuous period of more than 3 ms. Severity Index (SI) should not exceed 1000.

$$SI = \int_0^t a^{2,5}\,dt$$

Pelvis

The resultant pelvis acceleration should not exceed 600 m/s^2 for any continuous period of more than 3 ms. The maximum axial femur load should not exceed 7.6 kN.

Note – The injury criteria had been applied only to obtain comparable test results to other ESV's. Future legal injury criteria however have to be validated by more than biomechanic data and better anthropomorphic test devices.

In order to check the validity of our assumptions, a comprehensive test program composed of some six elements was performed.

[1]Corresponding with human tolerances as specified in MVSS 208, ESV specifications and VDA proposal.

- 40 mph Barrier Front
- 30 mph Fixed Pole Front
- 30 mph Rear Moving Barrier
- 30 mph Car to Car Side Impact
- 19 mph Side Pole (Moving)
- 30 mph Rollover

CRASH TEST RESULTS

The vehicles were loaded with four Hybrid II dummies which together with the test equipment resulted in a fully loaded condition.

The test results are listed in Figure 25 and show that—with few exceptions—the presently known injury criteria were fulfilled. Figures 26–31 provide more detail.

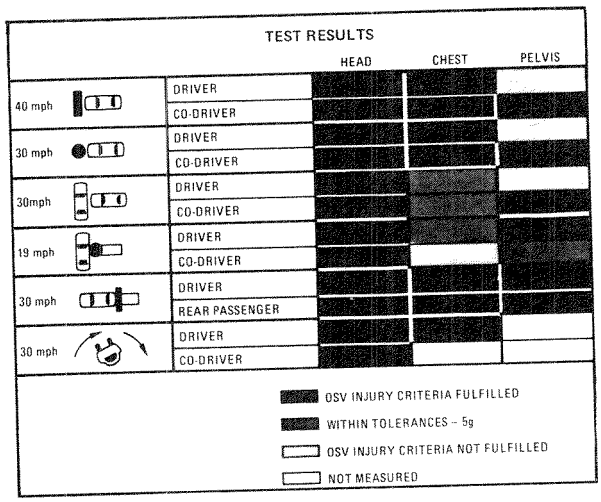


Figure 25

The experience from our test program showed that legal requirements which specify design criteria of single safety components are of little significance. Our goal is to save lives and reduce injuries. This can only be ensured by fulfilling defined injury criteria and not by meeting dimensional requirements. It should be recognized that the Hybrid II dummy, while the best device available, lacks biomechanic fidelity.

For an effective continuation of our program we urgently need biomechanic data, improved anthropomorphic dummies and generally accepted test specifications. We also need this information because the energy and resource situation will force us to check our concept again carefully.

CONCLUSION

The Opel Safety Vehicle showcased here in London is not an Utopian "exhibition vehicle" but a further advance based on an advanced production model. While it is a research vehicle this systematic development will permit Opel to check the effectiveness of the incorporated safety systems and their possible future use for production based on realistic cost benefit analyses. Further details of the OSV follow; they will be discussed in our technical speech "Structures and Restraint Systems."

65 kmh Front Barrier Impact

Test speed – v = 63.0 km/h
Vehicle weight – 1,161 kg
Length reduction – 500 mm

Test data:

Location	Front l.	Front r.	Rear l.	Rear r.
Head HIC I	784	500	608	–
Chest m/s² cumul.	500	510	–	–
Pelvis m/s²	–	470	–	–
Femur load (KN) r.	1.0	3.0	3.1	–
Femur load (KN) l.	2.3	1.8	4.8	–

Remarks:
Occupant protection requirements were fulfilled. All four doors could be opened without use of tools after the test.

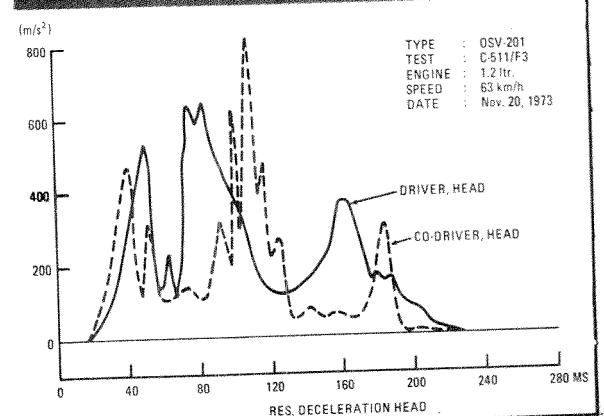


Figure 26

50 kmh Front Pole Impact

Test speed – v = 48.0 km/h
Vehicle weight – 1,160 kg
Intrusion at veh. center – 540 mm without bumper

Test data:

Location	Front l.	Front r.	Rear l.	Rear r.
Head HIC I	421	348	352	–
Chest m/s^2 cumul.	285	340	–	–
Pelvis m/s^2	–	417	–	–
Femur load (KN) r.	1.2	1.2	2.0	–
Femur load (KN) l.	1.4	1.2	1.2	–

Remarks:
Occupant protection requirements were fulfilled. In the front pole impact the crash behavior of the vehicle is of more importance than the occupant injuries. Therefore, the usefullness of this test will have to be checked for our further program.

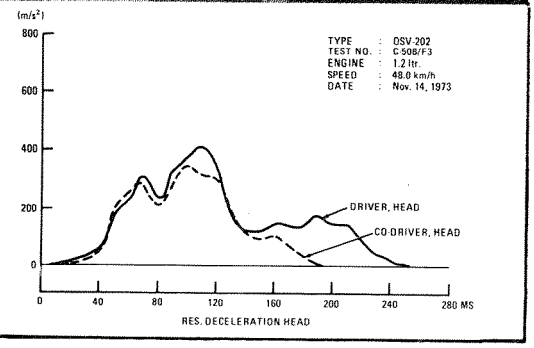


Figure 27

50 kmh Moving Barrier Rear Impact

Test speed – v = 50.1 km/h
Vehicle weight – 1,100 kg
Length reduction – 480 mm right side
– 590 mm left side

Test data:

Location	Front l.	Front r.	Rear l.	Rear r.
Head HIC I	97	–	56	80
Chest m/s^2 cumul.	180	–	200	–
Pelvis m/s^2	171	–	280	–
Femur load (KN) r.	0.8	–	2.2	–
Femur load (KN) l.	0.8	0.9	2.0	–

Average load of the shutter type headrest: 1.6 KN.

Remarks:
Occupant protection requirements were fulfilled. Tank and fuel system were not damaged. The installed shutter type headrest performed well.

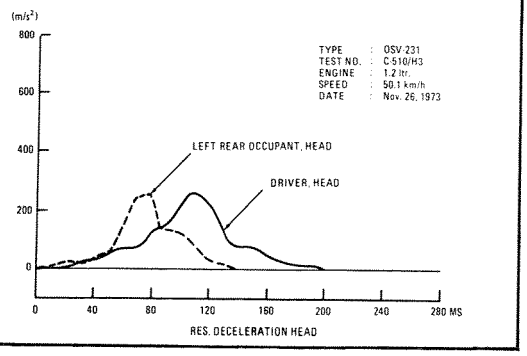


Figure 28

Moving Pole Side Impact

Test speed – 32.2 km/h
Vehicle weight – 1,170 kg
Impression at belt line – 180 mm

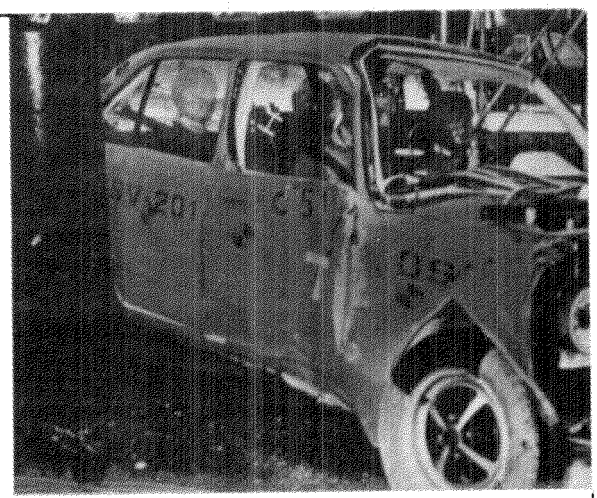
Test data:

Location	Front l.	Front r.	Rear l.	Rear r.
Head HIC I	*	8	*	–
Chest m/s^2 cumul.	150	–	70	–
Pelvis m/s^2	180	490	–	–
Femur load (KN) r.	0.5	3.8	–	0.4
Femur load (KN) l.	0.5	0.9	–	0.9

*Data too low to be evaluated – not measured.

Remarks:
Occupant protection requirements were fulfilled. The moving pole was aiming at the "H" point of the co-driver of the test vehicle.

The survival space for the occupants was maintained. The further usefullness of this test will have to be checked due to the extremely low injury data.



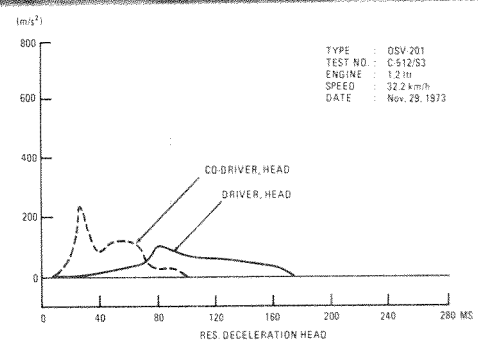


Figure 29

50 kmh Car to Car Side Impact

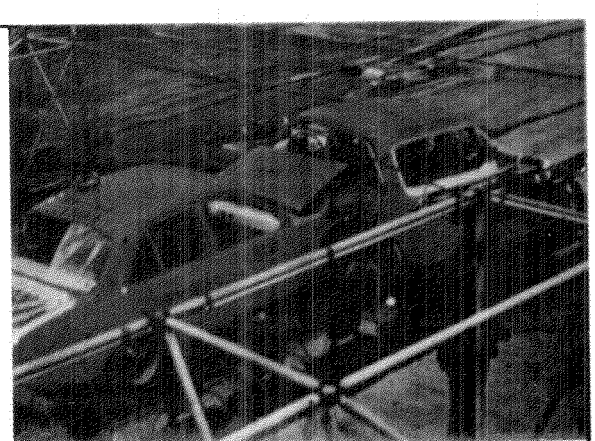
Test speed – v = 51.1 km/h
Vehicle weight – 1,180 kg
Impression at "H" point – 180 mm

Test data:

Location	Front l.	Front r.	Rear l.	Rear r.
Head HIC I	184	111	103	–
Chest m/s^2 cumul.	580	610	–	–
Pelvis m/s^2	–	–	–	–
Femur load (KN) r.	0.2	0.8	1.7	–
Femur load (KN) l.	0.4	0.6	0.9	–

Remarks:
Occupant protection requirements were fulfilled except for the cumul. chest deceleration of the co-driver. The center of the bullet car was aiming at B-pillar of the target car.

All four doors of the target car could be opened after the test without any use of tools.



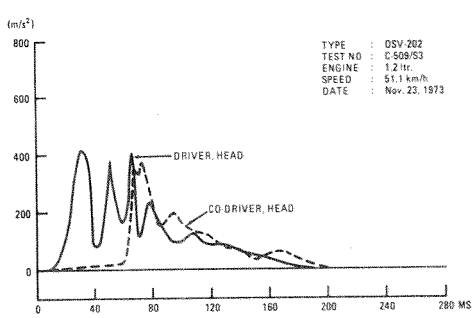


Figure 30

Rollover Test

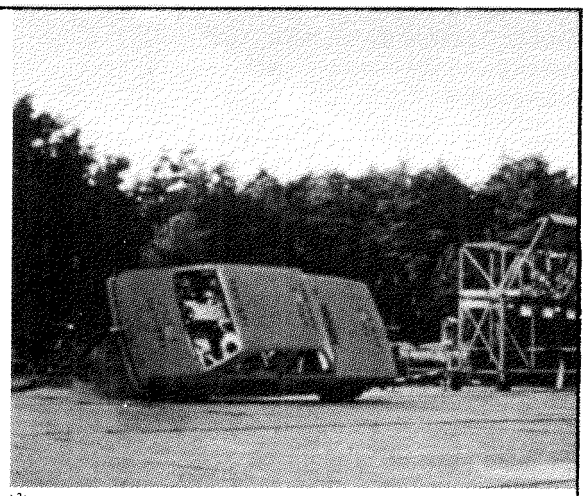
Dolly velocity — v = 48.0 km/h
Vehicle weight — 1,090 kg

Test data:

Location	Front l.	Front r.	Rear l.	Rear r.
Head HIC I	167	140	–	–
Chest m/s^2 cumul.	210	–	–	–
Pelvis m/s^2	–	–	–	–
Femur load $(KN)^{r.}_{l.}$	–	–	–	–

Remarks:
The vehicle was ejected from the dolly in a transverse position and rolled four times.

Occupant protection requirements were fulfilled. Decelerations and forces were only measured at driver and co-driver. The pelvis deceleration and the femur load were too small to be evaluated.



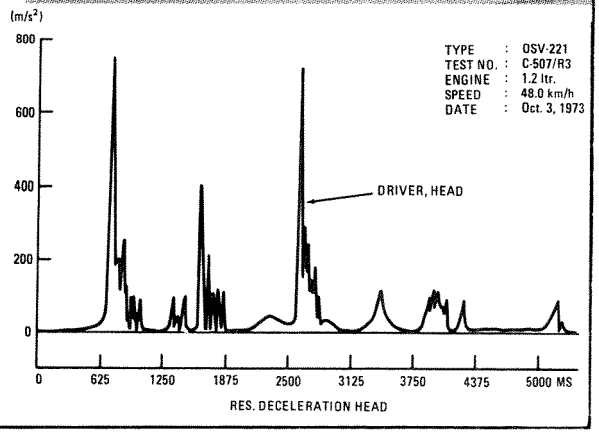


Figure 31

DR. W. LINCKE
Head of Research Department
Research and Development Center
Volkswagenwerk AG

The ESVW II is an experimental safety vehicle based on a production car of a lower weight class which has always been the emphasis of the VW model program.
The requirements in active and passive safety, fulfilled by the ESVW II, exceed all valid specifications for production cars.
The ESVW II Specifications were compiled by VW.
In its present condition the ESVW II is approx. 15% heavier than the standard car from which it was developed.
The more elaborate design and equipment would result in about a 30% increase in cost. The fuel consumption would increase about 15% due to the additional weight.
Since experience shows that the extra cost will not be accepted by the consumer, and the expediency has yet to be verified by cost/benefit considerations, production of the ESVW II is not expected.
Some safety features, which can be produced at realistic costs, and which have obtained all applicable certification, have already been incorporated in current production cars or are available as optional equipment.
With the ESVW II Volkswagenwerk AG will demonstrate techniques to fulfill extreme safety requirements for small cars.
Calculations of the extra costs of meeting such requirements will be used later in cost/benefit assessments of future specifications.

1. Technical data

overall length	3923 mm
width	1610 mm
height	1370 mm
wheelbase	2400 mm
track front	1390 mm
track rear	1358 mm
tyres	175/70–SR 13
curb weight	897 kp
engine performance	70 DIN PS
engine displacement	1,5 l

2. Abridged specifications

The following abridged version of the ESVW II specifications gives a survey of the tests to be conducted and the criteria for active and passive safety.
Likewise all valid standards for production cars must be met.

Passive safety
The specifications prescribe the following tests:

- Frontal impact against a fixed barrier at 64 km/h (40 mph)
- Frontal impact against a fixed pole at 50 km/h (30 mph)
- Car-to-car side collision at an angle of 45° with the front corner of the striking vehicle against the center of the door at 50 km/h (30 mph).
- Car-to-car rear end collision at 50 km/h (30 mph)
- Head-on collision between ESVW II and heavy car (4000 lb. category) at a closing speed of 100 km/h (60 mph)
- Rollover

Vehicle occupied by 50th percentile male dummies.
The following biomechanical limits must be observed in all tests:

Head:	H I C $\leq$ 1000
	S I $\leq$ 1000*
	Resultant head acceleration $\leq$ 80 g*
Chest:	S I $\leq$ 1000
	Resultant chest acceleration $\leq$ 60 g*
Upper thigh:	Femur load $\leq$ 771 kp

*) Note: values indicated * are given only to aid comparisons with earlier ESV tests.

Active safety
In the area of active safety the following series of tests are to be conducted:

- Steady state yaw response
- Transient yaw response
- Returnability
- Lateral acceleration
- Control at breakaway
- Directional stability
- Steering sensitivity
- Rollover immunity
- Braking performance

The limits to be observed are shown in the graphs.

3. Discussion of the specifications

The ESVW II Specifications differ essentially from the first US ESV Specifications in the requirements for destructive tests. The reason: the severe specifications for the US-ESV necessitate expensive, complex structures and restraint systems which result in large, heavy vehicles.
The extra costs of making a vehicle suitable for

higher collision speeds rise markedly with the speed whereas the benefit obtained from these investments increases only slightly.
Accident statistics show that the majority of accidents occur in the medium speed range and very few occur at either low or high speeds.
One of the most important tasks in our Research Safety Vehicle (RSV) project is therefore a detailed benefit/cost analysis based on extensive statistical material. One of its targets should be to help find the optimum use of safety measures, and beyond that, to enable definition of suitable test procedures for various types of collisions.
Building the ESVW II prior to finalizing the benefit/cost considerations should make possible more precise cost estimates of higher safety requirements.
The most frequent type of collision is the frontal collision, as accident statistics from Europe and the USA clearly show. Therefore we paid special attention to this type of collision. The relative accident frequencies according to impact direction are shown in figure 1.

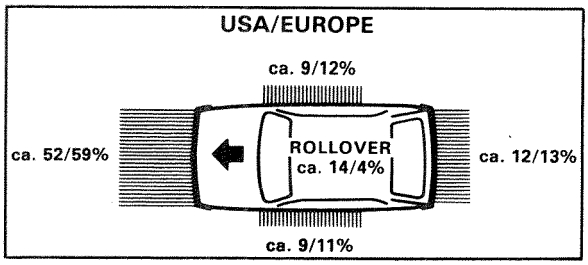


Figure 1 Relative accident frequency according to impact direction

The current requirement is that vehicle occupants must survive a 30 mph crash against a rigid barrier. With the ESVW I the occupants can survive even a 50 mph barrier crash.
Supported by the above considerations we therefore set the test speed for the frontal collision against a fixed barrier at 40 mph.
To get supporting data for future RSV work an impact speed of 30 mph has been selected for pole impact, side- and rear end collision.

4. Passive safety

The objective of our destructive tests was to fulfill the ESVW II specifications as well as all current standards.
The restraint systems described in part 4.1 were used in all tests. They are carefully matched to the modified structure of the ESVW II.

4.1 Restraint systems

The front seats of the ESVW II are fitted with two point belt-kneebar systems with automatic retractor, belt preloading and belt force limiter. This system is passive, which means that the front seat passengers are automatically protected by the system once the vehicle doors are closed, without doing anything themselves. This is accomplished by having the upper belt anchorage on the door, so that it moves when the door is opened or closed. With the door closed the force required to anchor the belt is transferred to the B-pillar of the ESVW II by a suitable latch arrangement. The automatic retractor is so arranged that the belt is guaranteed to make contact with the body at all times. Even so, normal body movements are in no way restricted. Only when the horizontal acceleration exceeds a given threshold will the belt lock.
The preloading feature of the restraint system ensures that the belted passenger immediately takes part in the passenger cell deceleration during a collision. Just after the commencement of a collision, sensors activate the preloading device. This reduces the belt slack to nil and tensions the belt to approx. 300 kp. The system developed for the ESVW II thereby guarantees optimum use of the available space for forward displacement of the body.

Furthermore, belt force limiters are incorporated in the restraint system, so that the belt forces cannot exceed a level which the human body can stand. The energy absorbing kneebar integrated in the instrument panel supports the lower torso of each front passenger in a collision. The structure of this padded kneebar together with the seats which are matched to it, ensures that the biomechanical load limits of the femurs are not exceeded.

The rear seat passenger belt system is a three point belt system with automatic retractor, preloading and force limitation. It is an active system and permits one-hand operation. Like the front passenger restraint system this one does not hinder the freedom of passenger movement.
Preloading and belt force limitation work with the same aim described previously for the two point belt-kneebar front seat system.

Figures 2 and 3 show the schematics of the restraint systems.

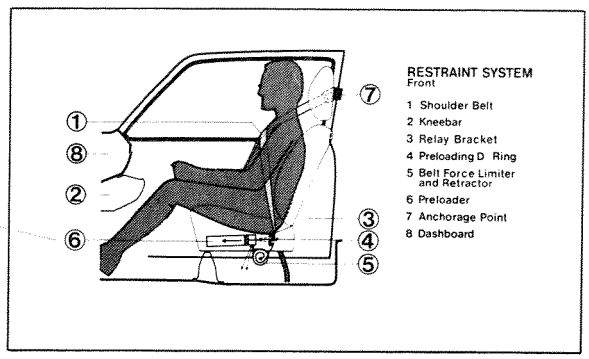


Figure 2 Restraint system front

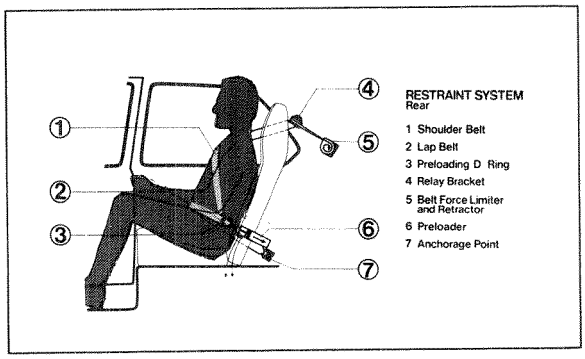


Figure 3 Restraint system rear

4.2
Frontal impact against a fixed barrier

at 64 km/h (40 mph)

During the crash the average passenger compartment cell deceleration was 24 g. The available deformation displacement at the front of the test vehicle was fully utilized according to the test requirements. The passenger compartment withstood the collision without any major deformation taking place. Figures 4 and 5 show the vehicle before and after the test.

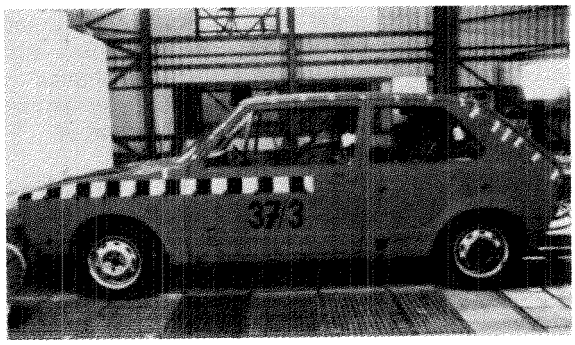


Figure 4

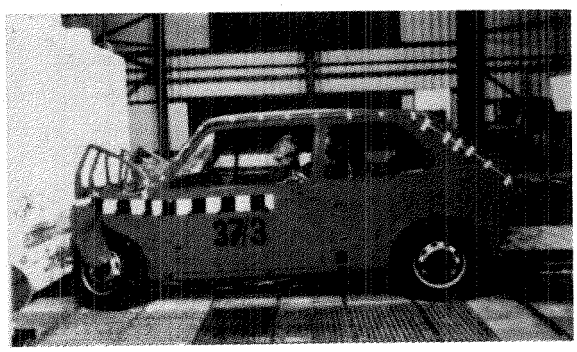


Figure 5

An inspection of the vehicle after the test showed that:

- The doors could be easily opened (they remained closed during the test)
- All the windows remained 100% within their frames, rear and side windows remained unbroken
- All seats remained in their anchorages

Load measurements were taken on the three occupants and they were below the given limits. (figs. 6 to 11).

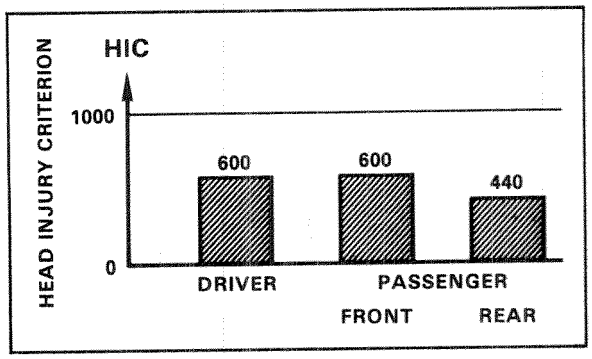


Figure 6 Frontal impact against fixed barrier at 64 km/h (40 mph)

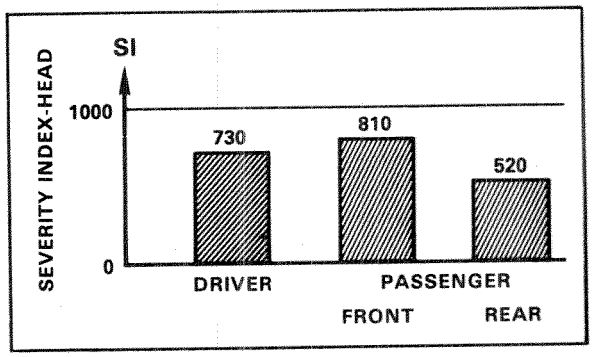


Figure 7 Frontal impact against fixed barrier at 64 km/h (40 mph)

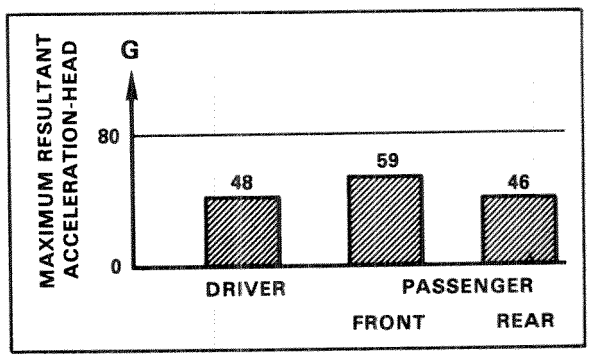


Figure 8 Frontal impact against fixed barrier at 64 km/h (40 mph)

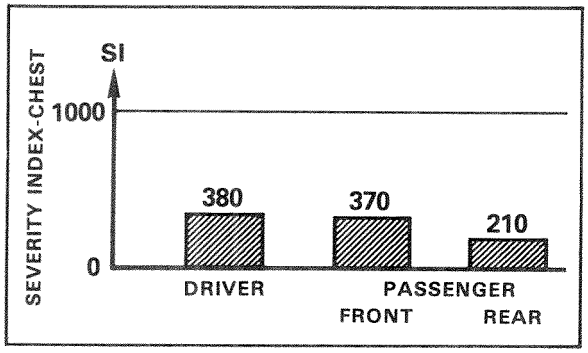


Figure 9 Frontal impact against fixed barrier at 64 km/h (40 mph)

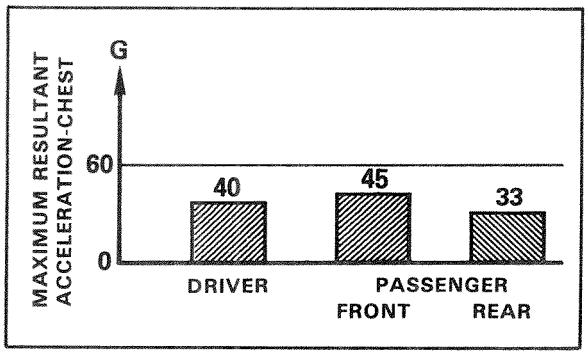


Figure 10 Frontal impact against fixed barrier at 64 km/h (40 mph)

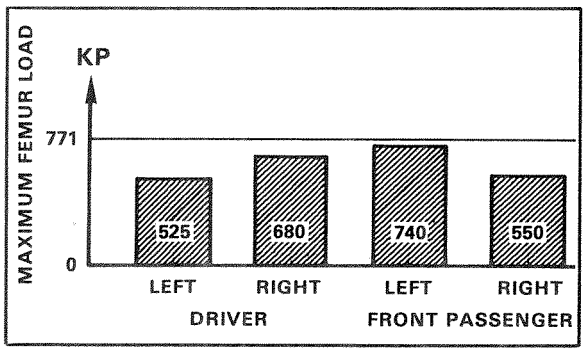


Figure 11 Frontal impact against fixed barrier at 64 km/h (40 mph)

4.3 Frontal impact against a fixed pole

at 50 km/h (30 mph).

Figures 12 and 13 clearly show that not only has the impact area absorbed energy but so has the complete front end structure. Figures 14 to 19 show further that all applicable standards were met and that the occupant loads remained well below the permissible limits

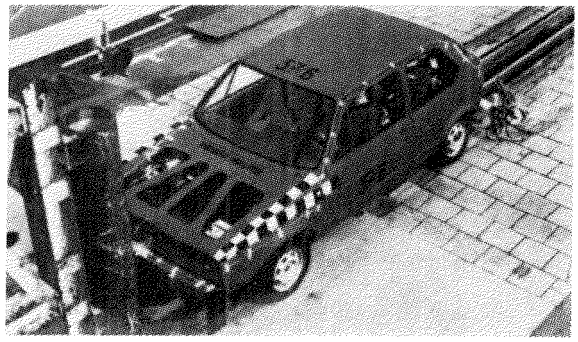


Figure 12

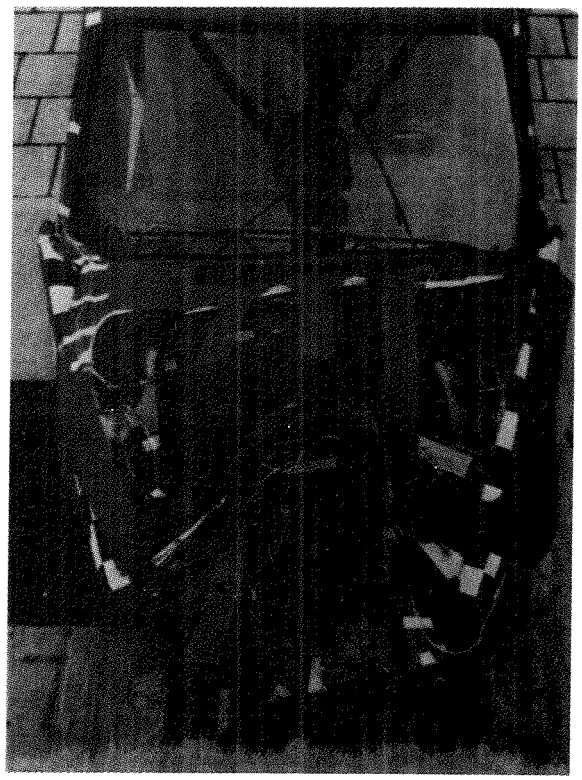


Figure 13

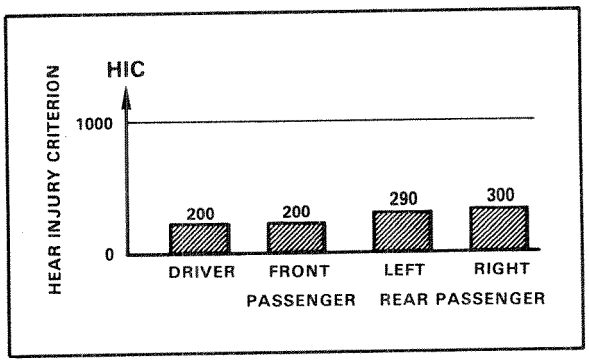


Figure 14 Frontal impact against fixed pole at 50 km/h (30 mph)

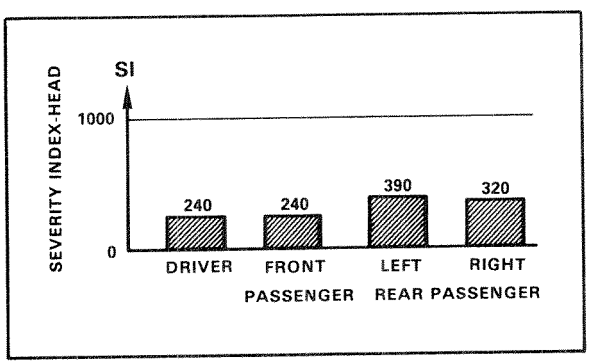


Figure 15 Frontal impact against fixed pole at 50 km/h (30 mph)

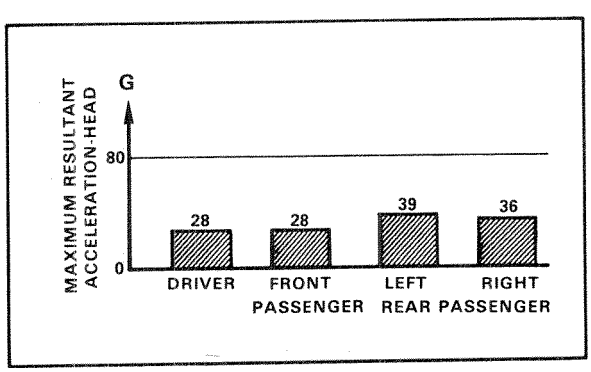


Figure 16 Frontal impact against fixed pole at 50 km/h (30 mph)

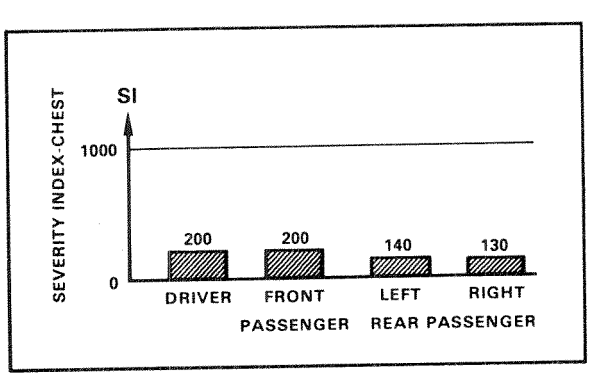


Figure 17 Frontal impact against fixed pole at 50 km/h (30 mph)

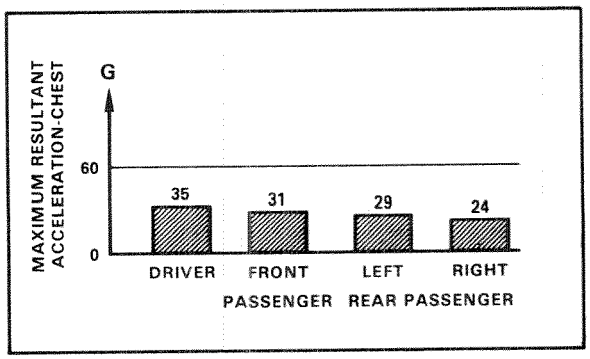


Figure 18 Frontal impact against fixed pole at 50 km/h (30 mph)

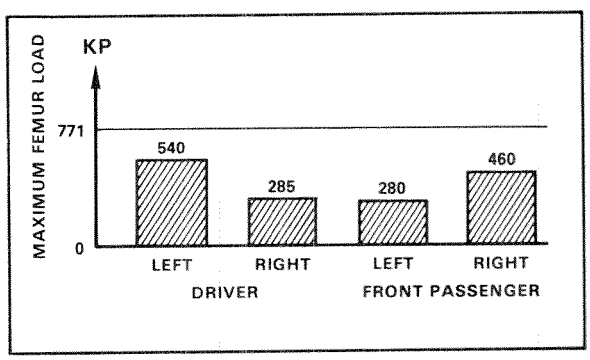


Figure 19 Frontal impact against fixed pole at 50 km/h (30 mph)

4.4 Car-to-car side collision

at 50 km/h (30 mph) with a vehicle of identical weight, with the vehicle corner striking the center of the stationary ESVW II door at an angle of 45°.

Once again all the applicable standards were met during this test and the occupant load limits were not exceeded.

Figures 20 and 21 show the configuration of the test before the collision and the deformation of the ESVW II after the collision.

Figure 21

Figures 22 to 25 show the occupant loads.
As on all other tests they are within the permissible range.

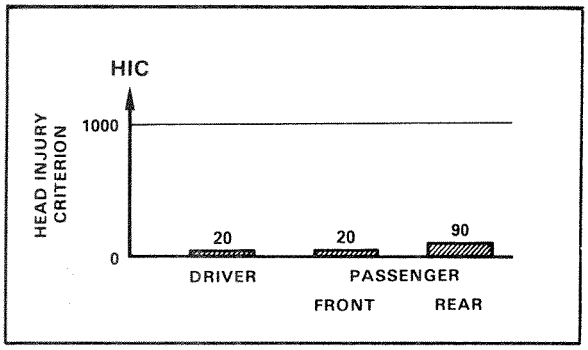


Figure 22 Car-to-side collision, 45° to center of door, at 50 km/h (30 mph)

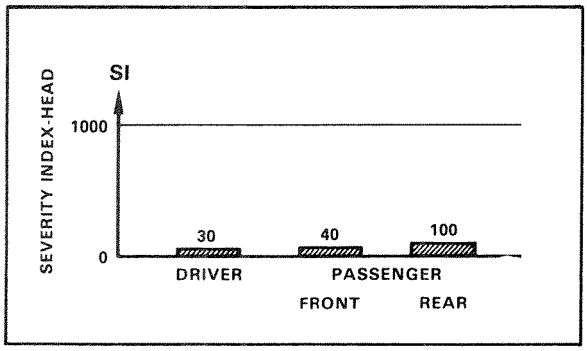


Figure 23 Car-to-car side collision, 45° to center of door at 50 km/h (30 mph)

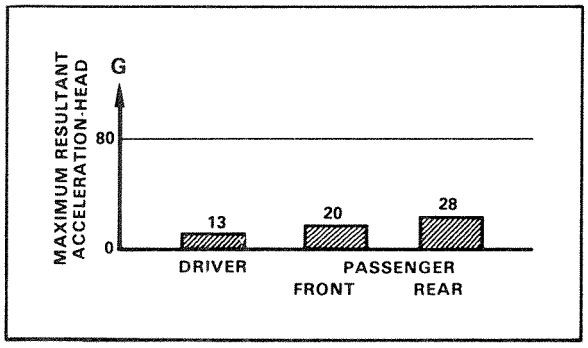


Figure 24 Car-to-car side collision, 45° to center of door at 50 km/h (30 mph)

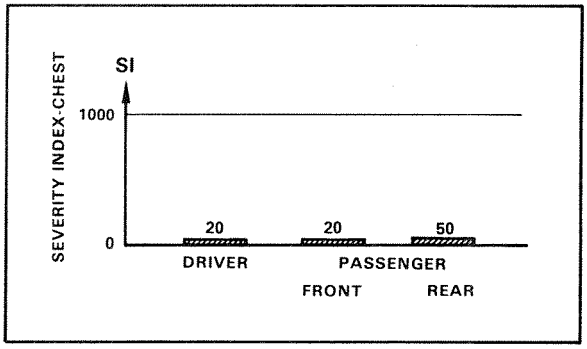


Figure 25 Car-to-car side collision, 45° to center of door at 50 km/h (30 mph)

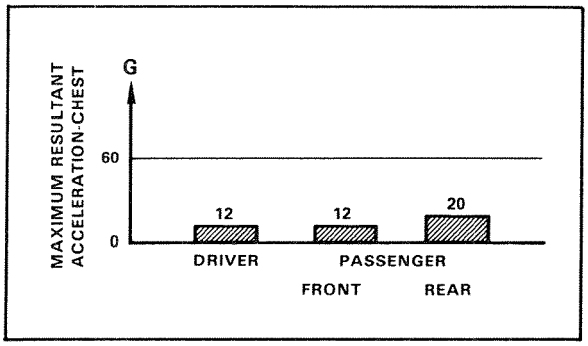


Figure 26 Car-to-car side collision, 45° to center to door at 50 km/h (30 mph)

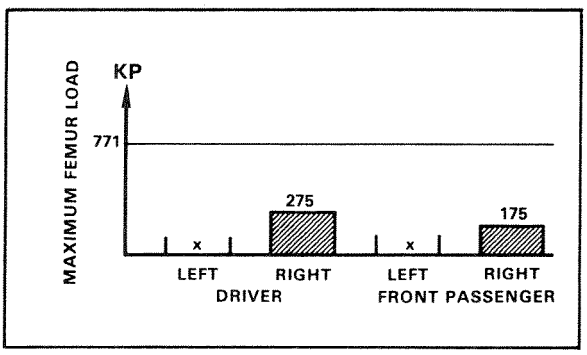


Figure 27 Car-to-car side collision, 45° to center of door at 50 km/h (30 mph)

4.5 Car-to-car rear end collision

with a vehicle of identical weight into the stationary ESVW II at 50 km/h (30 mph).

The test arrangement before the collision is illustrated on figure 28. Figure 29 shows the deformation to the ESVW II rear end after the crash.

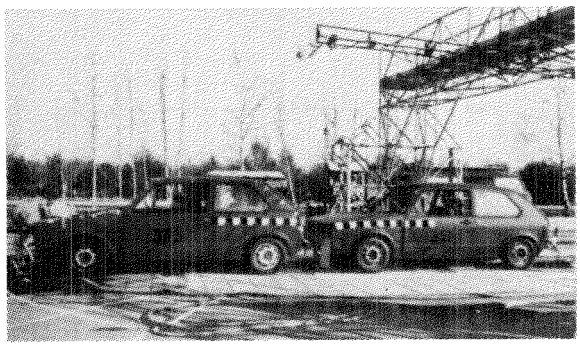


Figure 28

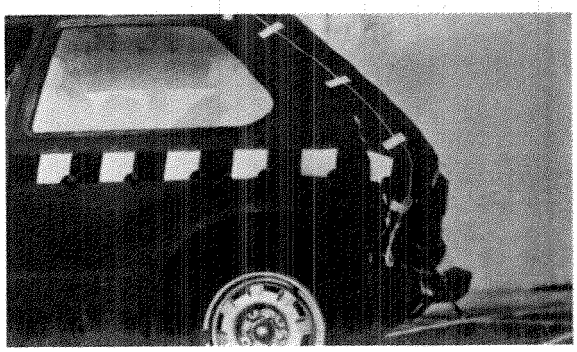


Figure 29

The figures 30 to 35 show that in this test as in others the occupants were effectively protected.

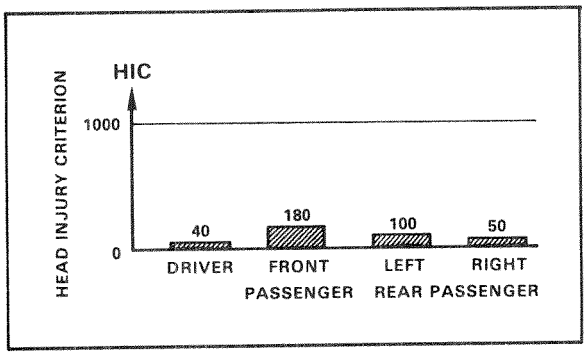


Figure 30 Car-to-car rear end collision at 50 km/h (30 mph)

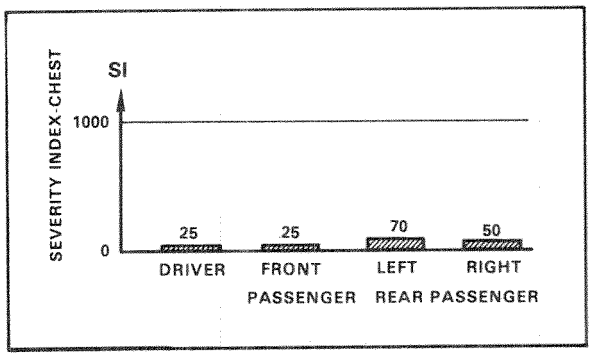


Figure 33 Car-to-car rear end collision at 50 km/h (30 mph)

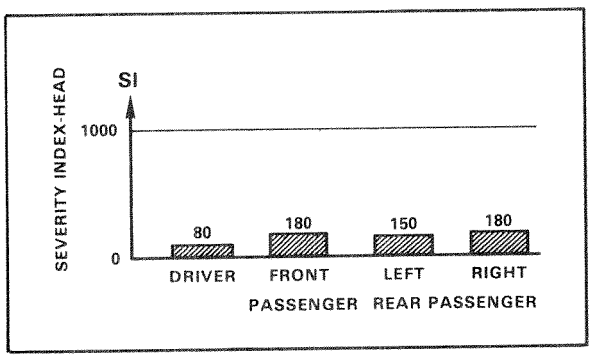


Figure 31 Car-to-car rear end collision at 50 km/h (30 mph)

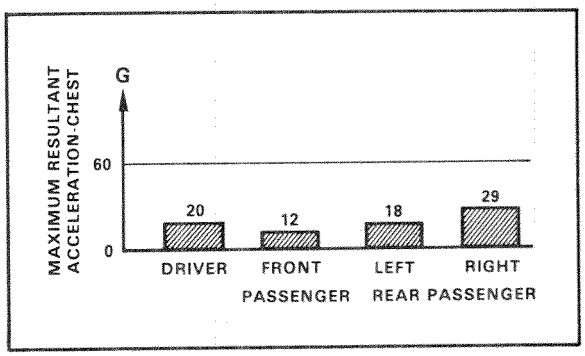


Figure 34 Car-to-car rear end collision at 50 km/h (30 mph)

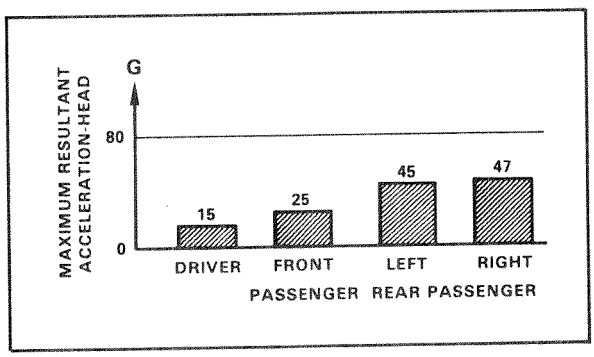


Figure 32 Car-to-car rear end collision at 50 km/h (30 mph)

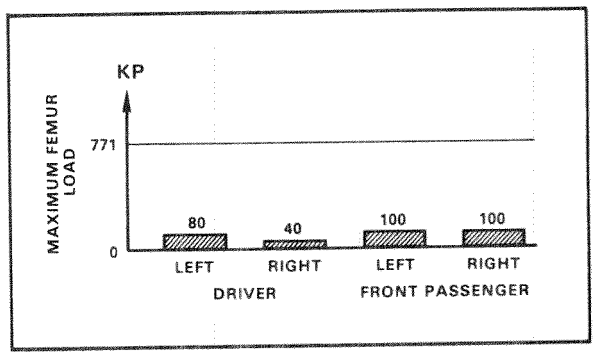


Figure 35 Car-to-car rear end collision at 50 km/h (30 mph)

4.6
Frontal collision of the ESVW II against a heavy car

at a closing speed of 100 km/h (60 mph)

To get information about its compatibility a head-on collision between the ESVW II (1100 kp) and a heavy car (1800 kp) at a closing speed of 100 km/h (60 mph) was performed.
The passenger compartments of the ESVW II and the heavy car withstood the collision without any major deformation taking place. Figures 36 to 38 show the vehicles before and after the test.

Figure 36

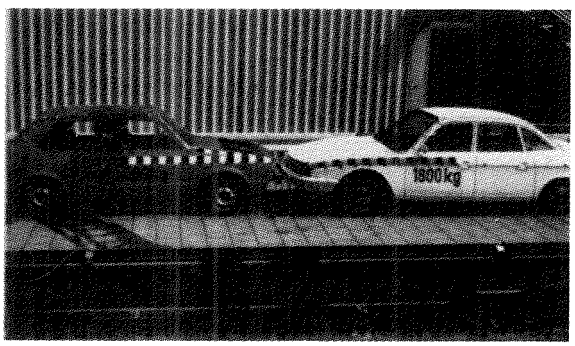


Figure 37

Figure 38

Load measurements were taken on the three occupants of the ESVW II and the two occupants of the heavy vehicle. All the measurements were well below the given limits. Figures 39 to 44 clearly show that for the three occupants of the ESVW II.

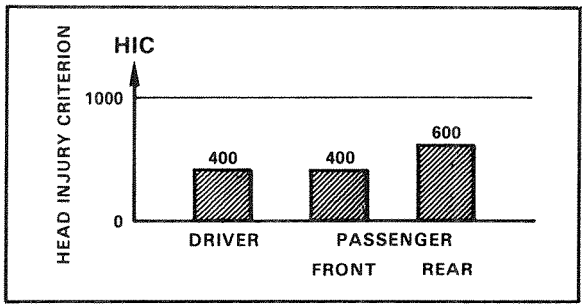


Figure 39 ESVW II (1100 kp) against heavy car (1800 kp) at a closing speed of 100 km/h (60 mph)

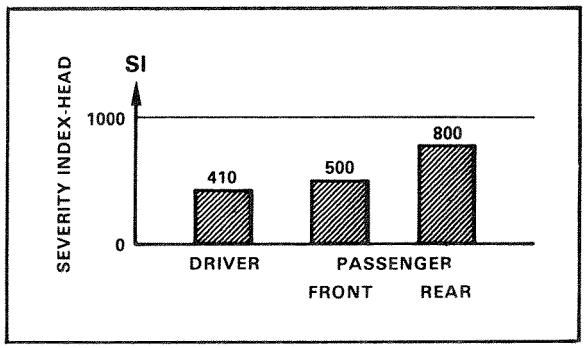


Figure 40 ESVW II (1100 kp) against heavy car (1800kp) at closing speed of 100 km/h (60 mph)

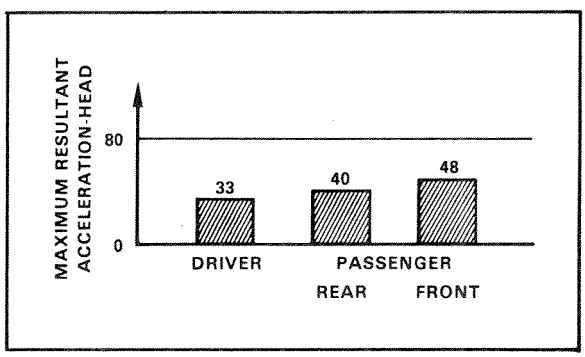


Figure 41 ESVW II (1100 kp) against heavy car (1800 kp) at closing speed of 100 km/h (60 mph)

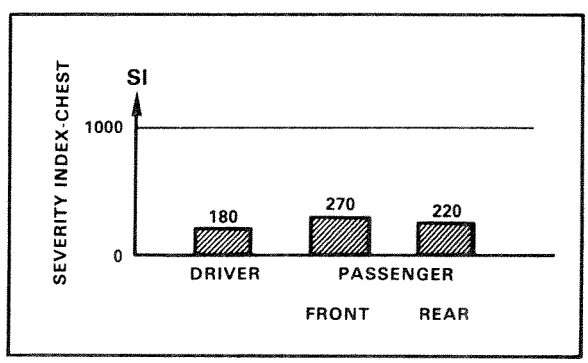


Figure 42 ESVW II (1100 kp) against heavy car (1800 kp) at closing speed of 100 km/h (60 mph)

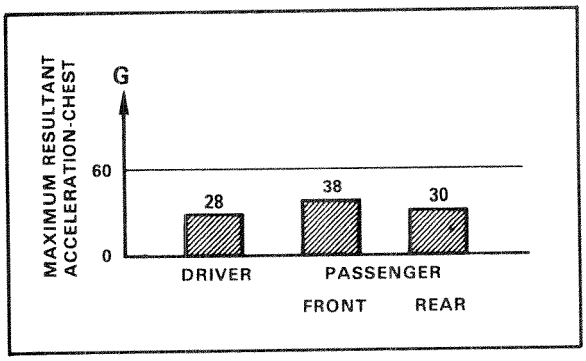


Figure 43 ESVW II (1100 kp) against heavy car (1800 kp) at closing speed of 100 km/h (60 mph)

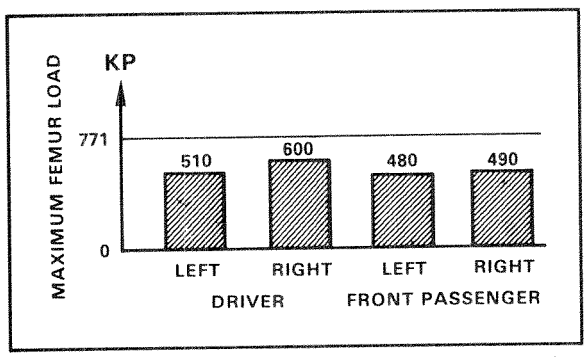


Figure 44 ESVW II (1100 kp) against heavy car (1800 kp) at closing speed of 100 km/h (60 mph)

4.7 Rollover

A rollover was performed with the ESVW II. The vehicle was driven over a ramp with approximately 70 km/h (43 mph).
After turning around its longitudinal axis for approximately 270 degrees the vehicle hit the ground. After a further 90 degrees turn the vehicle came into its normal position, slipped and came to a stop. The analysis of the movies showed that the FMVSS 208 requirements were completely fulfilled.

5. Active safety

The requirements of the ESVW II Specifications with respect to active safety adhere closely to the previous US ESV Specifications. A part of the ESVW II research work on this subject is intended to establish whether these requirements are suitable as a basis for future standards. The results obtained so far show the ESVW II fulfills the requirements of the specifications in every respect.

5.1 Steady state yaw response

Specification
The steady state yaw response at a lateral acceleration of 0.4 g should be within the limits shown in figure 45. The curves for higher lateral accelerations should be similar in shape and be below the curve for 0.4 g.

Results
Figure 45 shows that the ESVW II fulfills the requirements.

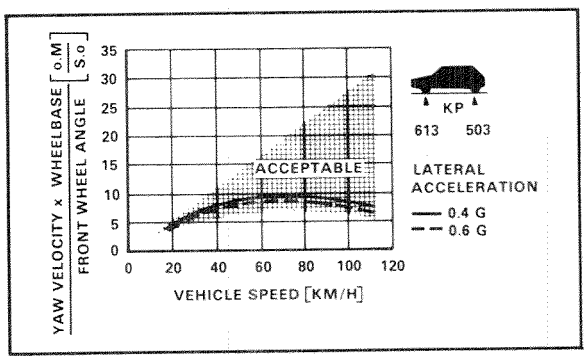


Figure 45 Steady state yaw response

5.2 Transient yaw response

Specifications
With a steering input applied at a rate of not less than 500°/sec which produces a lateral acceleration of 0.4 g at vehicle speeds of 40 km/h (25 mph) and 112 km/h (70 mph), the transient yaw response shall be within the limits shown in figure 46.

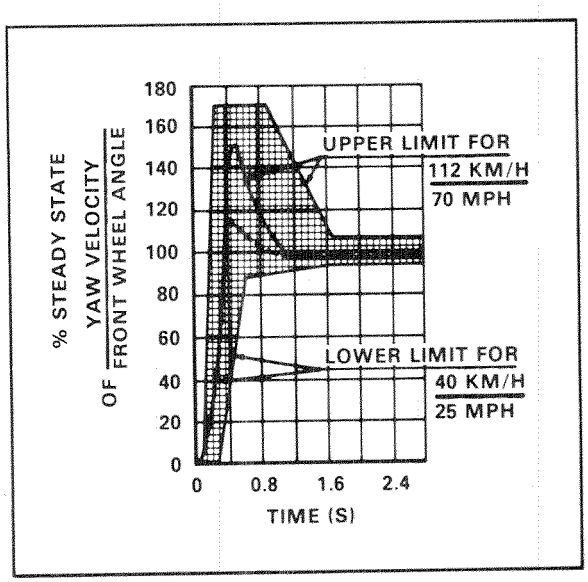


Figure 46 Transient yaw response

Results
At both speeds the yaw response was within the specified limits as shown in figure 46.

5.3 Steering returnability

Specifications
The vehicle shall be driven at 40 km/h (25 mph) and 80 km/h (50 mph) on a circular course at a steady state lateral acceleration of 0.4 g. The driver then releases the steering wheel suddenly and keeps the throttle in a constant position. After 2 seconds the yaw rate at a speed of 80 km/h (50 mph) must be less than 4°/sec. At 40 km/h (25 mph) the yaw rate after a period of 2 seconds must be less than 1°/sec.
Furthermore, the yaw angle at both speeds must be within the limits shown in figures 47 or 48, where the origin of the coordinates is the point in time when the steering wheel is released.

Results
The steering returnability requirements are fulfilled as shown in figures 47 and 48.

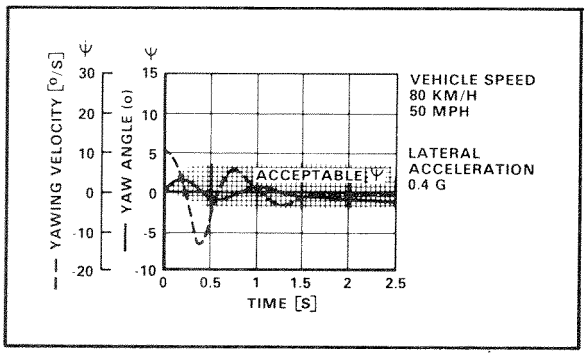


Figure 47 Returnability

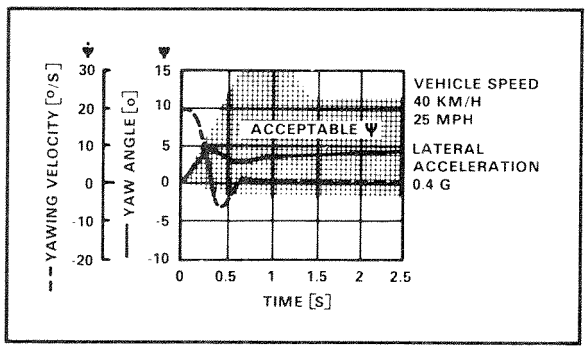


Figure 48 Returnability

5.4 Lateral acceleration

Specifications and results
The minimum lateral acceleration in the specifications for quasi-steady-state driving on a circular course with a steering correction of less than ±5° is entered in table 1 for various tyre pressures and road surfaces.
The values obtained with the ESVW II are considerably better than required by the specifications as shown in the table.

SURFACE	TIRE PRESSURE % OF DESIGN	LATERAL ACCELERATION (G) SPECIFIED	ACHIEVED
DRY CONCRETE OR ASPHALT	100	0.6	0.77
	120	0.6	0.8
	80	0.55	0.63
	120 FRONT 80 REAR	0.63	0.8
	80 FRONT 120 REAR	0.59	0.7
WET CONCRETE OR ASPHALT	100	PROPORTIONAL DRY TEST SKID NUMBERS	0.7

Table 1 Lateral acceleration, fixed control mode

5.5 Control at breakaway

Specifications
The vehicle shall be driven on a skid pad at the maximum possible lateral acceleration – below breakaway – on circular courses with a radius of 30.5 m and 69 m. The steering wheel is then held in a fixed position and the speed increases slowly until any point in the vehicle longitudinal axis between front and rear axles has moved outward 3.0 m from the initial path. At this moment the throttle shall be closed. Without using the brakes it must be possible to bring the vehicle back to the original course in less than 4 seconds with a steering wheel movement rate of less than 500°/sec.

Results
The ESVW II fulfills the specifications with times below the required 4 seconds. In every phase of the maneuver the vehicle is stable and easy to control.

5.6 Crosswind sensitivity

Specifications
The vehicle shall be subjected to a crosswind of 80 km/h (50 mph) over a distance of 6.1 m and a vehicle speed of 50 km/h (30 mph), 80 km/h (50 mph) and 112 km/h (70 mph). Two seconds after entering the crosswind section the course deviation shall not exceed the specified limits.

Results
Figure 49 shows that the deviation of the ESVW II was hardly detectable and thus well below the limit.

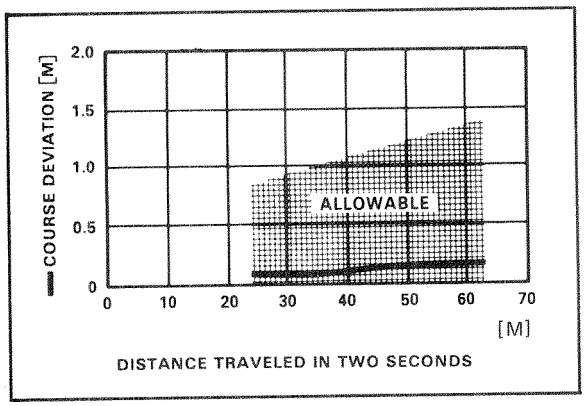


Figure 49 Crosswind sensitivity

5.7 Directional stability over pavement irregularity

Specifications and results
When driving over a halfround tube section with a radius of 2.5 cm at 30° to the direction of motion no measurable deviation occured at speeds between 48 and 112 km/h.

5.8 Steering control sensitivity

Specifications and results
The steering torque required for a yaw rate of 2°/sec was between 12 and 20 kpcm and thus above the specified lower limit of 5.8 kpcm.

5.9 Rollover immunity

Specifications and results
The pylon course of 305 m with pylon spacing of 30.5 m as specified was negotiated at more than the required 80 km/h and the vehicle remained stable and well under control at all times.

5.10 Braking performance

Specifications
The specifications prescribe that at 96 km/h (60 mph) the vehicle shall be capable of decelerating to a stop in a distance of 53.3 m without two weels or any one axle locking up while remaining in a lane 3.7 m wide.
The vehicle shall also be capable of decelerating to a stop in a distance of 25.9 m without wheel lockup or deviation from a 3.7 m lane from a steady state speed of 64.4 km/h (40 mph) on a radius of 108.8 m. Steering inputs up to 180° are allowed with rates not to exceed 500°/sec in this braking on cornering at 0.3 g lateral acceleration.
During failure of one circuit in straight ahead breaking the deceleration distance from 96 km/h (60 mph) to a stop shall not exceed 118.3 m.

Results
The values within the brackets are the required values.

Straight ahead braking from 96 km/h
- braking distance 51 m [53.3 m]
- average deceleration 6.9 m/s²

Braking at 0.3 g lateral acceleration from 64.4 km/h
- braking distance 25.1 m [25.9 m]
- average deceleration 6.3 m/s²

Straight ahead braking during failure of one circuit from 96 km/h
- braking distance 92.2 m [118.3 m]
- average deceleration 3.6 m/s²

No deviation from the initial path did occur on all the tests. Brake pedal forces met the requirements of the US ESV Specifications.

6.
Safety features of the ESVW II

The following tables and illustrations show the safety features of the ESVW II and to what extend they have been incorporated in production vehicles and their optional equipment.

	Standard equipment of the basic vehicle	Optional equipment available	Optional equipment proposed	ESVW II
General				
Visibility				
– large sized windows provide unobstructed overall visibility	■	■	■	■
– large washer wiper swept area	■	■	■	■
Driving comfort; reduced driving fatigue				
– convenient, comfortable and noise free interior	■	■	■	■
– comfortable ride – well matched springing/damping	■	■	■	■
Handling				
– good directional stability even under rough road condition	■	■	■	■
– high cornering capability under all driving conditions (integrated rollbar trailing arm suspension)	■	■	■	■
– very low crosswind sensitivity	■	■	■	■
Performance				
– ample reserve for passing due to wide torque range and well-chosen gear ratios	■	■	■	■
– meets all exhaust emission standards	■	■	■	■
Body				
Passenger compartment structural integrity for safety	■	■	■	■
Energy-absorbing front and rear structures	■	■	■	■
Controlled crash behaviour				
– rippled longitudinal frame sections for controlled collapse, controlled buckling of hood	■	■	■	■
– meets ESVW II specifications				■
Bumper system with hydraulic energy absorption			■	■
Padded front trim			■	■
Occupant restraint systems				
– 3-point belt system with one hand operation, front	■			
– 3-point belt system with automatic retractor, front		■		
– VW automatic restraint (passive restraint system), front		■	■	
– VW automatic restraint system with preloader and belt force limiter, front				■
– lap belt with automatic retractor, rear		■		
– 3-point automatic belt system with preloader and belt force limiter, rear				■
– increased interior side padding for greater energy absorption and shoulder supports in front			■	■
Interior mirror				
– breakaway mount, large field of view	■	■	■	■
– anti-glare feature		■	■	■
Exterior mirror				
– folding mount and large field of view	■	■	■	■
– remote adjustment			■	■

	Standard equipment of the basic vehicle	Optional equipment available	Optional equipment proposed	ESVW II
Seats				
– outstanding comfort, reduced driving fatigue and good side support	■	■	■	■
– high static strength (far exceeds current standards)	■	■	■	■
– head restraints		■	■	■
Dashboard				
– glare-free controls and ergonometrically arranged instruments	■	■	■	■
– impact areas energy absorbing and padded	■	■	■	■
Climate control ventilation				
– quick defrosting and defogging	■	■	■	■
– continuously adjustable heating and ventilation system	■	■	■	■
– air conditioning		■		
Windshield				
– laminated glass with bonded elastic mounting		■	■	■
Chassis				
Steering system				
– safety steering column	■	■	■	■
– safety steering wheel with large impact distribution area	■	■	■	■
Suspension				
– independent suspension; safety in all driving situations	■	■	■	■
– negative kingpin offset; positive tracking even in critical driving situations	■	■	■	■
Brakes				
– diagonally connected dual-circuit brake system; straight-line braking even if one circuit fails	■	■	■	■
– tactile disc brake pad wear indicator			■	■
Fuel tank				
– in the best protected location in front of the rear axle	■	■	■	■
– polymer fuel tank			■	■
Electric system				
Headlights				
– great illuminated area and distance	■	■	■	■
– halogen lamps		■	■	■
Headlamp washer system		■		
Foglamps front and rear		■		
Large warning lights		■		
Rear window defroster covering large area		■		
Rear window washer/wiper; wiper blade rests in heated area		■		
Centralized electrics and computerized diagnosis				
– high degree of operational safety, simplified check-out	■	■	■	■

DR. MAX DANNER
Research Scientist
German Association of Liability, Accident and Motor Traffic Insurers (HUK-Verband)

By virtue of the intensive work carried out on the ESV project by all manufacturers and governments, it has proved possible to enucleate the specific problems confronting future technical developments. Accident research has already provided the answers to many questions, but quite a number of problems still remain to be solved.

The HUK-Verband, which today probably has the most extensive research program in Europe on car accidents involving injuries to occupants, is able to report on the completion of the accident studies conducted in recent years and on the initiation of further studies in 1974. The total material comprising 30,000 car accidents, involving 52,000 occupants and 31,000 injured persons, is today fully compiled statistically and evaluated by a large number of various computer programs. We have thereby established a unique data bank on actual accidents, which is at the disposal of all interested parties. This data is all the more important, in that recent ESV conferences have clearly shown that effective safety specifications must be evolved not in theory, but against the actual background of day-to-day road traffic accidents.

In the light of the great volume of possible information, only central problems can be touched upon here. The question of the relation of vehicle mass in collisions and resultant injuries to occupants represents a focal point of our work today. Our material on vehicle/vehicle accidents is fully representative and thus able to furnish information as to the nature, severity and consequences of an accident in collisions between vehicles of differing mass. We have, during the course of our studies, found initial indications for the fact that specific types of vehicles are involved in specific types of accidents. The particularities are contingent not only on the vehicle itself and its characteristics as regards active and passive safety factors, but also to a great degree on the differing habits of driver categories handling heavy or light vehicles. These driver habits, in turn, are also correlated to the emotively recorded road characteristics of the vehicle.

A correct correlation of the various influential factors is, however, extremely difficult, and to some extent even impossible. In any event, the aggressiveness of the vehicles and the forces exerted on the occupants should not be defined in head-on barrier or in vehicle/vehicle tests only – the frequency of actual accident situations must also be taken into account.

With the assistance of this data bank, the efficiency of safety measures in serial production vehicles can also be controlled. While the majority of injured occupants statistically embraced by our past investigations had not fastened their seat belts, we also have data on more than 700 accidents in which the occupants did use their belts, and this enables us to clarify feature details to a great extent. In this connection, we have not found any accidents in which the wearing of seat belts has led to an increase in the injuries sustained. The decisive improvement in safety by the use of such belts has quite clearly been demonstrated. Our investigations have, however, also revealed that the protection afforded by seat belts is not exploited to the full if they are fastened too loosely or incorrectly. Occupants as a result, in actual accidents can be thrown head-first against the windshield even at relatively low speeds, which – according to automobile manufacturers – does not happen in experimental vehicle tests. The optimization of windshields is thus still an urgent problem, and the arrangement and fitting of safety restraint systems in relation to the stature of occupants must likewise be improved as soon as possible. Experimental studies on the flexibility of interior features, such as steering column, steering wheel, instrument panel, padding of the A stanchion, etc., must be augmented by results from actual accidents. In this connection, the data bank of the HUK-Verband can supply valuable basic material.

Accident studies must, however, be carried out continuously in order to be able, at any given time, to furnish a picture of the momentary accident situation and take new developments into account. For this reason, the HUK-Verband has – since January 1974, for a period of one year – been investigating all car accidents with injuries to occupants on an improved and internationally comparable basis. A great number of vehicles with uninjured occupants will also be embraced by the new program, thus rendering possible an even more comprehensive assessment of the safety factor in vehicles. By an exact comparison of accident severity and accident consequences in both vehicles in many thousands of cases, much more far-reaching results are expected from the investigations into vehicle aggressiveness. These new studies enable accident causes and evolution to be placed in direct relation to environmental conditions and the influence of weather, with the result that additional information in respect of anti-blocking systems and driving in rainy conditions (field of vision, misting of windows) and in darkness (polarized light) can be anticipated.

In addition to the scope of investigations conducted hitherto, we are also able to analyze the

influence of vehicles on other road traffic participants by incorporating car/pedestrian and car/2-wheeled vehicle accidents.

All in all, some 120,000 accidents will be investigated by a team of engineers of the HUK-Verband, assisted by medical experts. During the first few months since this new project was initiated, several thousands of accidents have already been evaluated, as a result of which essential data can also be quickly compiled for new models appearing on the market. Special procedures enable the computer evaluation to be so accelerated that a report on the work in hand can be issued every six months. Despite the extremely extensive volume of material, the HUK-Verband plans to largely complete these studies by mid-1976 and thus keep the inevitable time-lag between accident occurrence and accident evaluation as short as possible.

By specific concentration on central problems relating to vehicle safety – augmented by close coordination with all interested quarters – important results paving the way for future developments can be achieved in a considerably shorter period of time than hitherto.

STATUS REPORTS ON EXPERIMENTAL SAFETY VEHICLE DEVELOPMENT PROGRAMS

PART FIVE

THE UNITED KINGDOM

H. TAYLOR

Head of Safety Department Transport and Road Research Laboratory

Department of the Environment

The United Kingdom status report to previous conferences has outlined the British approach and highlighted important aspects of our program. These continue unchanged and our program has now progressed to the stage of building and testing prototypes incorporating various groups of safety features; a major objective has been the achievement of higher levels of safety at an acceptable cost in vehicles suitable for production. An important aspect of the program is not only to demonstrate improved safety performance but also to establish the trade-off between safety performance and cost. The compatibility between different sizes of cars is included, as is research into pedestrian safety features. The major part of our contribution today will therefore be made by colleagues from British Leyland and from the Ford Motor Company who will be reporting progress on their respective contracts with the Transport and Road Research Laboratory.

We strongly support the conclusions of the EEVC report, that most of the achievable gain in safety will in practice come from applying a few well chosen and relatively simple principles. The necessary measures must of course be agreed internationally and this will mean continuing and strengthening the collaborative efforts which have already contributed so much to the development of vehicle safety.

It may not be too early to look beyond this phase recognizing that there will be limits to the safety that can be achieved by car design alone. To progress beyond that point will direct attention once more to the environment within which cars operate and above all to the driver on whose judgement the safety of cars finally depends. These aspects are touched on in other British papers contributed to the conference.

CHARLES GRIFFIN
Operational Director of Vehicle Engineering
Austin Morris Group
British Leyland Corporation

British Leyland has prepared two technical reports to show the progress made on two separate contracts with Her Majesty's Department of the Environment. Both programs have arrived at approximately midpoint.

The first contract was based on the systems approach to safety research and allocated to the Body Engine Department of the Cowley plant. The second was directed toward achieving a measure of compatibility involving collision modes between vehicles of various sizes and towards improvements in primary safety. This work was allocated to the Austin Morris Central Engineering Department at Longbridge.

The Cowley work is based on the Marina vehicle. The Longbridge work is based on three front wheel drive vehicles, namely the 1100, 1300, and 1800 series cars. The Marina progress represents some 64 man years of effort and the front wheel drive works some 36. Both films are supported by technical papers so that reporting may be more detailed than would otherwise be possible in the allocation platform time.

The test and performance criteria used in our research were provided by the Transport and Road Research Laboratory and derived from their accident investigations. It must be stated that the knowledge of rural road accidents and of human tolerance needs continual expansion so that development criteria can be updated. This we regard as vital work in the business of governments. Our endeavor is to produce safe vehicles at realistic prices. So, therefore, a clear definition of identified benefits is vital if we are to avoid excessive cost and the wastage of materials.

We have learned a considerable amount in our programs and expose the fact that we have much more to learn. We are, however, suitably enthusiastic regarding our ability to improve vehicles with respect to better protecting occupants, offering some alleviation to the pedestrian problem, and making worthwhile gains in accident avoidance. Unfortunately we see little possibility of introducing many of our ideas into production vehicles through the evolving maze of regulations which are often ill-founded, seldom rationalized, and inevitably must give poor benefit return for cost expended.

Our work confirms that by using the belts provided, the car occupant is already well protected except in side collisions. The key solution to this later situation lies in locating bumpers at the correct height, so that an impacted vehicle is struck where its main structure exist and below its center-of-gravity. It is not surprising that such a height is found to be correspondingly correct if cars must strike pedestrians, as both situations and locations involve human dimensions. It appears from our development work that producing safer cars for the future must be impeded by the dimensional content of Federal Motor Vehicle Standard 215 which gives the child pedestrian no chance, and insures no possible economic solution to side collision protection.

BRITISH LEYLAND/TRRL EXPERIMENTAL SAFETY SYSTEMS CONTRACT

PETER M. FINCH
Safety Test and Development Department
British Leyland Motor Corporation Limited

INTRODUCTION

In the Spring of 1972 a contract was signed between the Transport and Road Research Laboratory and British Leyland. The contract was for the detailed development of six separate but interdependent safety projects; a 'Systems' approach in accordance with the first-phase requirements of TRRL's E.S.V. programme. (FIG. 1.)

Work on the entire contract was divided into three separate development periods:

A. 'STANDARD VEHICLE'. Involving an agreed series of impact tests on production vehicles, to establish the degree of modification necessary to meet performance targets.

B. 'PHASE I' covered the design, manufacture and testing of modified vehicles to meet the structural and injury requirements.

C. 'PHASE II' concerned with their further improvement and sophistication, and the incorporation of measures to mitigate the severity of pedestrian impacts.

In designing for these contracts our project objectives were related to frontal and side impacts only. The logic of this becomes apparent from a perusal of Figure 2, which shows that about 80% of all United Kingdom car accidents, and approximately 90% of their resulting fatalities, may be attributed to side or frontal accidents.

BRITISH LEYLAND
UK LIMITED–TRRL
SAFETY RESEARCH SYSTEMS

TRRL PROJECT Nos.

3	Frontal Impacts
4	Side Impacts–2 Door Vehicles
5	Occupant Protection–Head Impacts
6	Side Impacts–4 Door Vehicles–Fixed Seats
7	Occupant Protection–Knee/Leg Impacts
8	Occupant Protection–Steering Wheel Impacts

Figure 1

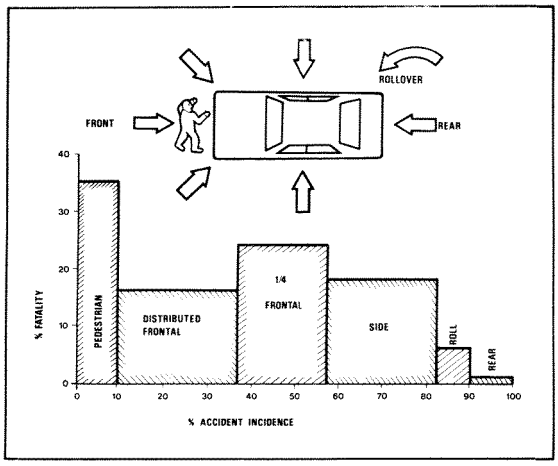


Figure 2

The elimination of injury is of course the whole purpose of the ESV programme. Ideally it should extend equally to all road users, but for the unprotected pedestrian or cyclist it is clearly impossible to provide the same degree of protection as can be achieved for the occupants of enclosed vehicles.

The only real answer is to completely segregate the various types of road user according to their accident vulnerability.

Since this too is obviously fraught with immense difficulties, we intend, under Phase II of our contract, to study the question of 'Pedestrian' versus 'Motor car' in some detail. Fortunately for car occupants the problem is less complex, two factors however are of paramount importance for the reduction of injury, irrespective of the accident situation.

Firstly, protection from crushing or ejection, by maintaining the structural integrity of the occupant compartment; and secondly protection during high velocity impact against the hard interior of the vehicle, by incorporating a combination of personal restraint systems and energy absorbing interior trim.

We have used two types of dummies for the assessment of injuries, they are the Alderson VIP 50A and a special TRRL dummy specifically designed for measuring side impact skeletal loads. (Fig. 3.) The required performance criteria, which was established by TRRL from their experiments into human tolerance levels, is more demanding than that recommended for the original ESV Programme. (TABLE I.)

Slight modifications to the original list of collision modes were also considered desirable, in view of the United Kingdom's accident statistics.

Frontal pole impacts on the centre line of the vehicle were felt to be less important than offset impacts into the radiused corner of a barrier, and so the front pole test was eliminated, but additional tests, in which one half and one quarter of the car overlapped the barrier face, were included. All the impact cars were equipped with two dummies generally representing a front and rear passenger, since the measurement of the steering column penetration precluded the use of a dummy in the driver's seat. We intend to include the driver in the Phase II tests, now that work on the reduction of column penetration has been completed.

With the exception of the project concerned with energy absorbing steering assemblies, it was decided that all development work be conducted on the 1.8 Morris Marina Saloon. This vehicle is fitted with a conventional engine and transmission layout, and has a kerb weight of 2030 lb for the two door version and 2060 lb for the four door. It was also decided that apart from the bumper/grille-surround area, no changes to the outer skin surfaces would be undertaken on the Phase I vehicles, all modifications being contained within the existing envelope.

Production vehicle tests were completed prior to the 1973 Kyoto presentation, and in April of this year the Phase I tests were finalised. Completely satisfactory results were obtained in the various impact modes with respect to both structural integrity and occupant injury, except for one instance where a possible hip fracture was indicated.

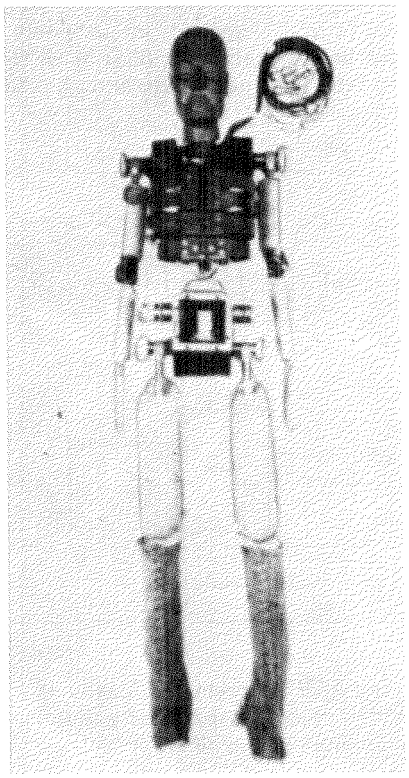


Figure 3

TOLERABLE LOADS AND INJURY CRITERIA SUGGESTED BY TRRL	
HEAD	H.I.C. < 1000
SHOULDER	6.0 KN (1350 lbf)
CHEST	60g for 3 millisec
RIBS	1.0 KN (225 lbf)
PELVIS	5 KN (1125 lbf)
FEMUR	4.4 KN (1000 lbf)

Table I

PHASE I DEVELOPMENT

TRRL PROJECT No. 3 – FRONTAL IMPACTS

Set against world road casualties British fatalities appear relatively small, even so a substantial proportion of them is preventable. (FIG. 4 MACKAY.)

Of the 257,000 annual British accidents involving private cars approximately 57% may be classified as frontal impacts of one sort or another.

Fortunately cars involved in 85% of European injury producing frontal accidents, suffer damage approximately equivalent to that sustained in barrier impacts up to 50 km/h (31 mph) and 95% at up to 64 km/h (40 mph). This upgrading by 33% in the barrier speed results in an improved accident coverage of only 10%, for an increase in the vehicle's Kinetic energy of 78%.

Country	TOTAL No. Injured	TOTAL No. Killed
West Germany	531,795	19,193
France (1969)	331,273	14,664
Italy	228,236	10,208
United Kingdom	355,866	7,498
E.E.C. Countries	1,660,068	58,151
Western Europe	1,958,623	71,545
U.S.A. (1971)	2,000,000	54,600
Canada	161,000	5,261
Japan (1967)	655,377	17,492

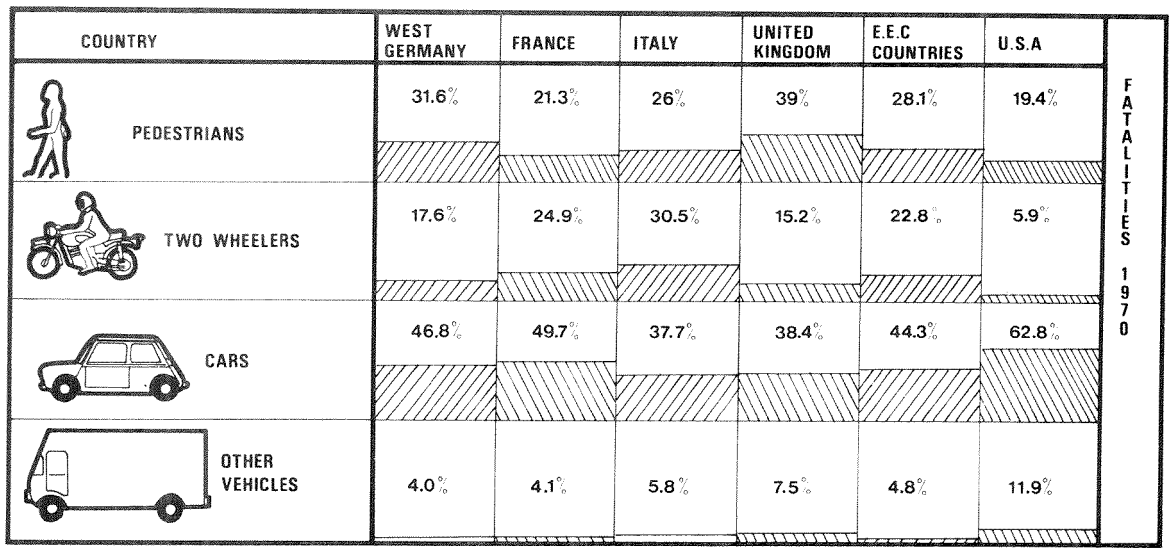


Figure 4

The disproportionate increase is because the Kinetic energy rises as the square of the speed of the vehicle. (FIG. 5.) If the equivalent barrier speed is amplified by 66% to 80 km/h (50 mph) the kinetic energy requiring to be absorbed by the vehicle structure increases by 178%, yet the percentage of fatalities covered improves by only 13%. (FIG. 6.)

To design a body capable of absorbing these higher kinetic energies would obviously result in a considerable increase in structural complexity and vehicle weight; both directly related to increased costs. It was therefore considered sensible to aim at an economically feasible design based on cost/benefits, rather than design an armoured vehicle that few could afford. For development purposes a frontal barrier impact speed of 64 km/h (40 mph) was therefore agreed.

However, since frontal car collisions involve a variety of obstacles and impact angles, a series of differing barrier configurations were devised as basic tests to assess both the standard vehicles, and the effectiveness of our subsequent designs.

These tests are:

(a) Low speed impacts, to determine repair costs.
(b) High speed full frontal impacts, at 64 km/h (40 mph).
(c) 30° angled impact at 50 km/h (31 mph).
(d) ½ offset barrier impact at 50 km/h (31 mph).
(e) ¼ offset barrier impact at 50 km/h (31 mph).

At this point it is perhaps worth recalling that a fixed barrier acts rather like a mirror, reflecting any object placed before it; similarly a vehicle impacting into a fixed barrier, suffers the same degree of damage as would result from two such identical vehicles colliding head-on; at a closing speed of twice that of the barrier impact speed.

ENERGY ABSORBING BUMPERS

The safety aspects of energy absorbing bumpers lie mainly in the reduction of pedestrian injuries and improved vehicle-to-vehicle compatibility. A third attribute, the elimination of damage in low speed impacts, cannot be classified directly under the heading of safety; but in the terms of costs to the community, vehicle repairs are of considerable economic importance. As a preliminary therefore to the

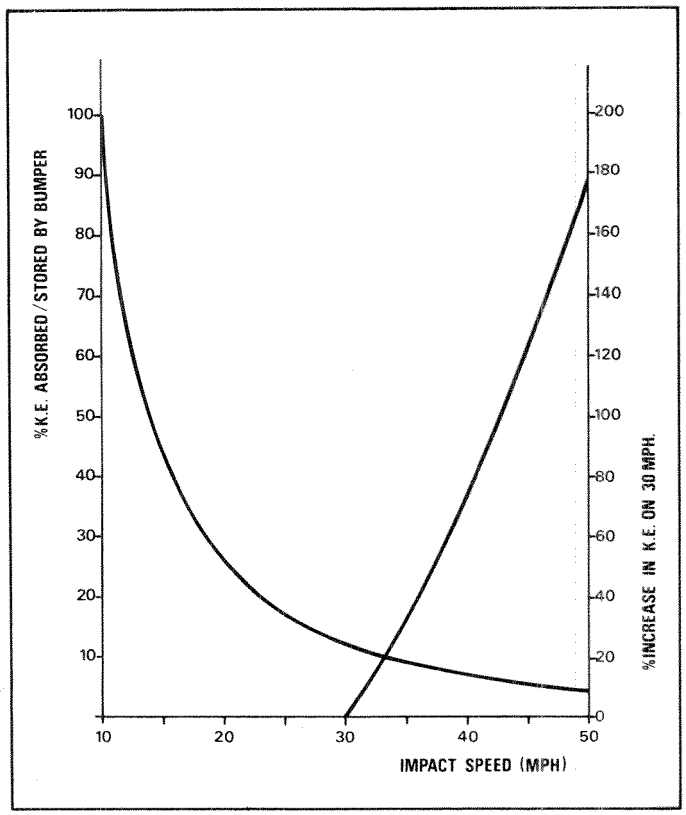


Figure 5

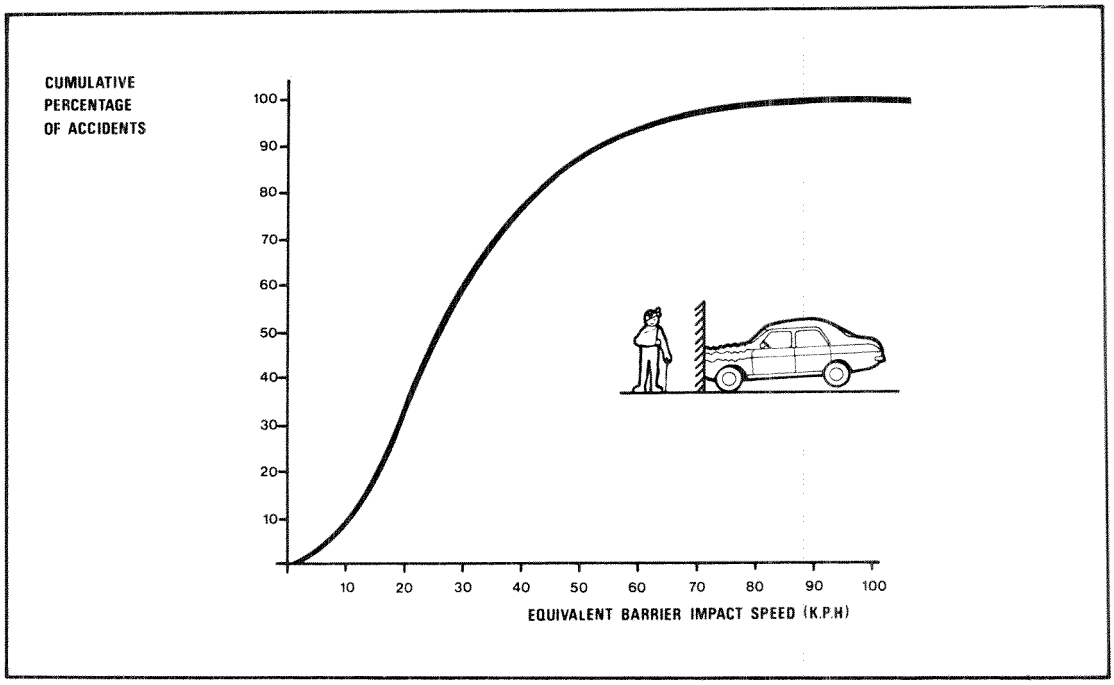


Figure 6

high speed safety tests, our programme included investigations into the reduction of damage resulting from low speed impacts.

Structural damage resulted from impact with the barrier, whilst the flexibility of the engine mountings allowed the engine to surge forward against the radiator. The former was overcome by the inclusion of an energy absorbing bumper system coupled with a stiffened frontal structure, and the latter by attaching the radiator to the engine and restraining their forward movement.

Two bumper systems have been tested and two more are in the development stage. One of those tested comprises two 'Dunlop Polymer' bonded rubber 'V' blocks combined with bolt-on energy absorbing cylinders. (FIG. 7.) The recoverable 'V' blocks cope with no-damage impacts up to 8 km/h (5 mph) and the non-recoverable cylinders accommodate limited damage impacts between 8 km/h (5 mph) and 16

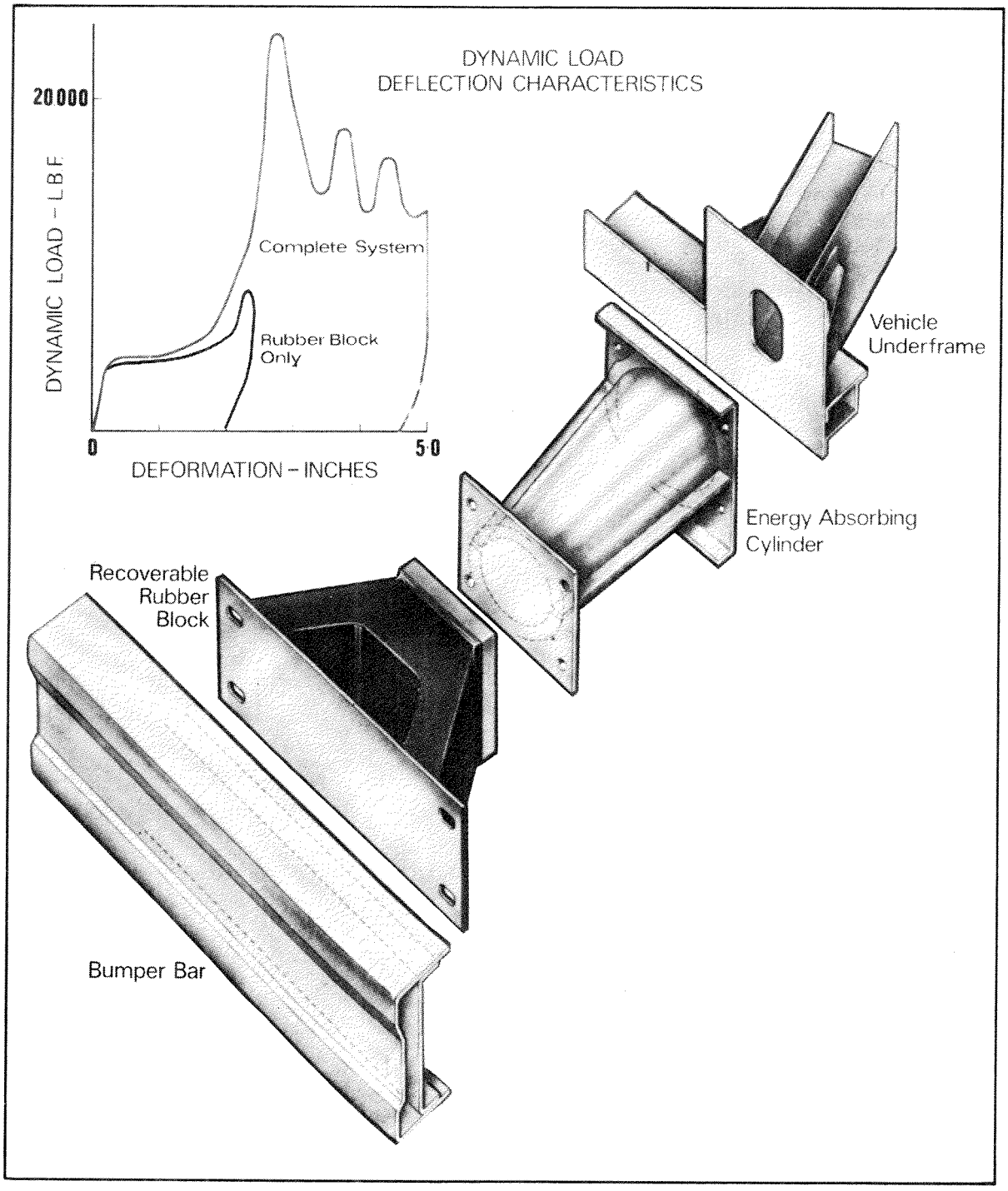


Figure 7

km/h (10 mph). The total damage being restricted to the cylinders which are designed as replaceable service items. The other system consists of an 'Enersorb' hollow section bumper comprising of layers of Nylon/Polyester fabric sandwiched in synthetic rubber and mounted on a Hy-Press steel armature: the stiffness of the bumper being dependent upon the distance between the fabric inter-layers. Incorporating this system with a Phase I structure enabled us to conduct barrier impacts up to 16 km/h (10 mph) without damage. This same combination offers improved vehicle-to-vehicle compatibility and was later successfully used in side impacts.

Complementary with both these systems, the Phase I vehicle incorporated moulded Urethane headlamp surrounds, with the intention of extending the no-damage frontal areas and reducing the vehicle's potential aggressiveness in corner collisions.

As the speed of impact increases the proportion of total kinetic energy absorbed by these bumpers is reduced: so that at a barrier speed of 64 km/h (40 mph) – equivalent to two Marinas striking head-on at 128 km/h (80 mph) – the energy absorbed by these non-velocity sensitive systems falls to about 6-7% of the whole. (FIG. 5.)

STANDARD VEHICLE SAFETY IMPACTS

The structural behaviour of the Standard Marina is particularly good in barrier collisions up to 50 km/h (31 mph), whereas at 64 km/h (40 mph) the damage is commensurate with the increase in Kinetic energy absorbed.

The major problems resulting from the 64 km/h (40 mph) impacts were concentrated in the degree of structural collapse of the engine compartment and floor assemblies, the intrusion of the passenger compartment, jammed doors and loss of windscreen, in addition to high dummy loads and accelerations; the remainder of the vehicle suffering little damage. (FIG. 8.)

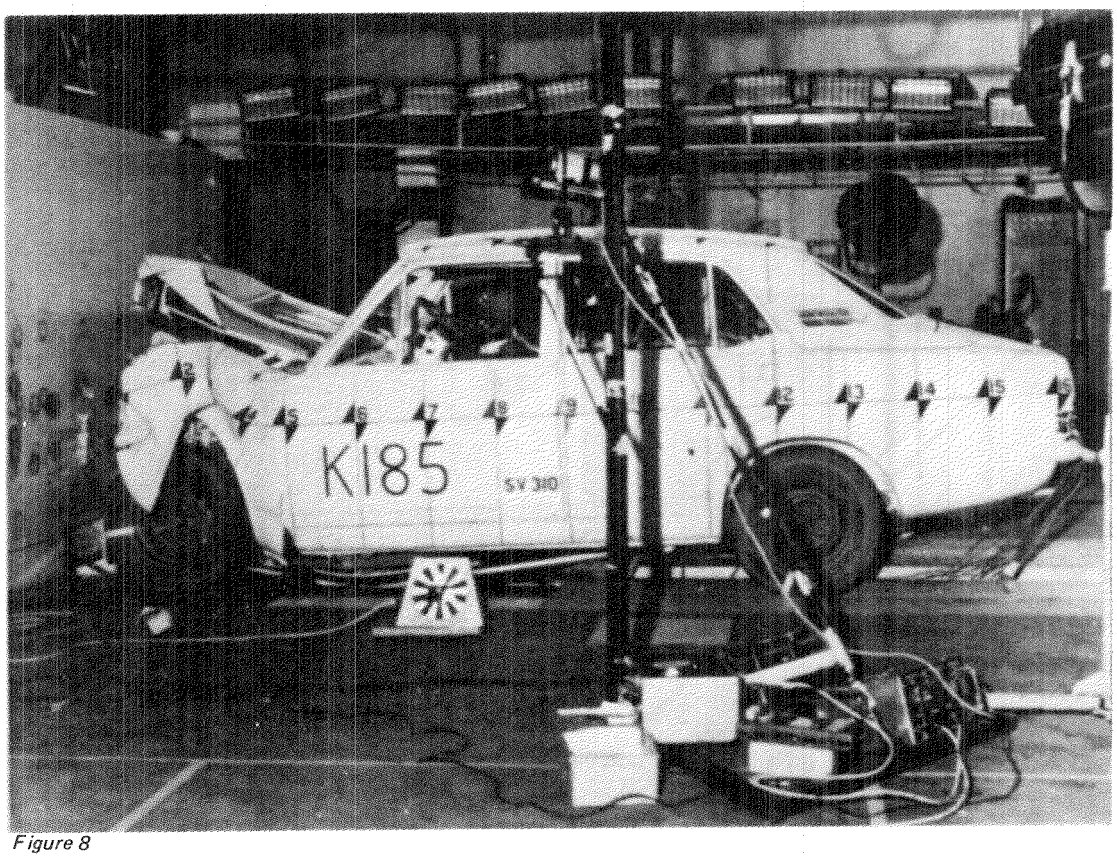


Figure 8

Results obtained from 30° angled (FIG. 9), ½ and ¼ offset impact at 50 km/h (31 mph) raised different problems, the latter test being structurally the more severe. (FIG. 10.)

Before attempting the redesign of the vehicle, it was first necessary to examine the sequence of events during the crashes and to comprehend the reasons for the impact severity.

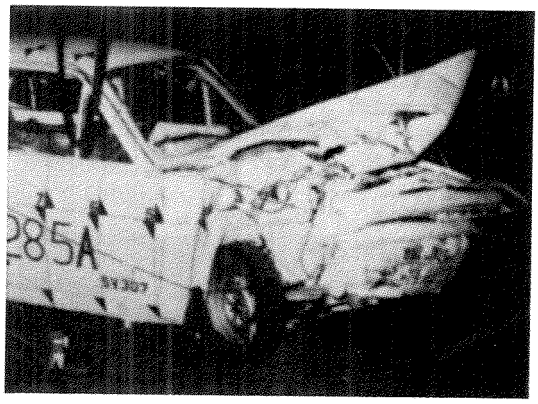


Figure 9

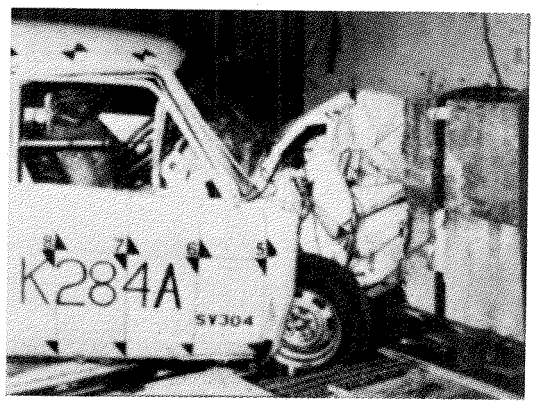


Figure 10

In this we were aided by instrumentation recordings and high speed film analysis. These studies emphasised a number of design features requiring modification, for example: the battery being located on the side of the valance panel forward of the rigid suspension tower prevented, to a large degree, the collapse of the adjacent structure, thereby transferring it rearward to the area of the decking panel; a factor which in turn contributed to the loss of the windscreen. Also, the engine penetrated the dash and forced the heater and facia assemblies into the passenger compartment, the remedy calling for improved structural design and engine relocation.

DEVELOPMENT DESIGN

The design of the entire structures, within the confines of the existing skin panels, was therefore appraised and the stiffness of each component and assembly assessed by calculation, crush or impact tests.

During these initial investigations, work commenced on a computer simulation model designed for full frontal impacts; this is now complete. With it our computer engineers are able to vary, within the model, the theoretical stiffness, location and mass of the vehicle structure and mechanical units; in order to obtain predictions of the probable behaviour under crash conditions of the projected designs and assist us in selecting the most favourable modifications.

Basically, the technique involves the solution of multi-mass spring system, in which the vehicle's major mass concentrations are linked together via a series of non-linear springs, representing the load deflection characteristics of the structure's principal crushable components. (FIG. 11.)

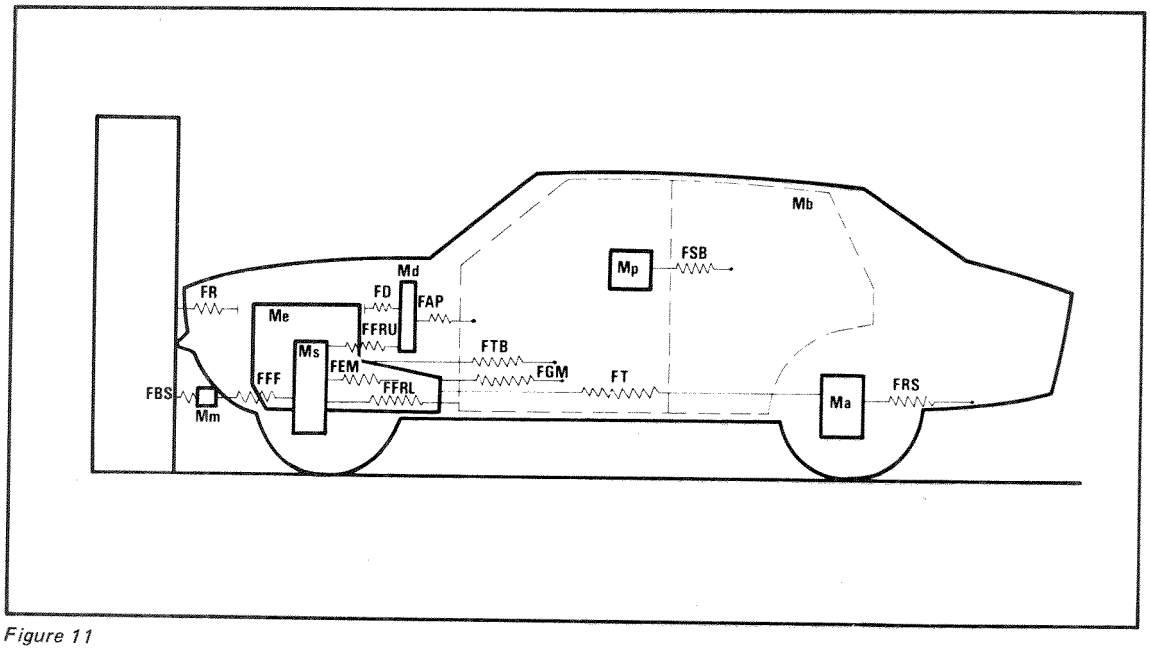


Figure 11

These stiffness characteristics are determined experimentally by means of crush rigs. (FIG. 12.) With simple components, such as longitudinals, direct end loading is perfectly satisfactory; but for complex structures a more sophisticated approach is often necessary. To obtain the stiffness characteristics of just one component of the computer model, for example the dash assembly (FD in FIG.11), the complete body shell was firmly supported in the rig and the engine loaded rearwards into the dash structure. Potentiometers and load cells measured the structural deflections and the imposed load distribution.

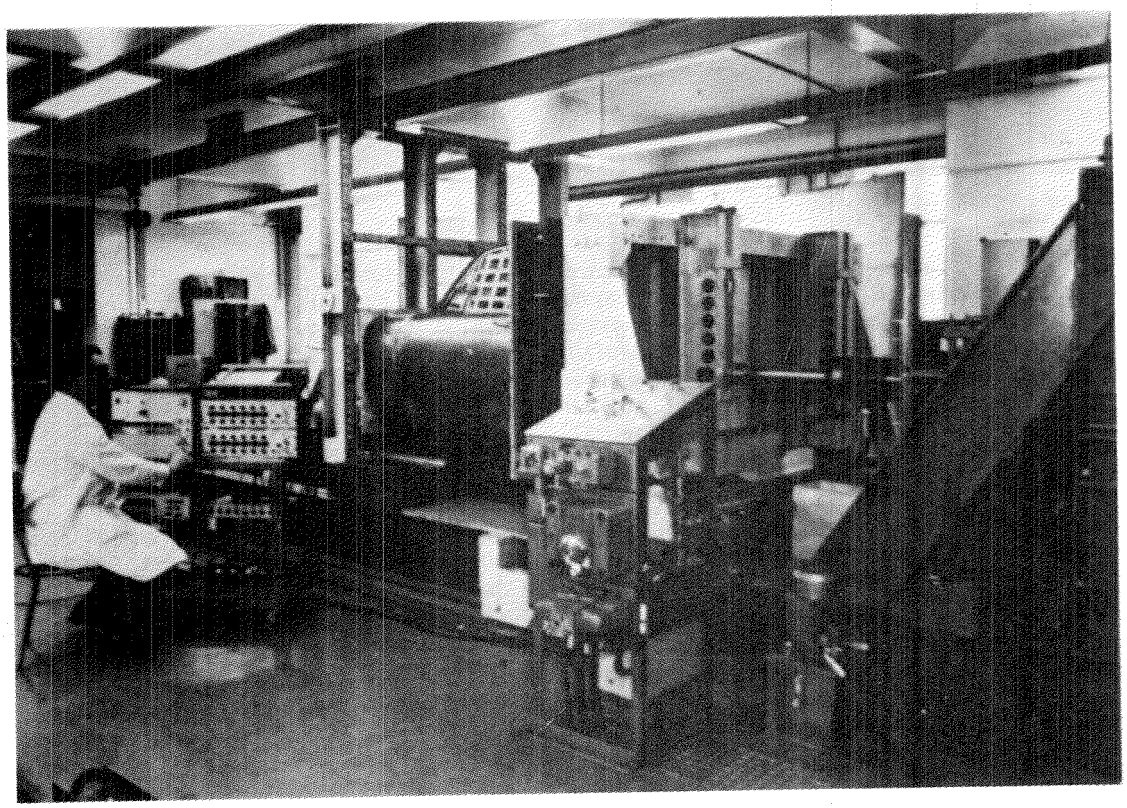


Figure 12

On occasions, we used crush testing as a direct means of solving design problems. For example, the Marina's front suspension is by torsion bars arranged horizontally fore and aft and parallel to the centre line of car; during the early part of the impact they acted as struts transmitting the impact load directly into the main cross-member, thus contributing to the floor damage. By introducing a slight curve into the bars, we were able to reduce their strut stiffness to be compatible with that of the redesigned cross-member, without loss of torsional ability. (FIG. 13.)

Investigations such as these enabled us to develop our Phase I design with some confidence. It became apparent that a radical redesign of certain areas was of the utmost importance if improved overall stiffness, coupled with the desired sequential frontal collapse, was to be achieved.

The first area requiring modification was the underframe/floor complex. The main longitudinals were redesigned to join with the sills in the area of the B-C post. Two smaller longitudinals joined with the main members under the dash and extended rearwards to the heel-board as part of the new tunnel assembly. Figures 14 and 15 compare the existing Marina underframe floor structure with the Phase I modifications incorporated in Projects 3 and 4.

The next items to consider were the side/door assemblies; here the sills were increased in sections the 'A' post to sill joint improved. Strut reinforcements were welded within the doors and posts to improve their fore and aft stiffness, and matching shims were added to the pillar faces to prevent jamming the doors by the collapse of the door apertures.

Finally the entire front end and dash assembly was redesigned, coupled with the relocation of the engine and mechanical components to improve clearances, and a project 5 facia was fitted. (TABLE II.) As a result of all these modifications, including an energy absorbing front bumper, the vehicle kerb weight increased by 163 lb to 2223 lb (8%) for the four door saloon version.

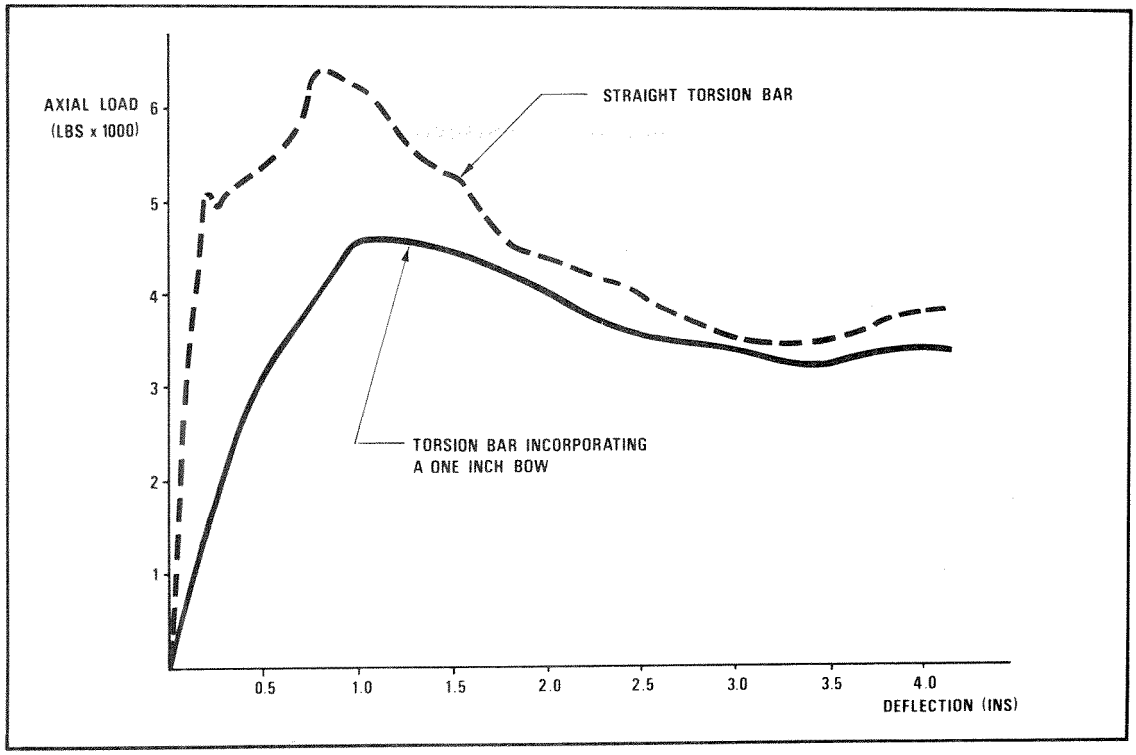


Figure 13

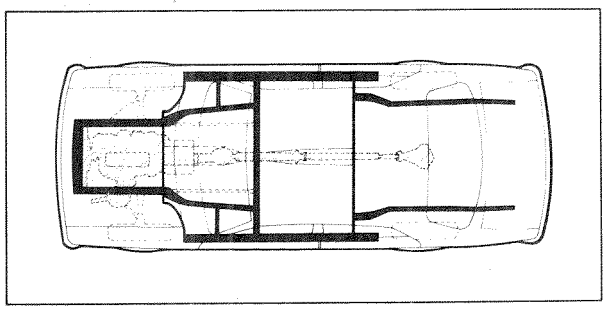


Figure 14

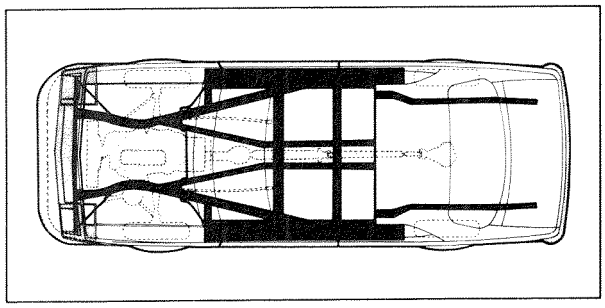


Figure 15

PROJECT 3 BASIC MODIFICATIONS

1. VEHICLE LENGTH EXTENDED 9".
2. REDESIGNED UNDERFRAME (SIDEMEMBERS, DASH, TUNNEL, FRONT FLOOR, VALANCES, CROSSMEMBERS ETC.) AND ADDITIONAL SILL AND 'A' POST REINFORCEMENT.
3. BEAMS IN DOORS AT WAIST LEVEL WITH SHIMS ATTACHED TO PILLARS.
4. FOAMED 'A' POST TO SILL JOINTS.
5. ENGINE MOVED FORWARD 2.5" AND DOWNWARD 1.5".
6. RADIATOR MOUNTED TO ENGINE.
7. ENGINE TO DASH LINKS.
8. HEAVY DUTY TORSION BARS INCORPORATING 1" BOW.
9. MODIFIED HEATER, HAND BRAKE, AND FRONT SEAT.
10. BATTERY RE-LOCATED ONTO DASH.
11. UNIVERSAL JOINT IN STEERING COLUMN LOWER.
12. 'ENERSORB' 10 MPH. FRONT BUMPER ASSEMBLY.
13. SELF SKINNING POLYURETHANE HEADLAMP SURROUNDS.
14. PROJECT 5 FACIA.

Table II

It should be reiterated that with our Phase I vehicles no styling studies were carried out, the objectives being exclusively the collection of acceptable crash design data.

PHASE I IMPACTS

Three modified vehicles were subjected to tests identical with those carried out on the standard vehicles. As before they included two dummies restrained by static seat belts, the front belt incorporating 1800 lb tear webbing in the diagonal loop and 1500 lb tear webbing in the lap.

The tests were completely successful in meeting all TRRL injury criteria and structural integrity requirements. The final results, with those obtained from the standard tests, are given in Table III.

COLLISION TYPE / Maximum Test Measurements	90° FRONT 40 MPH		30° ANGLED 30 MPH		½ OFF SET 30 MPH	¼ OFF SET 30 MPH	
	STANDARD	PHASE I	STANDARD	PHASE I	STANDARD	STANDARD	PHASE I
STRUCTURAL — FACIA INTRUSION	5.9 ins.	1.0 ins.	0.7 ins.	ZERO	2.1 ins.	3.4 ins.	0.1 ins.
TOEBOARD INTRUSION	11.7 ins.	1.7 ins.	5.4 ins.	1.1 ins.	9.0 ins	11.2 ins.	0.9 ins.
FLOOR INTRUSION	5.9 ins.	0.6 ins.	4.5 ins.	0.12 ins.	3.9 ins.	11.5 ins.	0.2 ins
STEERING COLUMN PENETRATION (DYNAMIC)	7.0 ins.	1.2 ins.	1.7 ins.	NOT MONITORED	2.8 ins. (STATIC)	8.6 ins.	0.42 ins
DOORS OPENABLE	NONE	FOUR	FOUR	FOUR	FOUR	TWO	FOUR
WINDSCREEN RETENTION	ZERO	100%	90%	100%	65%	50%	100%
FUEL INTEGRITY	OK	OK	SLIGHT LEAKAGE	OK	OK	SLIGHT LEAKAGE	OK
HEATER INTRUSION	11.0 ins.	0.5 ins.	2.3 ins.	ZERO	6.4 ins.	5.3 ins.	ZERO
FRONT PASSENGER DUMMY HEAD INJURY CRITERIA	1250	850	289	388	500	159	121
CHEST ACCELERATION	60 g . 0.5 m/sec.	53 g peak	25 g peak	25 g peak	60 g 0.8 m/sec	14 g peak	17 g peak
FEMUR LOADS (LH)	2050 lbf	988 lbf	INSTRUMENTATION FAILURE	388 lbf	INSTRUMENTATION FAILURE	150 lbf	184 lbf
FEMUR LOADS (RH)	1590 lbf	492 lbf	575 lbf	191 lbf	333 lbf	355 lbf	171 lbf

Table III

The deceleration pulse recorded at the B-C post, in the 64 km/h (40 mph) 90° Frontal impacts, is shown in Figure 16 together with the predicted pulse shape obtained from the computer simulation. Apart from slight changes in phase relationship it depicts a good correlation between the two traces; showing the value of computer simulation techniques in designing structures to meet impact requirements. These techniques permit an infinite number of variations to be conducted on the specified parameters, without resorting to continuous dynamic testing.

Following the simulation of these vehicle-to-barrier tests, we are now engaged in extending the model to incorporate car-to-car impacts involving vehicles of dissimilar structure and mass. The successful analysis of this, the most common form of frontal impact, will enable us to consider designs for vehicle-to-vehicle compatibility.

A comparison of the two front ends after the 64 km/h (40 mph) barrier impacts, illustrated the improvements obtained from the redesigned structure and shows the energy absorbing bumper system virtually undamaged. (FIG. 17 and 18.)

The windscreen, which was completely ejected on the standard vehicle (FIG. 19) was fully retained on the Phase I (FIG. 20). In both cases the screen was dry glazed and the windscreen aperture unmodified. Retention was assured by preventing distortion to the decking panel and windshield aperture.

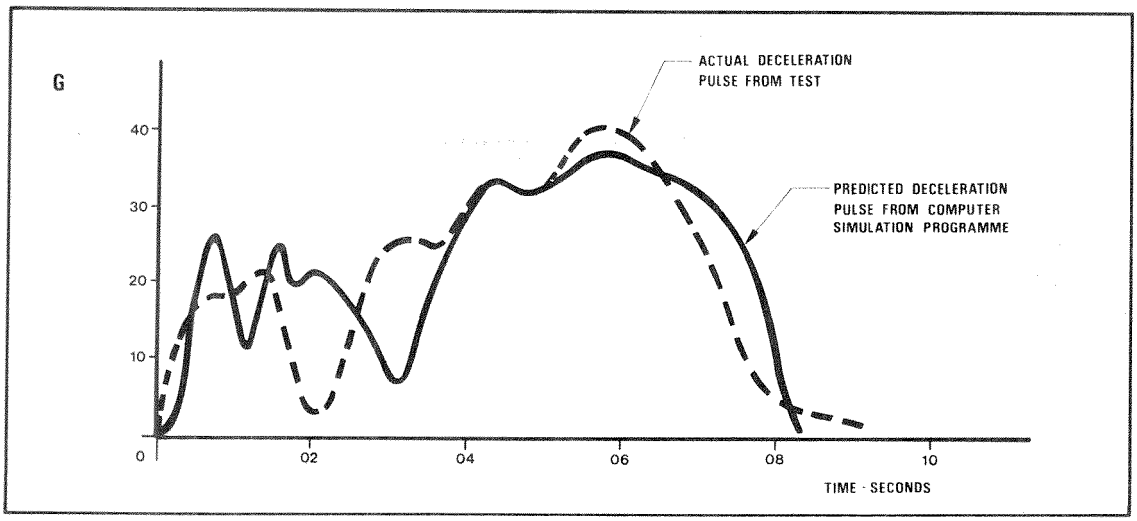


Figure 16

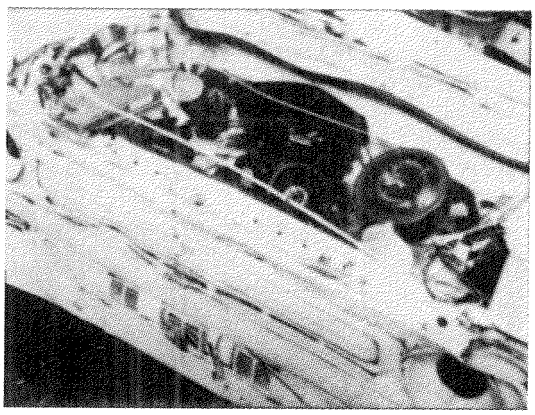


Figure 17

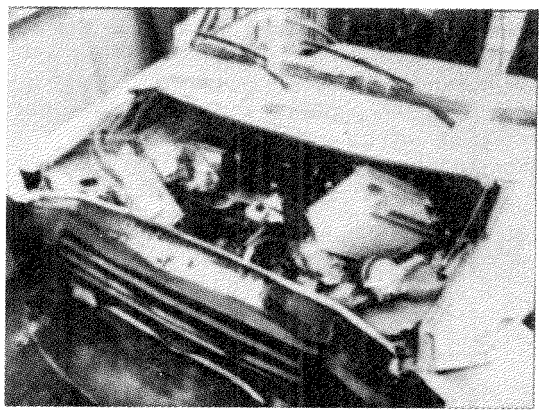


Figure 18

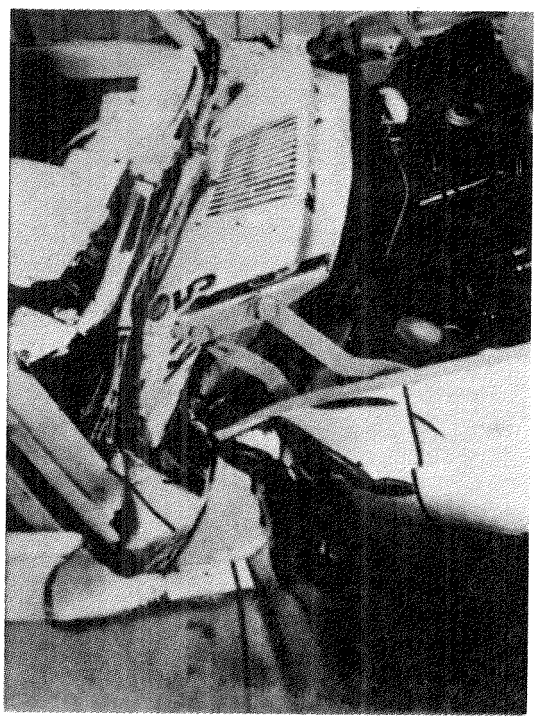


Figure 19

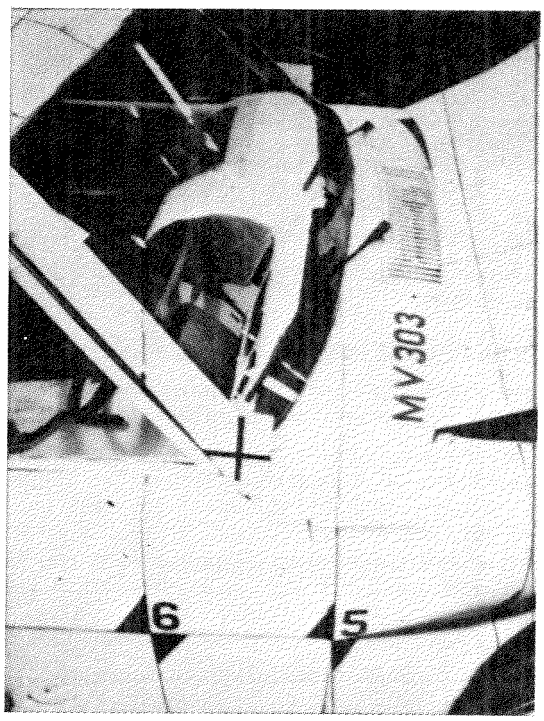


Figure 20

The Standard Marina (FIG. 21) suffered considerable intrusion of the passenger compartment from the facia, heater, steering column and floor panel; the corresponding damage to the Phase I vehicle (FIG. 22) was almost non-existent.

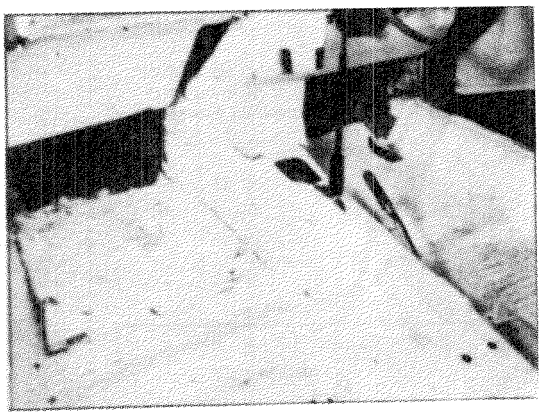


Figure 21

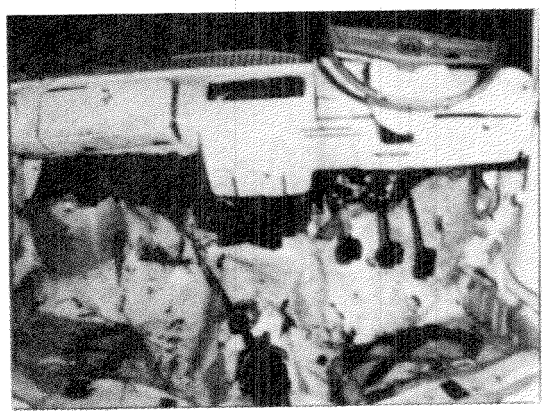


Figure 22

Similar improvements were obtained in the 30° angled barrier and the ¼ offset barrier impacts at 50 km/h (31 mph). Tests on the Standard Marina show that structurally, the latter was the more severe (FIG. 23); but in the identical Phase I test the damage was dramatically reduced, partly owing to the structural alterations, and partly to the considerable resistance offered by the bumper assembly. This tended to deflect the vehicle and prevent the structure from wrapping itself round the barrier as occurred with the standard vehicle. (FIG. 24.)

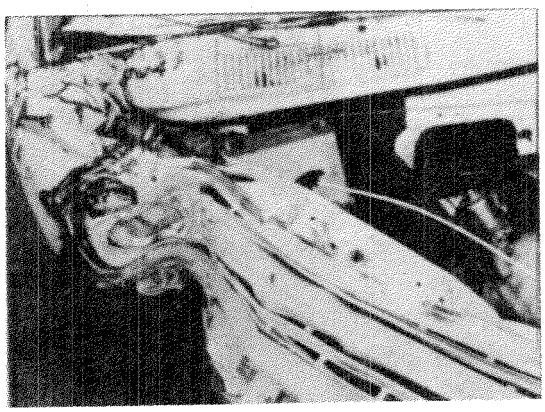


Figure 23

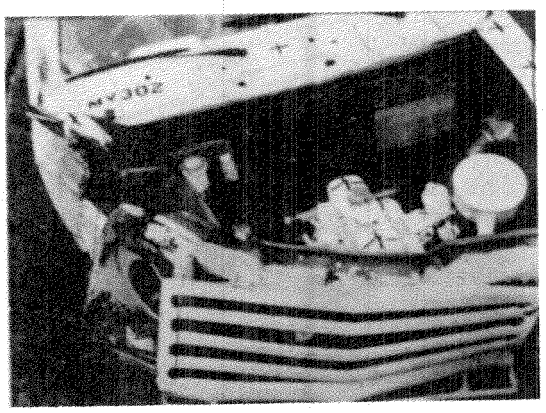


Figure 24

The ½ offset test was not repeated on the Phase I vehicle.

Following these frontal impacts all passenger doors on the Phase I vehicles could be opened without the use of tools.

TRRL PROJECTS 4 AND 6 – SIDE IMPACTS ON TWO AND FOUR DOOR SALOONS

The obvious similarity between side impacts involving two and four door saloons makes it relevant to discuss these two projects together.

Side impacts, whether involving two road vehicles or a single vehicle against a roadside obstacle, generally result in severe intrusion of the occupant compartment of the struck vehicle. This is due to the comparative lack of protection afforded by the car's door and side structure, coupled with the incompatible mass, shape and stiffness of large opposing objects. Seated occupants, particularly those immediately adjacent to the point of impact, may receive injuries directly as a result of this intrusion. With this in mind it was decided to conduct a dual investigation, Project 4 concentrating on the Marina two door and Project 6 the four door version. The overriding difference being, that in the two door the tip-up front seats were retained; whereas the seats in the four door were to be designed integral with the floor and side panel assemblies.

The latter approach appeared to offer the ultimate solution to the question of providing sufficient structural resistance to lateral impacts. But in both cases a second problem had to be considered.

Irrespective of intrusion, air bags and seat belt systems give insufficient protection for occupants adjacent to the impacted side. The deceleration forces, induced by the occupants sudden impact against the interior, can only effectively be reduced by the use of internal energy absorbing panels or cushioning. This has however the disadvantage of either taking up occupant space or increasing the overall width of the car – (see Projects 5 and 7).

In Great Britain side impacts constitute approximately 25% of all injury producing accidents, of which about 60% are car-to-car. The three most severe situations likely to be encountered in service were reproduced by the following laboratory tests:

(a) 45° car-to-car at 37 km/h (23 mph).
(b1) 90° car-to-car at 50 km/h (31 mph).
(b2) 90° mobile barrier-to-car at 40 km/h and 50 km/h (25 mph and 31 mph).
(c) Car-to-pole at 16 km/h and 24 km/h (10 mph and 15 mph).

The mobile barrier-to-car impacts were added in order to compare this type of experimental test with the 90° car-to-car test.

INITIAL SAFETY IMPACTS

In our first test, a 45° car-to-car impact at 37 km/h (23 mph), the bullet car complete with two dummies was aimed to strike a similarly loaded target car in line with the rear of the front door hinge pillar. The resultant damage was mainly confined to the door and 'B-C' post. The injury criteria recordings were within the TRRL requirements on all dummies.

Our next test, the 90° car-to-car at 50 km/h (31 mph) was infinitely more severe, the resultant structural damage extending over the entire side panel. The 'B-C' post was almost torn from the collapsed sill and cantrail, and the doors forced over the sills. (FIG. 25.) Compartment intrusion amounted to 14.25 inches. The front seat TRRL dummy recorded a series of severe fracture levels, probably amounting to an occupant death. On the other hand, levels recorded on the rear Alderson dummy were satisfactory.

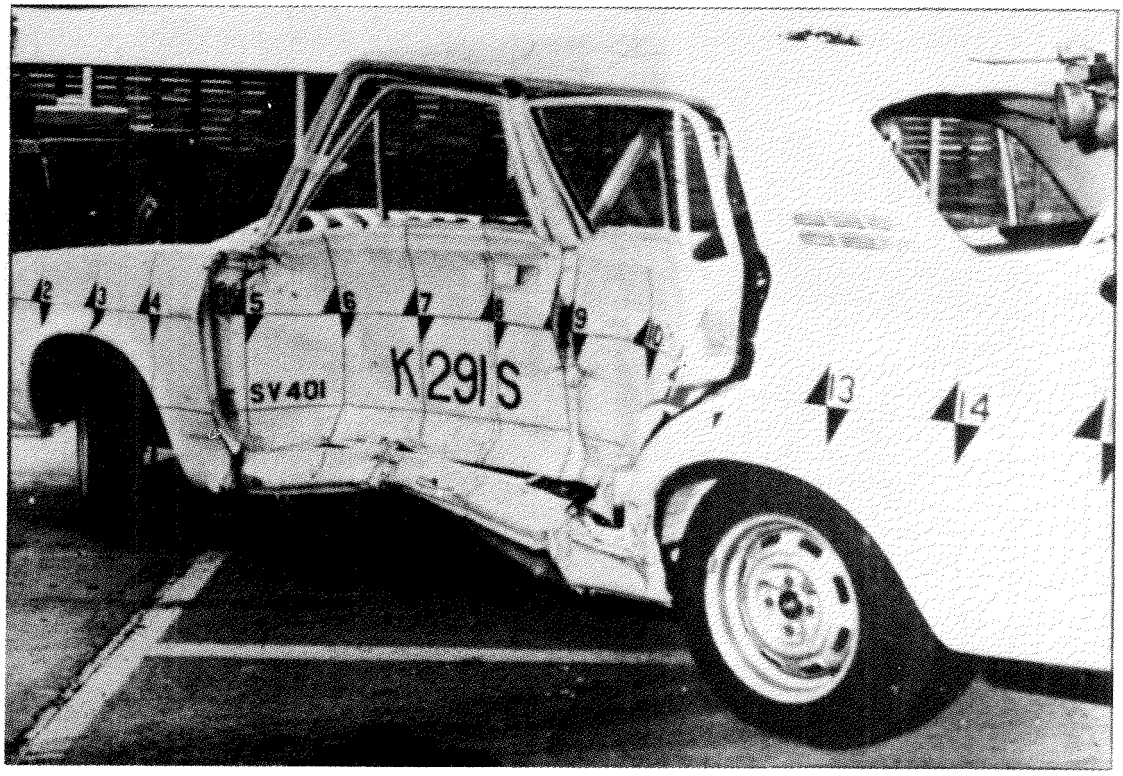


Figure 25

Comparative 90° tests using a mobile barrier and bullet car proved incompatible, due to the inflexibility of the rigid barrier face as compared with the collapsible frontal structure of the bullet car. An analysis of the two 50 km/h (31 mph) impacts shows that the severest occupant injuries and greater structural damage were caused by the mobile barrier; and only by reducing the speed of the latter to 40 km/h (25 mph) did the dummy loads become reasonably similar, although structural damage was still not representative.

After these initial investigations the use of the mobile barrier was discontinued. (FIG. 26.)

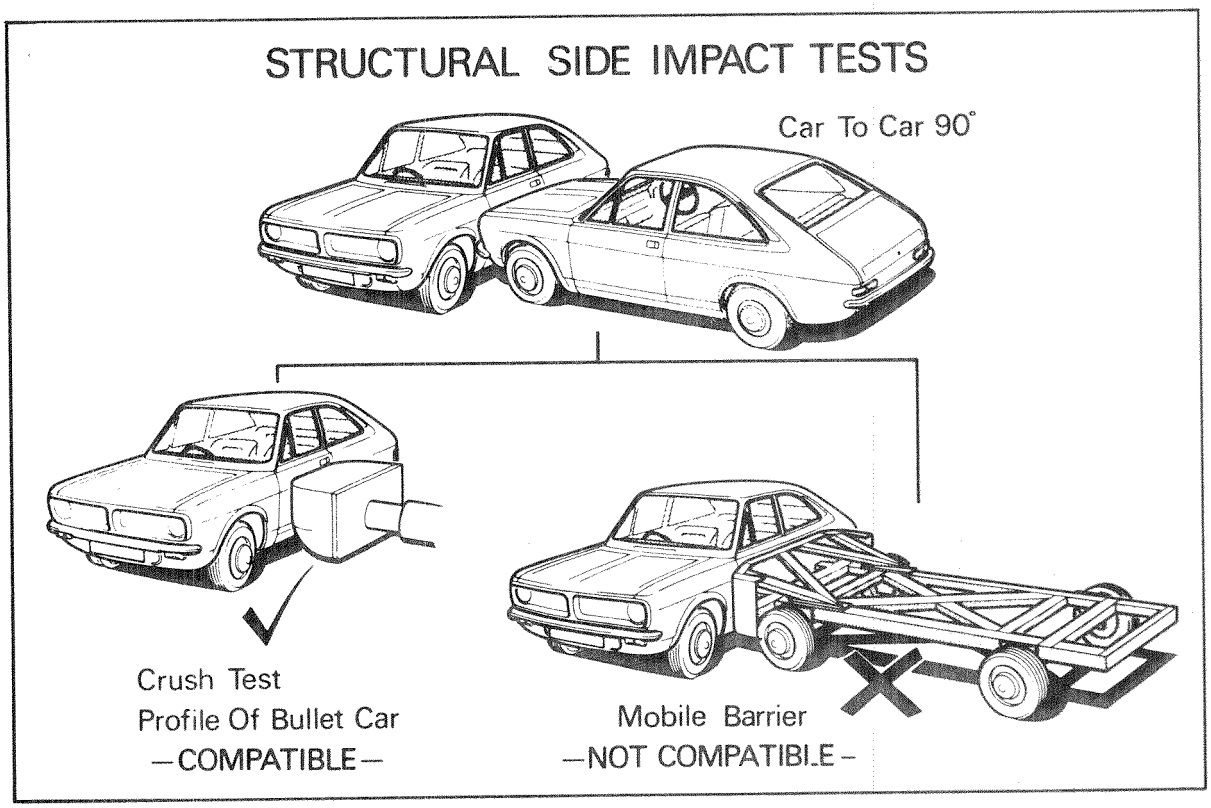


Figure 26

The final test series involved sliding cars laterally into a fixed 14 inch diameter pole at 16 km/h (10 mph) and 24 km/h (15 mph). (FIG. 27.)

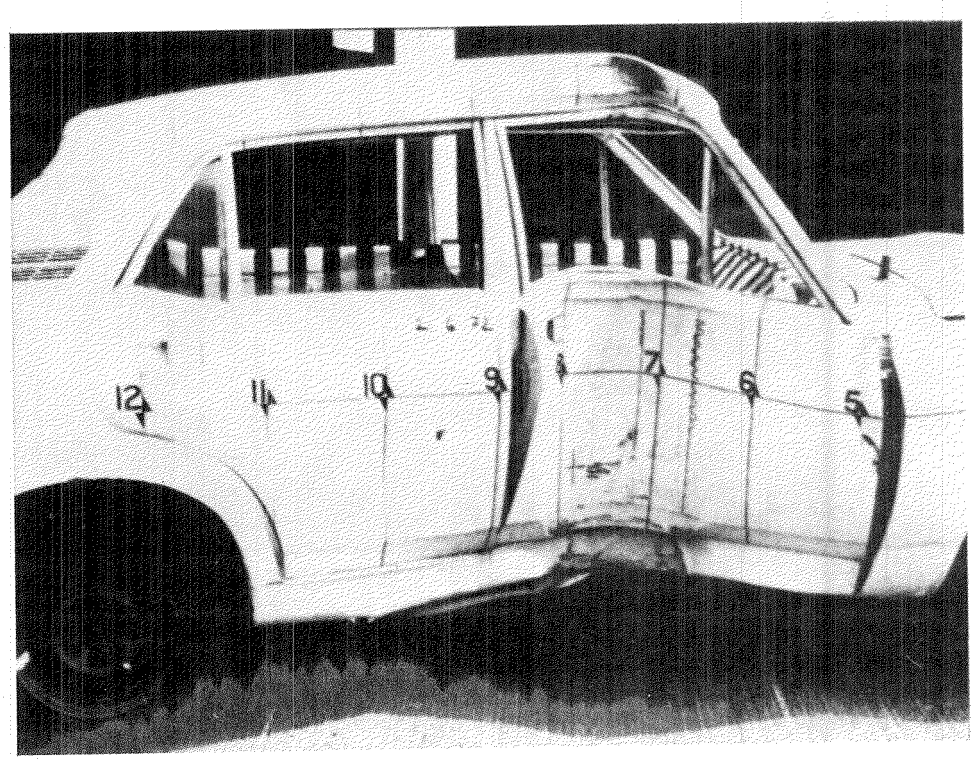


Figure 27

Structurally both cars simply wrapped themselves around the pole to a degree dependent upon the speed of impact. The maximum intrusions were 7.25 and 13.75 inches respectively. A possible hip fracture was recorded at 16 km/h (10 mph) and a very severe one, probably accompanied by internal injuries, at 24 km/h (15 mph). This increase in hip loading cannot however be entirely attributed to the speed differential, since in the former test the centre line of pole was located four inches forward of the dummy's hip, whereas in the latter test the centre line was relocated to coincide with this very vulnerable area.

DEVELOPMENT DESIGN

As computer simulation models were not available for side impacts, we proceeded with an extensive series of load tests on the side structures and transverse members, in order to assess the effect of various modifications on individual components. For example, it was obvious that the sills and 'B-C' post sections on the standard car would not offer adequate resistance to cars striking them at 90° and at speeds as high as 50 km/h (31 mph). Considerable improvements to these components were therefore essential.

The basic sill stiffness for example was determined by centrally loading a length of section to destruction. The section itself was increased in size and modified to include a vertical wall intended to resist the intrusion of the door inner panel. Finally combinations of the sill box sections were filled with rigid polyurethane foam. The results are shown in Figure 28.

With regard to the 'B-C' posts both the two and four door assemblies were redesigned to incorporate larger sections, increased metal gauge and greatly improved sill and cantrail ties.

The effect of various Project 4 'B' post modifications is shown in Figure 29; again illustrating the advantages of foam in delaying the collapse of small thin wall sections. However the improved sills and 'B-C' posts would not in themselves have been a sufficient barrier against side intrusion without improvements to the

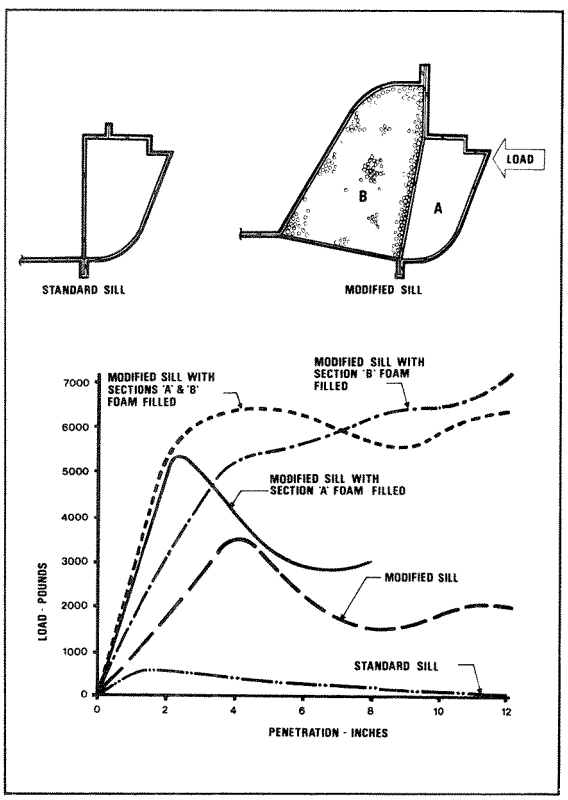


Figure 28

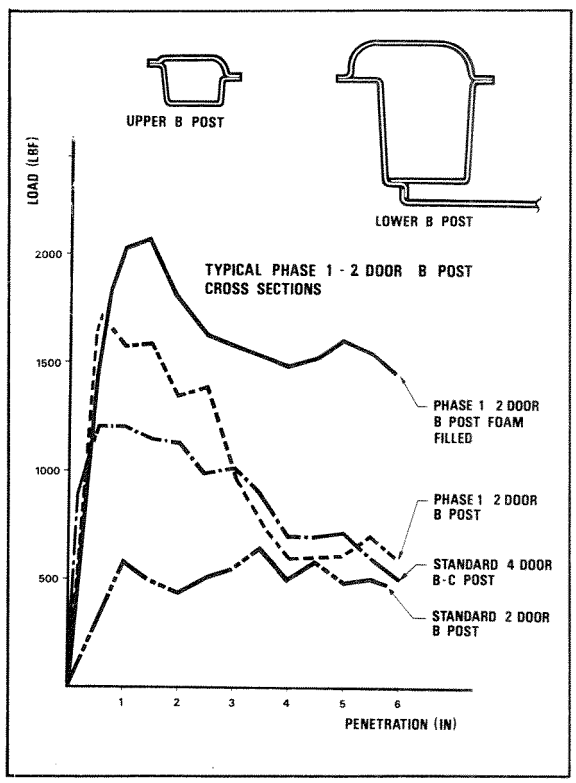


Figure 29

structure's transverse stiffness. This was achieved by the addition of a roof bow linking the cantrails at the upper 'B-C' posts and increasing the floor cross-members. A bolt-on assembly was added bridging the underside of the tunnel. (FIG. 30.) Finally guardrails and anti-penetration buttons were added to the door assemblies. (FIG. 31.)

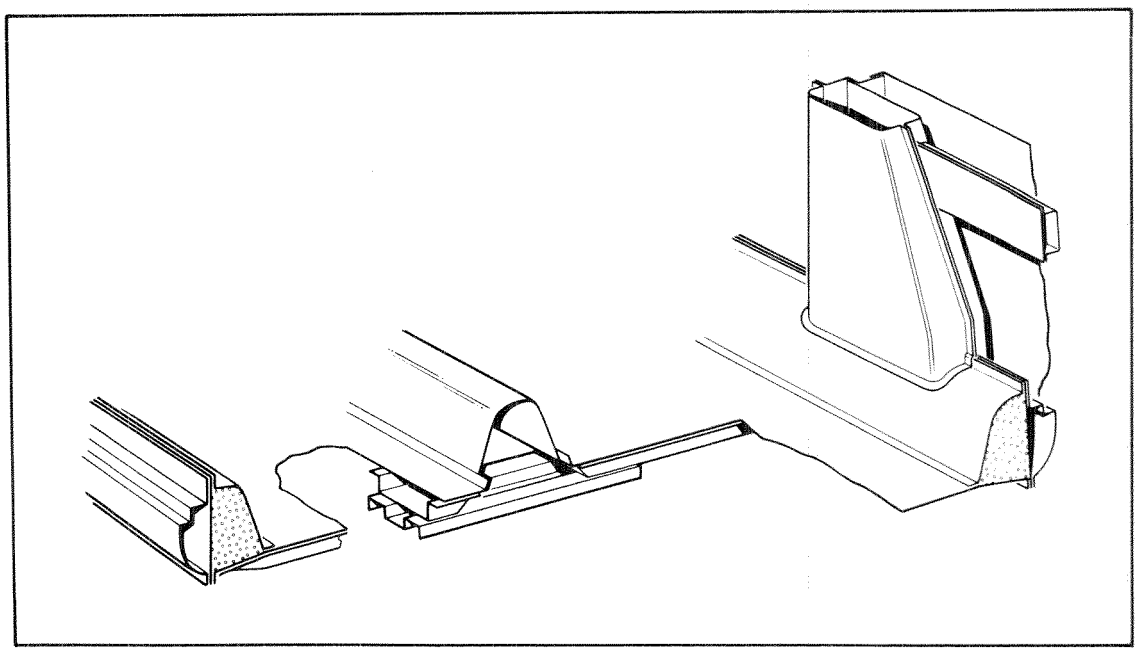


Figure 30

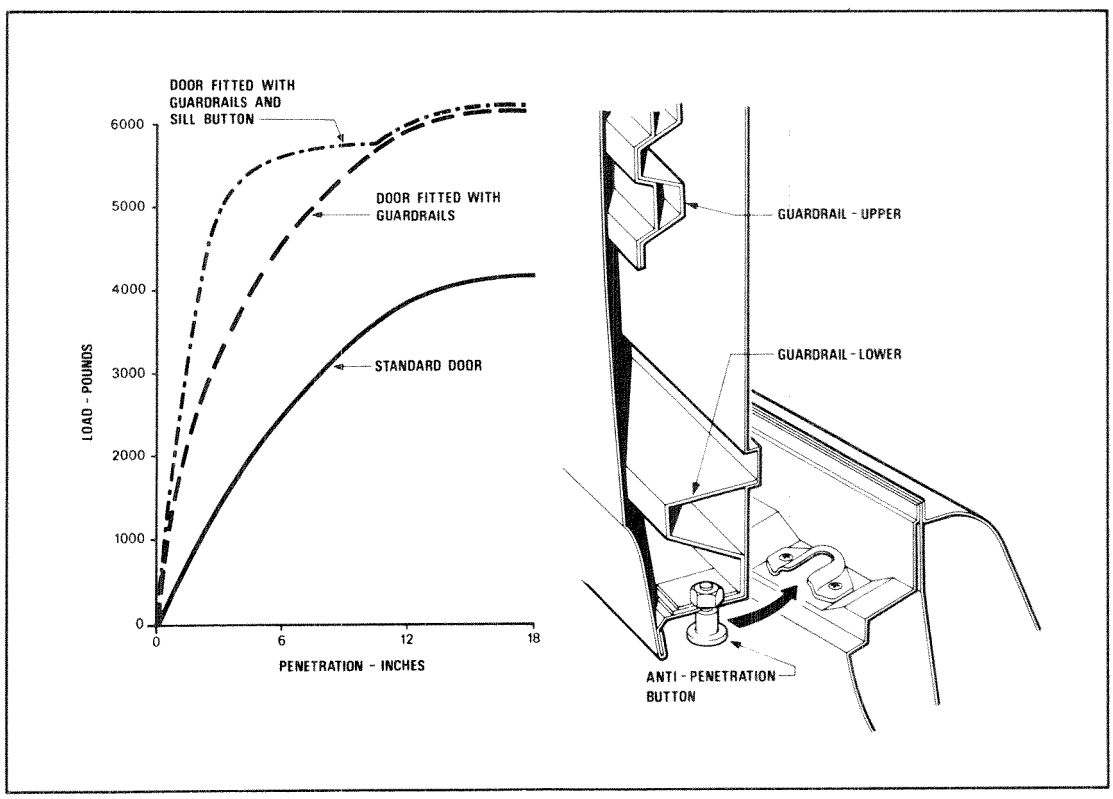


Figure 31

When these and other developments had been individually assessed, chosen modifications were built into body shells for final analysis prior to the resumption of dynamic testing. For the laboratory simulation of the 45°car-to-car impacts, we used a NHTSA side intrusion rig, and for the 90°impacts we adapted one of our crush rigs as an impact simulator. (FIG. 32.)

A shaped barrier-form, designed to reproduce the post-impact frontal profile of a bullet car, was mounted on the face of the crush rig. This was then forced, under load, into the side of a standard vehicle, until the same degree of penetration as that resulting from the dynamic test had been

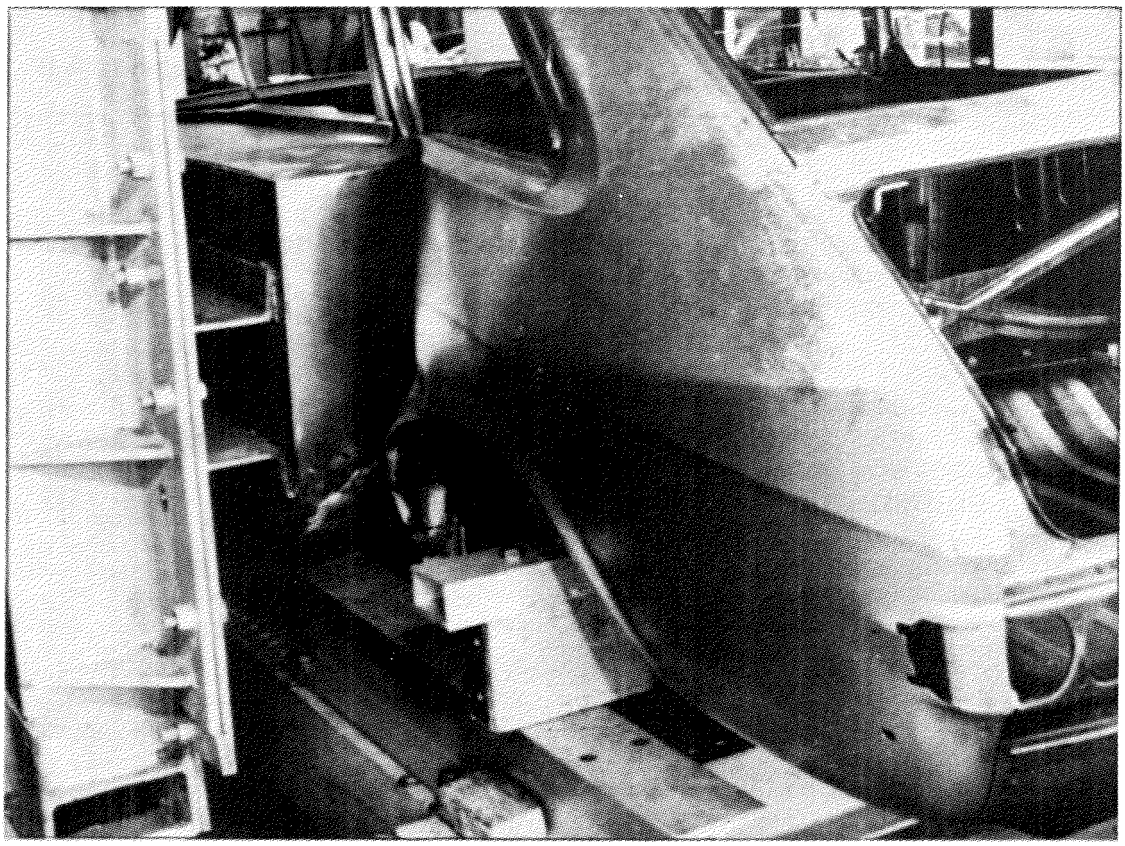


Figure 32

reproduced. The 'work done' in achieving this penetration was then calculated. By applying the same procedure and amount of 'work done' to a modified vehicle, it was possible to obtain a reasonable prediction of the degree of intrusion to be expected under dynamic conditions. (FIG. 33.)

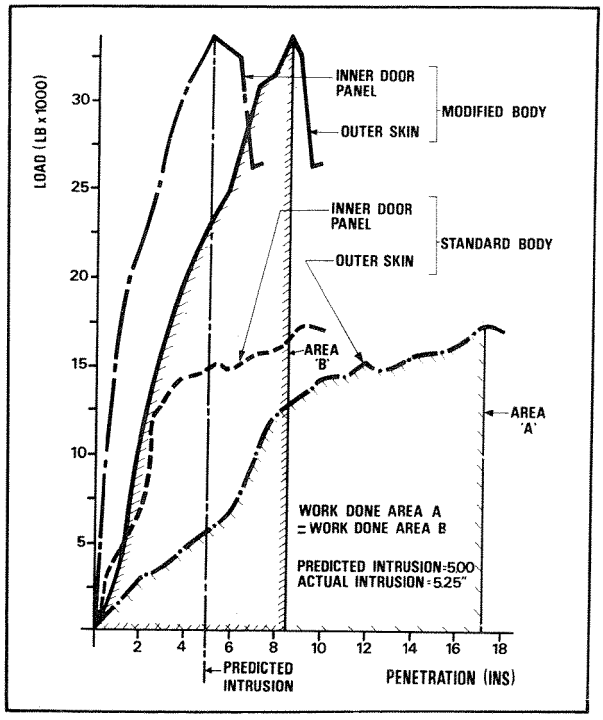


Figure 33

A similar approach was successfully adopted for the car-to-pole test.
The final modifications incorporated into the Project 4 test vehicle included those listed in Table IV.

PROJECT 4 MODIFICATIONS

1. 'B' POST REDESIGNED, FOAMED, AND JOINTS WITH SILL AND CANTRAIL IMPROVED. (NECESSITATED NEW DOOR FRAMES AND GLASSES.)
2. SILL DESIGN MODIFIED AND INNER SECTION FOAMED.
3. 'A' POST REINFORCEMENT PANEL ADDED.
4. ADDITIONAL CROSSMEMBERS FITTED TO THE FLOOR AND TUNNEL BRIDGES INCORPORATED.
5. TORSION BAR CROSSMEMBER TO SILL JOINT IMPROVED.
6. ANTI-PENETRATION BUTTONS FITTED TO DOORS IN LOCK AND SILL AREAS.
7. ROOF BOW ADDED IN 'B' POST AREA.
8. STRENGTHENING TUBE ADDED TO REAR SEAT HEELBOARD.
9. GUARDRAILS FITTED IN DOORS AND REAR TONNEAU PANELS.
10. FLOOR AND TUNNEL MODIFIED.
11. FRONT SEATS MODIFIED.
12. PROJECT 5 AND 7 OCCUPANT PROTECTION PADDING FITTED TO DOORS, REAR QUARTERS, 'B' POSTS AND CANTRAILS.

Table IV

Similar modifications were incorporated in the Project 6 vehicles, except here the foam filling was omitted from the sills, and instead two tubular sections were built into fixed front seats linking both side panels at the 'B-C' posts. (FIG. 34 and 35.)

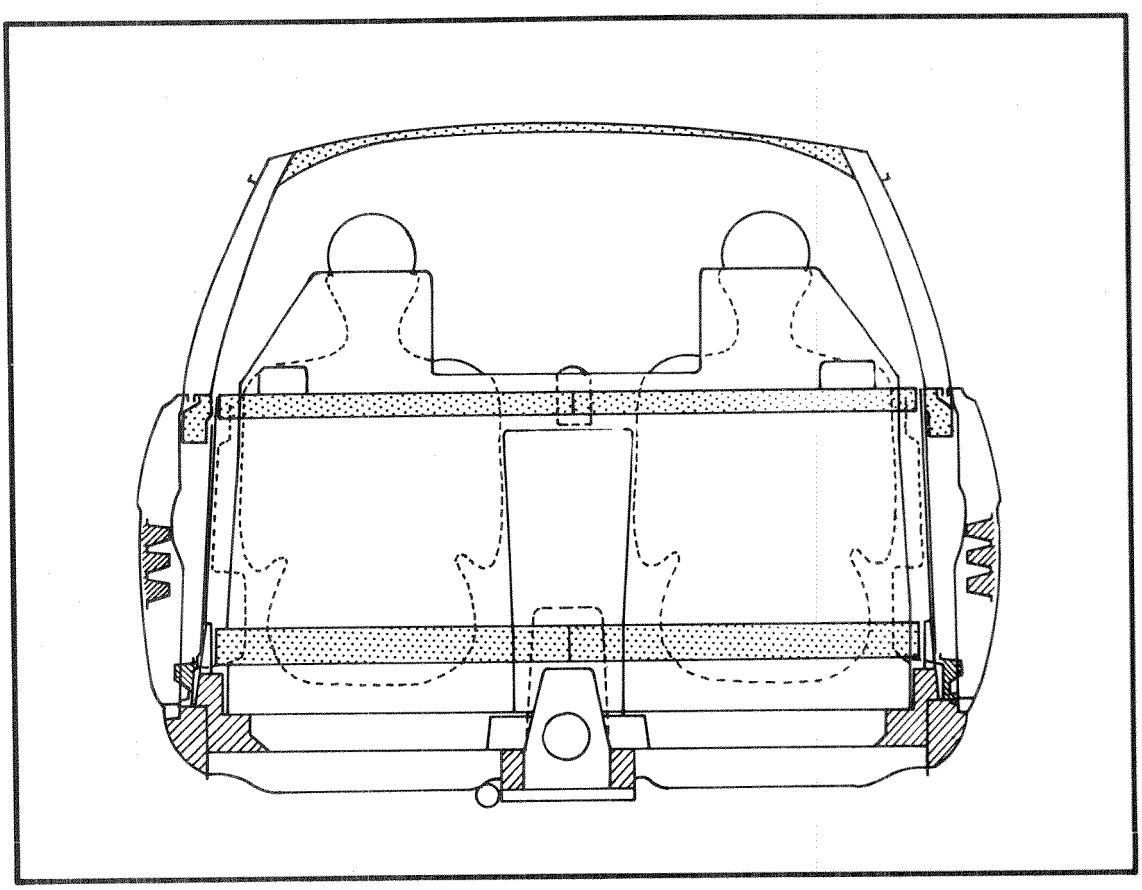


Figure 34

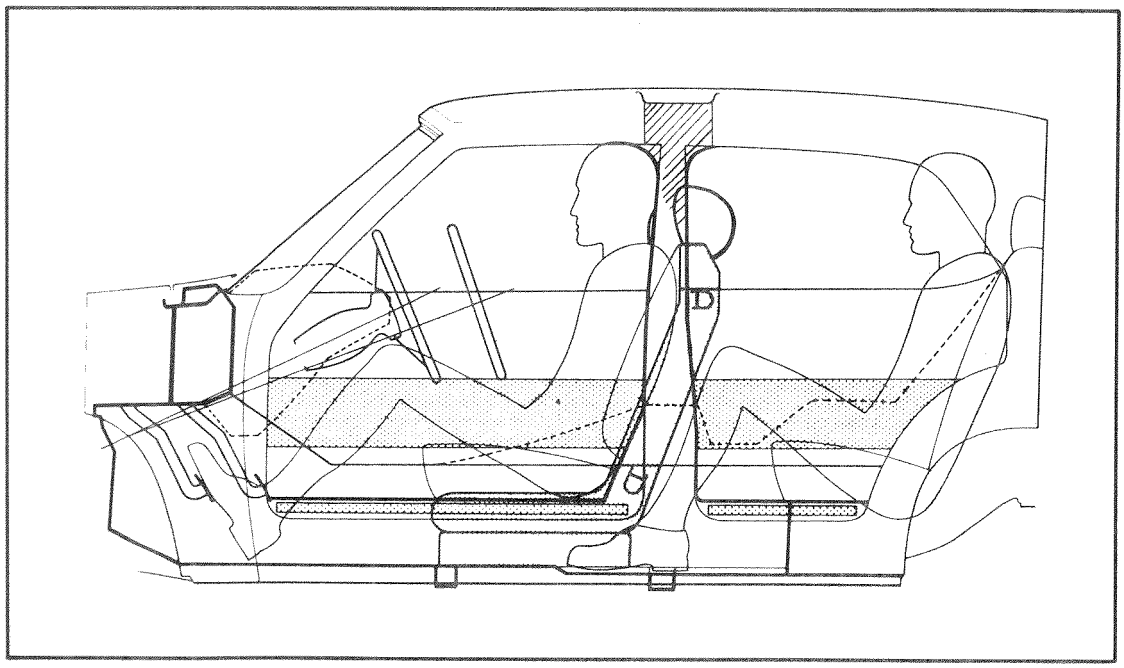


Figure 35

Inertia reel seat belts were anchored to the seat frames, providing the added advantage of ensuring optimum location for all sizes of occupants relative to the restraint system. (FIG. 36.) However, it does necessitate the provision of fully adjustable steering and foot controls together with the location of all other controls within the reach of the shortest driver. (FIG. 37.)

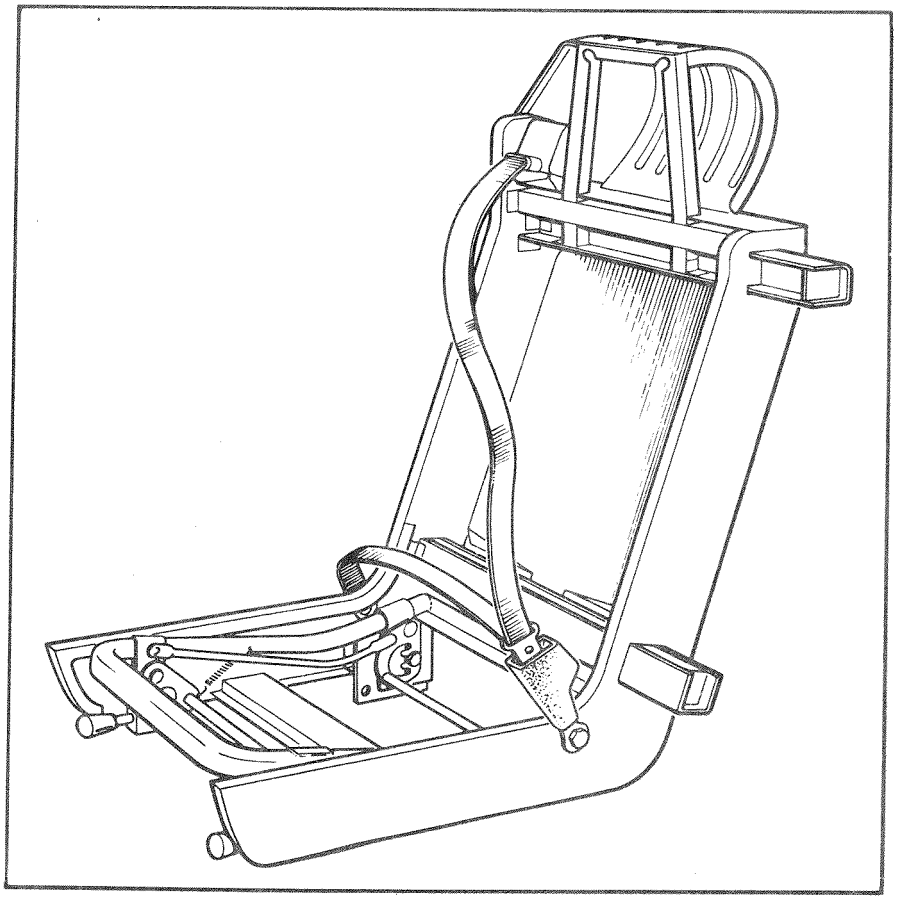


Figure 36

ADJUSTABLE CONTROLS — FIXED SEATS

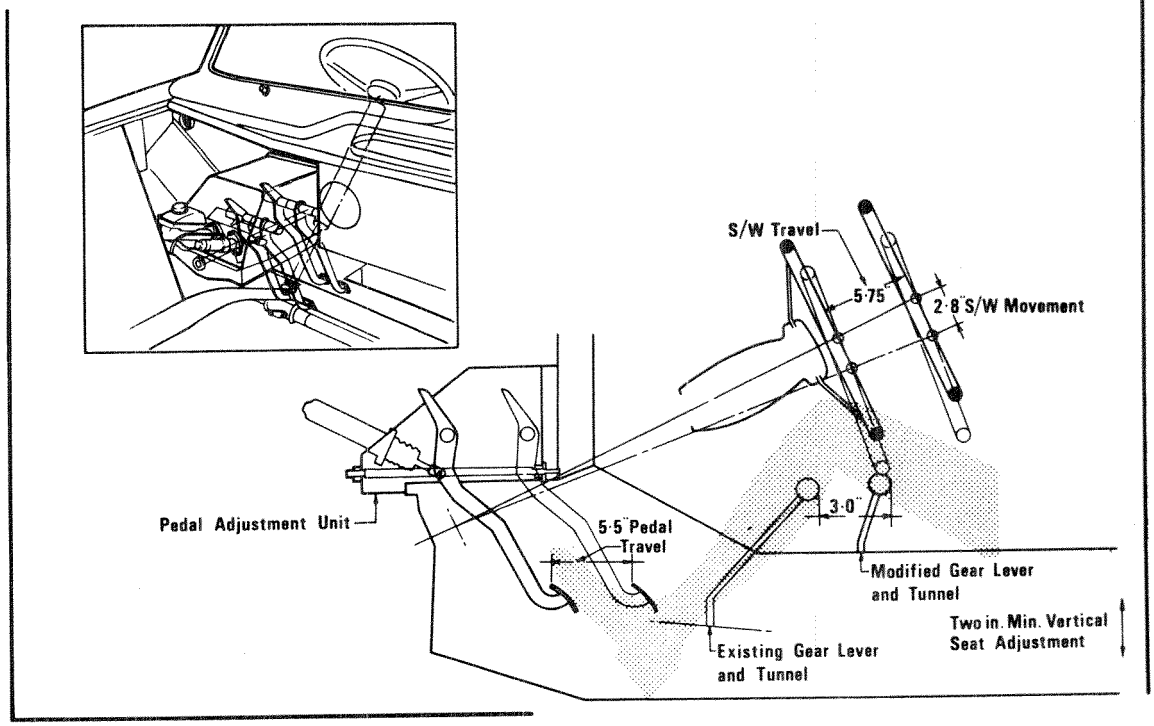


Figure 37

To accommodate the motorised pedal controls the dash was replaced by a Project 3 assembly, a requirement which had been anticipated during its design.

All these modifications increased the vehicle's weight by 175 lb (8.6%) for the two door and 262 lb (12.7%) for the four door.

The internal trim incorporated in the test vehicles will be described under Projects 5 and 7.

PHASE I IMPACTS

In describing the laboratory methods by which we judged the effect of possible modifications, it was stated that we anticipated being able to predict the degree of dynamic intrusion with reasonable accuracy. This expectation was realised when we finally came to conduct the Phase I impacts. The results obtained from the pole tests, together with the recorded injury criteria, are listed in Table V; a

INJURY CRITERIA		CAR TO POLE			
AREA	TOLERANCE LEVEL	STANDARD VEHICLE AT 10 MPH	STANDARD VEHICLE AT 15 MPH	MODIFIED 2 DOOR AT 15 MPH	MODIFIED 4 DOOR AT 15 MPH
HEAD	HIC < 1000	67.4	412.3	531.4	598.4
THORAX	GADD SI < 1000	79.1	127.5	316.8	168.6
	60 g RES < 3 ms.	34 g – 3 ms.	40g – 3 ms.	53 g – 3 ms.	42 g – 3 ms.
SHOULDER	< 1350 lbf	1344	840	521	548
ILIAC CREST / HIP	TOTAL < 1125 lbf	1201	4022+	1591	1908
RIB 1	< 225 lbf	47.5	58.5	144	123
RIB 2	< 225 lbf	7.5	65	88	213
RIB 3	< 225 lbf	136	40	114	171
RIB 4	< 225 lbf	200	32	48	184
COMPARTMENT INTRUSION	—	7.25"	13.75"	4.00"	6.50"

Table V

study of these shows that with the exception of a possible fracture to the hip, all other injury measurements are within those specified by TRRL. Indeed this particular fracture would probably not occur when the centre line of the pole is other than coincident with the occupants hip point.

Although considerable reductions in the degree of structural intrusion was obtained in both Phase I pole tests, a comparison of the results illustrates the greater improvement resulting from the partially foam filled sills of Project 4. (FIG. 38.) The point of impact on both cars was midway between two cross-members. The inclusion of an extra reinforcement in this area would have improved the situation on Project 6.

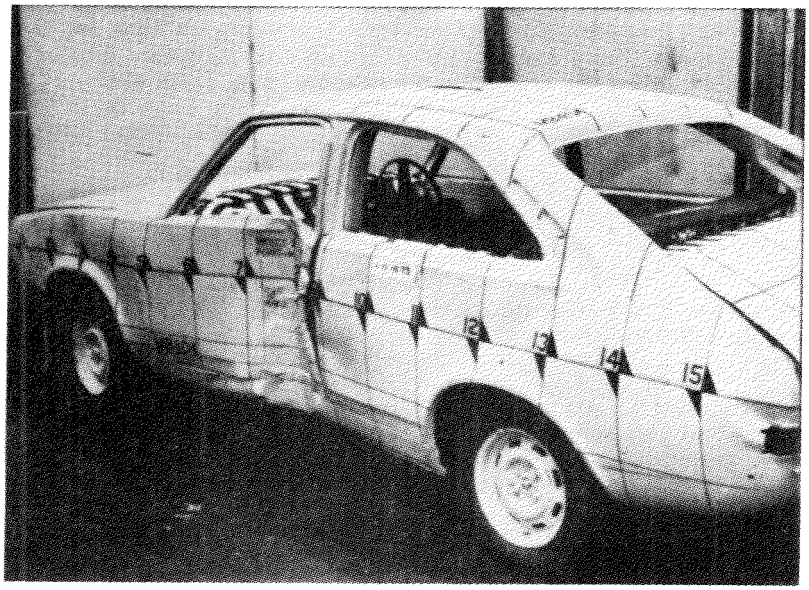


Figure 38

The results of the 90° car-to-car impacts are of particular interest since they illustrate, not only the improvements obtainable from structural and trim modifications on the target car, but also the additional advantages to be gained by improving the compatibility between both target and impacting vehicle by a reduction in the aggressiveness of the latter's frontal design. (Table VI.)

INJURY CRITERIA		CAR TO CAR AT 45° AND 20m.p.h.	CAR TO CAR AT 90° AND 30m.p.h.			
AREA	TOLERANCE LEVEL	STANDARD 2 DOOR	STANDARD 4 DOOR	MODIFIED 2 DOOR	MODIFIED 4 DOOR	MODIFIED 2 DOOR E.A. BULLET
HEAD	HIC < 1000	119.8	957.7	562.7	101	139.7
THORAX	GADD SI < 1000	60.26	453.4	279.7	132	71.6
	60 g RES. < 3 ms.	28.6 g – 3 ms.	60 g – 8.33 ms.	60 g – 5 ms.	41 g – 3ms.	26.4 g – 3 ms.
SHOULDER	< 1350 lbf	931	2807	2101	1094	800
ILIAC CREST / HIP	TOTAL < 1125 lbf	167	2037	1103	1442	713
RIB 1	< 225 lbf	55	84	271	89.6	168
RIB 2	< 225 lbf	46	169	213	72.3	90
RIB 3	< 225 lbf	65	352	358	81.8	74
RIB 4	< 225 lbf	18	321	128	79.2	66
COMPARTMENT INTRUSION	—	6.75"	14.25"	5.25"	2.75"	0.9"

Table VI

To investigate this experimentally we built two identical Project 4 Phase I vehicles and impacted them at 90°; one by a Standard Marina whose weight was increased to 2548 lb by the inclusion of dummies and ballast, and the other by a Project 3 Marina of the same weight.

Impacting with the Standard Marina resulted in a local intrusion of 5.25 inches, mainly resulting from impact by the leading edge of the bonnet against centre of the 'B' post. (FIG. 39.) This compares with 5.0 inches of intrusion predicted by the simulation crush test. (FIG. 33.)

Injury levels, though showing improvements over the base line tests, were still according to TRRL unacceptable. In contrast the substitution of the Project 3 vehicle as the bullet car, dramatically reduced both intrusion and dummy injury levels to insignificance. The maximum internal penetration being 0.94 inches, with all the injury levels well within the accepted limits. (FIG. 40.)

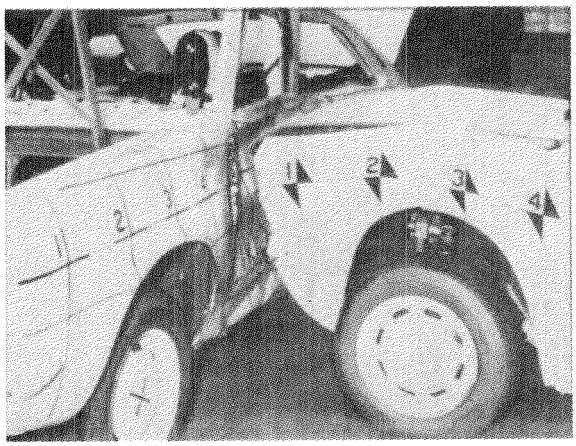


Figure 39

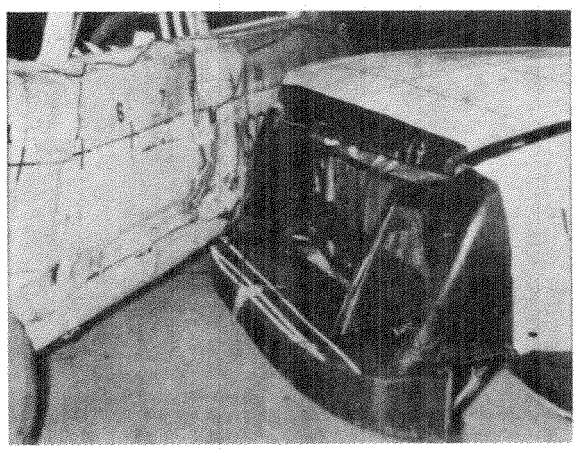


Figure 40

From analysis of the high speed films the difference in results appears to have been due to three main factors. Firstly the front end of the bullet vehicle incorporated soft headlamp surrounds and an energy absorbing bumper. This was responsible for dissipating approximately 20% of the impact energy prior to contact of the metal structures. The soft headlamp surrounds also assisted in the elimination of the hard contact points. Secondly the bumper was centred 14½ inches above ground level at kerb weight. In its lower position it contacted the overlapping lower door and sill panels, whose combined rigidity reduced the penetration to a minimal degree. Finally, because of the low bumper position, the target car tended to roll towards the bullet car and so delay impact of the occupant's head and upper torso with the interior. This delay contributed to the reduced occupant impact velocity.

In the previous test the standard bullet car, with its conventional bumper height backed by almost vertical front panels, spread the impact loads over the target vehicle's relatively weak side panels. Although this produced an even distribution of load; the deformation resulted in the dummies being struck by the collapsing inner panels, at a higher velocity than in the subsequent test. As a result, the recorded dummy loads were also higher.

The results of the Project 6 car-to-car impacts are also shown in Table VI. These illustrate the improvements gained by the inclusion of the fixed seat concept, over the conventional Project 4 vehicle, when both are struck at 90° by standard vehicles of the same weight.

Although showing considerable reductions in dummy impact loads, the front seat occupant still suffered a possible hip fracture. This would not have occurred had the car been impacted by a Project 3 vehicle.

In designing safer cars for survivable high speed frontal impacts, there is always a danger that the vehicle itself may become more aggressive and therefore a greater potential menace to other road users.

The example illustrates that a degree of compatibility, certainly between cars, is perfectly feasible and worthy of further and deeper study.

In conclusion, whether or not the occupant suffers physical injury as a result of a crash, there is always the problem of releasing them from the vehicle. Theoretically this is possible provided at least one door can be opened; in practice, particularly with regard to severely injured or very elderly occupants, it may prove extremely difficult.

Consequently as in the case of the frontal impacts, we endeavoured to provide for all the doors to be openable after each crash event, without undue effort. This aim was not achieved for doors impacted by the pole or by the standard bullet vehicles at 90°.

However, as a final illustration of the effectiveness of designing for reduced frontal aggressiveness, in the last test we were able to release the latch and open the struck door on our modified target vehicle, even though it had been directly impacted by a 2548 lb car at 50 km/h (31 mph).

PROJECTS 5 AND 7 – OCCUPANT PROTECTION

So far we have mainly discussed improvements to the structural properties of the vehicle, and certainly these are of vital importance in the reduction of occupant injury; but the most frequent danger to car occupants lies in their sudden deceleration against the interior of the vehicle during impact. In these circumstances a car may be brought to rest in a tenth of a second, whilst the occupants continue in the direction of impact at a speed commensurate with the closing speed of the car and the impacting object. If the interior of the vehicle is not designed to cushion the occupants impact at tolerable levels, contact may be violent and traumatic. It was to improve this situation that Projects 5 and 7 were commissioned, the former to deal with the reduction of head and thoracic injuries and the latter with injuries to the pelvis and legs. In addition Project 5 will, under Phase II, also deal with facial injuries caused by the occupant's penetration of the windscreen.

Statistics related to car occupant injuries differ considerably according to the peculiarities of the vehicle design, the severity of the accident and the individual impact tolerance levels of the occupants. In our laboratory tests these injuries were assessed by means of anthropomorphic dummies. Variations in the loads and accelerations were recorded in various parts of the dummies during the tests; unfortunately this method does not necessarily reproduce the skeletal and tissue damage sustained by their human originals with any degree of certainty.

However, from the severity of the results obtained from the standard vehicle tests, we concluded that in the majority of cases, front seat occupants would have been severely or fatally injured at the high impact speeds under consideration.

This assumption raised three problems, firstly could the vehicle's acceleration pulse be improved; secondly how much of the occupant's kinetic energy could be absorbed by an improved belt system; and finally to what extent could the occupant impact forces be reduced by the addition of energy absorbing trim?

The former problem was clearly associated with the structural design, and the somewhat conflicting need to tailor the vehicle's stiffness to provide complete integrity of the occupant compartment. The second requirement was the subject of a separate TRRL contract placed with Auto Restraint Systems Ltd., (see ESV Conference paper by Mr. W. M. Giffin).

The solution to the final problem commenced with an analysis of the high speed films obtained during the basic impact tests. These and subsequent sled tests enabled us to establish the directions, and areas of contact within the occupant compartment, of both restrained and unrestrained dummies.

The next essential was to reproduce the severity and directions of these impacts in the laboratory, and thereby assess a range of designs and materials prior to further testing. These laboratory investigations were conducted by sled tests using dummies and rig tests simulating head, shoulder, pelvis, leg and knee impacts. One example of this type of rig being illustrated in Figure 41.

RESTRAINT SYSTEMS

The proper use of seat belts can minimise, or even prevent, serious injury in frontal impacts. However, belts are designed to stretch to a considerable degree during high speed crashes, in order to reduce the belt loadings and decelerations of the wearer to survivable level, so that ultimate contact by even a belted occupant against the interior of the car remains a possibility.

In side impacts as we have seen, belts offer inadequate protection and violent impact against the interior is inevitable.

A fundamental requirement, in both of these situations, calls for a quantity of energy absorbing padding to be placed between the occupants and the interior structure, in order to cushion the impact.

Even so, the fitting of a personal restraint system remains an absolute necessity if significant reductions in the number of occupant injuries is to be achieved. Safety cars with their technological developments can considerably improve the chances of survival, but restraint systems, such as air bags or seat belts, remain a vital part of the whole. Under certain conditions they will be of greater importance than the padding, but both are essential for maximum protection.

HEAD AND SHOULDER IMPACTS

For the car occupant, the problem of head and shoulder impacts can be considered under two headings. Firstly the restricted areas, such as the upper 'A' post and cantrails, which preclude more than a limited modification because of conflicting vision and access requirements. Secondly the unrestricted areas, such as

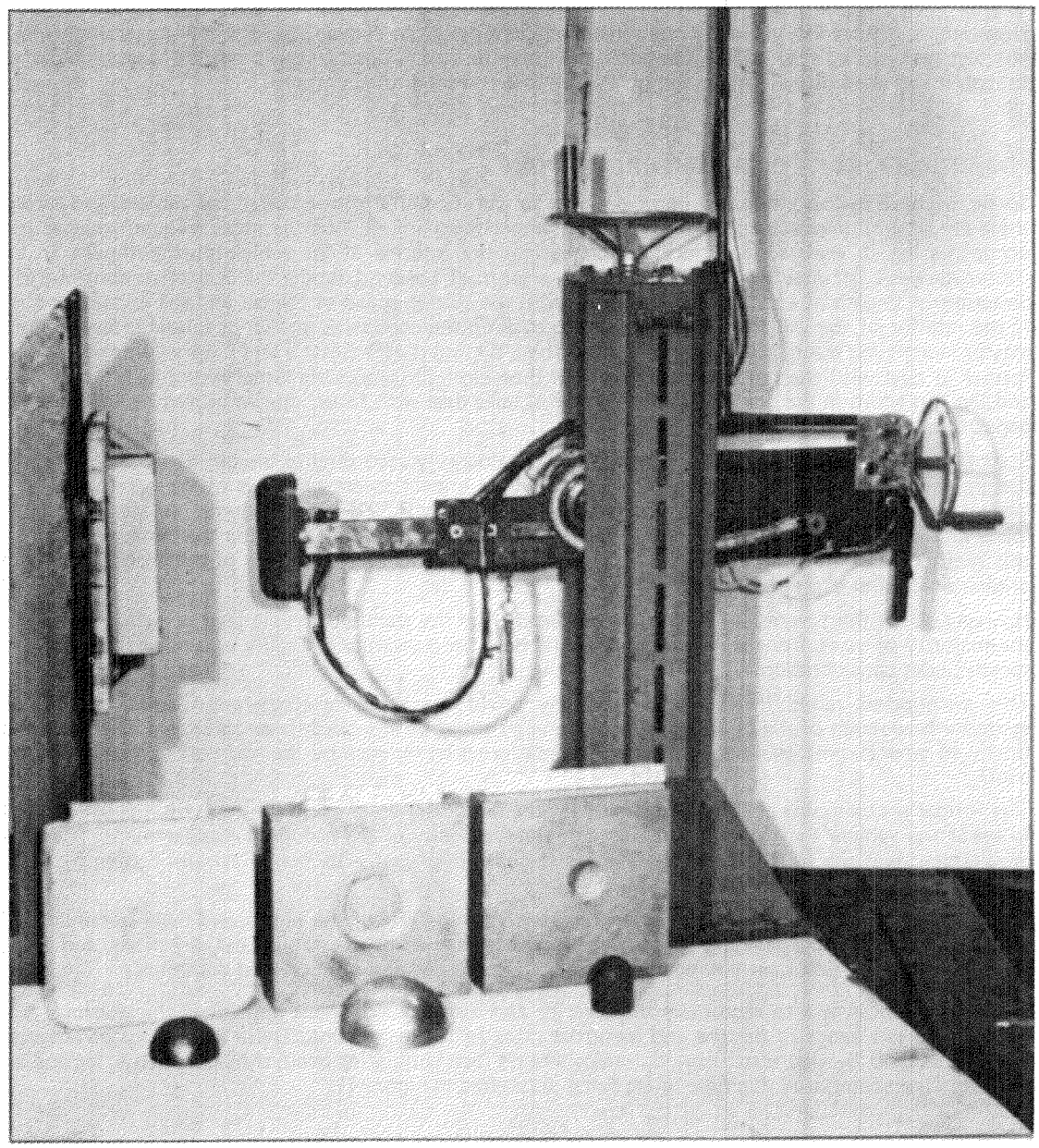


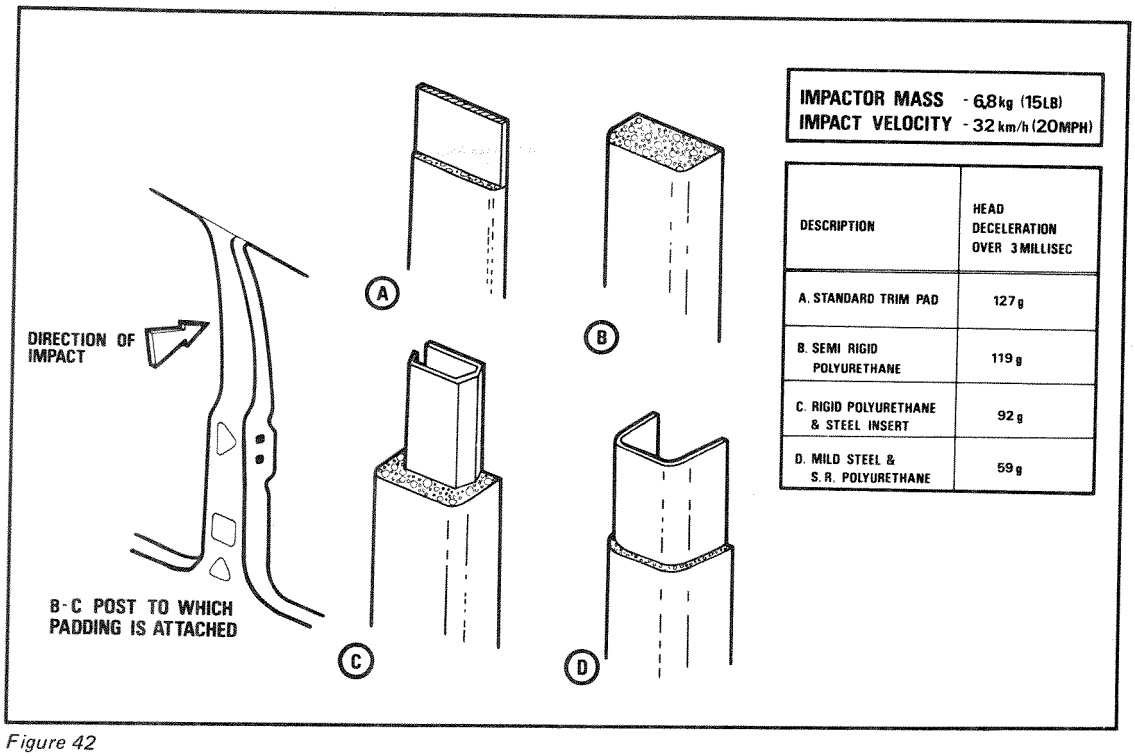
Figure 41

the facia panel, which permit major alterations to be undertaken. The majority of car interiors are not suitable for severe occupant impact, the thin layers of 'soft' padding offering little protection at high velocities. What is needed is padding of sufficient depth and resistance to allow the occupants to decelerate at a tolerable level during the crash period.

Our laboratory tests on highly resilient foams proved unsatisfactory because of their inherent high rebound velocities. Of the many non-resilient foams considered, the high density varieties tended to collapse into a solid block of material; with resulting low penetration and high decelerations.

Generally the most suitable foams were those possessing a rigid low density structure. However, unless backed by a second collapsible material having suitable complementary characteristics, a considerable depth of foam was necessary for satisfactory results. A further disadvantage with these very low density foams, was the difficulty of producing homogenous mouldings.

Laboratory testing indicated that functionally designed energy absorbing mouldings, manufactured from sheet steel and covered with a layer of force distributing foam, were generally superior to mouldings manufactured entirely of foam and directly applied to the normal structural members. As an example, the development stages in the design of an upper 'B-C' post moulding suitable for head and shoulder impacts is shown in Figure 42.



DESCRIPTION	HEAD DECELERATION OVER 3 MILLISEC
A. STANDARD TRIM PAD	127g
B. SEMI RIGID POLYURETHANE	119g
C. RIGID POLYURETHANE & STEEL INSERT	92g
D. MILD STEEL & S.R. POLYURETHANE	59g

Figure 42

Of course, the size, shape and gauge of steel panels has a profound influence upon their impact characteristics. Where necessary, this may be varied to advantage, by the addition of slots or holes in the panel surface to encourage the desired collapse pattern and lower decelerations. (FIG. 43.)

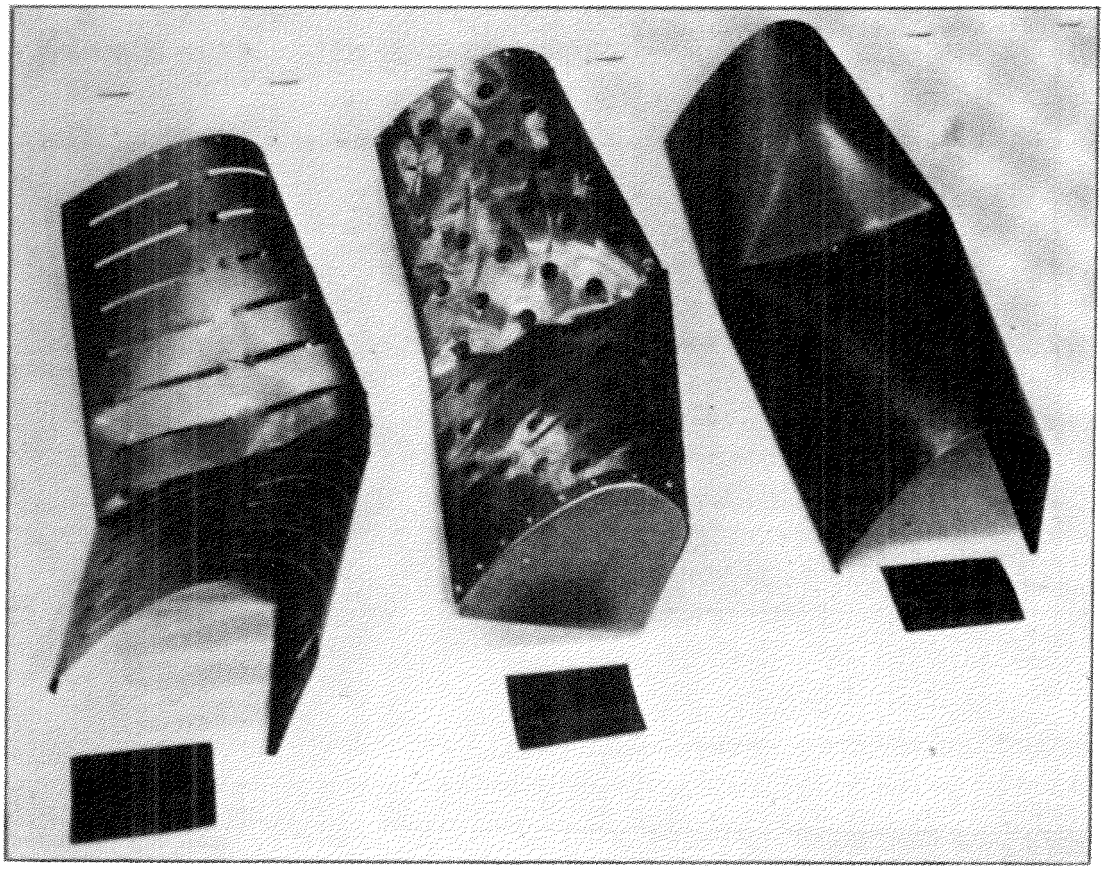


Figure 43

PANEL CONDITION	HEAD IMPACT DECELERATION OVER 3 MILLISEC.
PLAIN	38 g
HOLES	23 g
SLOTS	20 g

Figure 43—Continued

This same approach was applied to a number of panels, including the front seat head restraints, which were not only designed for the reduction of whip-lash resulting from rear impacts; but also for frontal crashes, during which they may be impacted by the rear occupants.

The comparative performance of the standard and Phase I structure within the head impact area, is shown in Table VII.

SIMULATED HEAD IMPACTS
6.8 kg (15 LB) HEADFORM IMPACTOR

AREA	IMPACT VELOCITY km/h	DECELERATION OVER 3 MILLISEC	
		STANDARD VEHICLE g	1ST PHASE DESIGN g
FRONT HEADER	32 (20 MPH)	78	52
FACIA	32 (20 MPH)	71	
	40 (25 MPH)		51
'A' PILLAR	30 (18.7 MPH)	136	62
CANTRAIL	32 (20 MPH)	144	64
B-C POST	32 (20 MPH)	127	59
FIXED SEAT HEAD RESTRAINT	40 (25 MPH)	–	70

Table VII

KNEE AND LEG IMPACTS IN FRONTAL ACCIDENTS

Our investigations into knee and leg impacts were initiated by a series of sled tests based on controlled 'submarining',' they were designed to establish the trajectory and relative impact velocities of the dummy's lower limbs against the interior of the car. This was achieved by restricting the forward movement of the dummy's chest by means of a pair of crossed diagonal seat belts. (FIG. 44.)

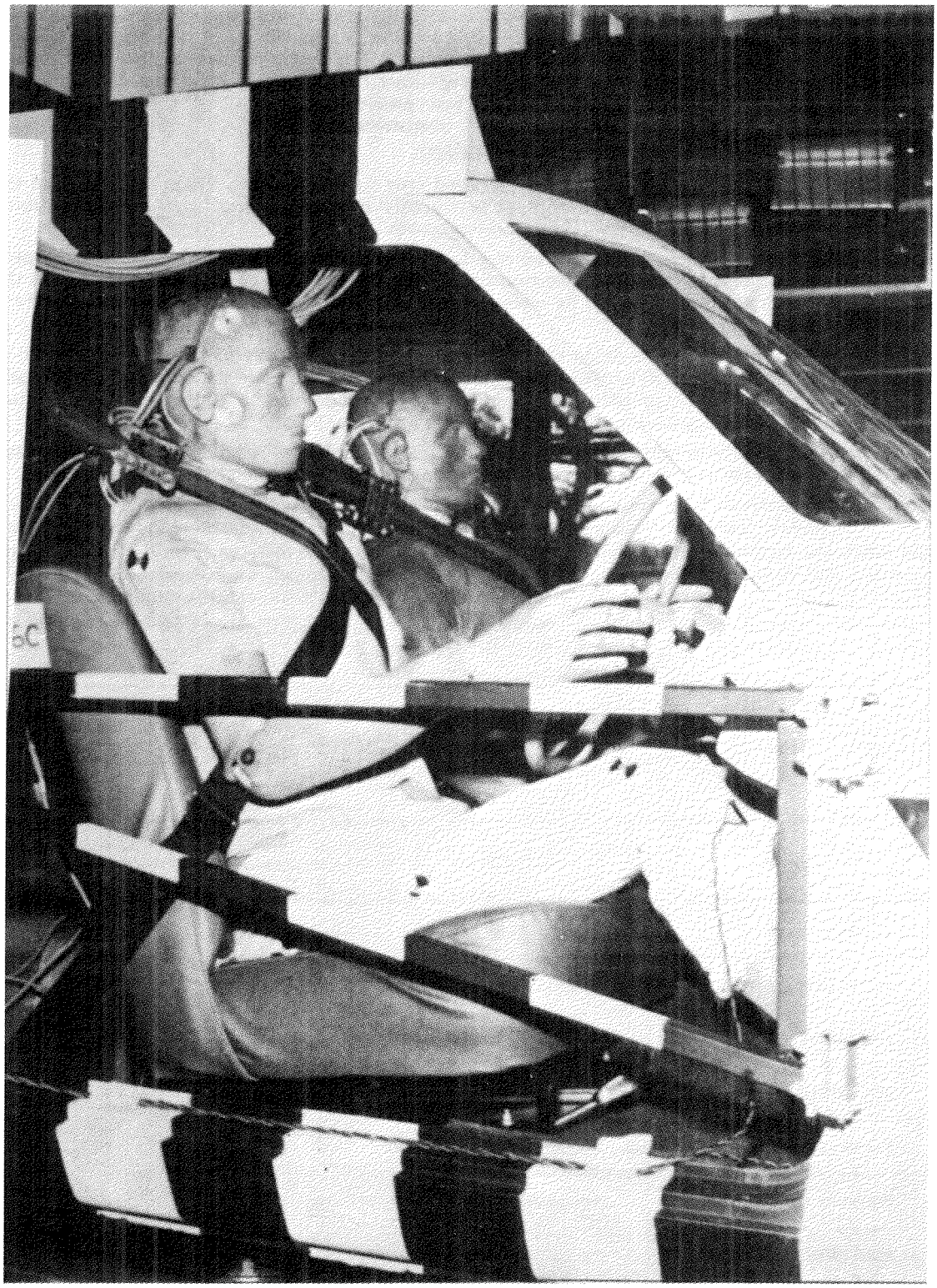


Figure 44

Measurements obtained during these tests, coupled with data supplied by TRRL, enabled us to establish laboratory impact tests. These confirmed that the principles evolved for the design of head impact panels, were equally applicable for the protection of the lower limbs.

Early in the contract we had decided to utilise the underside of the facia as a means of controlling the forward motion of the front seat occupants; similarly the backs of the front seats were to be used for

restricting the forward movement of the rear occupants. It followed from this decision, that whilst a parcel shelf could be suitably designed to serve this purpose, the possibility of it being used to carry heavy unyielding objects excluded this as a practical approach. For similar reasons no glove box has been provided, since its possible contents could drastically affect its collapsibility and therefore nullify its effectiveness. Alternative storage space has therefore been provided.

RIB, PELVIC AND LEG IMPACTS IN SIDE ACCIDENTS

Probable injuries to the ribs and pelvis were recorded only in side impacts. These resulted from the severity of occupant impact with the interior, gross structural collapse and insufficient energy absorbing door padding to accommodate the severe collision test speeds.

In side impacts, the solution to these problems is especially difficult because of the lack of space available, either for drastic structural modifications, or for the inclusion of adequate padding. On the Marina, the structural problems had to be solved within the contours of the existing outer skin and the inner surface of the door panels.

By refraining from modifying the depth of the side structure, we were able to utilise the full width of the occupant compartment in determining the maximum space available for the door padding. To gain full advantage of the restricted space, the padding over the doors was shaped to offer the greatest protection in those areas adjacent to the occupant's torso and lower limbs.

The resulting irregularity in outline and variations in depth, complicated the design of collapsible metal inserts. We therefore reverted to the idea of manufacturing the panels completely in trimmed foam.

Because of the relatively large areas involved we were able to design, within the periphery of the foam moulding, a hollow foam shell supported by a series of ribs or cones. Manufactured in moderately hard rigid foam, the excavations in the foam encouraged the collapse of the material at the desired loads, and prevented solidification of the cellular structure. This resulted in deceleration levels similar to those obtained from the ultra-light foams, but without their accompanying manufacturing problems.

Initial development work on these designs was conducted on an impact rig using a 25 lb hip form as the impactor. A selection of results from these tests is shown in Table VIII.

ENERGY ABSORBING MATERIALS
11 kg (25 lb.) HIPFORM, 24 km/h (15 mph) IMPACT VELOCITY
SAMPLE THICKNESS 76 mm (3 in.).

MATERIAL	CHARACTERISTIC	DECELERATION OVER 3 MILLISEC. g	MAXIMUM PENETRATION %
PRESSED FELT	7.8 lb./cu. ft.	96	77
POLYETHYLENE FOAM	2.2 lb./cu. ft.	72	73
POLYVINYL FOAM	5.6 lb./cu. ft.	68	93
POLYSTYRENE	1.0 lb./cu. ft.	66	77
PAPER HONEYCOMB	0.50 in. CELL SIZE	57	70
POLYURETHANE B.L.U.K.	3 lb./cu. ft. SOLID	93	47
POLYURETHANE B.L.U.K.	3 lb./cu. ft. CONED	50	80
POLYURETHANE B.L.U.K.	3 lb./cu. ft. RIBBED	42	96
POLYURETHANE	1.8 lb./cu. ft. SOLID	40	84
MILD STEEL	0.022 in. PANEL	30	86

Table VIII

The next stage was to conduct more advanced laboratory tests on a sled rig. A body shell containing a dummy was attached to a trolley and accelerated laterally at 24 km/h (15 mph). The trolley was then decelerated by means of 'Inver' tubes and the dummy allowed to impact the side of the padded door assembly. (FIG. 45.)

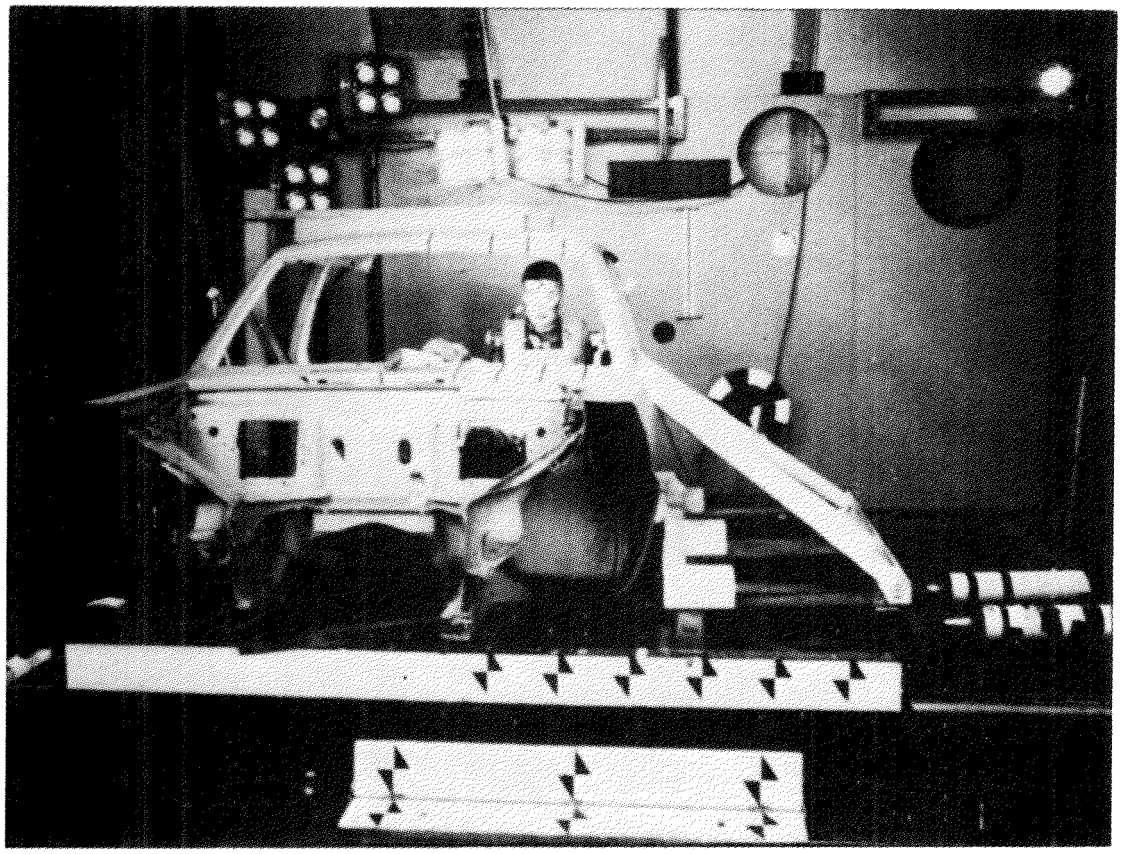


Figure 45

The same TRRL dummy was used for these tests as for the vehicle side impacts. In addition to measuring the head and chest accelerations, the dummy recorded loads in the shoulder, four ribs and the pelvis. A considerable number of tests were carried out to establish dummy repeatability, during which modifications to the shoulder and pelvis were shown to be necessary.

Finally, the true value of the development work on Projects 5 and 7 was not completely evaluated, until the designs had been incorporated in the Phase I vehicles and subjected to the full series of impact tests already described.

PROJECT 8 – STEERING COLUMNS

The project outlined by TRRL encompassed two distinct safety aspects relating to steering assemblies; one concerned with improvement of steering column penetration from multi-directional frontal impacts, particularly with regard to low rake columns; and the other with steering wheel collapsibility, of especial importance in the case of high rake columns.

A study of the Project 3 barrier tests shows by what margins the problem of column penetration has been solved. They indicate that controlled structural collapse can itself replace, to a significant degree, the necessity of providing complex collapsible columns. After all, from the point of view of energy absorption, the car's frontal structure may be considered as an extension of the steering column assembly. This is not to say that flexible shafts, universal joints etc., cannot be incorporated with advantage. Indeed minor modifications of this sort were included in our Phase I designs, but their contribution to the resulting penetration was probably comparatively small.

Reviewing the problem of driver's chest or facial injuries, the collapse of the upper column or steering wheel is preferable to collapse of the lower portion of the column. This is particularly so with transverse engined cars, many of which are fitted with high rake assemblies; these bend forward in a

crash situation and are thereby helpful in reducing chest injury levels. However, in some cases the rim of the steering wheel may be brought into a position of contact with the driver's neck and face. (FIG. 46.)

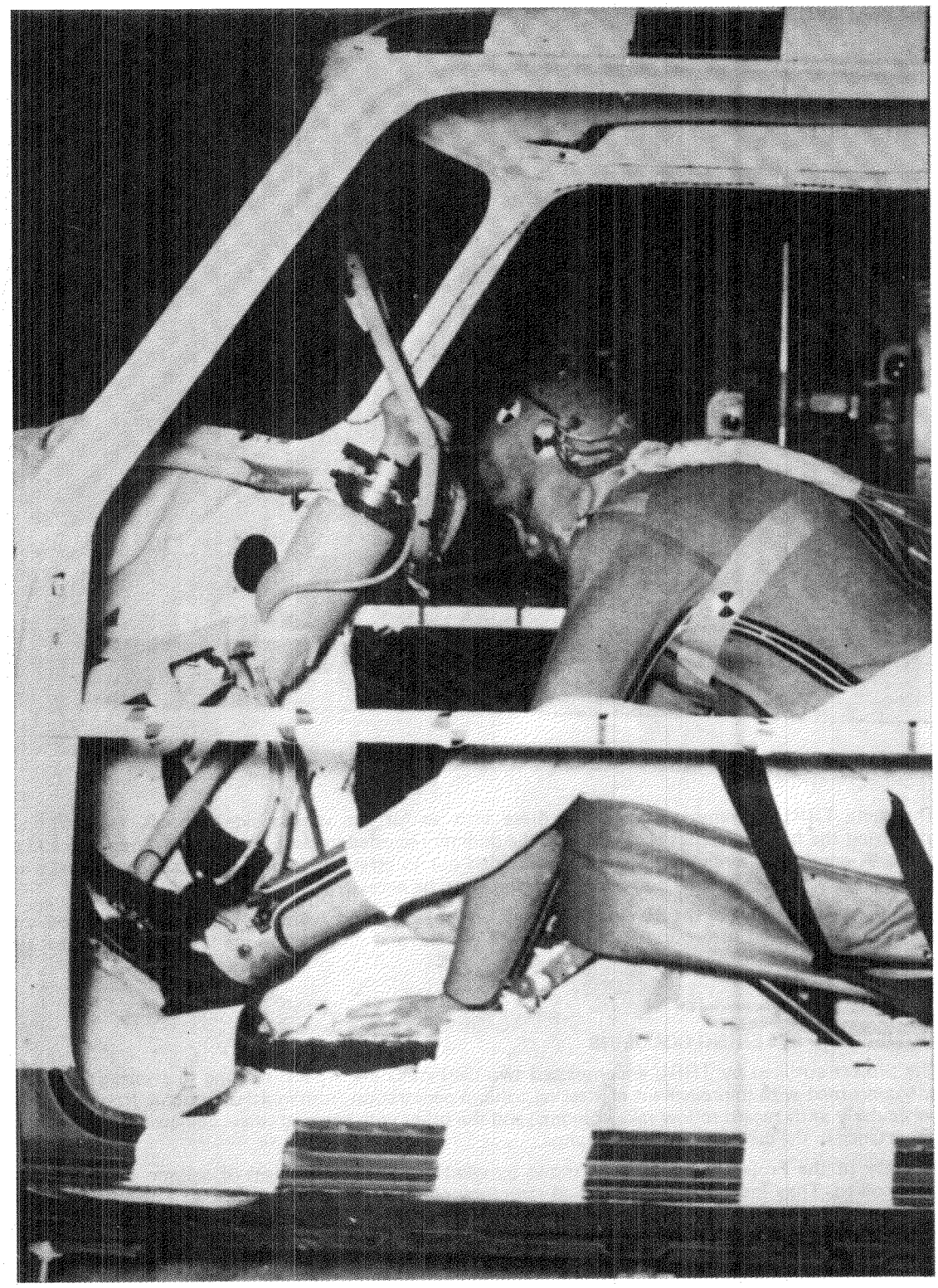
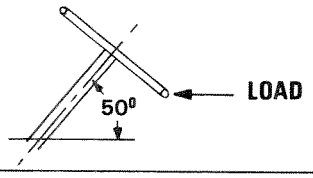
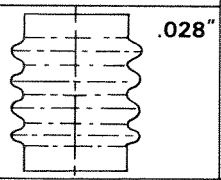
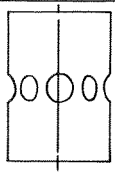
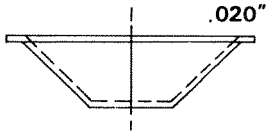
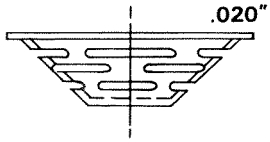
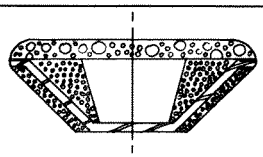
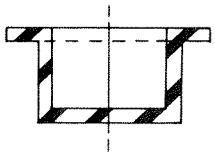
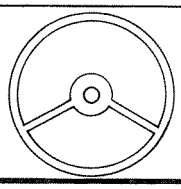


Figure 46

Under these circumstances, injuries to the neck and facial bones may be reduced by the provision of a heavily padded, easily collapsible wheel. This requirement is complicated by the corresponding need for the wheel to be sufficiently rigid to enable the driver to control the car under all conditions, and to support his weight when easing himself in or out of the seat.

Our initial designs replaced the conventional steel and rigid plastic moulding with a variety of devices having varying degrees of collapsibility. These included flexible spokes, convoluted tubes, sheet steel pressings and rubber mouldings etc. A selection of these and their load/impact characteristics are shown in Table IX.

TEST ITEM	STATIC LOAD	DECELERATION OVER 3ms.
		
CONVOLUTE TUBE 	670 LB	–
PLAIN TUBE (1/2" HOLES ON ℄) 	750 LB	–
STEEL CONE (PLAIN) 	960 LB +	–
STEEL CONE (3/8" PERIPHERAL SLOTS) 	DEPENDANT UPON GAPS BETWEEN SLOTS 60LB TO 200LB	23G TO 33G
PADDED STEEL CONE (SEMI - RIGID FOAM - TOP AND BOTTOM. 3/8" PERIPHERAL SLOTS) 	DEPENDANT UPON GAPS BETWEEN SLOTS 150LB TO 450LB	26G TO 38G
RUBBER TOP HAT 	500 LB	48 "G"
STANDARD WHEEL 	860 LB	41 "G"

NOTE:- COLUMN & S/WHEELS SET AT 50° TO HORIZ. FOR ALL TESTS

Table IX

One design yielding promising results, is based on a thin metal cone, slotted to provide uniformed collapse characteristics, and enveloped in a semi-rigid plastic foam moulding incorporating a face pad. (FIG. 47.)

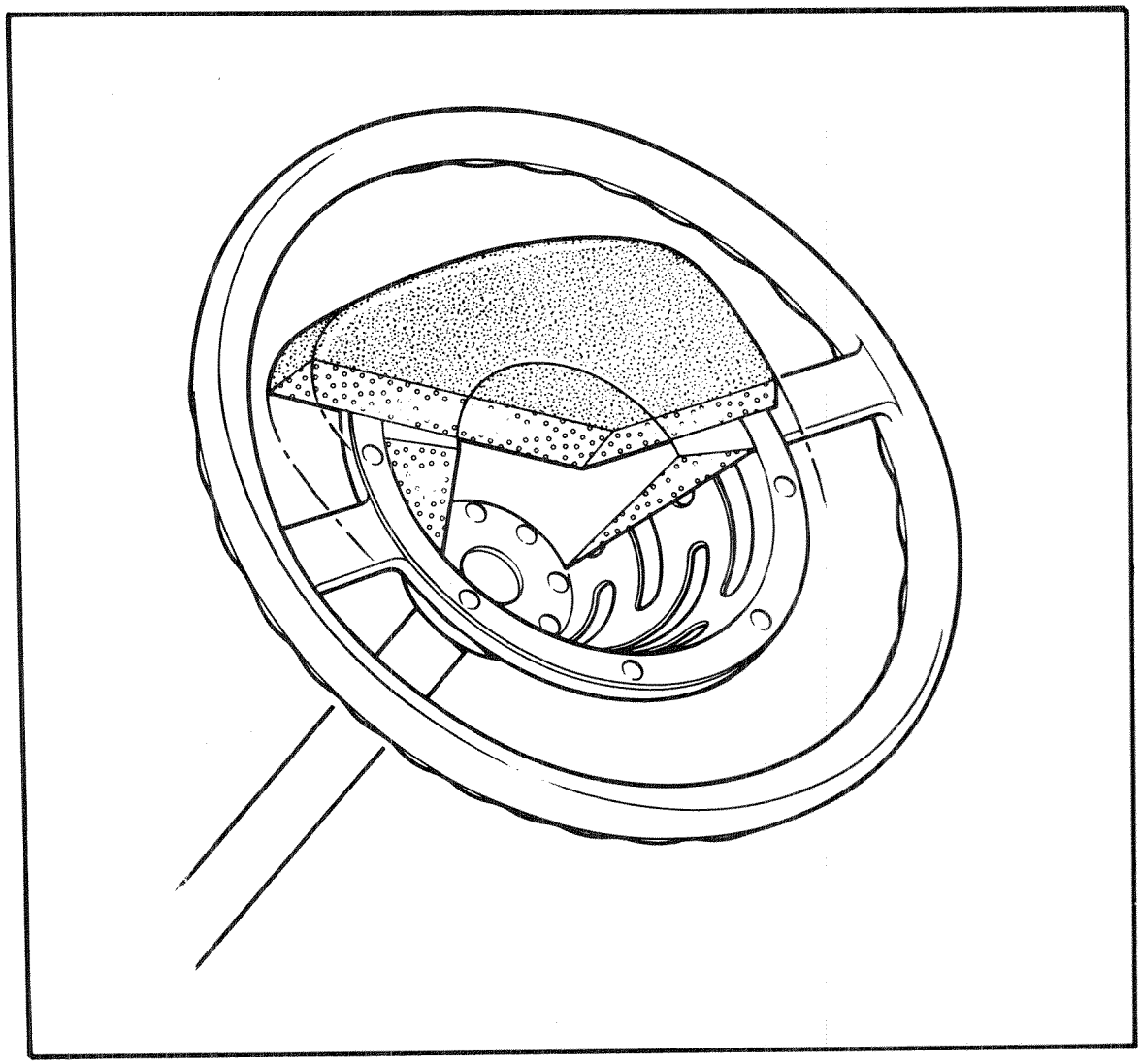


Figure 47

An entirely new design which will give adequate support under all driving conditions, yet collapse under a much lower impact load than even the foam covered metal cone, is currently under test.

Static and impact testing on all these designs is nearing completion, but sled and barrier tests using dummies have yet to be conducted.

S.R.V.1.

PHASE S.R.V.I. – TWO DOOR COUPE (FIG. 48 AND PHOTOGRAPHS IN FOLDER AND APPENDIX I)

The Phase I experimental safety vehicle on exhibition at the E.S.V. Conference is a modified two door Marina coupé, incorporating four of the six TRRL projects.

These are as follows:

1. Project 3 – Frontal Impacts.
2. Project 4 – Side Impacts – Two Door.
3. Project 5 – Interior Occupant Protection – Head and Thorax.
4. Project 7 – Interior Occupant Protection – Leg and Pelvis.

Details of the modifications and their effects on the test results are given in the preceding pages under their individual headings.

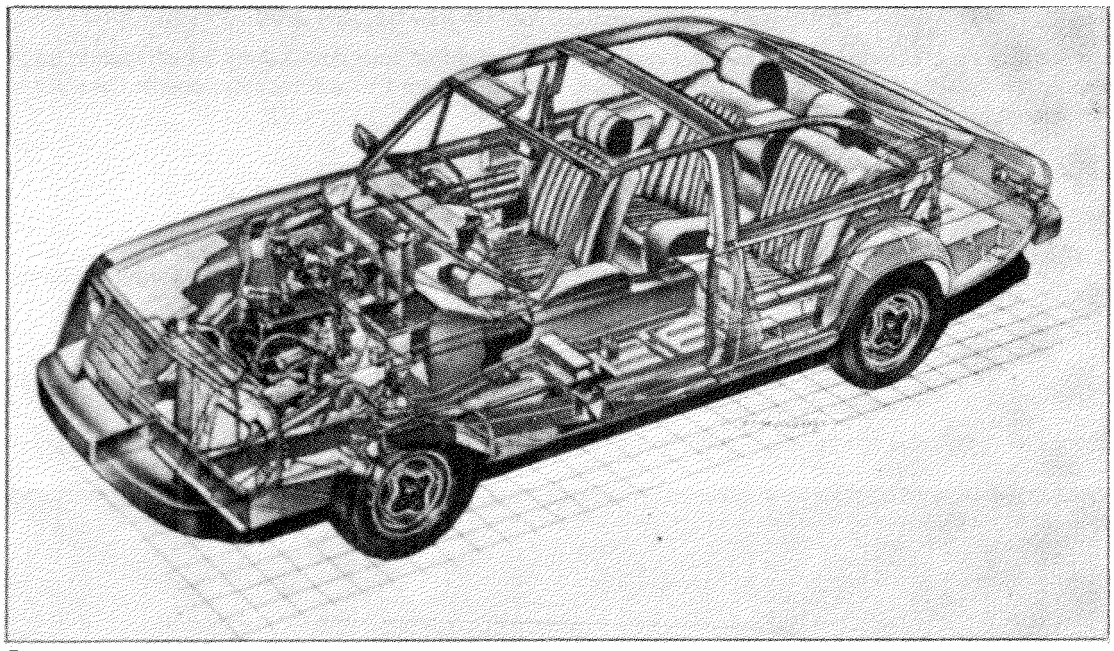


Figure 48

Throughout the projects we have been careful to incorporate only those designs, or materials, which are capable of mass production; schemes suitable for low production only, have been discarded. During the development of the Phase I vehicle some 900 separate tests were conducted on materials, assemblies and complete vehicles. Although modifications to the existing designs and mechanical layout have contributed to the improved test results, the main contribution has been derived from the additional structure and trim, resulting in an increase in overall weight. This inevitably will proportionally affect the cost to produce and market the vehicle.

Increased safety on the scale discussed in this report is synonymous with an increase in purchase cost. For this reason it is essential that legislators give careful consideration to the order of priority of new safety legislation, by considering not only the probable cost-effectiveness of a new regulation, but also whether its introduction may endanger other road users or partly nullify the requirements of existing safety standards.

The vehicle, complete with three dummies, is in 'Test Condition'. The front passenger has been omitted to allow viewers to inspect the front seat inertia lock. This device, based on a 'Kangol' inertia seat belt mechanism, automatically prevents the seats tipping forward in an accident or upon the sudden application of the brakes. It does however permit free access to and from the rear seats, without the need to physically activate or de-activate a locking mechanism. (FIG. 49.)

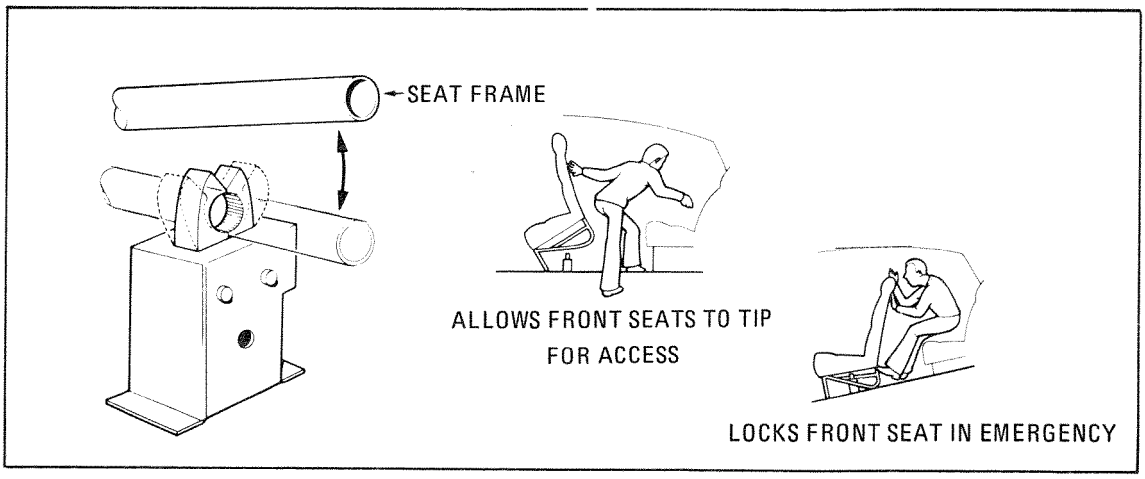


Figure 49

PHASE II – FOUR DOOR (FIG. 50 AND PHOTOGRAPHS IN FOLDER AND APPENDIX II)

The vehicle on exhibition at the E.S.V. Conference is a modified four door saloon which should be more accurately described as a Phase I/Phase II Hybrid, since the Phase I structural components have yet to be considered for further modification as a Phase II design.

The vehicle is Hybridised as follows:

A. **Phase I design**

1. Project 3 – Frontal Impacts.
   The areas incorporated include the understructure, floor, tunnel, door reinforcements, and modified mechanical layout.

2. Project 6 – Side Impacts.
   The areas here include, fixed seats, moveable controls, roof bow, side panels, cross-members and dash assembly.

Details of these modifications are given in the preceding pages under the appropriate headings.

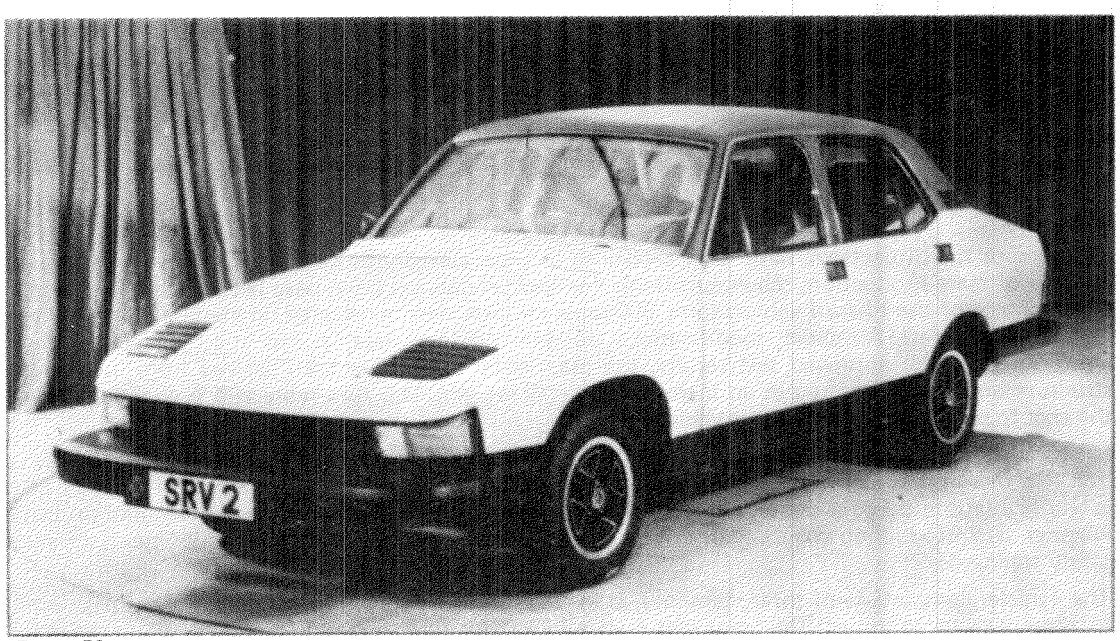


Figure 50

B. **Phase II design**

1. Front profile and Pedestrian protection:

In the United Kingdom about 90% of all serious pedestrian casualties occur in urban areas, at a mean contact speed of around 32 km/h (20 mph) (G. M. MACKAY). However involvement, in even low speed accidents, can result in multiple injuries from contact with the vehicle and road surface; such injuries being predominantly to the head and lower limbs.

Since over 90% of these casualties result from impact by the front of vehicles, the problem of the frontal design and its compatibility with the human body is of paramount importance. The struck pedestrian should be encouraged to fall onto the bonnet rather than onto the road, and having done so, we should endeavour to retain him on the vehicle during braking.

Generally, initial contact is made by an angular metal bumper, designed for vehicle rather than pedestrian protection; impact from its unyielding surface accounts for the high incidence of leg fractures in this type of accident.

To combat this, the Phase II bumpers are fitted with 'soft' load spreading leading edges. Additionally the bumper centre lines are considerably lower than required for the US Federal legislation. This was done for two important reasons; firstly, as described in the Phase I tests, to increase the vehicle-to-vehicle compatibility, particularly in side impacts; and secondly, by impacting below knee height, to reduce the possibility of incurring clinically serious joint fractures, with their unfortunate long term after effects.

Further protection is afforded by curving the rear of the decking panel upwards to cover the parked windscreen wipers and lower windshield surround. (FIG. 51.)

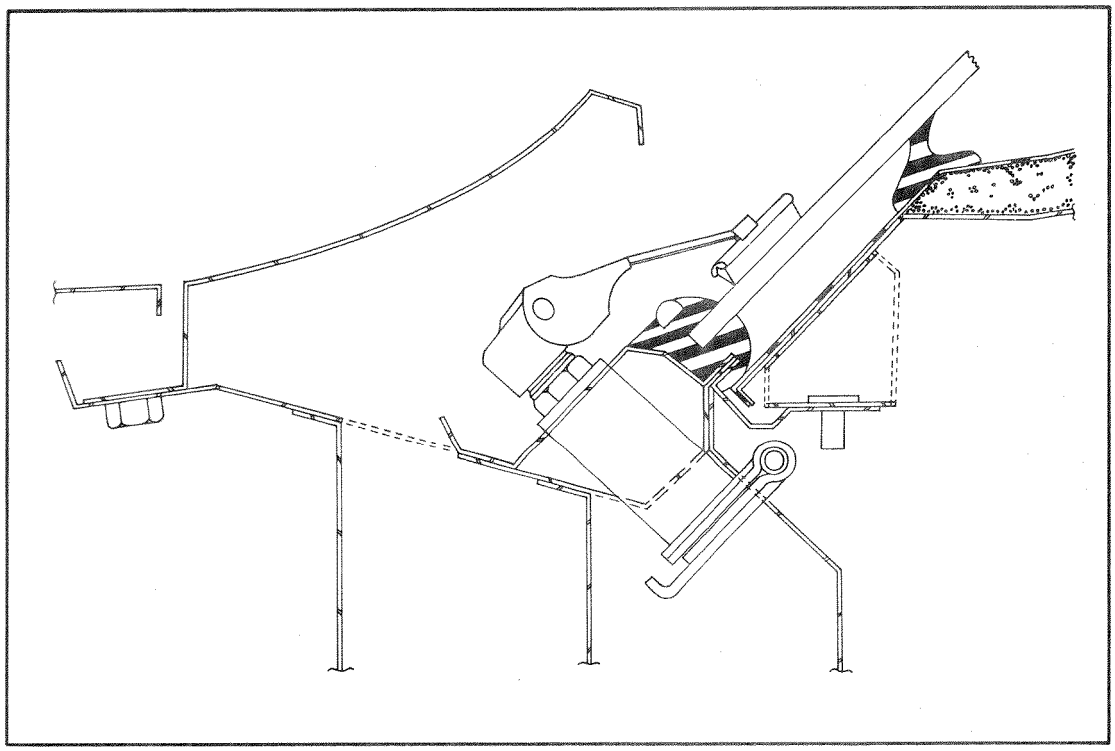


Figure 51

At low to moderate speeds, the struck pedestrian falls towards the impacting vehicle. The subsequent impact against a high profile bonnet can cause injuries to the pelvis, thorax and head; alternatively if the pedestrian is small or if the car brakes violently, the victim will be projected away from the car onto the road surface. A bonnet having a low frontal profile, on the other hand, will strike the pedestrian below his centre of gravity causing him to continue to fall towards and onto the car. (FIG. 52.) Ideally he should be retained in this position during braking, in order to save him from falling onto the road and suffering further injury. Provision has therefore been made in the design to incorporate, at a later date, a pedestrian retaining device similar to that developed by TRRL and fitted to the Austin 1300 demonstration vehicle.

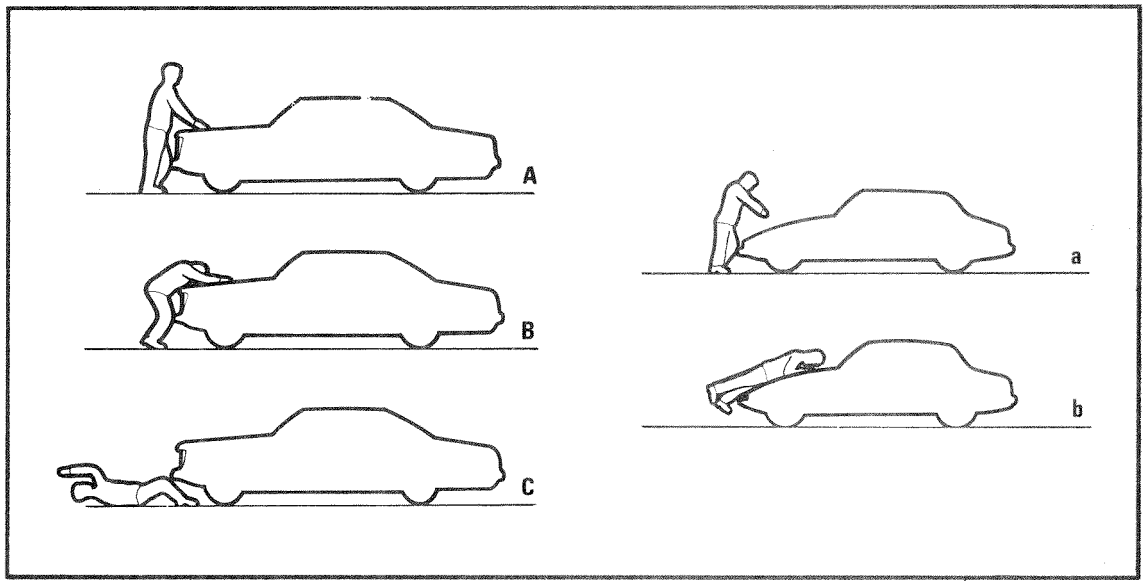


Figure 52

Unfortunately 30% of pedestrian accidents involve young children under 10 years. At this age the relationship between the child's centre of gravity, the height of the bumper and the location of the bonnet front is very critical if the 'pick up' approach is to work.

The Phase II Marina, as designed, should offer protection to the average six year old; this however has necessitated modifications to the Phase I bumper resulting in a reduction of the no-damage impact speed, from 16 km/h to about 10 km/h.

The provision of a pedestrian compatible bonnet, coupled with the necessity to comply with international lighting regulations, has led to the fitment of concealed headlamps. At night these lift up and thereby impose some degree of additional hazard. To ease this possibility, they are designed to collapse as separate modules under impact. The efficiency of the lights is maintained under inclement weather conditions by an integral washer and wiper system.

2. Interior Occupant Protection

The interior trim design is a restyled version of that fitted to the successful Phase I vehicle. Design changes, other than minor, are therefore not anticipated as a result of the proposed Phase II tests.

The two door version will of course be trimmed in a similar fashion to the four door model.

The exhibition vehicle includes experimental fixed seats and adjustable controls. The fixed driver's seat is fitted with a vertically adjustable seat cushion to accommodate occupants of different height; fore and aft movement of the steering wheel is hand operated, whilst similar adjustment of the pedals is by means of an electric motor actuated by a two way switch. Ignition interlocks are fitted to both the pedals and steering column lock to prevent their adjustment whilst the engine is running.

Linking the two side panel assemblies, via heavily reinforced fixed seats, offers the ultimate in lateral stiffness. Side impacts from conventional cars can be accommodated with a considerable reduction in intrusion. Our work on the Phase I vehicles shows that similar, or even better, results can be achieved more simply by alternative methods, provided the attacking car is designed for vehicle-to-vehicle compatibility. Since such a design can also be adapted to reduce pedestrian injuries, it is reasonable to insist that this approach, devoid of the complexities and additional costs inherent in the fixed seat design, is to be preferred.

The car is fitted with three point inertia belts in the rear and a 'passive' three point belt system, developed by Auto Restraint Systems Ltd., in the front. The system is based on the actuation of a central motorised lifting arm, coupled with the opening and closing of the doors. An adjustable anchorage plate on the door frame accommodates variations in occupant height. Collapse of this anchorage under load is prevented by means of an anti-burst loading strip attached to the 'B' post. (FIG. 53.)

Although the front seats are equipped with 'passive' belts, we have also considered the case of cars fitted with 'active' seat belts, which the occupants may choose not to wear.

A frontal impact under these circumstances, frequently results in the penetration of toughened glass windscreens by the front seat occupants. Fragments of glass inflict innumerable cuts to the facial tissues as the head ploughs through the shattered debris. Finally the face, in its downward path, may well smash into the jagged glass fragments retained in the lower windshield rubber.

We have endeavoured to overcome this problem by providing a large energy absorbing facia, in which the lower area is utilised as a knee restraint. When struck by an unrestrained occupant, the facia controls his kinematics and prevents facial contact with the lower screen aperture. Increased protection, to provide for variations in impact speed or occupant dimensions, is given by the extension of the glass below the upper surface of both decking panel and facia. This offers protection for pedestrians and occupants alike, by preventing head contact with any particles of glass remaining embedded in the rubber seal. The occupants are further protected since deformation of the facia also ejects the glass, by exerting a force on the remaining fragments.

3. Fuel Tanks:

The petrol tank has been redesigned and moved to a position between the rear wheels. Additionally, a solid fire-proof bulkhead separating the passenger compartment from the trunk has been incorporated.

4. Safety Wheels:

Avon safety wheels have been fitted.

5. Brakes:

A dual braking system has been incorporated.

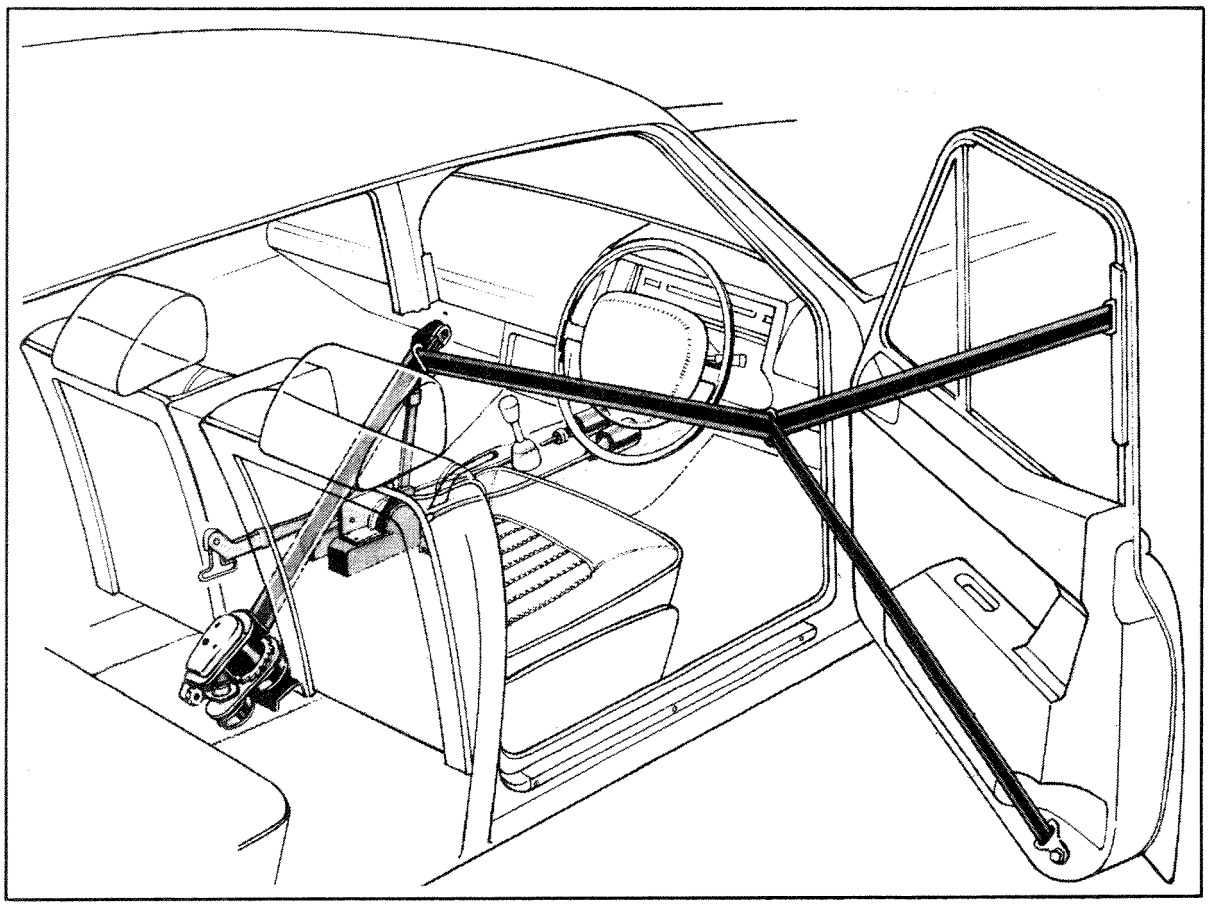


Figure 53

C. **Work Outstanding**

Work on the six projects is scheduled to finish in approximately nine months, by which time we hope to have completed the design and testing of the Phase II vehicles.

Investigations will include:

a. Pedestrian impacts.
b. Alternative bumper systems.
c. Continued occupant protection.
d. Computer simulation for occupant, side and vehicle-to-vehicle crash modes.
e. Reduction in material usage.
f. Further work on steering assemblies.
g. Cost analysis of the final designs.

Development work on a computer model simulating side impacts by mobile barriers is nearing completion. Later our computer engineers will study the possibility of combining the successful frontal model with that for side impacts, in an endeavour to continue our investigations into car-to-car structural compatibility.

The effects on the occupants will be studied on a separate programme, using the structural deformation and deceleration results obtained from the car-to-car analysis. (See BLUK Conference paper by Dr. W. C. Emmerson and Mr. J. E. Fowler.)

S.R.V.I
PHASE I VEHICLE – TWO DOOR
PREPARED AS TEST CONDITION

GENERAL SPECIFICATION

LENGTH	Wheelbase	96.00 in*	(2438 mm)
	Front overhang	37.30 in	(948 mm)
	Rear	40.70 in	(1032 mm)
	Overall	174.00 in	(4418 mm)
WIDTH	Track – front	52.00 in*	(1321 mm)
	Track – rear	52.00 in*	(1321 mm)
	Overall (maximum across rear wings)	64.36 in*	(1635 mm)
HEIGHT (unladen)	Overall	55.35 in*	(1406 mm)
	Bumper C.L. to ground – front	15.00 in	(381 mm)
	Bumper C.L. to ground – rear	16.50 in	(419 mm)
GROUND CLEARANCE (laden)	Minimum	5.52 in*	(140 mm)
	Ramp approach angle	20°	
	Ramp break-over angle	13°	
	Ramp departure angle	17°	
TURNING CIRCLE		31 feet*	(9.45 mm)
BODY CAPACITY	Occupants – front	2*	
	Occupants – rear	2*	
	Luggage	16.8 cu. ft.*	(0.476 m³)
FUEL CAPACITY	Tank	11.5 galls.*	(52.3 litres)
WEIGHTS (estimated)	Total – kerb (ready for road, full tank)	2380 lb	(1080 kg)
	Normal laden (2 up + 50 lb luggage)	2730 lb	(1240 kg)
	G.V.W. (4 up + 100 lb luggage)	3080 lb	(1400 kg)

POWER UNIT	Four cylinders, 1798 cc, OHV 'B' Series with single SU type HS6 carburetter.
TRANSMISSION	4-speed manual gearbox, all synchromesh, remote control gear lever.
FINAL DRIVE	Hypoid bevel crown wheel and pinion.
SUSPENSION	Front – independent by curved torsion bars and lever type dampers. Rear – semi-elliptic leaf springs and telescopic dampers.*
WHEELS/TYRES	Pressed steel disc wheels with 165/70R-13 radial ply tyres.
STEERING	Rack and pinion*; column-mounted on energy absorbing bracket; wheel – energy absorbing.
BRAKES	Hydraulic, with servo assistance*, 9.79 in (250 mm) diameter discs at front, 8.0 in (200 mm) diameter drums at rear.
LIGHTS	
Headlamps	7.0 in (178 mm) diameter incorporating side lamps, mounted behind polyurethane headlamp surround.
Flashers	Front – above bumper mounted in grille between headlamp surrounds. Rear – above bumper at body extremities*.

* Denotes Standard Production Marina Specification.

Tail lamps	Above bumper, adjacent to flashers*.
Stop lamps	Above bumper, adjacent to tail lamps*.
Side Marker Lamps	Front – forward of front wheels mounted in wings. Rear – above bumper in rear lamp cluster.
Reverse lamps	Dual, inboard of stop lamps.
REAR VIEW MIRRORS	One interior, one exterior on driver's door.
INSTRUMENTS/WARNINGS/CONTROLS	
Speedometer	Standard Marina*, Triple Pack instrument cluster.
Controls	Dip, horn, flasher, screen wipe/wash, column mounted*. Supplementary switches mounted for reach below instruments and on central console.
VENTILATION	Through-flow ventilation. Heater mounted on front of dash, controls central. Twin fresh air vents in central console.
INTERIOR OCCUPANT PROTECTION	
Restraints	Front – lap and diagonal static seat belt system, incorporating energy absorbing tear webbing in the upper diagonal and outer lap belts. Rear – lap and diagonal inertia reel seat belts, with tear webbing for energy absorption.
Energy absorbing panels/padding	Interior to Phase I design. Facia panel upper and lower. Header panels front and rear. Cantrails. 'A' and 'B' pillars. Upper and lower rear quarters. Front door and lower rear quarter panels with shoulder, rib and hip protection.
Seats	Front – adjustable fore and aft and tip up for rear seat ingress/egress. Automatic inertia lock prevents seats from tipping when vehicle deceleration exceeds 0.4g. Seats incorporate head restraints. Rear – individual with head restraints. Central hip and shoulder restraints, between seats.
GLASSES	Screen – side and back-light of toughened glass. Back-light heated.*
WASH/WIPE	Screen*.
DOORS	Lock control mechanism, flush fitting*. Structure – anti-collapse, via struts front to rear, with door gap shims, and sill and 'B' post dovetails. Side intrusion guard rails.
BUMPERS	Front and rear – low mounted, rubber/fabric bumper, energy absorbing. Mounted on metal armature to body longitudinals.
BONNET	Standard Marina*, but Polyurethane headlamp surrounds and energy absorbing bumper provide low speed no-damage protection.
RADIO	Fitted in central console, aerial front offside.
BODY STRUCTURE	All steel unitary construction, incorporating Phase I 64 km/h (40 m/h) impact frontal structure described in Table II of conference paper, and Phase I 48 km/h (30 mph) car-to-car side impact structure described in Table IV of conference paper.

* Denotes Standard Production Marina Specification.

S.R.V.2.

PHASE II VEHICLE – FOUR DOOR

GENERAL SPECIFICATION

LENGTH	Wheelbase	96.00 in*	(2438 mm)
	Front overhang	38.40 in	(975 mm)
	Rear	42.20 in	(1072 mm)
	Overall	176.60 in	(4485 mm)
WIDTH	Track – front	52.00 in*	(1321 mm)
	Track – rear	52.00 in*	(1321 mm)
	Overall (maximum across rear wings)	64.56 in*	(1640 mm)
HEIGHT (unladen)	Overall	56.12 in*	(1425 mm)
	Bumper C.L. to ground – front	14.50 in	(368 mm)
	Bumper C.L. to ground – rear	15.50 in	(394 mm)
GROUND CLEARANCE (laden)	Minimum	5.52 in*	(140 mm)
	Ramp approach angle	19°	
	Ramp breakover angle	13°	
	Ramp departure angle	17°	
TURNING CIRCLE		31 feet*	(9.45 mm)
BODY CAPACITY	Occupants – front	2*	
	Occupants – rear	2*	
	Luggage	15 cu. ft.	
FUEL CAPACITY	Tank (over rear axle between wheels)	8.5 galls.	(38.7 litres)
WEIGHTS (estimated)	Total – kerb (ready for road, full tank)	2480 lb	(1126 kg)
	Normal laden (2 up + 50 lb luggage)	2830 lb	(1285 kg)
	G.V.W. (4 up + 100 lb luggage)	3180 lb	(1443 kg)

POWER UNIT	Four cylinders, 1798 cc, OHV 'B' series with single SU type HS6 carburetter*.
TRANSMISSION	4-speed manual gearbox, all synchromesh, remote control gear lever*.
FINAL DRIVE	Hypoid bevel crown wheel and pinion*
SUSPENSION	Front – independent by curved torsion bars and lever type dampers. Rear – semi-elliptic leaf springs and telescopic dampers.*
WHEELS/TYRES	Avon Safety Wheels and Avon Tyres. 165/70 SR 13 radial ply.
STEERING	Rack and pinion*; column – adjustable, mounted on energy absorbing tripod; wheel – energy absorbing.
BRAKES	Dual braking system, hydraulic, servo-assisted, operated by adjustable pedals (5.5 in (140 mm) adjustment); Front – 9.79 in (250 mm) diameter discs; rear – 8.0 in (200 mm) diameter drums; Parking – cable to rears, central hand control.
LIGHTS	
Headlamps	7.0 in (178 mm) diameter with wipe and wash, recessed below bonnet level (daylight condition), raised for night driving by electrical system. Module designed to break away (pedestrian protection).
Side lamps	Above bumpers, adjacent front flashers.
Flashers	Above bumpers at body extremities, for 100° side viewing and coupled for hazard warning.

* Denotes Standard Production Marina Specification.

Tail lamps	Above bumpers, adjacent to flashers.
Stop lamps	Above bumpers, adjacent to tail lamps.
Side marker lamps	Above bumper, in rear lamp cluster.
Reverse lamps	Dual, inboard of stop lamps.
REAR VIEW MIRRORS	One interior, one exterior on driver's door.
INSTRUMENTS/WARNINGS/CONTROLS	
Speedometer	Strip type, incorporating choke, oil, flashers, main beam, brake and ignition warnings.
Controls	Lights, Dip, Horn, Flasher, Screen Wash-Wipe, Headlamp Wipe-Wash and Steering Lock, column-mounted. Supplementary switches mounted for reach below instruments and on central console.
VENTILATION	Through-flow ventilation. Heater mounted on front of dash, controls central. Twin fresh air vents in central console.
INTERIOR OCCUPANT PROTECTION	
Restraints	Front – lap and diagonal passive seat belt system, incorporating tear webbing for energy absorption. Upper diagonal anchorage adjustable for occupant shoulder height. Outer lap and diagonal anchorages mounted on door. System actuated by door opening and central electrically-operated arms mounted in tunnel lift belts clear of seats. Rear – lap and diagonal inertia reel seat belts, with tear webbing for energy absorption.
Energy absorbing panels/padding	Interior to Phase II design. Facia panel upper and lower, header panels front and rear, cantrails. 'A' and 'B-C' pillars and rear quarter panels. Front and rear doors with shoulder, rib and hip protection.
Seats	Front – fixed (incorporating side impact beams). Driver's adjustable for height. Central hip and shoulder restraints between seats. Head restraints also incorporate protection for rear occupant impact, and back panels designed for rear occupant knee impact. Glove box located beneath passenger's seat. Rear – individual with head restraints. Central hip and shoulder restraints between seats.
GLASSES	Screen – toughened (Triplex 10/20 intended but not available). Lower edge recessed below facia to prevent occupant facial lacerations. Side – toughened. Back-light – toughened, heated*.
WASH/WIPE	Screen and headlamps.
DOORS	Lock control mechanism, flush fitting*. Structure – anti-collapse, via struts, front to rear, with door gap shims and sill dovetails.
BUMPERS	Front – low mounted, 'soft' facing for pedestrian protection. Energy absorbing, mounted on metal armature to body longitudinals. Rear – low mounted, 'soft' facing, height compatible with front. Energy absorbing, mounted on metal armature to body longitudinals.
BONNET	Low profile, designed for pedestrian protection.
RADIO	Fitted in central console, aerial rear offside.
BODY STRUCTURE	All steel unitary construction, incorporating Phase I 64 km/h (40 m/h) impact frontal structure which included modified understructure, floor, tunnel, door reinforcement and modified mechanical layout. Front end and bonnet profile to Phase II design. Side structure to Phase I 48 km/h (30 m/h) car-to-car impact design which included fixed seats; roof bow, modified side panels, cross-members and dash assembly.

* Denotes Standard Production Marina Specification.

W. J. DANIELS
Chief Design/Development Engineer
Austin Morris Vehicle Safety Research Project Section
British Leyland Motor Corporation Limited

The object of this paper is to re-define and expand on the themes and strategies depicted in the associated film presentation and to cover progress during the first year of the 2¼ year contract.

PROGRAMME OBJECTIVES

This ambitious programme, devised jointly by TRRL and British Leyland, seeks to apply impact criteria from accident research, towards development of **three different sized FWD vehicles,** with the broad objective of **'Producing Safer Vehicles at Realistic Costs.'**

VEHICLE SELECTION

From our current range of FWD vehicles, of which family some 8 million have been sold, we have selected for development the Mini Clubman 1000, the Austin Morris 1300 and the Austin Morris 1800. These vehicles represent a fair cross-section of the European scene, and from the numbers involved data now accruing from accident research should have a real significance.

CONTRACT REQUIREMENTS

Improvements are sought in each of the following broad sub-sections:—

1. Primary Safety (accident avoidance).
2. Secondary Safety (occupant protection and vehicle to vehicle compatibility).
3. Pedestrian Protection (rated as almost equal in importance to 2 above).

For datum purposes, the programme requires that specimens of each of the vehicle types be submitted to controlled dynamic testing in each of the major crash modes, supplementing this work where necessary with both static crush and computer simulations.

The development cycle to follow in train with vehicles incorporating the latest condition of up-date being made available for this conference.

PROGRESS

Chart 1 – Summarises the extent of preliminary testing now completed.

DATUM TESTS COMPLETED					MINI	1300	1800
Full Frontal	Barrier	60 kph	2 up front	Restrained	✓	✓	✓
Full Frontal	Barrier	25 kph	Full complement	Unrestrained	✓	✓	✓
¼ Frontal	Barrier	50 kph	2 up front	Restrained	✓	–	✓
90°Side	Bullet	30 kph	2 up front	Restrained	–	–	✓
Car/Car	Target	Static	1 front 1 rear impact side	Restrained	✓	–	✓
45° Side	Bullet	30 kph	2 up front	Restrained	–	–	✓
Car/Car	Target	Static	1 front 1 rear impact side	Restrained	✓	–	✓
90° Side	Pole	15 mph	1 front 1 rear impact side	Restrained	✓	✓	✓
Roll-over	Standard No 203	50 kph	1 front 1 rear driver side	Restrained	✓	–	✓
Static Crush	–	–	–	–	✓	–	✓
Pedestrian	–	5-15 mph	–	–	✓	✓	✓

CHART 1

Chart 2 – Summarises development progress during the first year of the contract.

MODEL	MINI	1300	1800
Test Full Frontal 60 kph – Stage 1 Reconstruction	✓	✓	✓
Test (90° Side Impact Target – Stage 1 Reconstruction	✓	–	–
(90° Side Impact Bullet – Stage 1 Reconstruction	–	–	✓
Test Static Crush – Stage 1 Reconstruction	–	–	✓
Continuous Development – Pedestrian	–	✓	–
Continuous Development – Brakes	✓	–	✓
Continuous Development – Handling	✓	–	✓
Continuous Development – Components	✓	✓	✓
Demonstration Vehicles – Stage 2 Reconstruction	✓	–	✓

CHART 2

PROGRESS – SECONDARY SAFETY

Reverting to the sequence as shown on film, occupant protection will be first considered, opening this chapter with a review of the accident statistics relating to it.

Figure A ii Shows how the various accident types involving motor cars are apportioned in the UK.

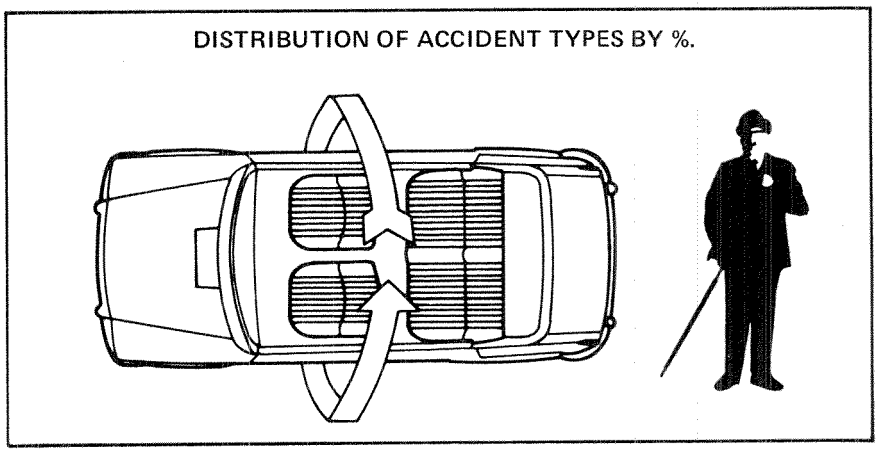


FIGURE A i

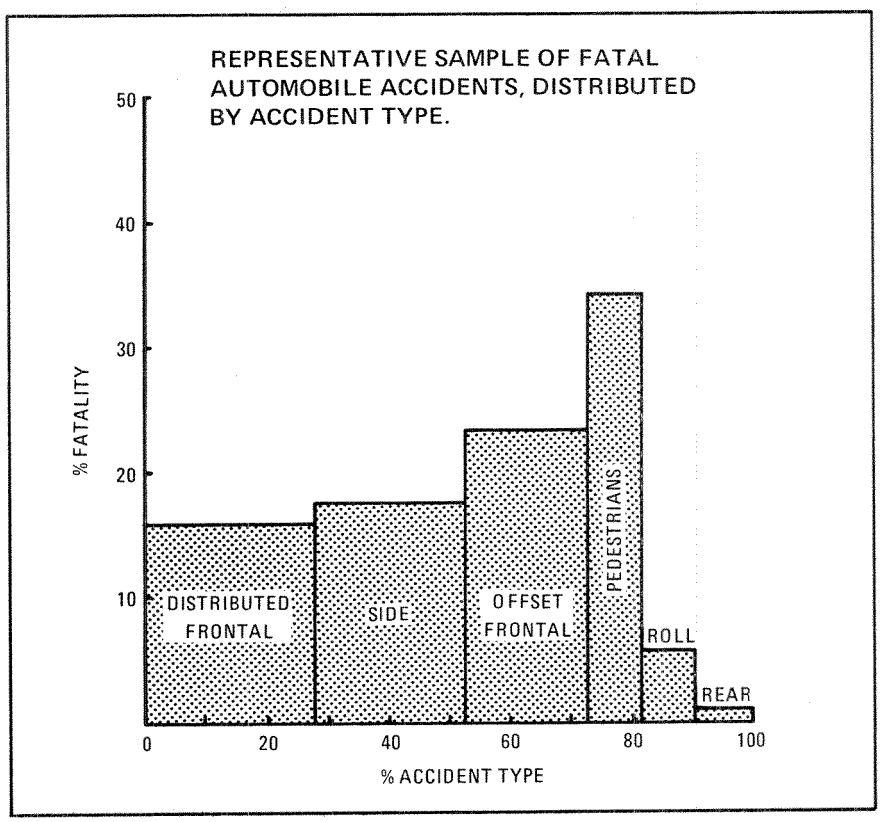


FIGURE A ii

This diagram is quantified as follows:—

	Accident involvement	Total fatality involvement
Block 'a' — Distributed Frontal	28%	16%
Block 'b' — Side Impacts	25%	18%
Block 'c' — Part Frontal	20%	24%
Block 'd' — Pedestrians	9%	35%
Block 'e' — Roll-overs	8%	6%
Block 'f' — Rear-End Shunts	10%	1%

From these figures it can be see that 16 + 24 = 40 per cent of occupant fatalities, together with $\frac{9}{10}$ x 35 = 32 per cent pedestrian fatalities, sum up to 72 per cent of fatalities being associated with the front of the car. Consequently, the vehicle front is the key location from which design and development studies must generate. The pattern is similar for serious injuries.

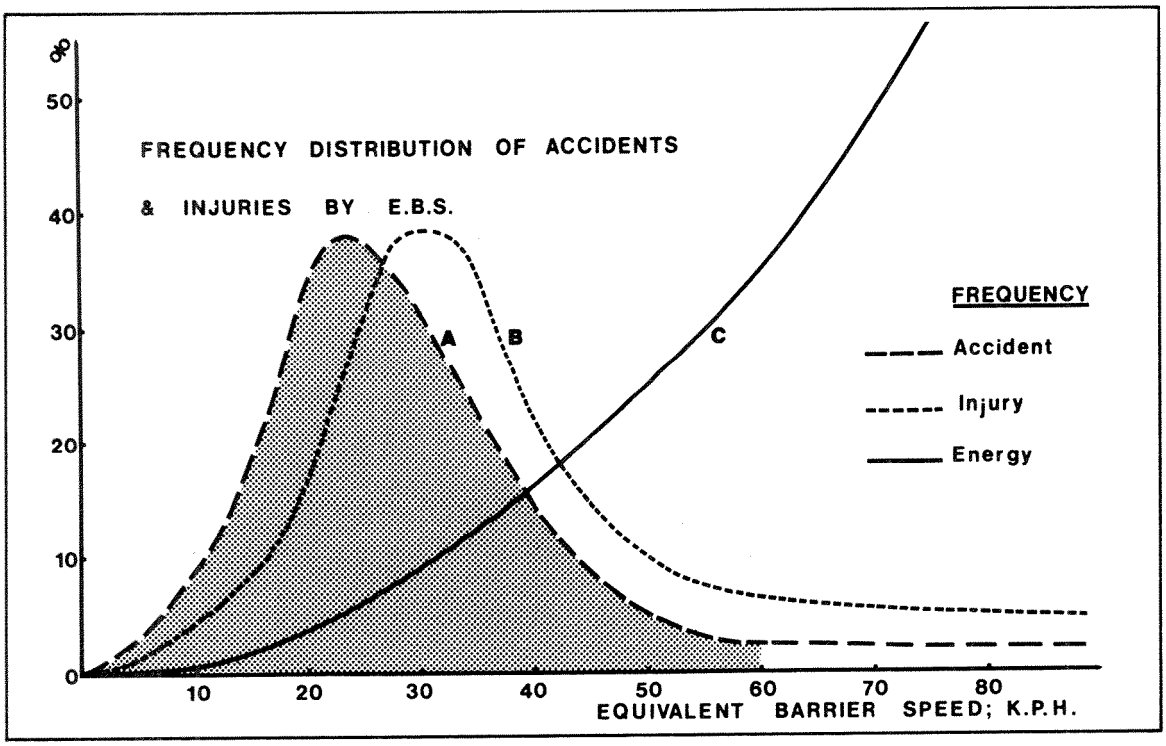


FIGURE B

Reviewing the frontal impact factors in respect of occupants,

Figure B — illustrates the following significant factors:—

Curve 'a' — The incidence of frontal accidents rises rapidly to 25 kph (indicating higher incidence of urban accidents) and declines thereafter.

Curve 'b' — At the same time, injury frequency increases up to around 30 kph, with a decline thereafter, although severity will tend to increase as a function of speed.

Curve 'c' — Energy levels (implying weight and cost) increase as the square of the speed.

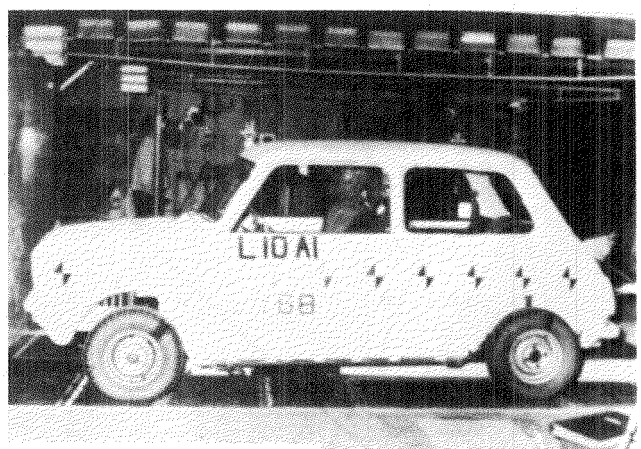
A study of this information reveals that of all accidents producing some kind of injury, 95 per cent occur within a 60 kph band-spread, beyond which the return is so marginal as to be unsupportable in terms of the cost/benefit relationship. Our contract stipulates 60 kph for frontal impact testing. To allow a margin for experimental error, cars are put into the block at a nominal 65 kph, and the following trio of performance comparisons derive therefrom.

Full Frontal Sequence

Starting with the small car:— Mini Specimen — Datum vehicle

No L10 A1

Base vehicle weight	1314 lb (596 kg)
Test weight, 2 up	1654 lb (751 kg)
Crush — dynamic	28.5 inches (72.3 cm)
static	26.44 inches (67.1 cm)



External view before impact

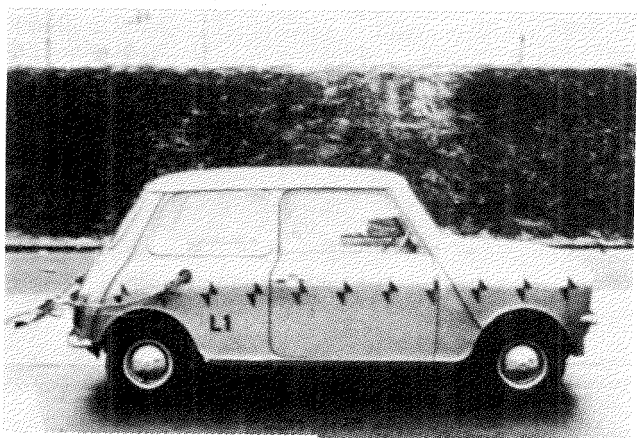
External view after impact

Comment: Occupant space inadequate. Occupants experienced decelerations above survival limits at both head and thorax.

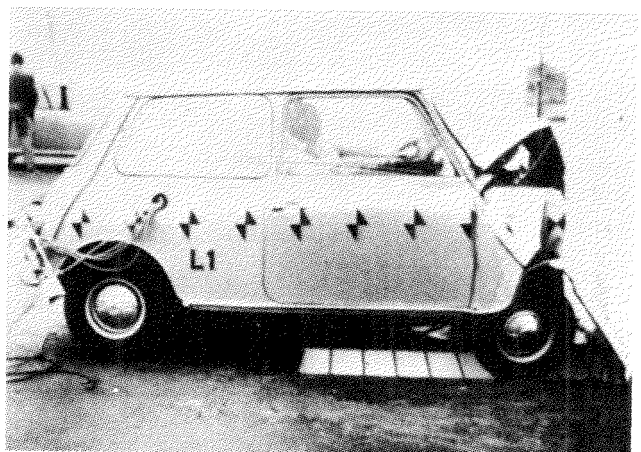
Internal view after impact

Mini Specimen — Stage 1 Reconstruction
No L10 B1

Base vehicle weight	1296 lb (588 kg)
Test weight, 2 up	1636 lb (743 kg)
Crush — dynamic	24.5 inches (62.2 cm)
static	20.25 inches (51.4 cm)



External view before impact



External view after impact

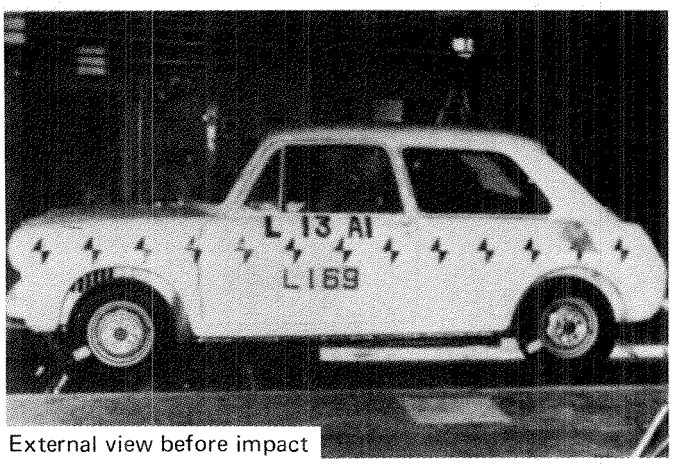
Comment: This specimen incorporated our first attempts toward satisfying occupant space criteria. To reduce intrusion hazards, the engine and road wheel complex was first set forward within the front profile, the bulkhead-mounted ancillaries moved outboard and the car heater moved from its central position inside the vehicle onto the port side valance, with the joint object of removing impediments which violate the car interior, and providing optimum space for ride-down. Additionally the frontal understructure was re-routed behind the wheels and grouted directly into the body sill structure. Bracing members were introduced in the doors and from the 'B' post to the rear wheel arch.

Internal view after impact

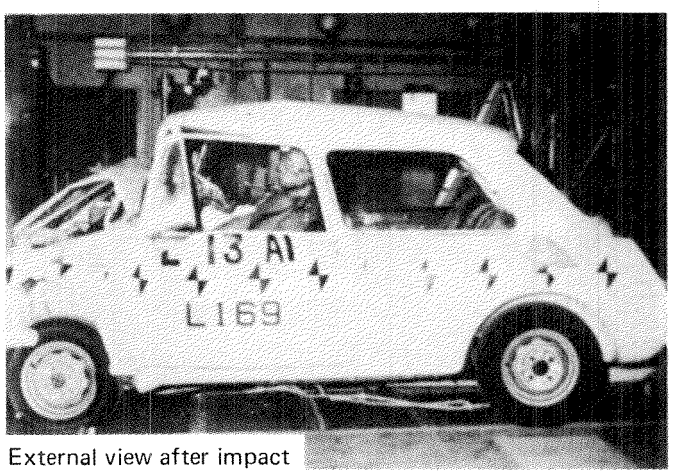
The outcome — a marked improvement in occupant space, less severe folding of the floor and windscreen retention. Occupants — thorax decelerations within the criteria; no reference available regarding head due to instrument failure.

Following with the 'medium' car:— 1300 Specimen — Datum vehicle

No L13 A1	Base vehicle weight	1730 lb (785 kg)
	Test weight, 2 up	2070 lb (940 kg)
	Crush — dynamic	29.0 inches (73.7 cm)
	static	25.6 inches (65 cm)



External view before impact



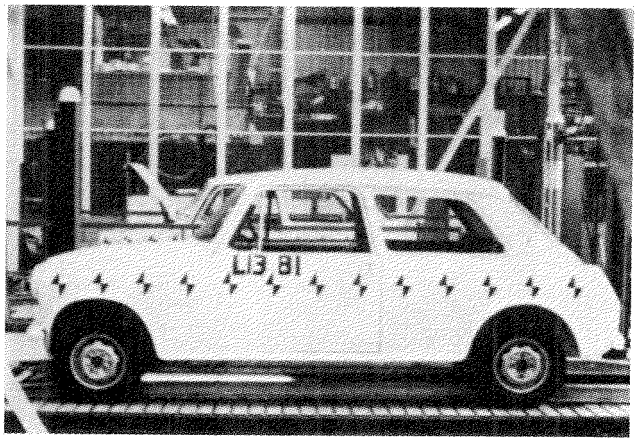
External view after impact

Comment: Occupant space inadequate, with severe intrusions of fascia, steering, front floor, and pedals. Occupants — failures recorded for both head and thorax.

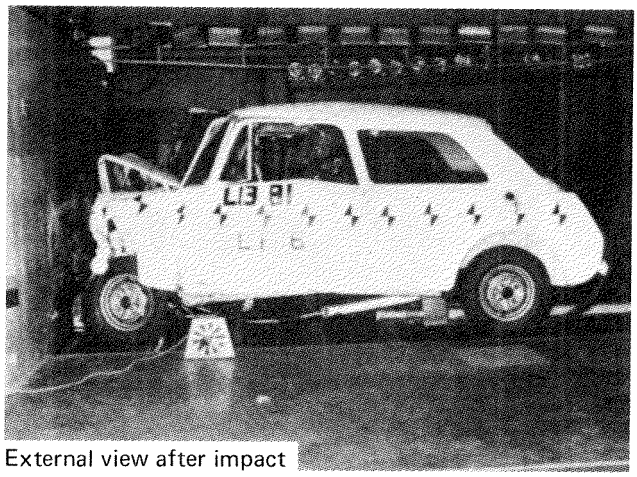
Internal view after impact

1300 Specimen — Stage 1 Reconstruction
No L13 B1

Base vehicle weight	1725 lb (783 kg)
Test weight, 2 up	2065 lb (935 kg)
Crush — dynamic	25.0 inches (63.5 cm)
static	25.6 inches (65 cm)

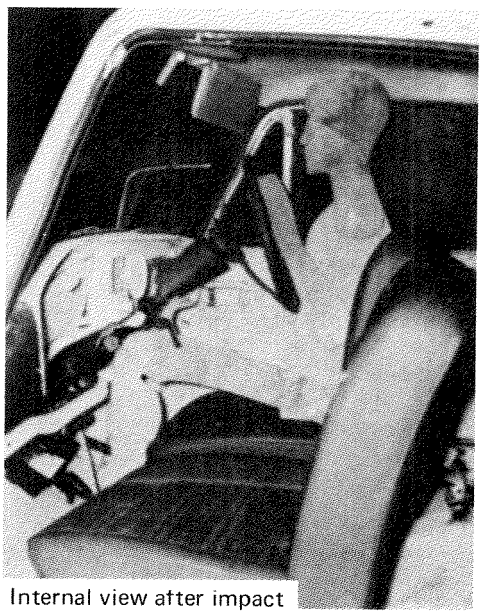


External view before impact



External view after impact

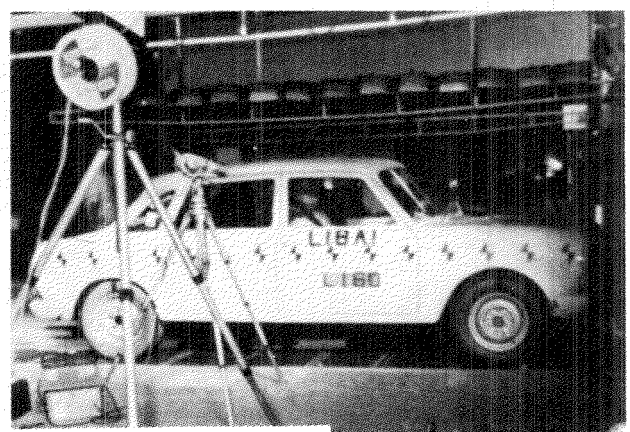
Comment: This specimen followed the same theme as employed in the earlier Mini, with the object of confirming the general principles. Again, a marked improvement in occupant space integrity was achieved, but failure still registered for occupant head and thorax.


Internal view after impact

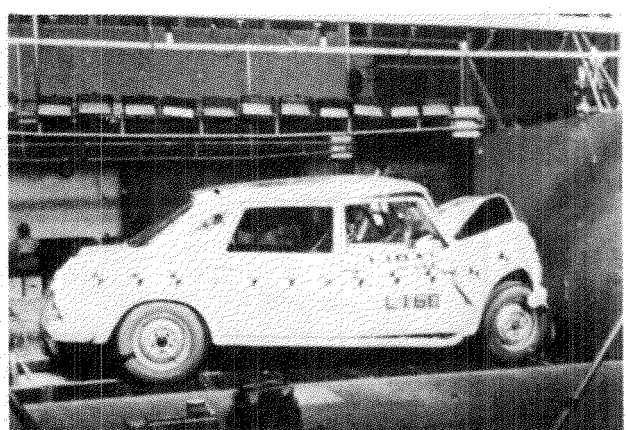
Next in sequence the 'large' car:— 1800 Specimen — Datum vehicle

No L18 A1

Base vehicle weight	2436 lb (1106 kg)
Test weight, 2 up	2776 lb (1260 kg)
Crush — dynamic	31.0 inches (78.7 cm)
static	24.75 inches (62.9 cm)

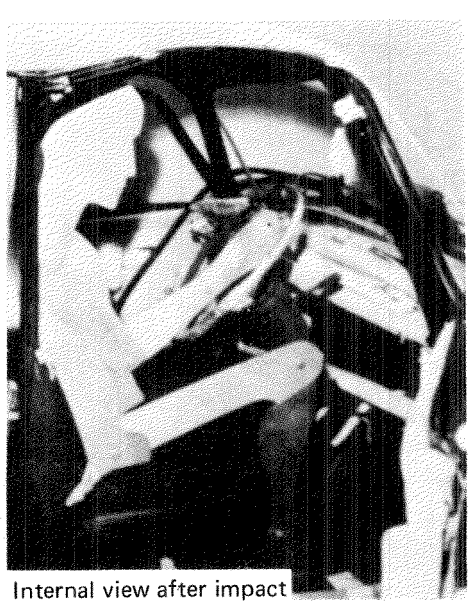


External view before impact



External view after impact

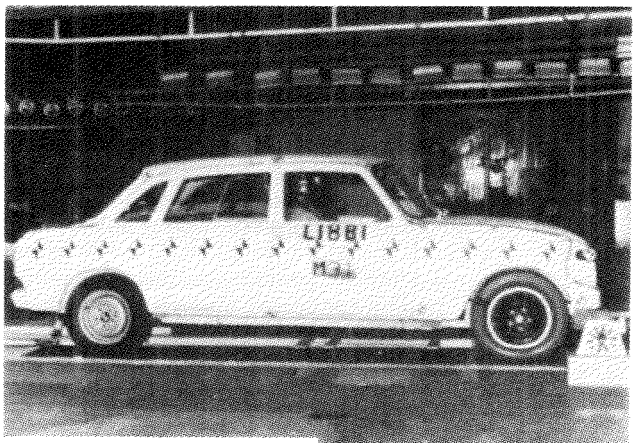
Comment: Occupant space barely adequate. Occupants — failures at head and thorax.



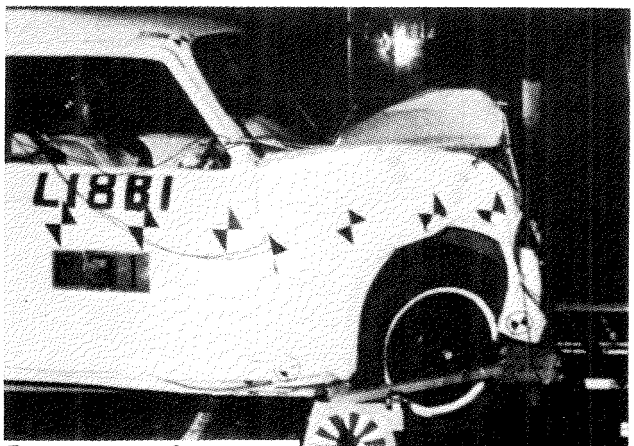
Internal view after impact

1800 Specimen — Stage 1 Reconstruction
No L18 B1

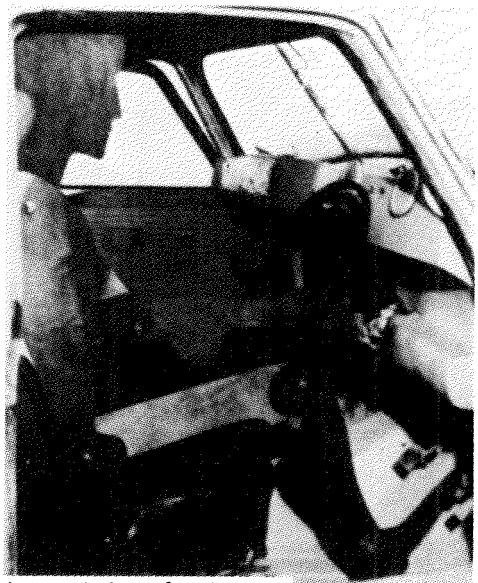
Base vehicle weight	2352 lb (1068 kg)
Test weight, 2 up	2692 lb (1222 kg)
Crush — dynamic	27.5 inches (69.8 cm)
static	16.5 inches (41.9 cm)



External view before impact



External view after impact


Internal view after impact

After studying the characteristics of the preceding impacts both from high speed film records and the finished state of the vehicles, the engineering of this specimen was arranged in general conformity with the Stage 1's of the smaller vehicles, together with certain new innovations.

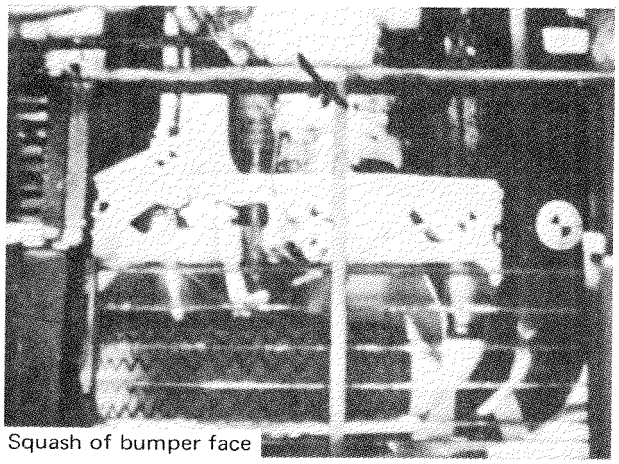
To start, the engine and wheels were moved forward and the bulkhead mounted ancilliaries moved outboard beyond the engine silhouette, then the underframe was swung outward behind the front wheels to abut the body sill members. In consequence of the wheel re-disposition of this model, the suspension media was re-arranged to position its spring element vertically above its upper wishbone, making possible removal of the large circular crossmember currently sited behind the engine, thereby enhancing the engine to bulkhead clearance. A forward movement of the steering rack follows the wheel re-disposition, and this has allowed inclusion of a rack to column link made flexible in the bend mode, whilst the column itself is rigidly supported in a robust fascia member, which is in turn grouted only to the 'A' post structure to divorce it from any intrusion of the bulkhead panel. The toeboard area also received attention, with the object of so positioning the front occupant that in an impact his knee will drive into padding at a level above his hip joint, thereby prescribing his further trajectory. Opportunity was also taken to introduce first thoughts on the pedestrian problem, and a low mounted 'soft-nosed' bumper was installed. Since our contract involved the fitment of total mobility tyres, these also were fitted so that their behaviour in the crash mode could be studied. As a partial offset against the increased weight of these new components, the more modern 'E' series engine was substituted. Doors were also strutted as on the other prototypes.

(The general theme, as applied to the 1800 Stage II Demonstration vehicle, is illustrated at Appendix I – Sketches 1-6 inclusive.)

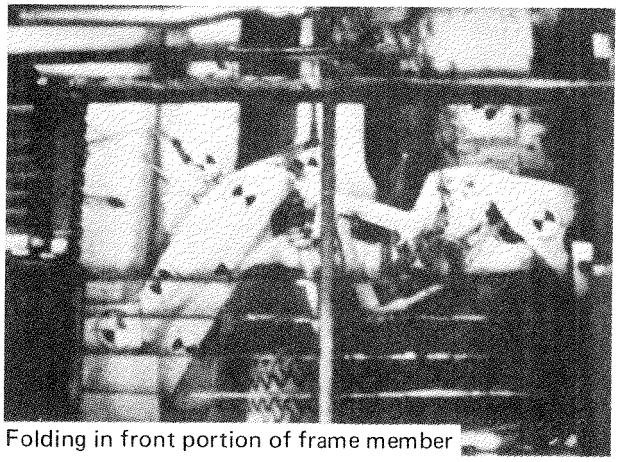
The result:–

Occupant space virtually intact
Steering penetration 1¼ inches
Screen retention 100 per cent
Occupant's decelerations reduced, but the gain insufficient to register pass levels – indicating that considerable work remains to be done on restraints.

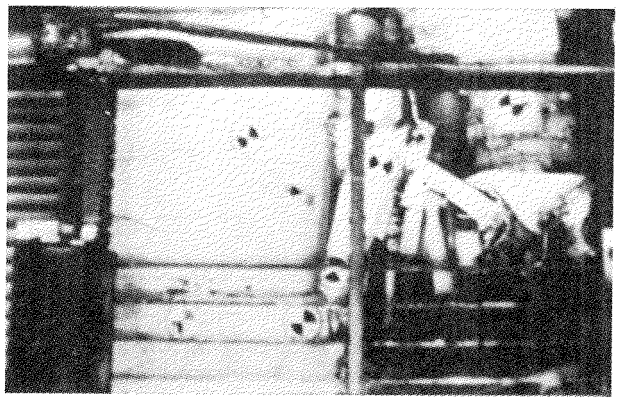
For additional information the following photographs (extracted frames from high speed film of the underside of this vehicle during impact) illustrate the sequential collapse pattern of its understructure:—



Squash of bumper face



Folding in front portion of frame member



Buckling of rear portion of frame member at maximum excursion.
Note the flattening against the toe-board without penetration.

Conclusion: Design theory well substantiated in respect of occupant space, with scope for development both in respect of restraints and optimisation of material usage.

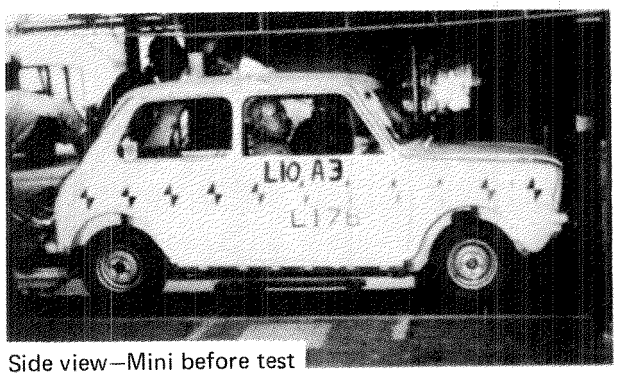
Future related programmes: Sled testing for restraints and paddings. Static Crush testing and computer simulations for material economy. Confirmatory dynamic test on up-dated specimen.

Part Frontal Sequence

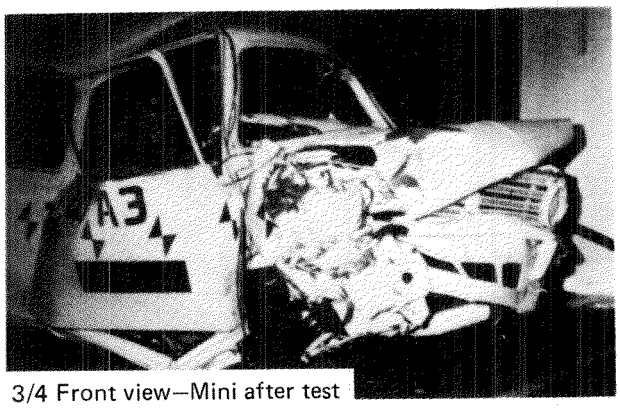
To supplement this full frontal programme, our contract includes a representation of the more typical collision mode, i.e. oblique or part frontal.

The specification defines: ¼ frontal overlap into barrier at 50 kph.

Datum testing has been completed on each of the 3 vehicle types, and the following photographs illustrate how the small vehicle fares.



Side view—Mini before test



3/4 Front view—Mini after test

Interior view—Mini after test

Severe penetration extended for some 42 inches, and occupant space for the driver is very restricted — surprisingly, the instrumented readings suggest his survival, and the passenger would seem to have only minor injuries.

Comment: Further investigations in respect of this accident mode are being deferred, pending conclusion of the Full Frontal mode, when some alleviation would be anticipated.

On the basis of the evidence now available, we submit that this '¼ overlap' test seems too severe. To simulate more closely typical part front impacts, some amendment of this test specification seems desirable.

Side Impact Sequence

Our specification calls for Side Impact Tests to be performed both at 45° and 90° angles, at a speed of 30 kph.

Efforts to date have been concentrated on the small car (Mini) as the target, and the large car (1800) as the bullet.

The evidence of the two datum tests suggests that the 90° impact is the more severe, and it is our intention to concentrate developments only on this method hereafter.

The following sequences are of 90° impacts at 30 kph:—

TARGET	— Datum vehicle	
Mini Specimen	Base vehicle weight	1314 lb (596 kg)
No L10 A2/4	Test weight, 2 up	1654 lb (751 kg)
(Previously used in 25 kph unrestrained test)	Occupants restrained	1 front, 1 rear on impact side
BULLET	— Datum vehicle	
1800 Specimen	Base vehicle weight	2436 lb (1106 kg)
No L18 A4	Test weight, 2 up	2776 lb (1260 kg)
	Occupants restrained	2 front

3/4 Side view of target before impact

3/4 Side view of target after impact

Front view of bullet before impact (previously used in 45° side impact test)

Test result:—

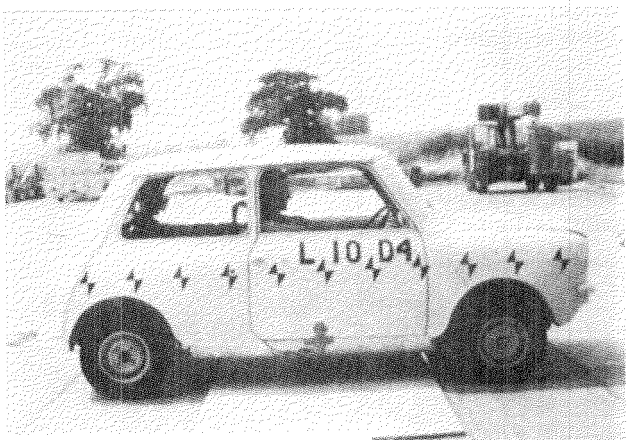
Target —	Penetration maximum, adjacent 'H' point	External — 7½ inches Internal — 5½ inches
	Occupant: Front	Severe injury to shoulder and hip was indicated
	Rear	No injury indicated
Bullet —	Damage	Minor—vehicle has robust front
	Occupants	No injury indicated

Comment: Penetration on the target car was due in large measure to override of door to sill, in combination with rigidity of the bullet front end with its bumper at around 'H' point level.

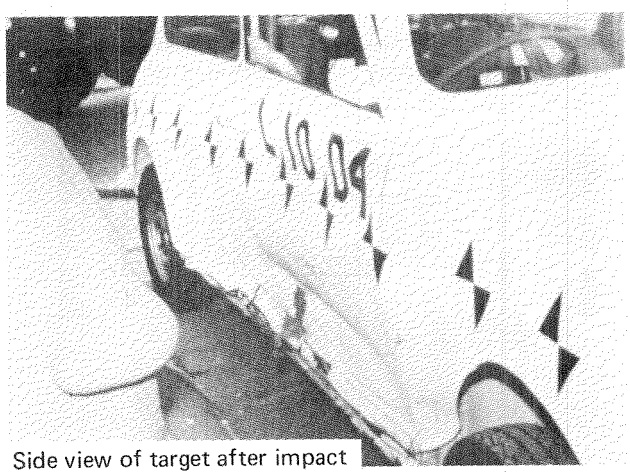
For the second phase in this section of the programme both problems were tackled as follows:—

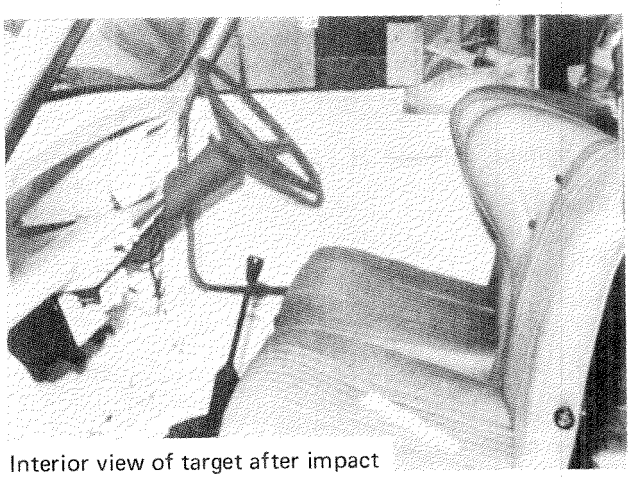
Target — The sill was raised to the highest level consistent with ingress to create an overlap, stiffened in bending, and so re-shaped to generate an interlock of the panelling within the overlap during the actual impact. The front seating was re-formed to achieve shoulder side support, with mating padding at the door.

TARGET	— Stage 1 Reconstruction for side impacts	
Mini Specimen	Base vehicle weight	1344 lb (610 kg)
No L10 D4	Test weight, 2 up	1684 lb (764 kg)
	Occupants restrained	1 front, 1 rear on impact side



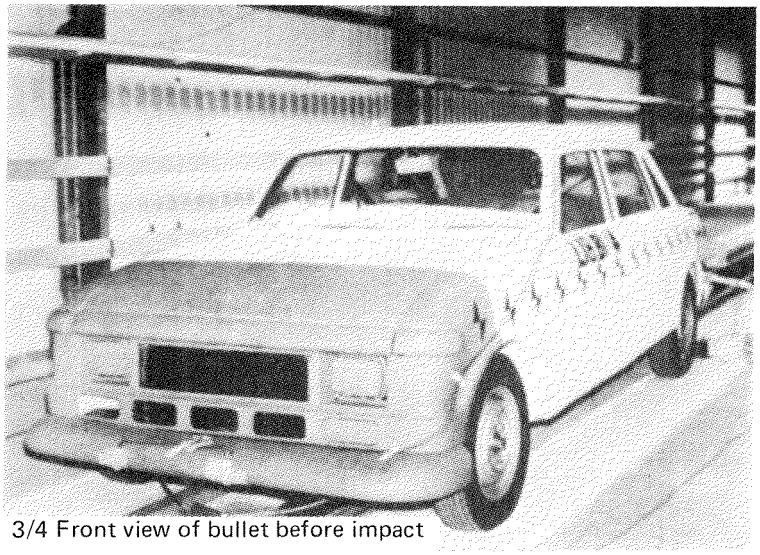
Side view of target before impact


Side view of target after impact


Interior view of target after impact

Bullet – This vehicle was precisely as described for Specimen L18 B1. The key elements in this case being the soft-nosed bumper, located low to strike the target at sill height, together with a frontal body structure less aggressive than on the datum car.

BULLET	–	Identical to L18 B1	
1800 Specimen		Base vehicle weight	2352 lb (1065 kg)
No L18 B1/4		Test weight, 2 up	2692 lb (1222 kg)
		Occupants restrained	2 front



3/4 Front view of bullet before impact

Test result:–

Target	–	Penetration maximum, adjacent 'H' point	External – 4 inches Internal – 2½ inches
		Occupant: Front	Pass
		Rear	A general pass, but failure on one rib
Bullet	–	Damage	Relatively minor
		Occupants	No injury indicated

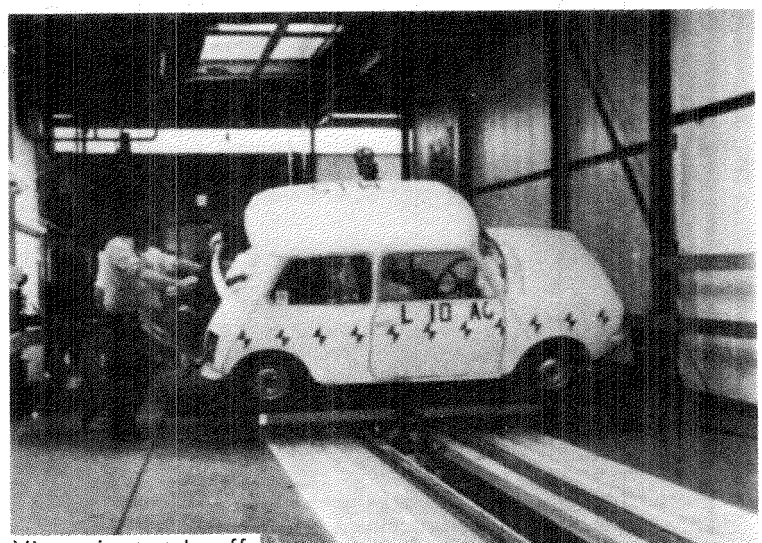
Comment: Door to sill interlocking on the impact side of the target car was as anticipated, whilst the far door opened without effort. A 55 per cent reduction of internal penetration, accompanied by no increase in injury severity for the occupants, is justification for the principles. It should be noted that the initial trajectory of the target car is influenced by the level of the first strike, and in this case the target first leans toward the bullet to delay and reduce the effect of the critical first contact, head to vehicle. For the future programme, an increase of impact speed seems to be indicated. The more difficult sequence to cater for is the 'Side Pole', for which we have as yet only datum evidence for the three cars.

Roll-over Sequence

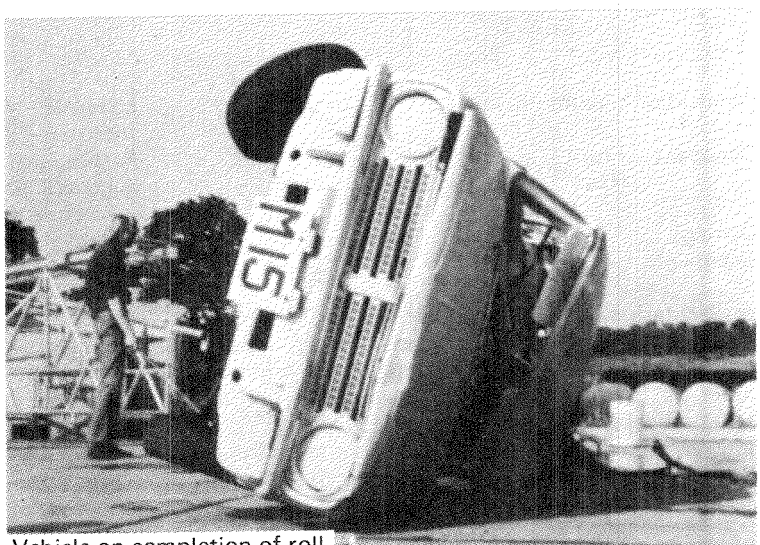
Our specification calls for Roll-over testing to be conducted by the 'FMVSS 208' method, with an initiation speed of 50 kph. The occupants, 1 front and 1 rear, are both restrained and seated at the driver's side, which makes first contact with the ground.

To date we have only datum evidence on the Mini and 1800 to offer.

Mini Specimen No L10 A6	– Datum vehicle	
	Base vehicle weight	1314 lb (596 kg)
	Test weight, 2 up + ...	1689 lb (767 kg)
	Occupants restrained	1 front, 1 rear (driver's side)



View prior to take-off



Vehicle on completion of roll

Test result:–

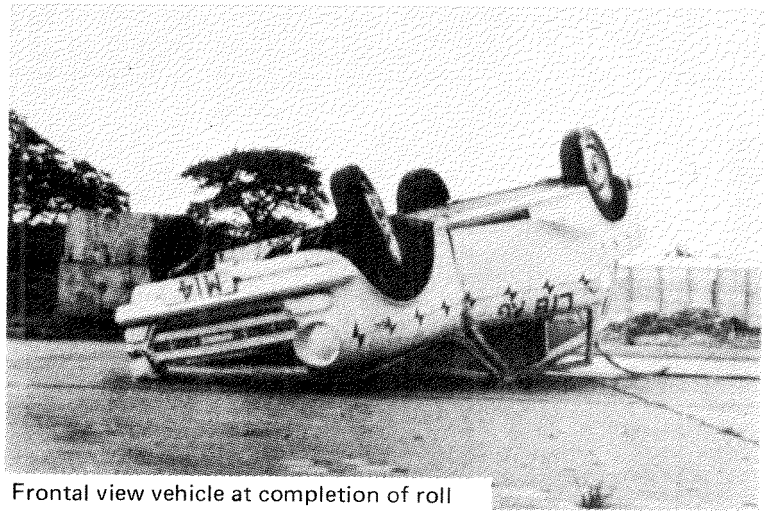
Maximum intrusion after test: 10 inches
(Against specification requirement of 7.9 inches)

Injury criteria: Apparently satisfactory, but with doubt arising in respect of repeated side head impacts.

Comment: The test clearly defines the areas of body structure needing attention, e.g. screen surround, upper 'A' post, doors, and door lock mechanism. In this instance one door jammed shut, due to local distortion at the external handle from first ground contact.

1800 Specimen No L18 A6	– Datum vehicle	
	Base vehicle weight	2436 lb (1106 kg)
	Test weight, 2 up	2819 lb (1280 kg)
	Occupants restrained	1 front, 1 rear (driver's side)

Frontal view prior to take-off



Frontal view vehicle at completion of roll

Test result:–

Maximum intrusion after test: 12.9 inches
(Against specification requirement of 7.9 inches)

Injury criteria: Apparently OK, subject to the same reservation as for the Mini.

Comment: This test highlighted the short-comings of protruding door handles, which in this case made contact with the ground, freed a front door lock, and allowed the door to swing open, permitting the driver's arm to swing out into space. Without his restraint, total ejection with consequent serious injury would have been the probable result.

Again, as on the Mini, the areas of body structure needing attention were clearly defined.

PROGRESS – PEDESTRIAN PROTECTION

As on the film, this chapter will open with a review of the statistical guide lines.

Chart 3 – shows in the left-hand columns the UK figures for 1972. The right-hand columns are projections from these figures to indicate how the pedestrian will feature on the charts, once the use of some restraint becomes mandatory.

	1972		Projected with compulsory Seat Belt usage	
	Fatalities	Serious Injuries	Fatalities	Serious Injuries
Pedestrians	3,089 (39.7%)	24,200 (26.5%)	3,089 (47.6%)	24,200 (37.25%)
Motor Cyclists and others	1,665 (21.1%)	25,800 (28.2%)	1,665 (26.0%)	25,800 (39.75%)
Car Occupants	3,034 (39.2%)	41,300 (45.3%)	1,699 (26.4%)	14,883 (23.0%)

Summarising for the fatalities, this indicates that of the reduced number of deaths, occupants will register about 26 per cent, whilst pedestrians will head the list at about 48 per cent.

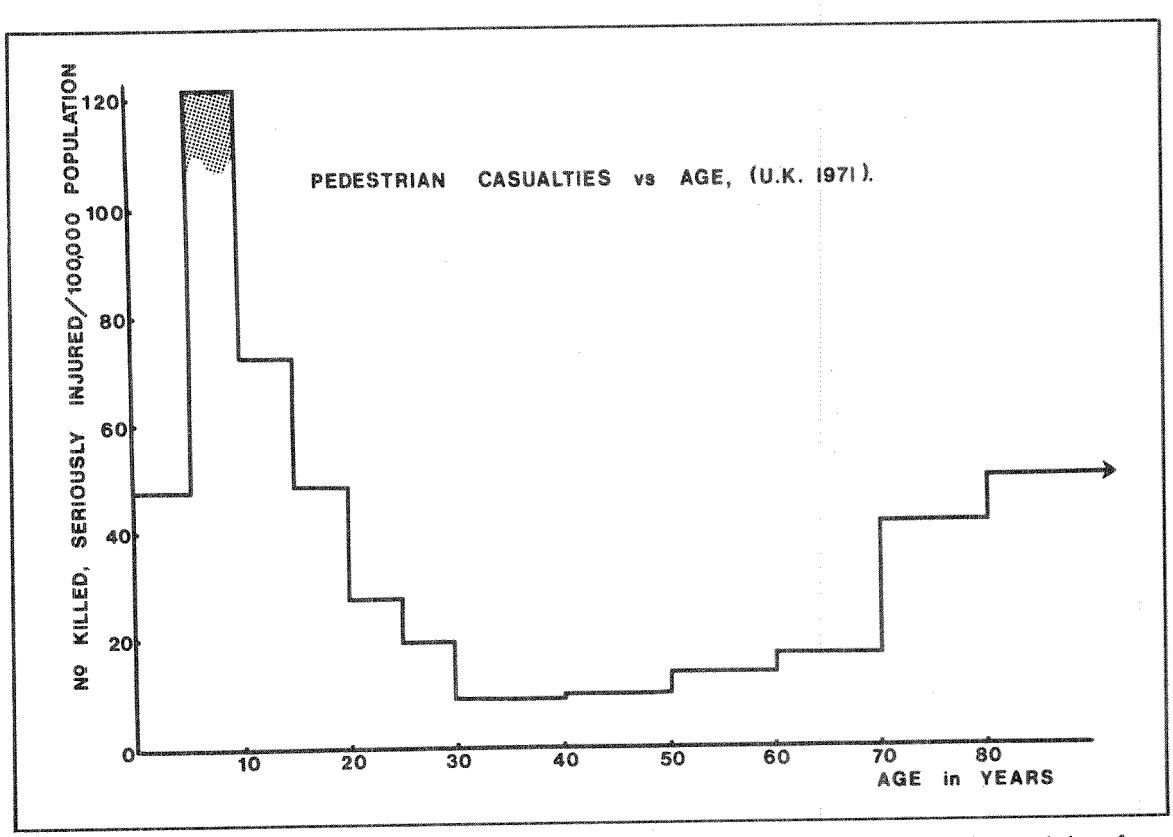
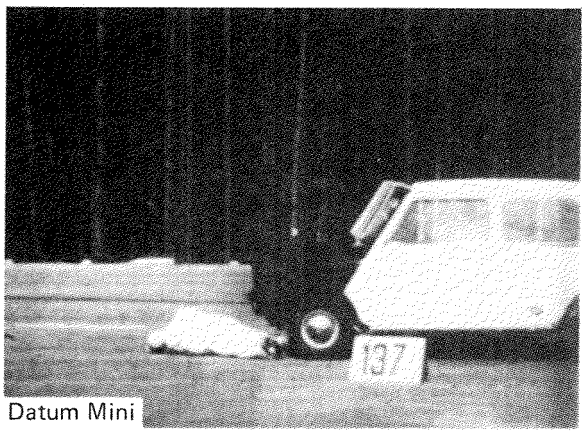
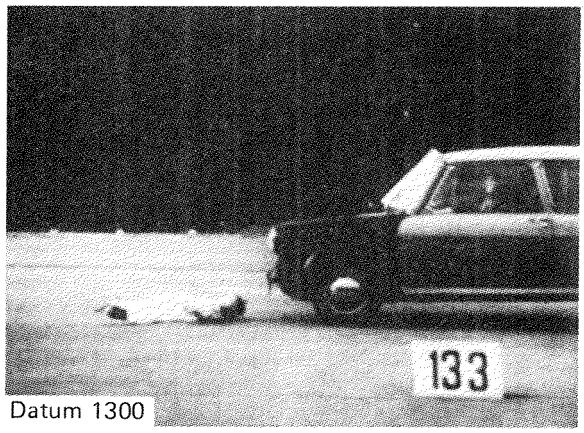


Figure C – shows how children in the five to nine year age group dominate the statistics, and therefore merit prime consideration.

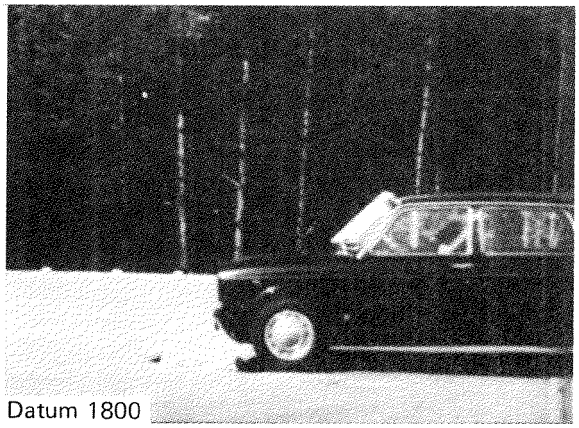
The following photographs illustrate how a child pedestrian fares in impacts at around 16 kph:—



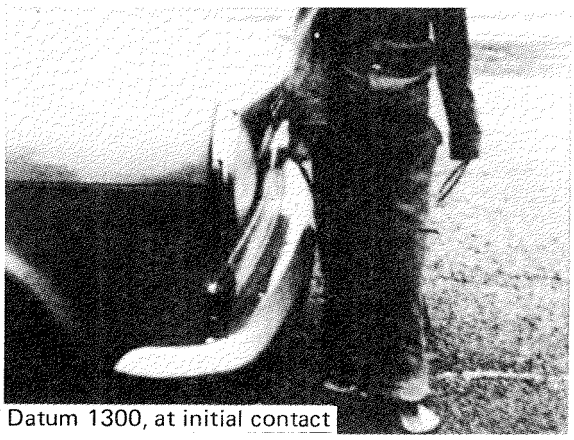
Datum Mini



Datum 1300


Datum 1800

The following photograph identifies the basic problem:—


Datum 1300, at initial contact

Note: The bumper makes first contact with the child in the leg area, well below his centre of gravity. This promotes his rotation toward the oncoming vehicle (i.e. anti-clockwise for the view shown).

Immediately thereafter a second contact will occur, with the aggressive prow of the vehicle at around the child's centre of gravity height, and this impact will throw him violently to the ground, with critical head to ground impact and ensuing consequences.

One method of tackling this hazard suggested by TRRL has been investigated in principle, concentrating on our 1300 vehicle as a base-line.

The following photograph sequence shows how this system works:—

Modified 1300, child's initial contact

Modified 1300, child's trajectory

Modified 1300, child held by catcher

These photographs being frame extracts from high speed film record need no further explanation, except to emphasise that any such addition leaves the vehicle buyer with a substantial **on-cost from which he personally will derive no benefit.** Segregation of traffic and pedestrians wherever practical seems to be a better answer.

Clearly the efforts so far are somewhat embryonic, but development continues apace, and some more concrete proposal should emerge in due course.

As an interim measure, it can be postulated from existing evidence that the combination of low bumper and low profile bonnet would alone produce a worthwhile contribution toward reducing pedestrian injury.

Our two exhibits have been shaped to represent this theme.

Departing now from the associated film text we shall next consider the various aspects of:

PRIMARY SAFETY

Although primary safety may be defined in qualitative terms it is an aspect that is difficult to quantify. It is widely recognised that any advances achieved in this field are rapidly absorbed by the driver 'taking up the slack'.

There is an urgent need for extended research into these fundamental issues, in lieu of which one must at this time resort to the common sense approach, backed up by 'seat of the pants' experience and reinforced by scientific measurement where this is practical.

Our specification calls for a number of features within this category, which break down into several sub-groupings, as follows:—

Handling

The overall aim is to develop handling characteristics superior to those of the present day. The specification requires:—

1. No drastic change in characteristic when the vehicle is used at the extremes of loading, tyre pressures and tyre wear.
   To this end, constant mobility tyres such as the Dunlop Denovo offer a major contribution.

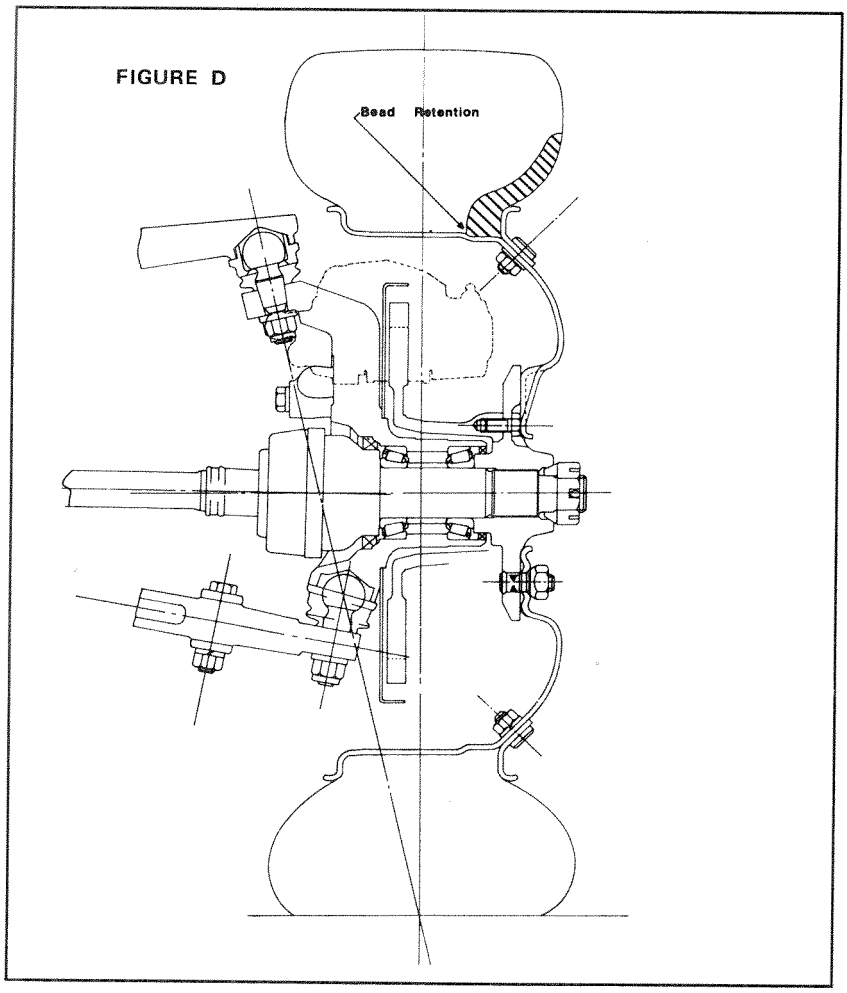


Figure D — shows in section the front suspension unit as arranged for our 1800 vehicle. **Note:** the zero offset and the wheel configuration, both for retaining the Denovo tyres and optimising brake space, with tyre concentricity implicit from the rim and nave in one piece. Some measure of energy absorption in severe impacts is also sought from the nave shaping.

2. No sudden change in behaviour, with lateral accelerations approaching limit conditions.
   Vehicle roll offers a graduated warning; the difficulty being to engineer the suspensions so that no wheel shall lift prior to break-away of the tyres, whilst holding total roll within an acceptable level.
3. No sudden change in behaviour with power on and off. The vehicles being Front Wheel Drive, we advocate a 'zero offset' where the centre lines of wheel and king pin converge at the ground.
4. Lateral digression in wind gusts is minimal.
   This characteristic is not a major problem on our FWD vehicles. Checks on this must await the post conference era, when the accumulated effect of vehicle revisions can be examined.
5. Maintenance of control, in the presence of unbalanced forces from either brakes or surface adhesion.
   'Zero offset' as in 3 above is considered the best all round compromise.

Brakes

The specification requires that the large vehicle at least be fitted with the best equipment reasonably available, together with warnings for the driver in the event of any major fault. Cost/effectiveness to be determined in due course. The criteria against which alternative systems be evaluated are too extensive for inclusion in this paper.

Current indications are that whilst a full anti-lok system is essential to fulfil the total requirement, it may prove acceptable only on the large car, in terms of cost. Our 1800 demonstration vehicle is so fitted, using the AP 'Lockheed Antilok' system as representing the first of several to be tried.

For the Mini version, we intend to explore the 'grey area' that currently exists between the orthodox and full anti-lok systems.

Instrumentation

The specification calls for fitment of a 'Head-up Display' Speedo. This has not yet been implemented.

For this conference, our 1800 vehicle is equipped with a 'Strip-type' Speedo, mounted for peripheral vision above the rim of the steering wheel, but still within the eye-level silhouette of the bonnet. On either side of this instrument are banks of warning lights and switches, all within peripheral vision and the latter within reach. One warning remains inoperative, since the various means so far explored fail to reveal an economic solution to the 'Low Tyre Pressure' hazard. As normal British Leyland practice, the Horn, High/Dip Beam, Flasher, and Screen Wiper/Washer controls are all contained in column mounted stalk switches, with Column Lock/Ignition Switch adjacent.

Lights and Lighting

In accordance with specification the equipment is as follows:—

1. Car lighting system — in general conformity with UK, ECE and EEC requirements, with some latitude where there is conflict with any specific objective, e.g. headlamp location — to circumvent pedestrian hazard. On our vehicles these lamps have been set back to simplify fitment of a Wiper/Washer system and to cater for headlamp levelling.
   This involves a marginal relaxation in current regulations.
2. **Flasher lights** — outboard viewing angles increased from the regulation 80° to a minimum 100°, with 4-way hazard signalling incorporated.
3. **Stop lamps — 2-level system** — the outer pair in addition to normal function are required to 'lock-on' when the brakes arrest the vehicle, switching off only when the vehicle moves off or the ignition is cut. An inner adjacent pair to illuminate when braking decelerations reach 0.4 g.
   The 1800 vehicle complies, but due to space limitation some latitude has been taken with the placement of these lamps on the Mini.
4. **Reversing light** — Fitted as required on large car.
5. **Failure warnings** — for side, rear and outer brake lights, visible to seated driver.
   Provision made for visual warning at fascia on large car. No such provision on Mini.

GENERAL

So far no mention has been made of 'front-pole' or rear-end impacts. The former does in fact appear within our total programme, but its timing is well after this conference.

As regards rear-end shunts, our programme presently excludes such a requirement on the basis of its relative insignificance when considered alongside the other accident modes. It is, however, clear that

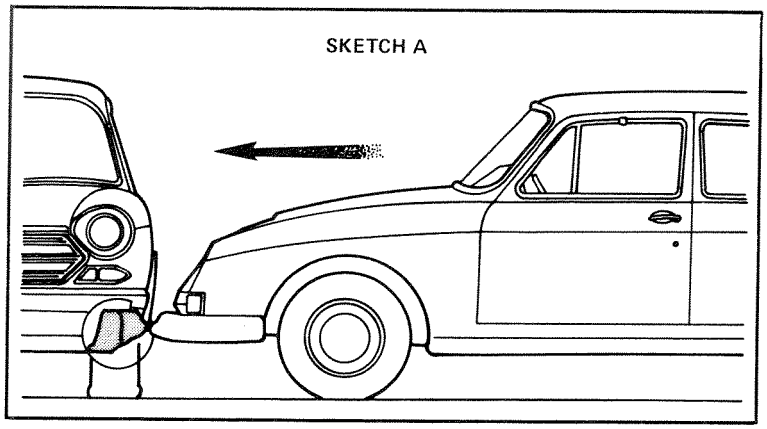
some of the fatalities occurring in rear-end impacts are associated with rupture of fuel tanks and the consequential fires. Some effort has been made to minimise this hazard by siting the fuel tank in the region most remote from the majority of accidents, i.e. under the rear seat. Our intention was, and remains, to produce a collapsible form of tank needing no direct vent to atmosphere; so that it can operate as a sealed system. To date the inherent problems have not been solved, and, in consequence, our demonstration vehicles have been fitted with rigid tanks for expediency.

The screen glass too is an area where an advance was expected by incorporation of the Triplex 10/20 laminate, but unfortunately the suppliers found themsleves unable to provide screens of requisite size and shape in time for this conference.

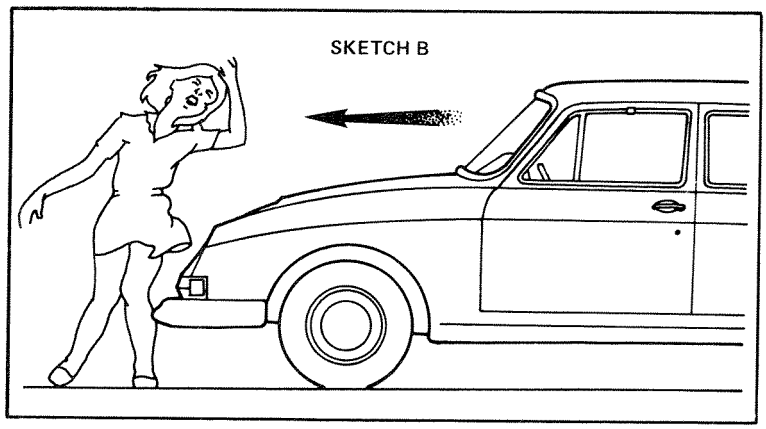
SUMMARISING

Our developments thus far are too embryonic to offer more than a few hints for consideration. Taking these in logical sequence:—

1. PRIMARY SAFETY — We all know what this means in general terms. How then can it be properly evaluated???
2. SECONDARY SAFETY — Both in respect of the vehicle itself and its restrained occupants, it seems likely that frontal impacts equating to a 60 kph barrier crash can be sustained with only marginal increase of vehicle weight and/or cost. For side impacts, a low mounted bumper system offers considerable advantage, together with a measure of compatibility. (Reference Sketch 'a'.)



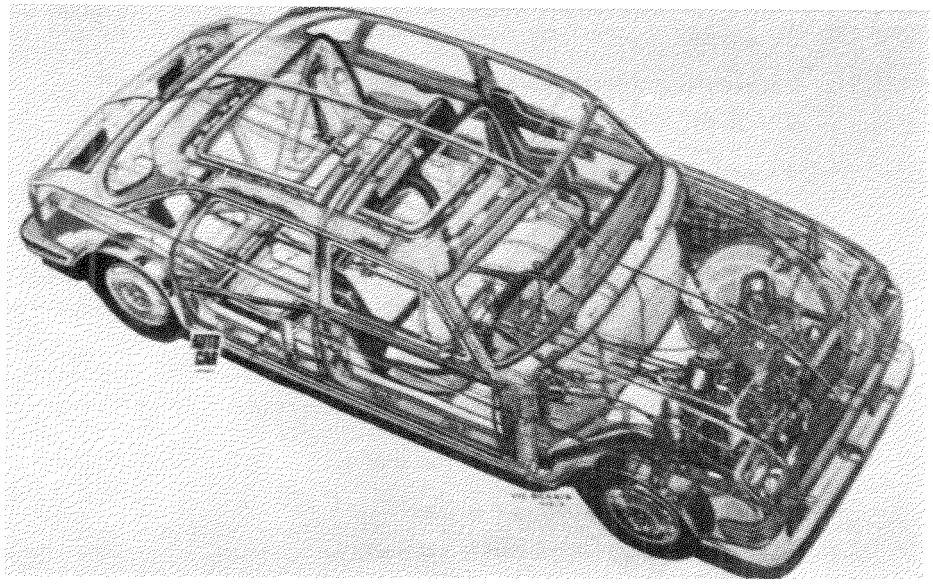
3. PEDESTRIANS — The same low bumper system is a key element in determining the trajectory of a pedestrian when struck, and associated with it a low profile bonnet top is another essential. (Reference Sketch 'b'.)



An associated catcher device confers additional benefit, particularly during brake activations, but the work to date is somewhat elementary, leaving many problems to be overcome, not the least of which will be owner acceptance.

The Mini and 1800 Stage II vehicles on display at Crowthorne have been updated toward the pedestrian considerations, but exclude the catcher device. In other respects they embrace most of the concepts featured in this paper. The total programme has at least another year to run, after which time more positive statements should generate from the extended experience.

BRITISH LEYLAND 1800, FRONT WHEEL DRIVE SAFETY CAR (STAGE II RECONSTRUCTION – SRV 3)



GENERAL SPECIFICATION

LENGTH:	Wheelbase	112.50 in (2860 mm)
	Front overhang	30.71 in (780 mm)
	Rear	31.84 in (810 mm)
	Overall	175.05 in (4450 mm)
WIDTH:	Track – front	55.68 in (1413 mm)
	Track – rear	54.50 in (1386 mm)
	Overall (maximum across front wing eyebrows)	66.75 in (1698 mm)
HEIGHT: (Normal laden – rolling)	Overall	56.30 in (1430 mm)
	Bumper C.L. to ground front	13 in (330 mm)
	Bumper C.L. contact face to ground rear	13.50 in (340 mm)
GROUND CLEARANCE (Normal laden – rolling)	Central between wheels	8.5 in (216 mm)
	Ramp approach angle	24.0 degrees
	Ramp break-over angle	9.0 degrees
	Ramp departure angle	20.0 degrees
TURNING CIRCLE (Estimated)		39 feet (11.89 m)
BODY CAPACITY:	Occupants–front	2
	Occupants–rear	2-3
	Luggage	18.5 cu ft (0.5235 m^3)
FUEL CAPACITY	Tank (under rear seat)	11.5 galls (52.28 litres)
WEIGHTS:	Total–kerb (ready for road, full tank)	2580 lb (1171.3 kg)
	Distribution: front	1600 lb (726.4 kg)
	Distribution: rear	980 lb (444.9 kg)
	Normal laden (3 up + 75 lb luggage)	3105 lb (1409.7 kg)
	Distribution: front	1805 lb (819.5 kg)
	Distribution: rear	1300 lb (590.2 kg)

G.V.W. (5 up + 125 lb luggage)	3455 lb (1568.6 kg)
Distribution: front	1878 lb (852.6 kg)
Distribution: rear	1577 lb (715.0 kg)

POWER UNIT: Transverse mounted 4 cylinder, 1750 cc, OHC 'E' series with single down-draught carburetter.

TRANSMISSION: 5-speed manual gearbox and final drive, integral with power unit.

C.V. joints to drive shafts.

COOLING: Frontal radiator with thermostatic controlled electric fan.

EXHAUST: Dual silencer system.

SUSPENSION: Front: Transverse double wishbone system, arranged for zero offset king pin to wheel C.L. at ground. Drive via wide angled C.V. couplings. Supporting subframe grouted into body sills.

Rear: Trailing arm, compliance mounted at sills.

Springing: Hydragas, interconnected front to rear.

WHEELS/TYRES: Dunlop Denovo total mobility (no spare).

STEERING: Rack and pinion: mounted on subframe.

Column: rigidly mounted in fascia member.

Connection: cable element, flexible in bend mode.

Wheel: energy absorbing with swash element.

BRAKES: A.P. 'Lockheed Anti-Lok' dual line hydraulic power system with 'L' split.

Disc front, drum rear.

Parking: cable to rears; central hand control.

LIGHTS:

Headlamps:	Rectangular, with wash and wipe and manual levelling.
Side Lamps:	Separate above bumper, adjacent front flashers.
Flashers:	Above bumpers at body extremities, for 100° side viewing, and coupled for hazard warning.
Tail Lamps:	Below flashers. 100° side viewing.
Stop Lamps:	Dual.
1	Inboard from tail lamp; light cancels only as vehicle moves off or ignition cut.
2	Inboard, adjacent Stop Lamp 1; registers only for braking above 0.4 g.
Reverse Lights:	Dual, inboard of stop lamps.

INSTRUMENTS/WARNINGS/CONTROLS

Speedometer:	Strip type, mounted for peripheral vision above steering wheel rim, including flasher warnings.
Warnings:	Light clusters mounted for peripheral vision outside steering wheel, at 2 and 10 o'clock.
Controls:	Dip, Dim, Horn, Flasher, Screen Wash-wipe, and Steering Lock, column mounted. Supplementary switches mounted for reach and peripheral vision outside steering wheel, at 3 and 9 o'clock.

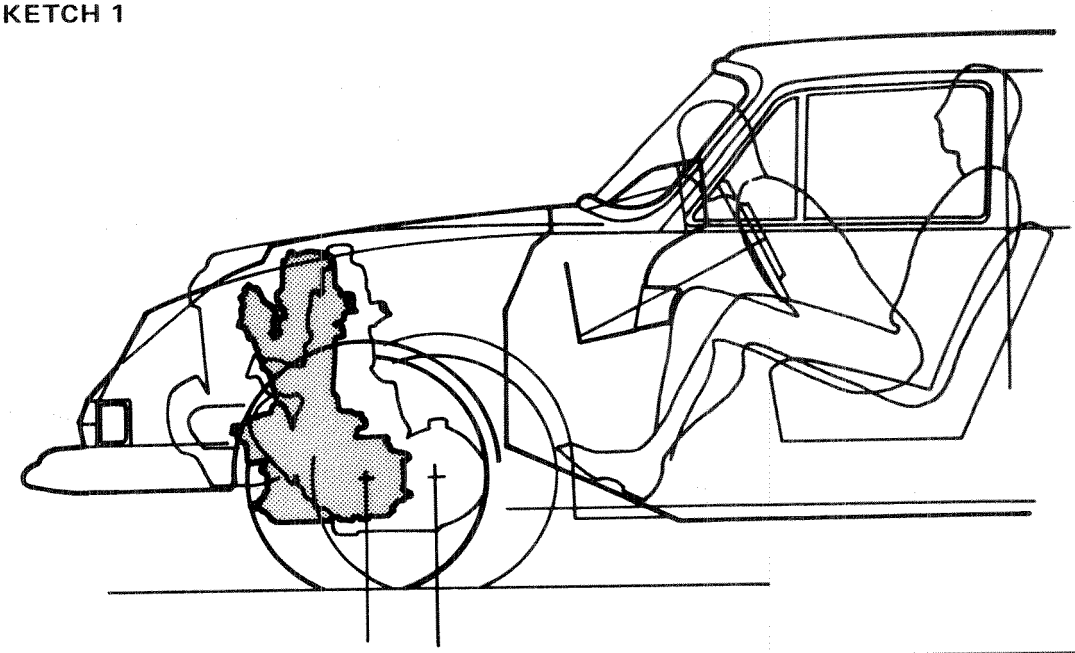
VENTILATION: Heater mounted on port valance outside body space, distributing via fascia member. Controls central. Twin fresh air vents adjacent 'A' posts. Air pick-up from front, via wing/valance box structure.

RESTRAINTS/PADDING:

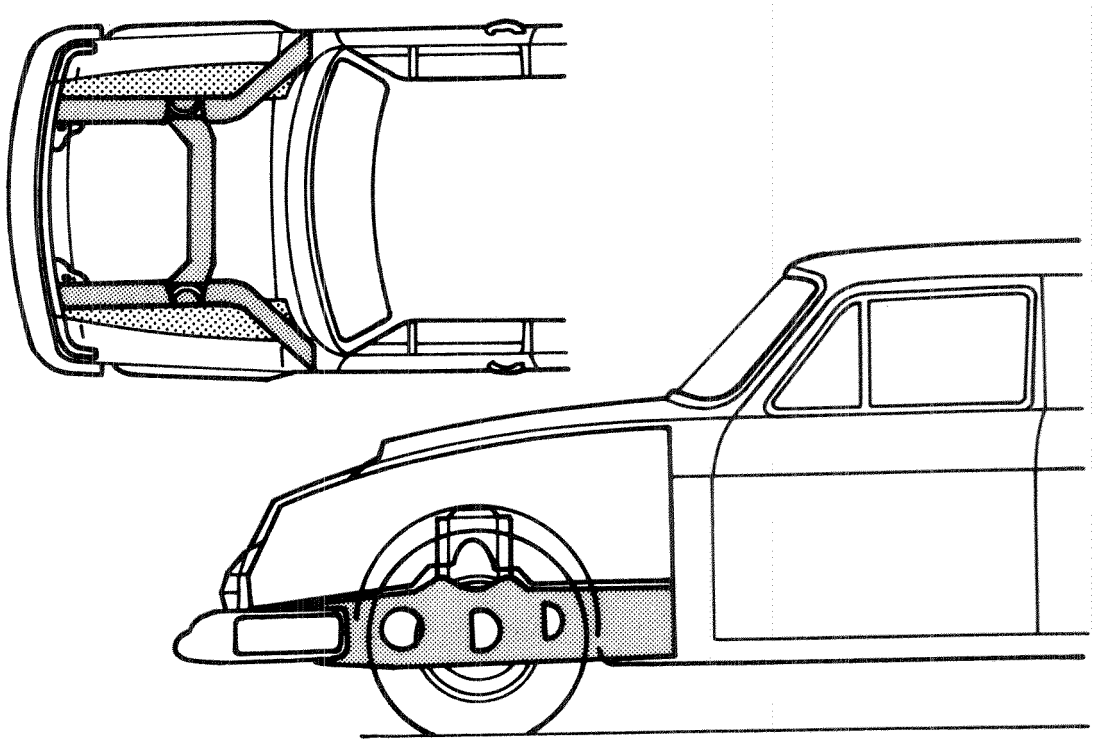
Front: Seats with head restraints. 3-point inertia reel seat belts, with tear-webbing for energy absorption.

Rear:	3-point static belt on outers. Lap belt in centre position.
Paddings:	Fascia rail, top and bottom. Front door, shoulder and hip. Screen top, cantrails and roof centre stiffener.
GLASSES:	Screen: Laminated (Triplex 10/20 intended but not available). Side: Toughened. Backlight: Toughened, heated.
DOORS:	Lock control mechanism, flush fitting. Anti-collapse, via struts front to rear, with proximity pads and sill dovetails.
WASH/WIPE:	Screen, headlamps and backlight.
BUMPERS:	Front: Low mounted, 'soft' facing on rigid armature, backed by E.A. device, aiding pedestrianisation. Rear: Mounted at height compatible with front at normal, laden condition. Section rigid, with abbreviated 'soft' facing, on compliance mounting.
BONNET FRONT:	Low profile, aiding pedestrianisation.

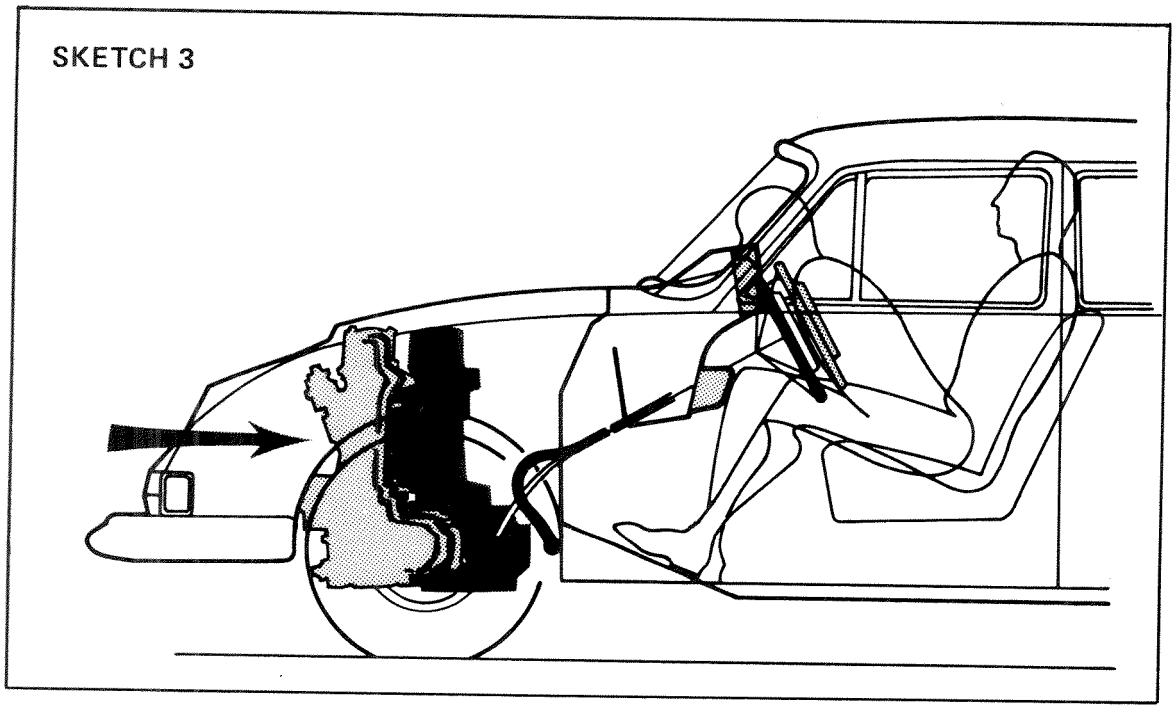
SKETCH 1



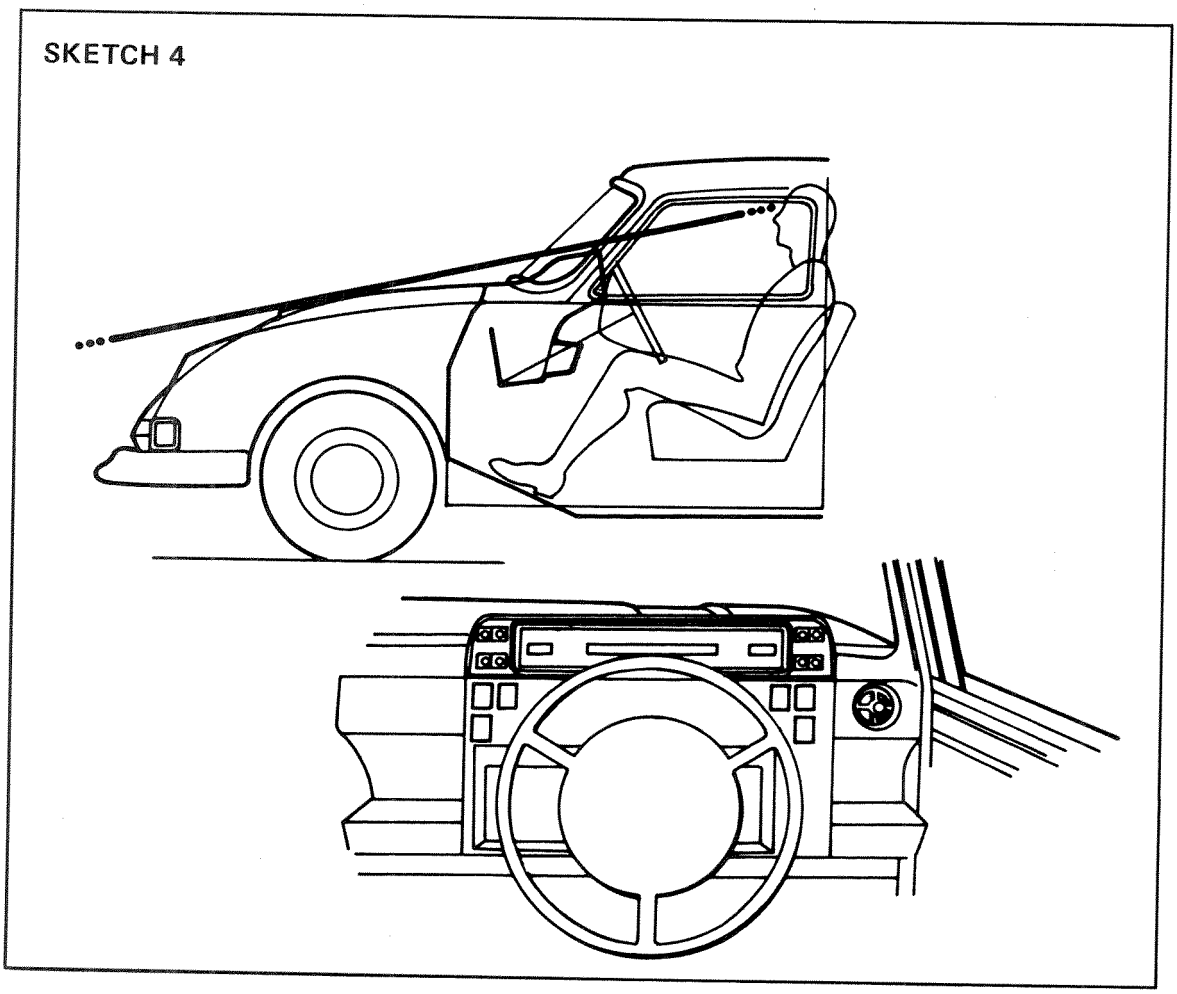
SKETCH 2



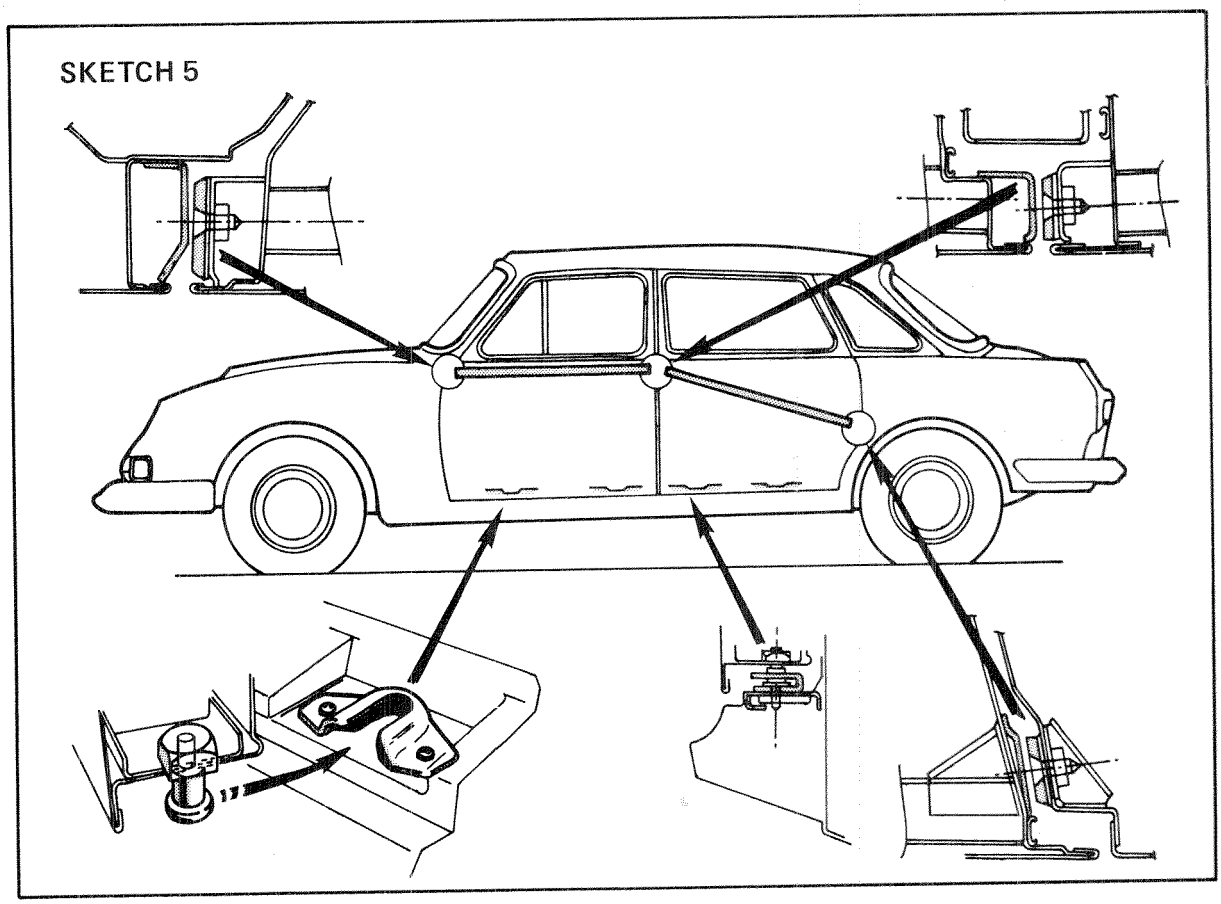
SKETCH 3



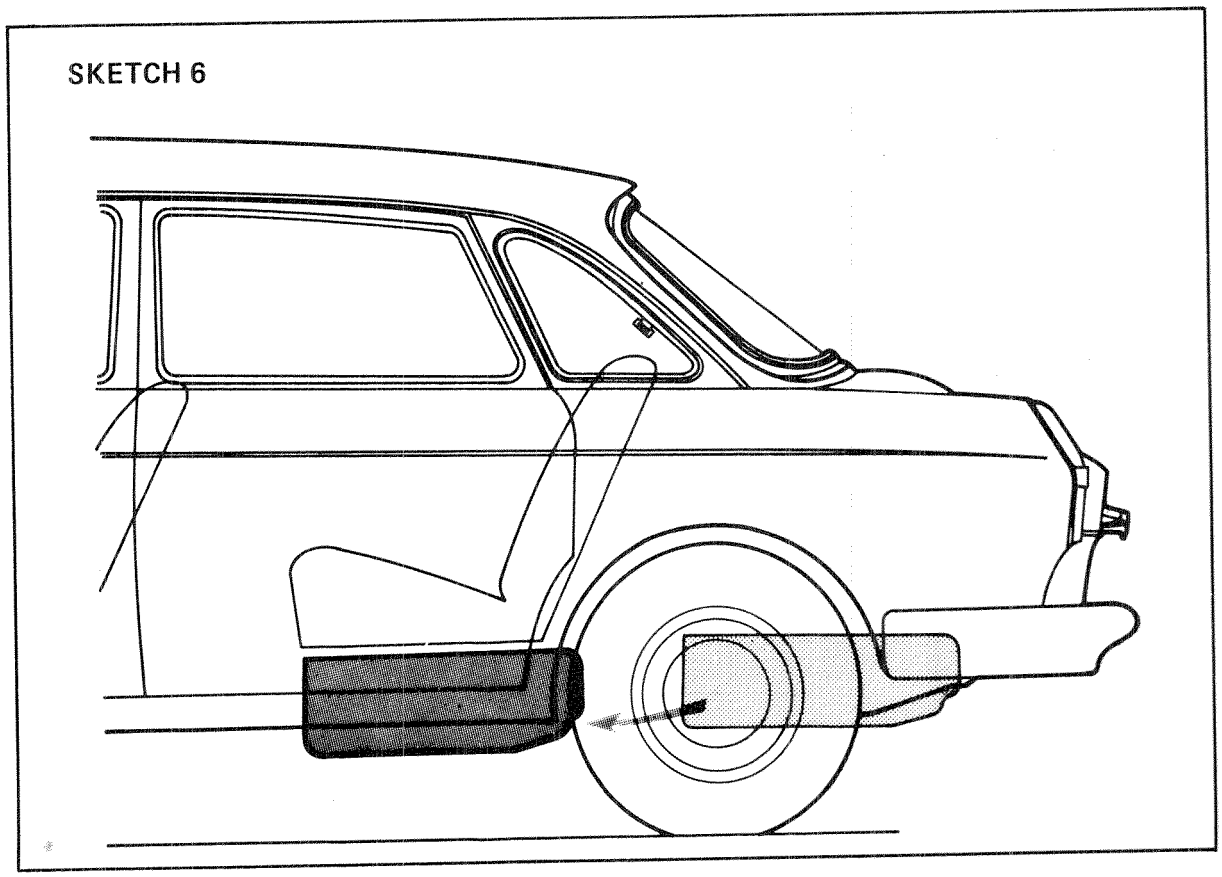
SKETCH 4



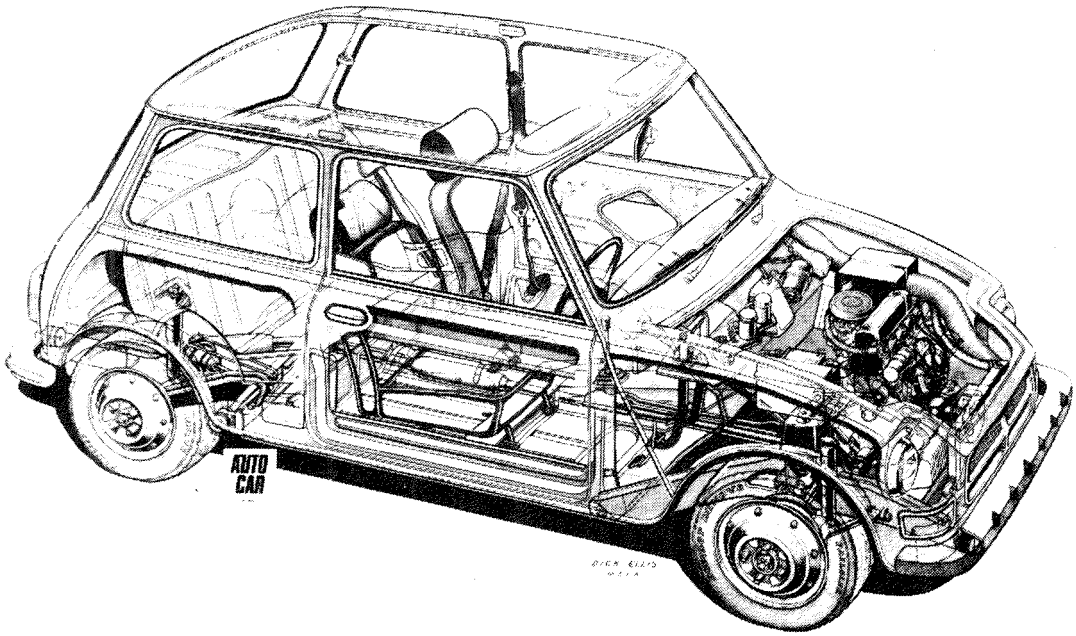
SKETCH 5



SKETCH 6



BRITISH LEYLAND MINI, FRONT WHEEL DRIVE SAFETY CAR (STAGE II RECONSTRUCTION – SRV 4)



GENERAL SPECIFICATION

LENGTH:	Wheelbase	85.50 in (2171.70 mm)
	Front overhang	24.50 in (622.30 mm)
	Rear overhang	21.75 in (552.45 mm)
	Overall	131.75 in (3346.45 mm)
WIDTH:	Track – front at kerb	48.68 in (1236.50 mm)
	Track – rear at kerb	46.00 in (1168.40 mm)
	Overall (maximum across doors)	55.50 in (1409.70 mm)
HEIGHT: (Normal laden – rolling)	Overall	51.75 in (1314.45 mm)
	Bumper C.L. to ground front	12.25 in (311.15 mm)
	Bumper C.L. to ground rear	12.25 in (311.15 mm)
GROUND CLEARANCE (Normal laden – rolling)	Central between wheels	6.40 in (162.56 mm)
	Ramp approach angle	23.30 degrees
	Ramp break-over angle	10 degrees
	Ramp departure angle	25.50 degrees
TURNING CIRCLE (Estimated)		29 feet (8.839 m)
BODY CAPACITY:	Occupants – front	2
	Occupants – rear	2
	Luggage	7 cu ft (0.1981 m^3)
FUEL CAPACITY	Tank (under rear seat)	6.5 galls (29.57 litres)
WEIGHTS: (Estimated)	Total – kerb (ready for road, full tank)	1412 lb (641.1 kg)
	Distribution: Front	968 lb (439.5 kg)
	Distribution: Rear	444 lb (201.6 kg)
	Normal laden (2 up + 50 lb luggage)	1762 lb (800.0 kg)
	Distribution: Front	1115 lb (506.2 kg)
	Distribution: Rear	647 lb (293.8 kg)
	G.V.W.	

(4 up + 100 lb luggage)	2112 lb (958.8 kg)
Distribution: Front	1155 lb (524.4 kg)
Distribution: Rear	957 lb (434.4 kg)

POWER UNIT: Transverse mounted 4 cylinder, 1000 cc, OHV 'A' series, with single down-draught carburetter.

TRANSMISSION: 4-speed manual gearbox and final drive, integral with power unit.
C.V. joints to drive shafts.

COOLING: Frontal radiator with thermostatic controlled electric fan.

EXHAUST: Dual silencer system.

SUSPENSION: Front: Transverse double wishbone system, arranged for zero offset king pin to wheel C.L. at ground. Drive via wide angled C.V. couplings. Supporting subframe grouted into body sills.

Rear: Semi-trailing arm, compliance mounted at sills.

Springing: Rubber cone springs, with tele-dampers.

WHEELS/TYRES: Dunlop Denovo total mobility (no spare).

STEERING: Rack and pinion: mounted on subframe.

Column: rigidly mounted in fascia member.

Connection: cable element, flexible in bend mode.

Wheel: orthodox 3-spoke (E.A. means to be introduced later).

BRAKES: Disc front/drum rear.

Hydraulics coupled for diagonal, split half system.

Parking: mechanical via cable to rears, central hand-lever.

(Note: Developments contemplate optimisation of the simple system by newly patented means, but progress to date is not sufficiently advanced to merit inclusion in this vehicle specification.)

LIGHTS

Headlamps:	Circular, set back within front profile as pedestrian safeguard.
Front Flasher:	Above bumper, at body extremity for 100° side viewing. Coupled for hazard warning.
Side Lamp:	Inboard/adjacent flasher.
Rear Cluster:	At body extremity for 100° side viewing.
Upper:	Flasher.
Lower:	Tail/Stop 1 — Stop cancels only as vehicle moves off or ignition cut.
Middle:	Stop 2 — registers only with braking 0.4 g plus.
Reflector:	Separate, inboard from tail cluster.

INSTRUMENTS/WARNINGS/CONTROLS

Speedometer:	Circular, 3.5 inches, calibrated mph/kph.
Combination Gauge:	Circular, 3.5 inches, including Fuel and Temperature Gauges, Ignition, Oil and High-beam warnings.
	Dual flasher warnings.
Controls:	Dip, Dim, Horn, Flasher, Screen Wash-wipe, Choke, and Steering Lock, column mounted.
	Supplementary switches: Lights, Hazard, Brake Fail, Backlight Heater, and Fog, mounted on centre panel.

VENTILATION: Heater mounted on port valance, outside body space, distributing via fascia member. Controls central. Twin fresh air vents adjacent 'A' posts, air pick-up from front via wing/valance box structure.

RESTRAINTS/PADDING

Front:	Tip-up seats with safety latch. Head restraints fitted. 3-point inertia reel seat belts (E.A. means to be introduced later).
Rear:	3-point static seat belts.
Paddings:	Fascia rail top and bottom. Front door, shoulder and hip.
GLASSES:	Screen: Laminated (Triplex 10/20 intended but not available). Side: Toughened. Rear: Toughened, Heated.
DOORS:	Lock control mechanism, flush fitting. Anti-collapse, via struts in doors, and impact keying overlap at sills. Strutting carried through to rear wheel arches adjacent outer skin.
WASH/WIPE:	Screen. (Headlamp may be introduced later.)
BUMPERS:	
Front:	Low mounted, simulated 'soft' facing on rigid armature, backed by compliance mounting, aimed at pedestrianisation.
Rear:	Mounted at height compatible with front at normal, laden condition. Section rigid, on compliance mounting.
BONNET FRONT:	Low profile, aimed at pedestrianisation.

HUMAN TOLERANCE TO IMPACT

Project guide-lines as set by TRRL; Ref. LF 311, with additions.

FRONTAL IMPACT TYPE INJURY

Body Area	Injury Type	Tolerance Level
Head (brain)	Frontal bone impacting a plane surface	Wayne State University tolerance curve, supplemented by Head Injury Criteria = 1000
Face	Zygomatic area (cheek bone) — one square inch contact area	1 kN
Neck	Larynx — one square inch contact area	400 N
	Hyper extension	80°
	Lateral flexion	60°
Chest	Impact on steering wheel and column assembly mounted at approximately 45° to the horizontal	6.5 kN
	With adequate distribution of loading between sternum, ribs and clavicles	8 kN
	Impact by a 150 mm diameter flat target centred at the level of the 4th interspace	65 mm chest deflection
	Chest deceleration at the vertebral column at the level of the 4th interspace	60 g for 3 ms
Knee-thigh-hip complex	Knee impacts, without localised loading causing skeletal injury to hip joint	4.44 kN
Tibia and fibula	Plateau fracture	5 kN

SIDE IMPACT TYPE INJURY (related to TRRL Side Impact Dummy)

Body Area	Injury Type	Tolerance Level
Head	—	Wayne State University tolerance curve
Shoulder	Transverse	6 kN
Chest	Transverse on each of the four ribs of the dummy	1 kN
Iliac Crest	Transverse	4.5 kN
Pelvis at hip joint	Transverse	4.5 kN

PEDESTRIAN TYPE INJURY

Body Area	Injury Type	Tolerance Level
Shin	10 kg shin form, striking bumper at 25 km/h	2 kN

OCCUPANT PROTECTION IN FRONTAL IMPACT (PASSIVE RESTRAINT), REAR IMPACT AND ROLLOVER

M. RODGER
Research and Engineering Center
Ford Motor Company Limited

INTRODUCTION

As part of Ford's overall safety program we have taken up a contract with the United Kingdom Government to develop certain safety features. This contract covers three quite separate projects, all aimed at improved occupant protection. The projects are on:

- Rollover
- Passive Restraint
- Rear Impact.

The selected projects were to some extent influenced by our own research requirements and it can be seen that these three cover a very wide range of vehicle safety problems in the crashworthiness side. For example, rear impact protection is largely a problem of vehicle structure design and the control of vehicle deformation during impact. By contrast, the passive restraint project is an exercise in occupant protection to which the vehicle interior and its restraint systems are the key. This project does not include any attempt to influence the vehicle structrual design. Finally, the rollover project has elements of both structural design and interior protection, coupled with the difficulty of correlating much more complicated vehicle dynamics with a standardized test format.

The formal starting date for all three projects was 1 February 1973 and all the projects are scheduled to finish before the end of 1974.

ROLLOVER

Rollover is only a small contributor to the statistics of serious injury in actual road accidents (Figure 1), but it is neither humane nor necessarily cost-effective to ignore it for that reason. Overall statistics are not everything and in depth accident studies show no clear cause and effect relationship between vehicle damage and occupant injury in rollover. Our study is aimed at finding out how rollover occupants get hurt, how important is roof strength and how important is interior padding. The limitation of accident studies is of course that investigation commences long after the rollover has finished, and so a laboratory test is needed. Our earlier work on rollovers suggested the lateral trolley

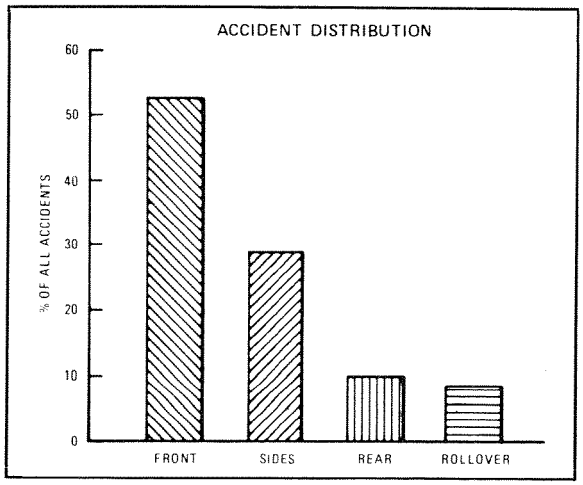


Figure 1

method for the dynamic roll (Figure 2), supplemented by a roof crush test to measure roof strength (Figure 3).

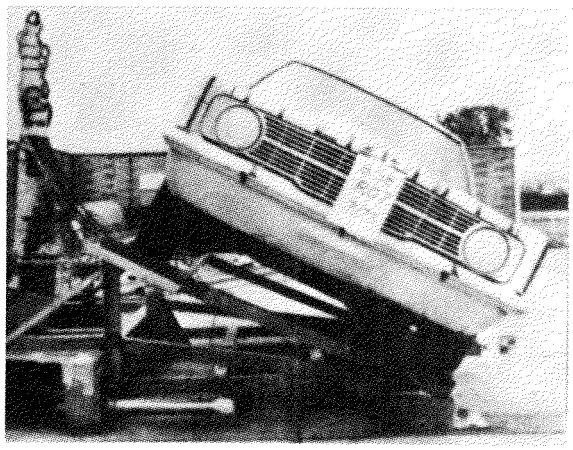


Figure 2

Dynamic rollover tests are notoriously unrepeatable but can, nevertheless, be used to show trends of behavior. For example, the highest occupant accelerations we have yet recorded occurred when a metal and plastic dummy head hit a metal and plastic dummy elbow – that is not repeatable. On the other hand, the deformation patterns shown in these three rollovers on the same model vehicle are recognizably the same. (Figure 4, 5, and 6)

The project divides into 3 main phases (Figure 7):

Phase 1 – Baseline rollovers on 4 different model vehicles each with unrestrained front seat dummies.

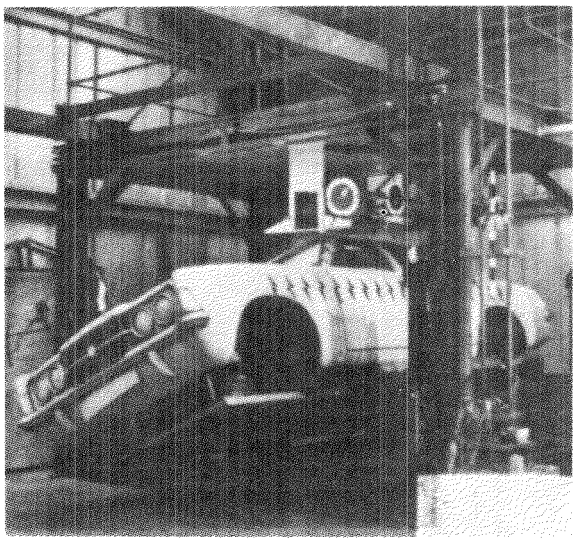


Figure 3

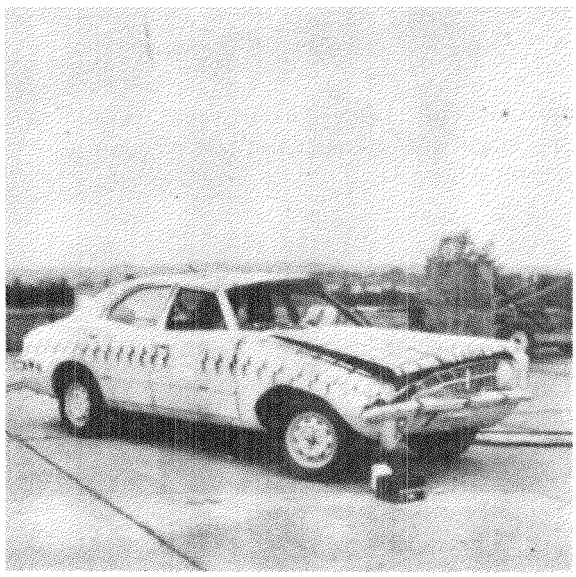


Figure 4

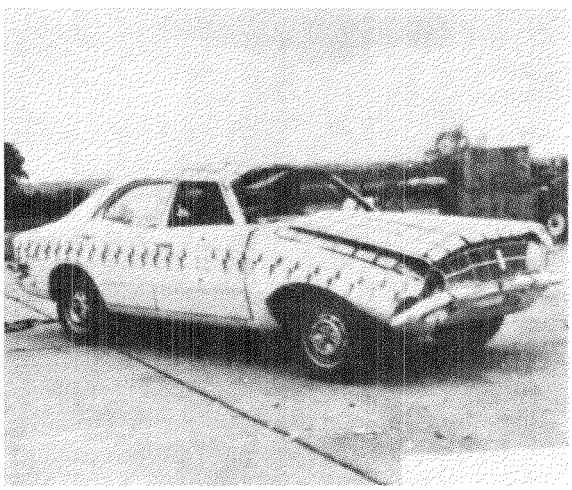


Figure 5

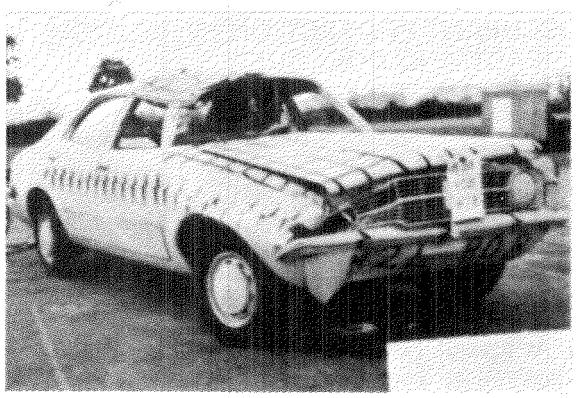


Figure 6

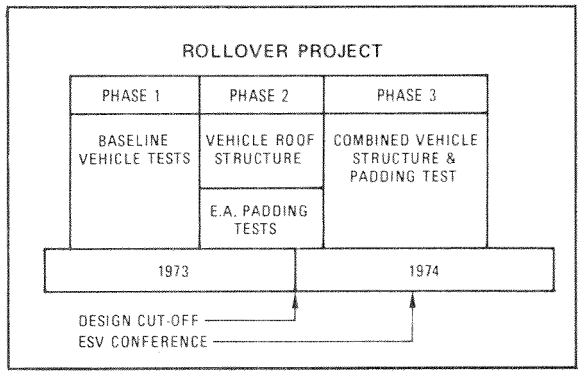


Figure 7

Phase 2 — Concentrated on one model, the Mk III Cortina. Rollovers were made with both unrestrained and fully belted occupants and with strengthened and weakened roof structures. Phase 2 also included investigations of padding materials for interior padding.

Phase 3 — Is still under way and consists of a number of rollovers to determine the most effective combination of structural treatment and interior padding. Results to date suggest that the 30 mph lateral roll test is unrealistically severe since it typically produces 1-1/2 to 2-1/2 rolls (Figure 8), whereas in practice, about 90% of rollovers are for one complete roll or less.

PASSIVE RESTRAINT

There is no need to repeat the requirements for occupant restraint systems in frontal impact. The problem really is the choice and development of alternative methods. This passive restraint system arose as an alternative to the air bag. It consists of a

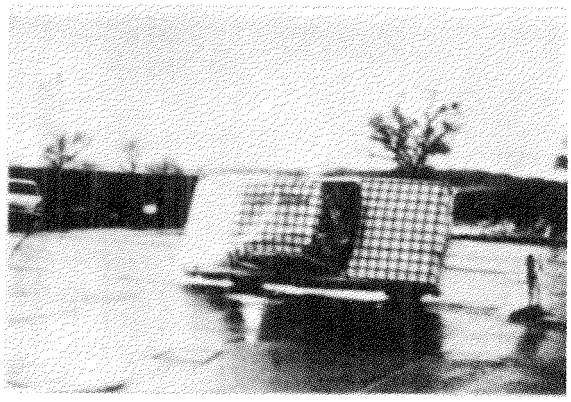


Figure 8

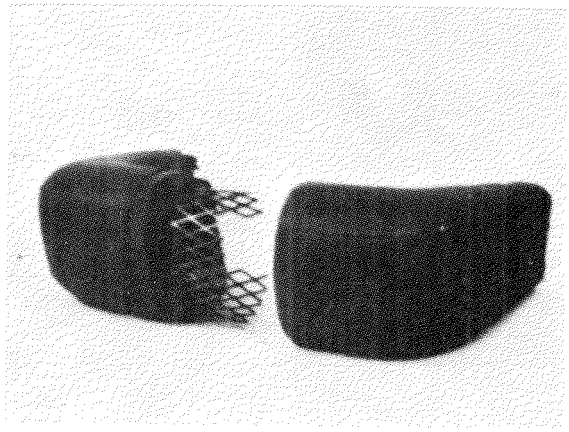


Figure 9

large crash pad made of foam plastic and expanded metal (Figure 9).

It makes no noise, needs no electronic devices to make it work and cannot operate inadvertently. As installed in a vehicle (Figure 10) it simply provides crash pad surfaces for chest and knees capable of bringing an occupant to rest in a 30 mph barrier impact without exceeding widely used injury criteria (Figure 11). It is comparable to a properly worn lap and diagonal seat belt and is infinitely better than a belt which is not being worn at all! Protection at higher speeds would be possible but would require a larger restraint, closer to the occupant. Alternatively, the system has potential for use with belts or in a mixed arrangement using seat belts for front seats and passive restraints for rear seats.

As you will see from the film shots later, this passive restraint system allows the head of the front seat occupants to hit the screen (Figure 12). However, correct control of head rotation can eliminate excessive decelerations and reduce facial lacerations. Further reduction of laceration is expected from the improved screens now under development.

This project was originally set out in 3 parts (Figure 13).

Part 1 – Development of passive restraint system for the front passenger.

Part 2 – Carry over and adaptation of front seat systems to rear seats/positions.

Part 3 – Extension of testing and development away from the 30 mph frontal to include up to 30° angled impacts and rollover.

In practice it proved easier to put in a rear seat passive restraint system than the front seat one and Part 2 "overtook" Part 1.

The project is fairly test oriented, making use of static tests to compare and develop the basic restraint element structures; sled tests to investigate their basic dynamic performance with anthropometric dummies

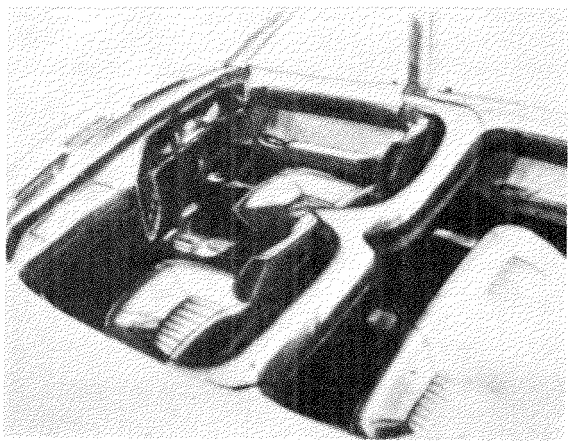


Figure 10

DUMMY ACCELERATION AND LOAD DATA
48km/h FRONTAL IMPACT

		HEAD GADD	CHEST PEAK G (ms over 60_g)	FEMUR LOAD kN RH	LH
SLED	FRONT	780	74 (3.5ms)	3.3	9.7
	REAR	850	54	3.0	3.0
BARRIER	FRONT	1620	41	6.5	6.5
	REAR	983 778	48 32	1.8 2.7	1.8 2.7
INJURY	LIMITS	1000	60 (3cms)	7.6	7.6

Figure 11

(Figure 14); and vehicle barrier tests for final prove-out of each stage (Figure 15).

REAR IMPACT

Rear impacts, like rollover, make a relatively small contribution to total injuries and fatalities. This is

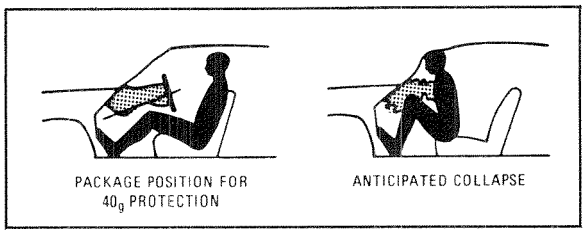


Figure 12

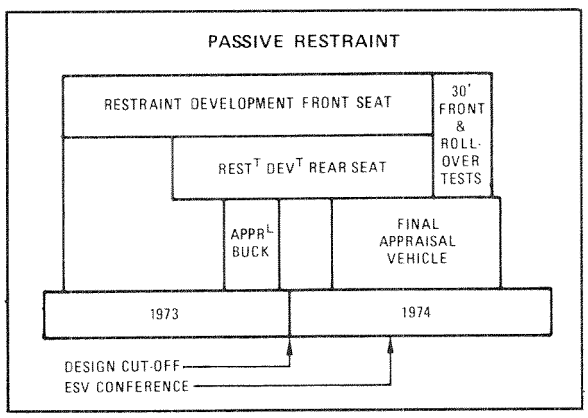


Figure 13

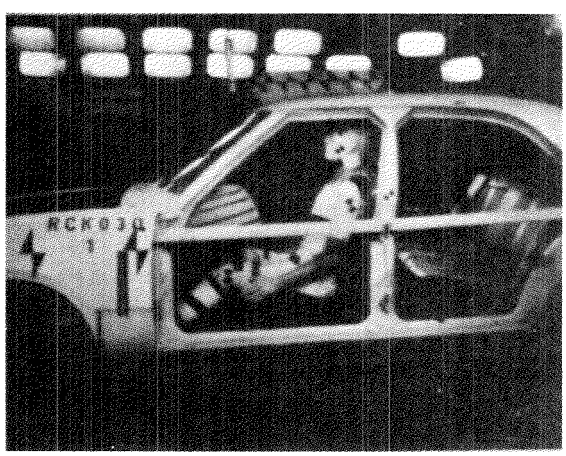


Figure 14

partly because rear impacts are, in general, less severe than frontals and the occupants are better protected in their forward facing seats. Nevertheless, there is no cause for complacency, the "multiple crash" or "motorway pile-up" seems likely to become more common and in this type of crash, rear impact protection becomes as important as front. We set a target equivalent to a vehicle to vehicle impact of 50 mph (80 km/h). Figure 16 shows the first vehicle modified, prior to test.

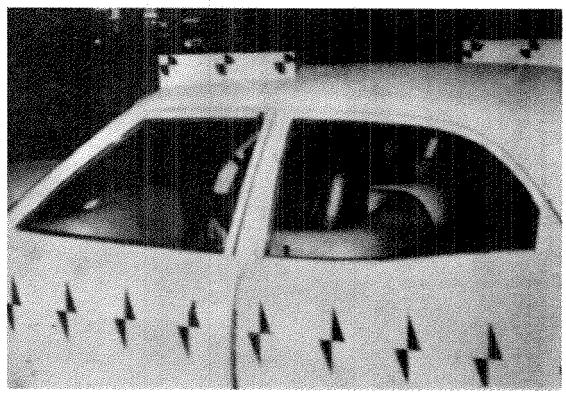


Figure 15

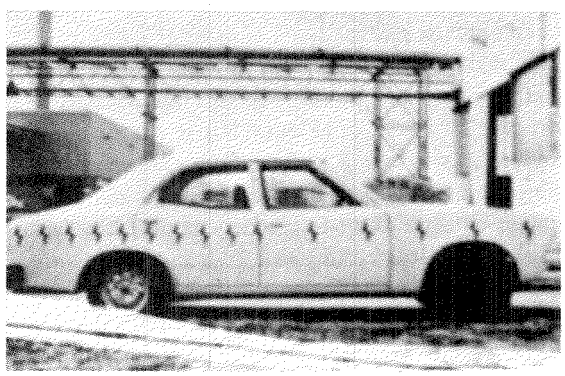


Figure 16

For the type of vehicle being considered, Rear Impact protection falls into two parts –

- Protection of the occupant.
- Protection of fuel system.

Occupant protection depends largely on a seat and head restraint design with adequate stiffness. Vehicle accelerations are relatively low and all that is required is to properly support the occupants.

Fuel system protection depends upon tank location and in recognition of current trends, towards Estate Cars, 3 door coupes and hatchbacks, etc., we chose to consider a tank placed under the rear luggage compartment floor. The design problem is then to locate the tank in a nondeforming area and to control the vehicle collapse to leave sufficient space for it. Figures 17 and 18 show the first vehicle modified, after test.

The project also included a 5 mph (8 km/h) "no damage" bumper although the precise advantages of these devices with regard to either safety or cost seem doubtful.

The first stage of the project aimed at a complete vehicle rear end, fuel tank, bumper design for barrier test at the target levels, (Figure 19). The objectives

Figure 17

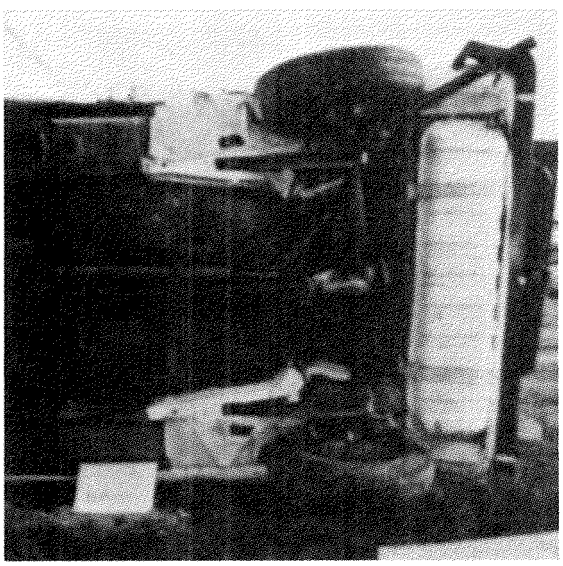


Figure 18

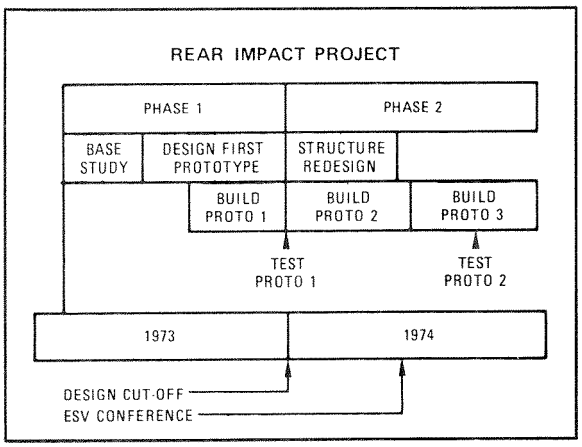


Figure 19

for fuel leakage and occupant injuries were in fact met and the second stage now in progress is simply the refinement of design from the lessons learned in the first stage.

GENERAL

It will be apparent from the way these projects are being tackled that we have adopted the subsystem approach. This approach enables much valuable work to be done without the expense and complication of building complete vehicles. It avoids the waste that can otherwise occur when a single vehicle cannot be properly tested in more than one or two of the many impact modes for which it may have been designed.

However, having briefly reviewed each of our three projects I have to say that each of them has now reached a stage where although the potential of the work can be clearly recognized, it can only be fully demonstrated by a wider redesign of the vehicle than can be contained in the subsystem approach.

DR. AUGUSTO SIRIGNANO
Director-General
Ministry of Transport and Civil Aviation

In my opinion, it must be recognized that the conference held at Kyoto last year represented, as compared to the original ESV conference, an important turning point, in the consideration of certain new trends which did not intend to signify the final freezing of the program, but its reasonable orientation towards more concrete objectives and more real interests.

On that occasion, we did not fail to underline that the search for better safety features should proceed at the same rate as a proper evaluation of cost/benefit factors, the respect of which was, among other things, imposed by the necessity of encouraging, as far as possible, the social diffusion of the vehicle.

In this field, it would be easy to forecast an inversion of trend be designed with a weight and size incompatible with the financial possibilities of the average user. We also pointed out at Kyoto that whatever its technical and constructive features might become one day, the so-called safety vehicle should be compatible in road traffic with vehicles of different weights and sizes.

We also remarked that there could be no guarantee that vehicles particularly equipped as far as passive safety was concerned, would not present some disadvantages which are not typical features of excessively heavy vehicles.

Some events having occurred after the Kyoto conference, in particular the trend toward the study of prototypes of 3,000 pounds, and the consequent launching of the wider RSV program, seem to have confirmed our doubts. It is therefore possible to conclude that the invitation has been conditionally accepted to take into consideration a research program based on a more realistic evaluation of the problem.

Of course, this does not prevent the first stage of the ESV project from having its own valid and profitable content, since it focused on the problem of safety and called it to the attention of governments and industries which until then had been unable to give it all the required interest.

It should therefore be recognized that considerable efforts have been made by all those working in this field at various levels in specific research to collect a precious mass of information, which not only can be profitably utilized but also can facilitate the course of further research studies in the RSV program.

We must recal' in this connection that our industry, though not sharing "in toto" the basic criteria of the initial specifications of the ESV program, has proved able to implement prototypes which, even though their weight is much below 4,000 pounds, can respond to almost all the requirements foreseen for heavier vehicles. This provides substantial proof that the weight element should not and can not be considered as a determinant, if not exclusive component – of course within certain limits – for reaching the desired goal.

In other words, our industry, instead of just expressing its reservations, however understandable, also preferred, in order to support its opinion, to complete the studies and researches commissioned to it, while giving at the same time a practical demonstration of the foundation and consistency of its standpoint. We have reasons to believe that this type of cooperation, which avoids adopting a position on an *a priori* basis, has been rightly appreciated.

We must also state that our industry is not available, for obvious and understandable reasons, without deadlines and the prior establishment of precise and reliable objectives, to undertake further construction of ESV prototypes. This does not mean, however, that in our country there has been no urge to deal with, or expand, research activities in view of improving the technical and safety features of the vehicles.

We consider it useful to give a short account of the activity which is being carried out in various sectors, ranging from the Testing Centre of the Ministry of Transport to the scientific laboratories of highly specialized institutes and organizations, and of course by those set up by industry.

Within the Ministry of Transport – Direction General for Motorization – the High Research and Testing Centre of Rome has designed a model of an energy-absorbing sliding seat, contracting its implementation and testing to private industry. In

this connection, we wish to state that a sample of this device has been mounted on one of the safety vehicle prototypes which Fiat has put at the disposal of the U.S. Department of Transportation for carrying out tests at the Operational Center of Phoenix. Precise information on this subject will be given during the 3rd Seminar on passive safety. The DOT representatives will report directly on the results of the tests carried out in the United States.

Researches are also being developed on nonlocking brake devices on the basis of new concepts, as well as on an aerodynamic-action brake.

Alfa Romeo has continued along the way already indicated during the previous conference, when it illustrated the results of studies and, following tests concerning a mathematical model simulating vehicle behavior, and took into account the intervention of the driver. The first encouraging results led to further improving the model, considering other parameters relating to the behavior both of the driver and of the vehicle, and thus permitting the simulation of more and more varied and complex maneuvers. The model also enables simulation of braking during curves.

We would like to announce in this connection that a paper on this particular subject will be submitted to the seminar segment of this conference. The contribution which Alfa Romeo believes it can give through these studies is all the more important, if we consider that the development of valid mathematical models enables enormous savings of money and, mainly, time, as compared to experiments carried out following traditional systems; in fact, the mathematical model serves as a guide for testing and for the interpretation and extension of its results.

Concerning passive safety, during the previous conferences, Alfa Romeo has reported on studies applied to restraint systems, attributing primary importance to them, and considering the problem as complex and difficult as that of structural behavior of the vehicle. The solutions indicated have formed the subject of further studies on the theoretical experimental level, mainly as regards body dynamics of the vehicle's occupant.

Alfa Romeo has also gone deep into the study of the structural behavior of the vehicle, and in the seminar on crashworthiness this company will present a paper illustrating the results of the investigation carried out in this connection.

Evidence of the serious commitment of Alfa Romeo in the field of safety, over and above the efforts – not only financial – supported in developing the above-mentioned studies and tests, is to be found in the setting up of a large new laboratory, where it is possible to carry out all the tests necessary to focus on the various problems of passive safety, as well as to thoroughly analyze them.

As regards interventions promoted by Fiat, we believe that it is useful to recall that this company has pursued its studies and research work in the field of passive safety, with particular attention to the structural and restraint system behavior.

Concerning the first item, information will be given during this conference on the tests both on compatibility between vehicles of different models (front and side collisions) and between vehicle and pedestrian; this is the firm conviction that compatibility – as we had the occasion to emphasize at Kyoto – must be considered as one of the key points of future regulations on safety.

Regarding the second item, useful information will be given on tests pertaining to the restraint system consisting of belts and other mechanical devices. It will be pointed out, among other things, that the utilization of the present belt has to be considered unsatisfactory already in case of a crash at roughly 50 km/h, while it becomes more effective when adopting some corrective measures, among which is the utilization of energy dissipators mounted "in series" on the actual strap of the belts seems to have a positive influence.

To confirm the spirit of cooperation which has been until now largely and repeatedly witnessed, we wish to recall that Fiat has promptly responded to the invitation of the U.S. Department of Transportation by forwarding four ESV prototypes to be submitted to the tests which have been recently completed at the Phoenix Center.

Still in the framework of ESV studies, the Ministry of Transport has contracted to the Pininfarina Body-maker research of an aerodynamic nature concerning the safe driving of vehicles in transversal gusts of wind.

This research is carried out by utilizing the Pininfarina wind tunnel, on the basis of a plan of tests which will tell us with more precision the importance of the aerodynamic coefficients of a basic vehicle and of four variants of it, and in particular, for each model tested, how the position of the center of lateral thrust changes following the variations of the yaw angle.

We are confident that these tests will yield valid suggestions and orientation criteria to follow in the planning of vehicles having a safer behavior in the presence of transverse gusts of wind.

To complete this picture of the activities developed to obtain improved safety performances in vehicles, we consider it useful to report briefly on the studies which are being carried out by specialized laboratories. Among these, of undoubted importance, is the study developed by the National

Electrotechnical Institute Galileo Ferraris, concerning a device of stop lights, which should have the requirements of good perceptibility, no glare, and the possibility of indicating the deceleration intensity.

This research aims at the construction of a prototype stop light designed for the glare limits which are tolerable under the characteristic conditions of night hours, keeping into proper account the possible changes of the vehicle's attitude. The goal is thus to obtain the best effectiveness both in the night and in the day, with results even perhaps superior to those obtainable through the "double level" system; a limitation of the height of assembly of the device on the vehicles is also under consideration. The indication of the deceleration intensity could be entrusted to a variation of the area illuminated by the device, related to the pressure exercised on the brake pedal, or directly to the vehicle's deceleration. This study has reached the final design stage, and within a few months the first tests on prototypes should start.

On its own account ISAM – Experimental Automobile and Motor Institute – has pursued its study, already partly presented and described in the previous ESV conferences of Washington and Kyoto, namely the experimental research of vibrations present in the passenger compartment of a vehicle, the bibliographic research on the characteristics of the vibrations prejudicial to the human organism, and then, in synthesis, a parallel analysis of the results of the two researches. A final report will then be developed indicating which frequencies and amplitudes of the vibrations limit active safety because they affect the psycho-physical conditions of the driver.

At present, experimental research, supported by a first bibliographic study, demonstrates that there are vibrations in the passenger compartment of a vehicle of a particular frequency which can adversely affect the human organism. The experimental research has been carried out on a vehicle which is very popular in Italy, under different driving conditions, while for the bibliography, studies already made in many European countries are being consulted. It is hoped that, after completion of this research, it will be possible to propose reasonable ways capable of eliminating the causes of such vibrations, or to limit their significance.

Having said all this, it seems possible to state that the initiative undertaken in a more general context by CCMS, and, during its initial stage, by the ESV project, has stimulated a climate of cooperation to studies on safety which extends beyond national boundaries.

This orientation was also mentioned in the final resolution of the 10th Plenary Session of CCMS which expressely recommended the utilization of the results so as to incorporate them, if necessary, into the technical regulations relative to the features and the requirements of vehicles.

In this field, it is worthwhile to recall that our country has always appropriately participated in the initiatives promoted at the international level, with its contribution of knowledge and experience. In fact, our present Road Traffic Code was considered initially and still remains one of the most complete and severe regulating systems as concerns technical and construction specifications for vehicles, with the consequent imposing on manufacturers of safety requirements which have involved non-negligible and burdensome commitments.

The Ministry of Transport has never failed to illustrate its position during discussions of the safety program promoted by CCMS, and its recommendations have been broadly accepted.

In view of these considerations, we wish to confirm our intention to pursue the activity within the European Experimental Vehicle Committee (EEVC). We are convinced that this body, which is free from any procedural pattern can contribute significantly to a more realistic formulation of the problems pertaining to the experimental safety vehicle while keeping into account the European reality.

Within that Committee, we intend to direct some of our research toward a vehicle designed to be more harmonious with the environment, making it not only safer, but also less noisy and less polluting.

Following such guiding principles, the Ministry of Transport, over and above the initiatives undertaken so far as the problem of the safety vehicle is concerned, has decided to carry out interesting and specific researches also in the field of noise and pollution.

For this purpose, through its own phonometric laboratory, the Ministry of Transport has developed:

- a study on low-frequency components present in the noise spectrum made at the vehicle exhaust;
- a study on the audibility of sound signalling inside the passenger compartment of speedy vehicles, and the formulation of regulations for sound signalling of high sound power;
- a study on sound directionality and the formulation of regulations for supplementary alarm sound signalling, to be mounted on vehicles on emergency service;
- a study on sound penetration inside the crash-helmets of motorcyclists (to avoid that the effect of passive safety of such devices could jeopardize the needs for active safety represented by sound signalling and other traffic noises).

As concerns pollution caused by exhaust emissions, having recognized that a fight against this problem has to be led mainly through valid controls, which in turn call for the solution of a whole series of hard technical problems, studies have been concentrated on the following subjects:

- choice of a testing procedure for the determination of the emissions of carbon monoxide in low gear;
- formulation of requirements for approval of analyzers, also portable ones, of carbon monoxide;
- formulation of requirements for approval of opacimeters of diesel motor exhausts;
- research and implementation of an experimental opacimeter of limited sizes, for measurement on the road of smokes emitted by diesel motor exhausts.

Finally, we consider it useful to express the advice of our Delegation concerning the problem which inevitably arises, of the relationship between the studies for the ESV and RSV programs, and the technical to be enforced. In our opinion these should remain the responsibility of the international bodies (ECE and EEC) which for some time have been dealing with the formulation of the requirements relating to the construction features of vehicles and their devices destined for mass production in order to avoid duplications.

Coming back to the project of constructing ESV and RSV prototypes in the 3,000 pound class, while confirming that on the Italian side there is no intention of undertaking further engagements in this connection, it should be considered that a safety vehicle of this class, bearing in mind the increased weight imposed by increased safety features, can only come from a medium-sized European vehicle of 1,000-1,100 kg. Should such a trend assert itself, it will not be possible to ignore the tradition and the tested experience of the European industries which – the Italian one in particular – will not fail to offer their convinced and tangible contribution, even in the present difficult economic cycle.

This outlook, from whatever standpoint the matter is considered, has strong arguments for asserting itself, and this is confirmed, quite aside from the otherwise inevitable break of the delicate cost/benefit balance, by the need – which unfortunately has not a contingent nature – for reducing fuel consumption to indispensable levels, which is obviously linked to the weight factor.

To conclude, our Delegation does not deviate, concerning the main topic of the conference, from the guiding principles indicated in the spring of last year at Kyoto.

We stated then, and now, that no practical result would be achieved by implementing a model with very high safety features, but of such a high cost as to be inaccessible to the average user.

We consider it correct, therefore, to encourage the apparent trend of bringing back ESV studies and researches from an abstract level – though of a very high scientific interest – to a reality within the dimensions and limits of which the more general question of motorization will necessarily have to fit in and adapt itself.

Even if there is widespread opposition to this viewpoint, we still feel that the vehicle remains an irreplaceable means of individual transport, without which the enormous progress of our civilization over the last 70 years would not have been possible; it remains an expression of the individual's ideal of freedom and autonomy.

VITTORIO MONTANARI
Director for Applied Research
Department of Motor Cars

INTRODUCTION

At the ESV Conferences in Washington and Kyoto, FIAT showed the collision tests – frontal against barrier, rear with moving barrier, side pole and rollover (drop) – run according to the procedures laid down by the US-DOT Specification.

The tests had been conducted on the three FIAT ESV models – 1,500, 2,000 and 2,500 pounds – during the finalization phase of the several prototypes.

In the months following the Kyoto conference and as soon as the "Impact Resistance against Rigid Obstacle" phase was over, the conclusive research and testing work was performed by emphasizing the passive safety areas. The FIAT research work paid specific attention to compatibility and restraint systems (Figure 1).

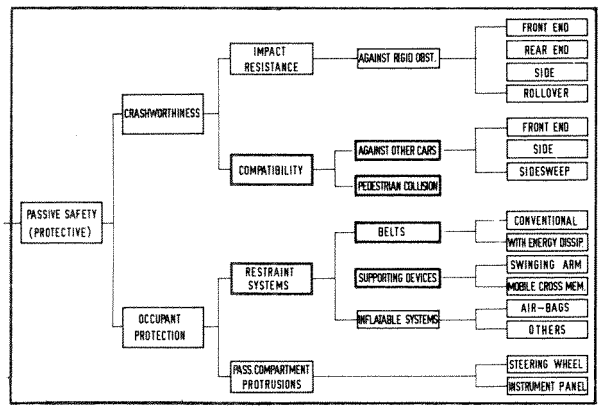


Figure 1

Compatibility Tests

- Frontal collision between the three types of FIAT ESVs with closure speed of 75 mph.
- Side collision between the three types of FIAT ESVs with impact speed of 22 mph.
- Sidesweep, at increasing angles from 5 to 10 degrees and speeds up to 18 mph between both FIAT ESV prototypes *and* FIAT standard cars.
- Pedestrian collision.

Restraint Systems

- Conventional seat belts.
- Seat belts with energy dissipators of the friction, deformation and preloading types.
- Mechanical supporting devices of the swinging arm and mobile crossmember types.
- Combination of seat belts with energy dissipators and retractile steering wheel.

This report illustrates the results obtained at the conclusion of the FIAT work in the ESV field.

COMPATIBILITY

The car must be compatible both with the other vehicles against which it collides and with the pedestrians struck by it. As regards car-to-car impacts, we continued the frontal collision tests (already shown in Kyoto) and ran side impact tests. The survey was completed with an investigation on ESV performance, and, as in the case of sidesweep, even at a low energy level.

Frontal Collision Between the Three FIAT ESV Models

Tests were run at a closure speed of 75 mph and demonstrated that the various combinations of different front end profiles (Figure 2) caused the interface to change intentionally, case by case, at the instant of impact initiation.

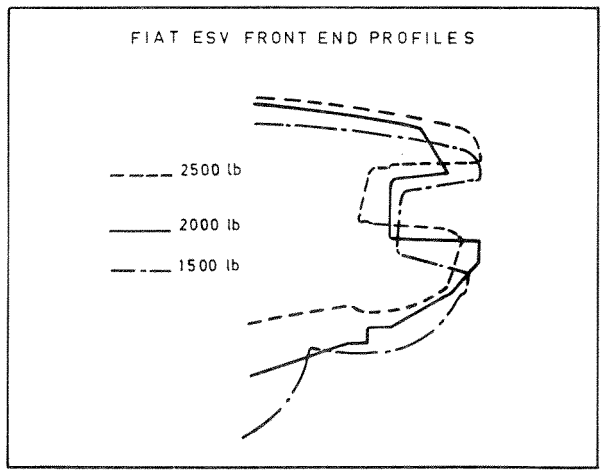


Figure 2

The cars travelled on guided paths at the same speed and, on impact, were free from directional restraints. The photographs of Figure 3 show the results of these tests; it can be seen that in the impact between the 1,500 and 2,000 pound ESVs, deformations to the front end are limited on both cars and no deformation to the interior is evident. In the impact between the 1,500 and 2,500 pound

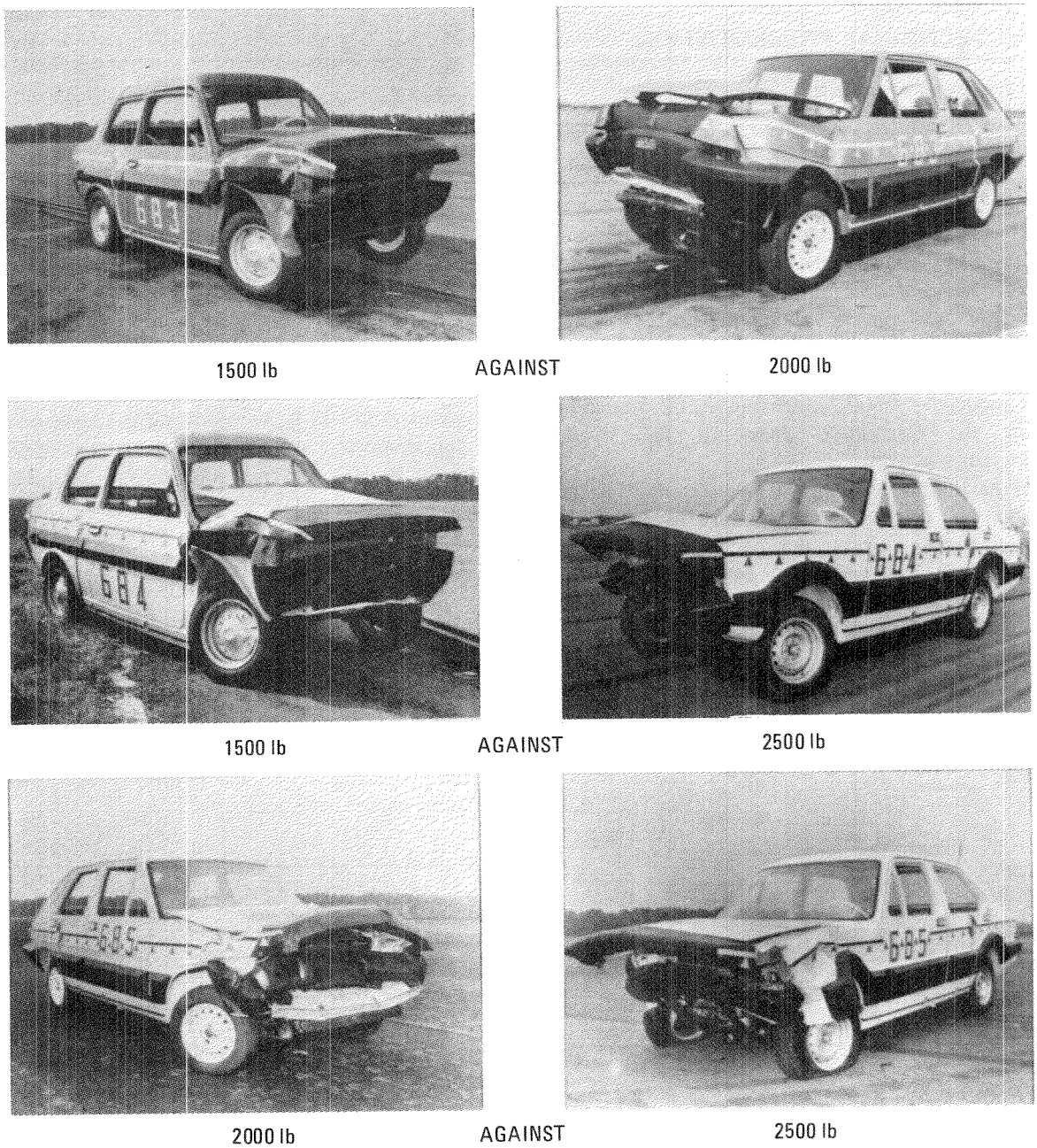


Figure 3

vehicles an unfavorable interpenetration of front ends occurred. Noteworthy, in particular, is the fact that the top part of the 2,500 pound model front end remained unharmed and deformation was confined to the frontal area; the passenger compartment was undamaged on both the 2,500 and 1,500 pound models.

In the impact between the 2,000 and 2,500 pound ESVs, the upper front end part of the latter escaped deformation – damage confined, in any event, to the front end on both cars – while the passenger compartment remained unaffected.

In conclusion, the outcome in terms of frontal collision was determined to be:

- On all models, deformation occurred in the front end only and the passenger compartment remained unaffected.
- Of the three profiles shown in Figure 2, the one inclined backwards and the one almost vertical gave better results.

Side Collision Between the Three FIAT ESV Models

The side collision tests were run in such a manner that

- The collision path angle was 90°.
- A car travelling at 22 mph struck a transversally placed standing car.
- The passenger compartment became the struck area.

The different side profiles of the three models (Figure 4) came into contact with the different front end profiles illustrated previously (Figure 2) so that the interface changed intentionally case by case. Tests have been run with seats removed so that intrusion

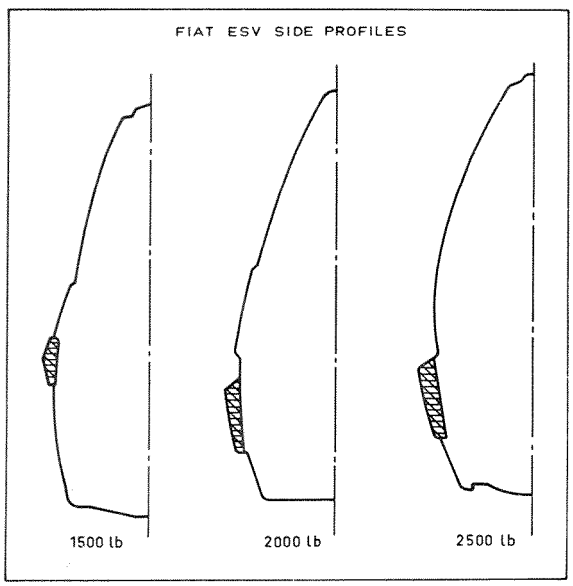


Figure 4

into the passenger compartment could be measured also during the impact. The photographs of Figure 6 show the results of these tests: the deformation of the side outer panelling is in all cases much greater than internal intrusion.

On the 1,500 pound unit struck by the 2,000 pound car (Figure 6a) a 5 cm internal intrusion (dynamic value) was measured. No damages were suffered by the striking car. In the impact of the 2,500 pound car against the 1,500 pound unit (Figure 6b) it was noted that the latter tended to overturn. Dynamic intrusion on the impacted car was 7.6 cm. No damage was suffered by the striking car.

This test cycle was concluded with the collision between two 2,500 pound ESVs (Figure 6c). In this case the impacted car showed a tendency to overturn while dynamic intrusion on the impacted car was 7.3 cm. The striking car suffered no damage. In no case (Figure 5) are intrusions conspicuous. The striking cars are absolutely unharmed, thanks to the particular configuration of their front ends. Decelerations measured in the passenger compartment are contained within 10-15 g on the striking car and 20-25 g on the struck car (Figure 5).

Tests paying specific attention to nose outline (Figure 2), have shown how the 2,500 pound car front

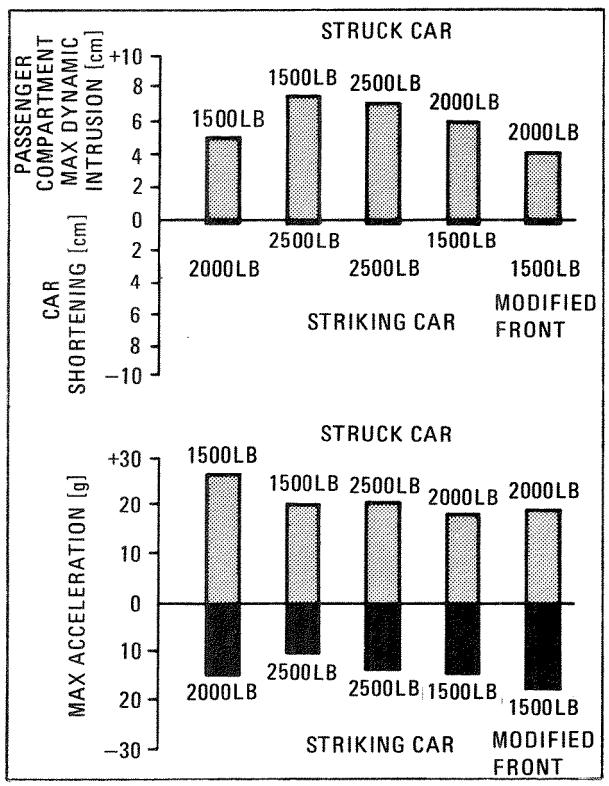


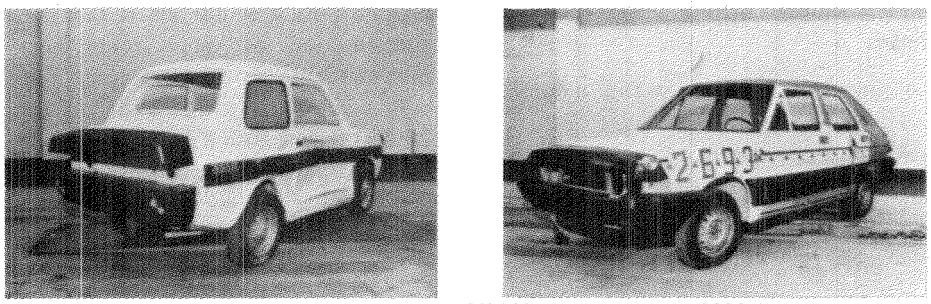
Figure 5

end – having a forward-inclined profile so as to protrude at top – not only suffers greater deformations at beltline, or at occupant thorax height, but also tends to push over the struck car. The 2,000 pound car front end – having a backward-inclined profile so as to protrude at bottom – suffers deformations greater in body side lower section but still quite appreciably at beltline.

A test was conducted to investigate the effect of the nearly vertical front end of the 1,500 pound ESV by having it strike the side of the 2,000 pound model (Figure 6d). A maximum dynamic intrusion of 6 cm results; it was nearly equal at top and bottom. This means that contact was more uniformly distributed, although intrusion at the top still called for some reduction.

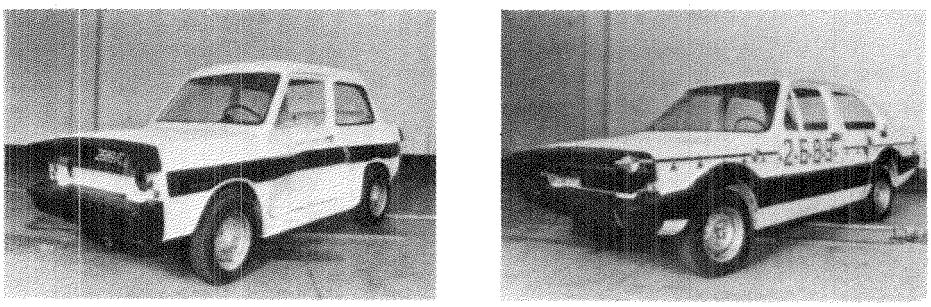
Towards achieving an improved design:

- The profile was changed by provisionally removing the bumper upper element.
- The bumper height was lowered through the artifice of varying car attitude.



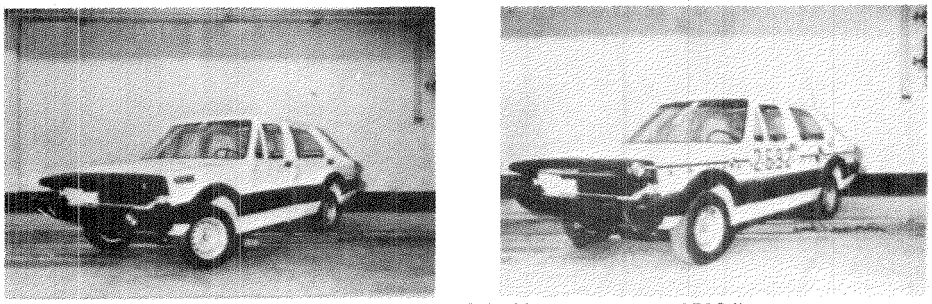
1500 lb STRUCK BY 2000 lb

6(a)



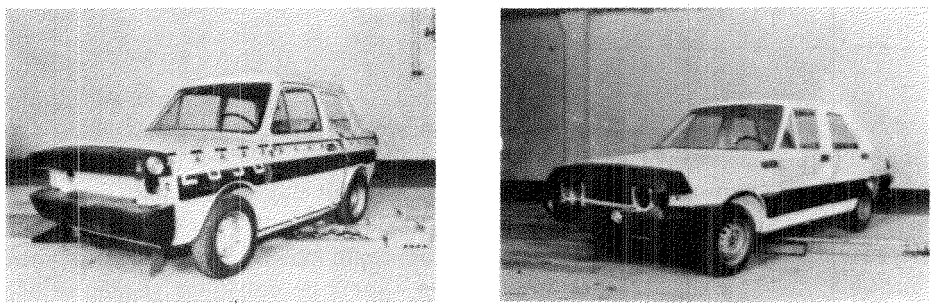
1500 lb STRUCK BY 2500 lb

6(b)



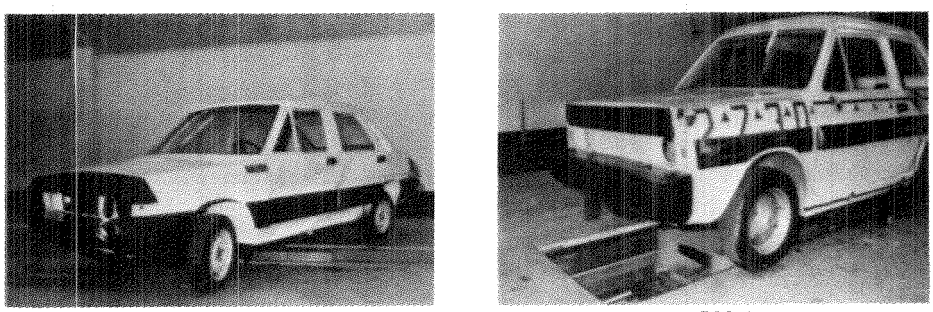
2500 lb STRUCK BY 2500 lb

6(c)



2000 lb STRUCK BY 1500 lb

6(d)



2000 lb STRUCK BY 1500 lb

6(e)

Figure 6

The side impact test was repeated. The 2,000 pound car was now struck by the 1,500 pound unit with modified front end (Figure 6e). Contact on the side improved. Intrusions no longer occurred in the interior at belt height and the maximum value was reduced to 4 cm (Figure 5). Again, the striking car was undamaged. It should be noted that the intrusion value decrease is accompanied by a negligible decrease in passenger compartment deceleration.

The following can be inferred from the above side collision tests:

- Front ends with synthetic bumper material are effective as they contain damages to both the impacted and impacting cars; in crashes at speeds up to at least 22 mph, the striking car is undamaged.
- The more viable front end configuration is the slightly backward-inclined profile which has less rigidity at top and more at bottom. Such a design concentrates the load prevailingly in the sturdier section of the body (side rails and seat base) and prevents the occurrence of dangerous intrusions in the upper area at a height in line with occupant thorax.

Sidesweep

This particular car-to-car impact condition was investigated, realizing that the magnitude of consequences of sidesweep impact depend substantially on two factors: impact conditions and characteristics of the vehicles involved.

Impact severity is a function of both collision course angle and speed. The significant vehicle characteristics are: side configuration and structural rigidity. On our ESVs the profile in elevation is characterized by a new approach to locating the deformable bands of synthetic material (Figure 4); side contactability of the three models is therefore different.

Tests were run at increasing course angles by launching a car against the side of a standing car at an increasing speed for each course angle. Dynamic and permanent deformations in all contact areas were measured (Figure 7).

In a first set of tests, a 1,500 pound ESV was launched against a car of the same model. Running at narrow impact angles and low speeds, the striking car sidesweeps, then glances off. No permanent deformations are evident; only at 18 mph and at an angle of 10 degrees did a permanent deformation of the struck car door beam begin to show up. By repeating the test at increasing angles and speeds it was possible to plot the curve of the maximum dynamic deformations (Figure 8). The chart shows the critical value – the angle representing, at the

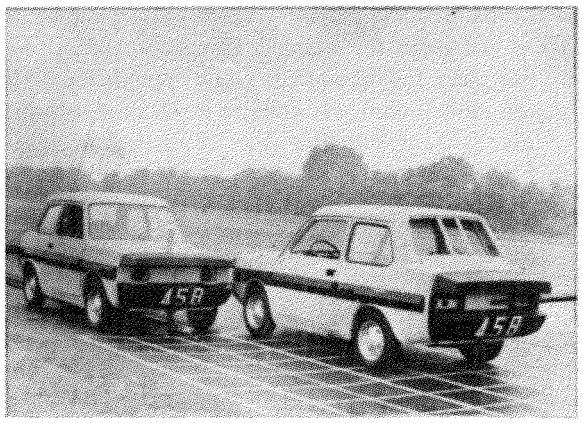


Figure 7

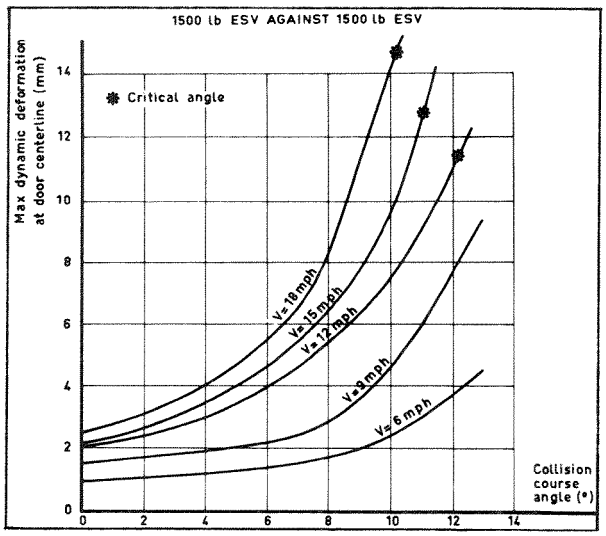


Figure 8

several speeds, a permanent deformation of at least 3-4 mm.

Another set of tests employed a 1,500 pound ESV to strike a 2,000 pound ESV. The dynamic

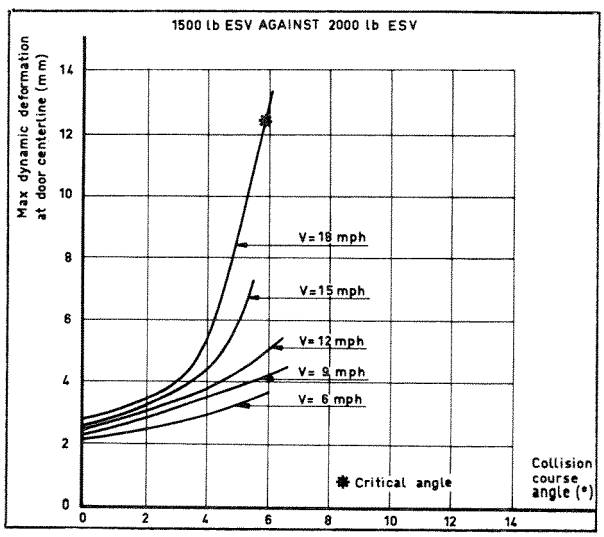


Figure 9

deformation curve pattern (Figure 9) is similar to the previous one but the critical angles are smaller.

For the 1,500 pound car impact against the 2,500 pound model, the curve (Figure 10) is similar to that of the 2,000 pound ESV, but the critical angles are slightly larger.

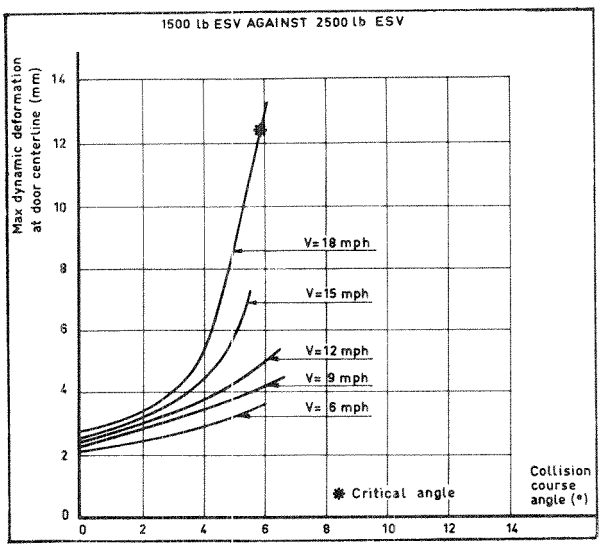


Figure 10

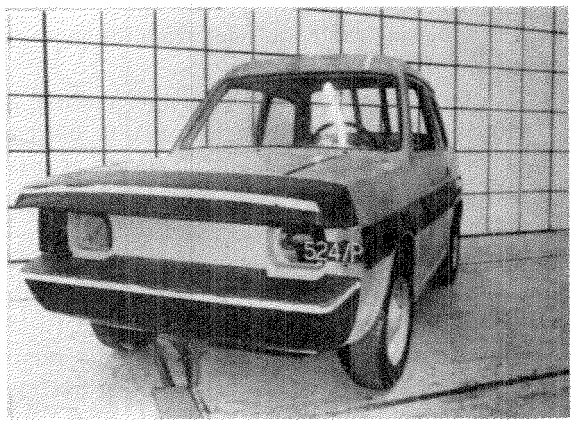
This first set of tests demonstrates that at collision angles from 5 to 10 degrees, depending on the model, sidesweep speeds of up to 18 mph can be reached before permanent deformations of a few millimeters occur in the impacted car. No damage is suffered by the impacting car.

To make comparison terms available, the same tests were run on similar, standard production cars (Figure 12). A 126 model is launched at 18 mph and at a 10-degree-angle impact against another 126 model. When collapse is imminent, conditions are the same as set earlier for the two 1,500 pound cars. Damage is limited to the left front end corner of the striking car but involves, quite notably, the door and rear side panelling of the struck car (Figure 12a).

Another 126 model was speeded at 18 mph and at a 5-degree impact against the side of a 128 model; conditions are the same as in the previous collision between a 1,500 pound and 2,000 pound car, at collapse initiation. The sides of both vehicles are damaged: in particular, the concentration of damages is evident on projections such as, for instance, the wheelbox edge contour (Figure 12b).

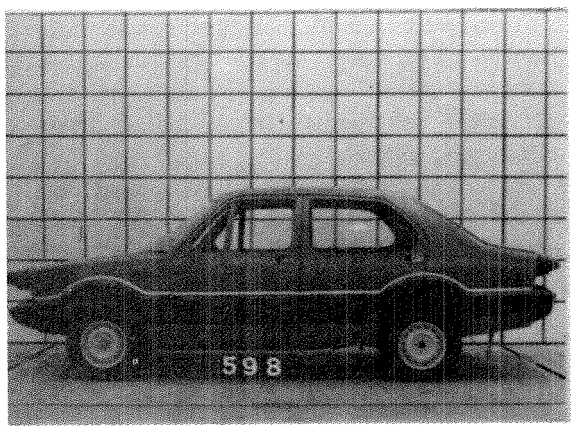
Finally, a 126 model was sent to strike the side of a 124 model at 6 degrees and 18 mph, i.e., the same conditions set for collapse inception in the 1,500 pound/2,000 pound test. Damages are more evident, not only on projections, but also in those areas where rigidity variations are present, namely, where a plain panel meets an area with underlying structural members: for example, in proximity of the A and B pillars on the 124 model (Figure 12c).

A comparison of the results from these tests shows how sidesweep collisions at angles from 5 to 10 degrees and speeds up to 18 mph for standard production cars, cause enough damage to call for quite costly repairs, whereas the ESVs (Figure 11),



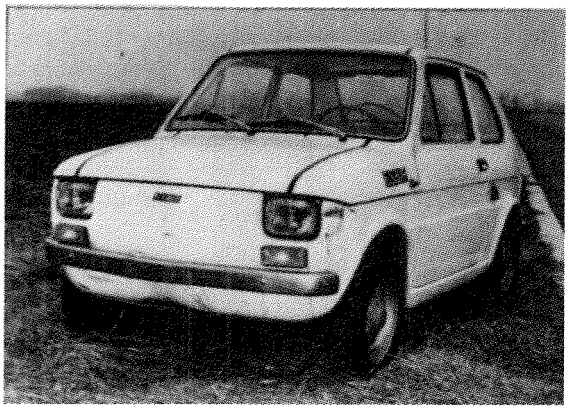
1500 lb SIDESWEPT BY 1500 lb
(COLLISION COURSE ANGLE: 10° – SPEED: 18mph)

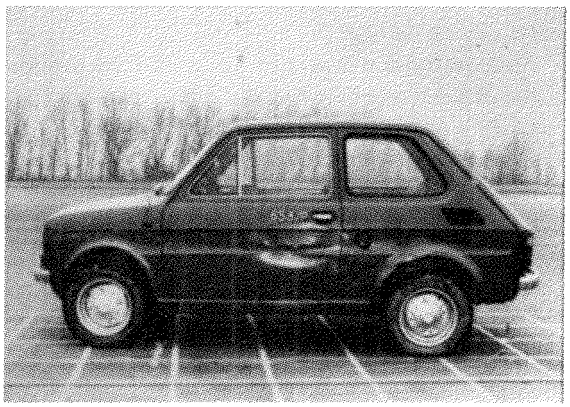
2000 lb SIDESWEPT BY 1500 lb
(COLLISION COURSE ANGLE: 5° – SPEED: 18 mph)



2500 lb SIDESWEPT BY 1500 lb
(COLLISION COURSE ANGLE: 6° – SPEED: 18 mph)

Figure 11



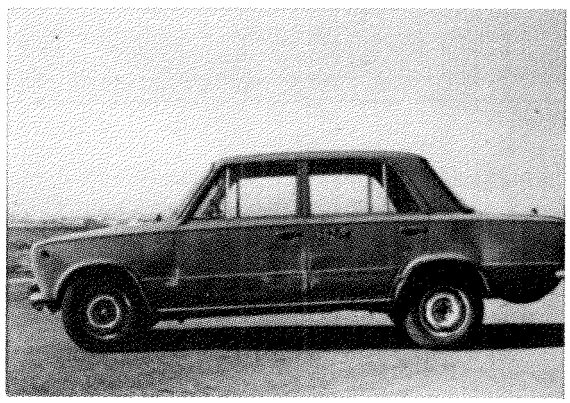


126 AGAINST 126

(A) (COLLISION COURSE ANGLE: 10° – SPEED 18 mph)

126 AGAINST 128

(B) (COLLISION COURSE ANGLE: 5° – 18 mph)



126 AGAINST 124

(C) (COLLISION COURSE ANGLE: 6° – 18 mph)

Figure 12

provided with lateral protection bands, suffer practically no damage.

An investigation was also made into the problem of the interactions between the body "contour" and "rigidity" parameters. To this end we have fitted reinforcements in the 2,000 pound ESV door (Figure 13) – the model having the smallest critical angle – and have run a collision test between two 2,000 pound ESVs. By plotting again the chart of maximum dynamic deformations versus speed (Figure 14) and

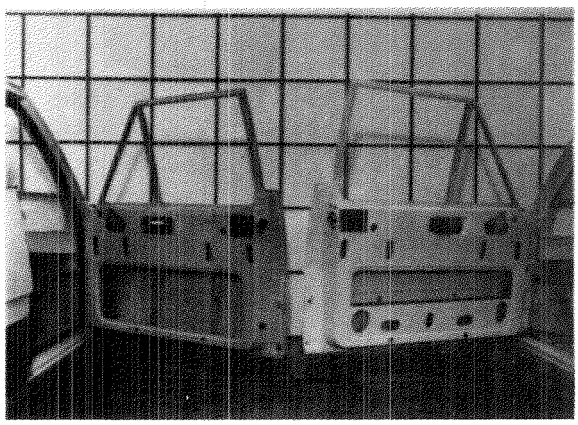


Figure 13

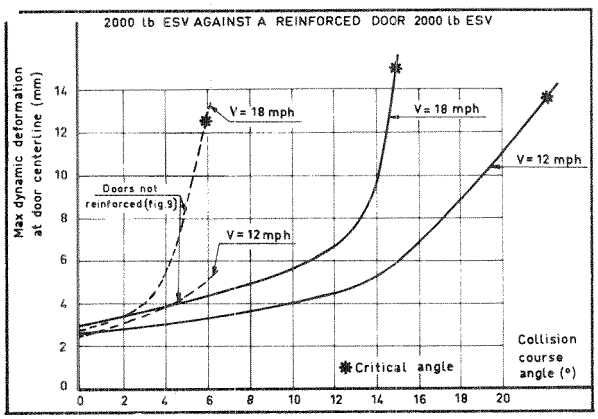


Figure 14

placing this chart over the one plotted for the 2,000 pound car struck by the lighter 1,500 pound unit (Figure 9), the benefits obtained through the application of local reinforcements are evident and significant.

Pedestrian Collision

Another aspect of compatibility considered concerns pedestrians: a car which is safer for occupants should not become more dangerous for nonprotected road users. To this purpose, we ran a set of impact tests to study mainly the trajectory of the dummy (pedestrian) when impacted by our ESVs.

Pedestrian collision (Figure 15) is a complex phenomenon which is characterized by four factors: impact conditions, characteristics of the striking car, impacted pedestrian, and the road surface on which the accident occurs.

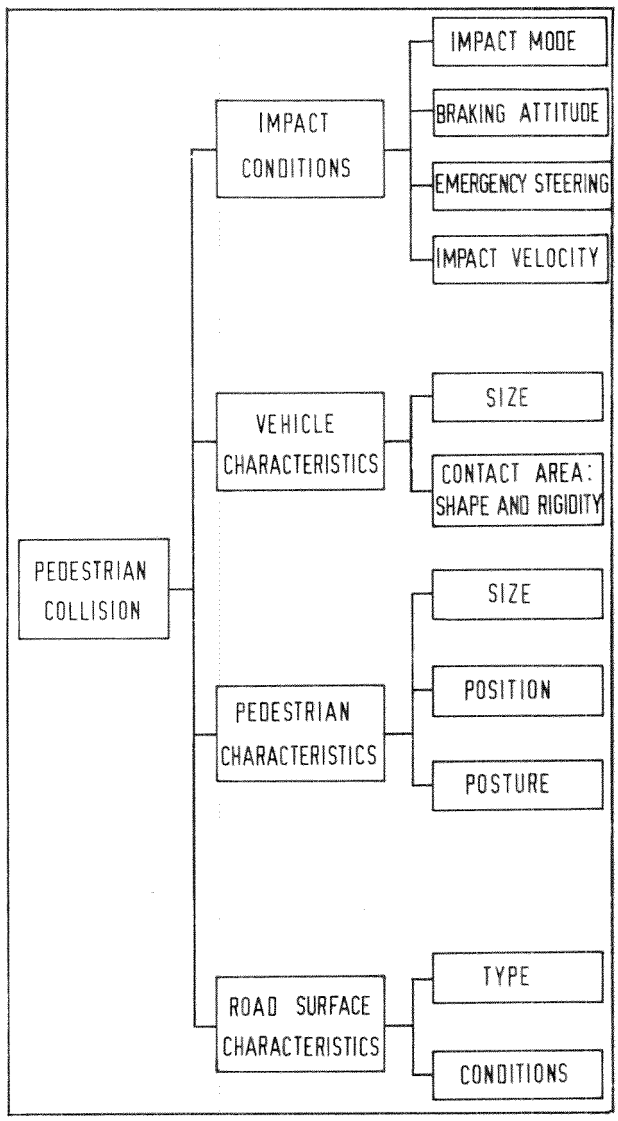


Figure 15

Impact conditions are linked primarily to the impact mode; according to the few statistical data available, we find over 75 percent of injured pedestrians are struck by the vehicle front end. Accordingly, only frontal impact tests against the dummy were conducted. Two series of tests were run with the car braked (panic stop) immediately after the collision (assuming the driver had not seen the pedestrian) or immediately before the collision (assuming the driver was unable to stop the car in time).

In real accidents the driver can steer abruptly in the attempt to avoid the pedestrian, thus changing

the car course and contact point. To simplify the test, the course was maintained straight and the dummy was hit by the car front end centerline. The parameter having the greatest influence on the impact condition is the speed at the instant of impact. From American, British and Australian statistical data it appears that most pedestrian collisions occur at low speeds, up to 18-25 mph at the most. Our tests were run at 6, 12, 18, and 25 mph.

As far as the vehicle characteristics are concerned, we ran the tests in the size range from small to medium sized cars, using our ESVs and standard production units. The contact area shape on the three ESV types (Figure 2) is purposely different to find out how it affects collision dynamics.

Car/pedestrian contact is influenced by front end shape and stiffness. The Washington ESV conference had furnished information on how our ESVs have plastic bumpers to provide for a softer and more uniformly distributed contact with pedestrians' legs. Substantially, pedestrian characteristics are:

- Size: a 50th percentile male dummy.
- Position: one set of tests was run with the dummy struck frontally by the car and one set with the dummy struck laterally.
- Posture: the dummy was standing erect, with closed legs, and sustained in a vertical position by a string strong enough to barely hold up the dummy – but not at all contributory as a restraint.
- Test site road paving characteristics: dry asphalt surface.

A first series of tests was conducted on the dummy struck frontally by the 1,500 pound ESV at increasing speeds of 6, 12, 18, and 25 mph, with braking immediately after the impact.

The test results demonstrate:

- At 6 mph, speed is so low that the dummy just folds forward and then falls into a seated position facing the car (Figure 17).
- At 12 mph, the dummy falls onto the hood on which it lies horizontally until it slips off and falls face down onto the ground (Figure 17).
- At 18 mph, the impact dynamics are similar but the dummy hits the windshield (Figure 18).
- At 25 mph, the dummy spins in the air and lands on its head and shoulders first (Figure 18).

It should be noted that all the varying dummy postures in the different tests can practically be reduced to the most severe, basic trajectory, i.e., the one of the impact at 25 mph. The higher the velocity, the more completely the dummy will follow the trajectory mode. Random factors or basic parameter variations may cause departures of the actual from the basic trajectory (Figure 16). For instance, at 6 mph the energy is still so low that the dummy cannot keep up its motion and the trajectory departs starting from position B; at 12 and 18 mph the energy is greater, and so the departure begins at position D. By repeating the series of tests at increasingly higher speeds with the 2,000 and 2,500 pound cars, it is found that dummy trajectories relevant to these vehicles have much in common with those recorded for the 1,500 pound car.

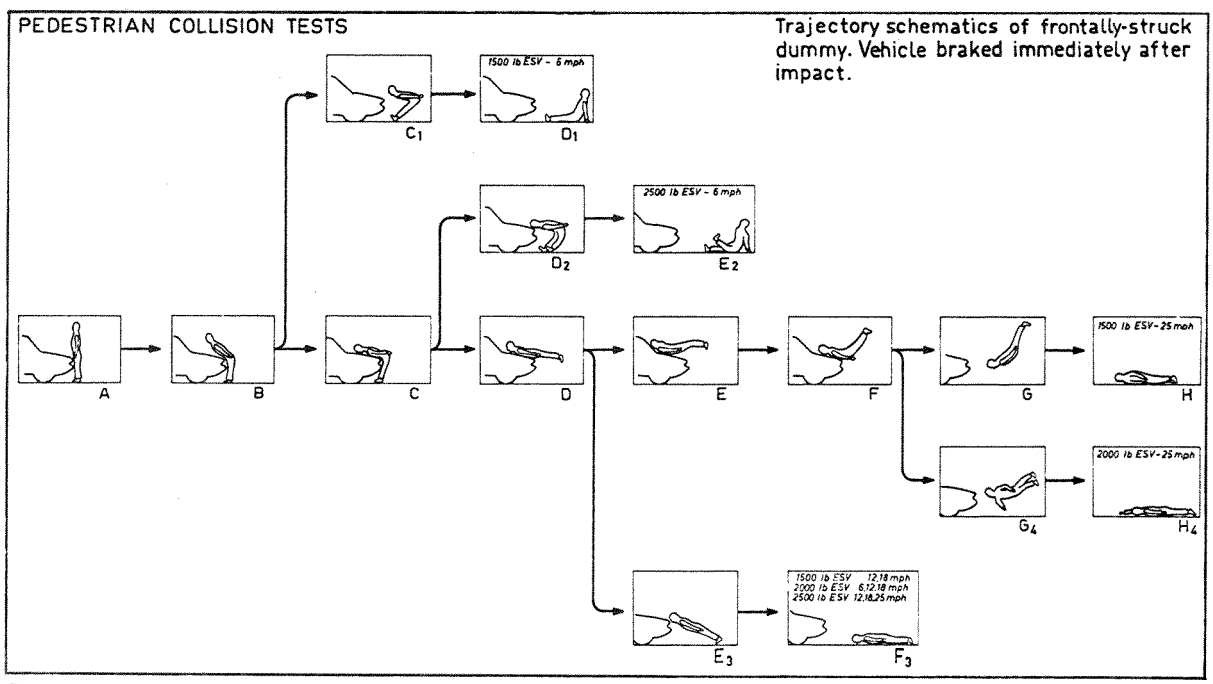
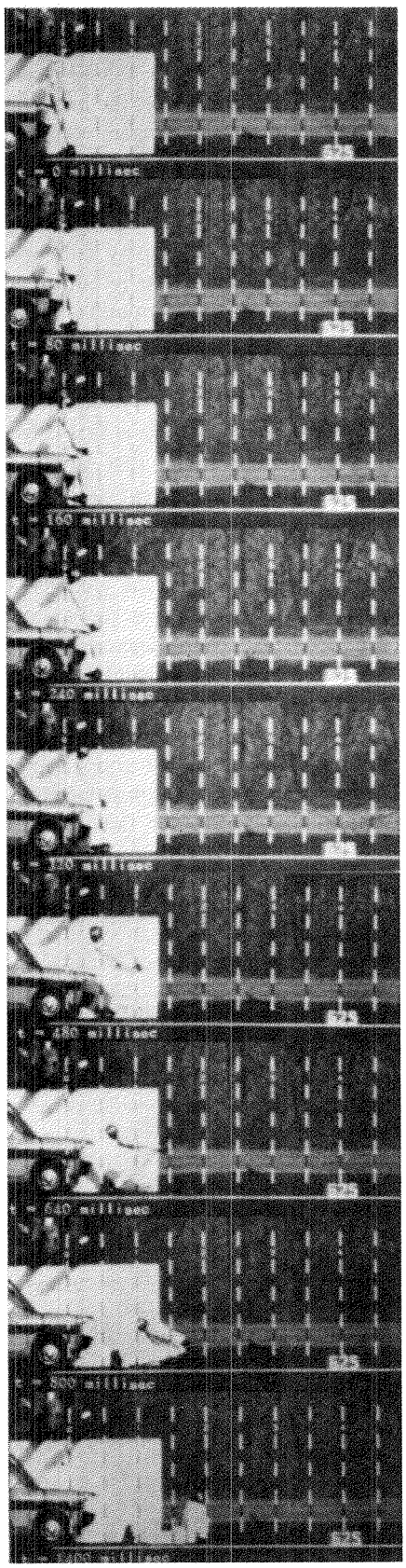


Figure 16

PHOTO REFERENCE NUMBERS FOR CORRECT SEQUENCE

190029

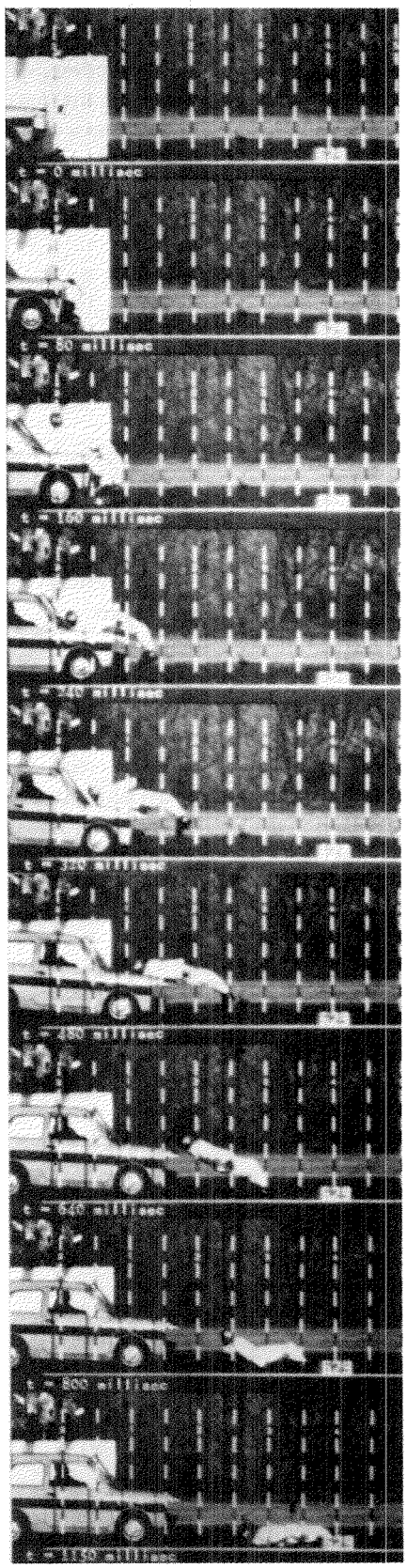
190030



1500 lb ESV HITS PEDESTRIAN AT 6 mph

190035

190036



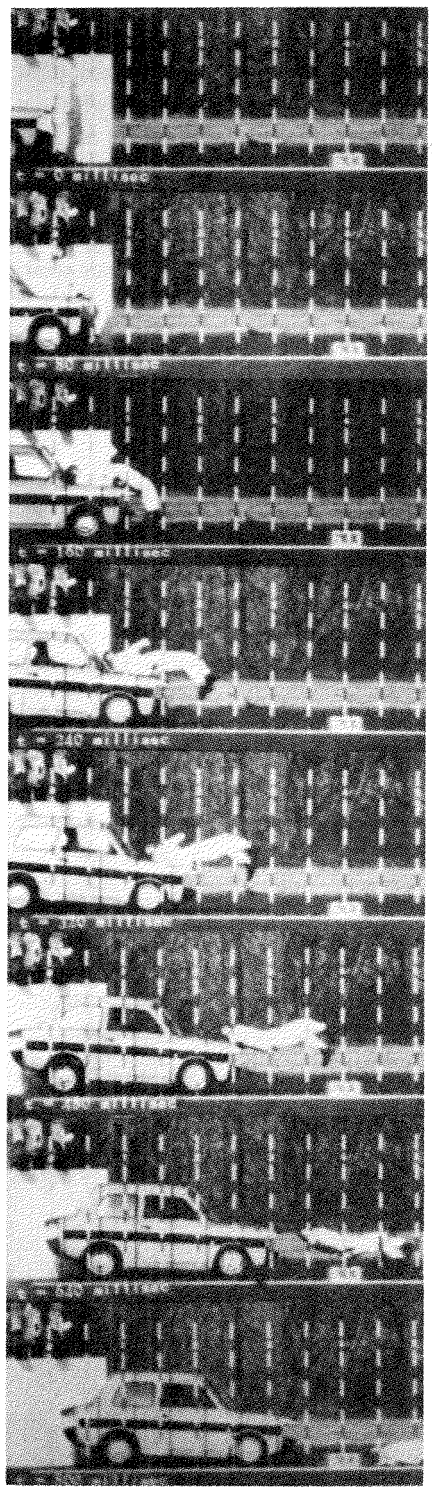
1500 lb ESV HITS PEDESTRIAN AT 12 mph

Figure 17

PHOTO REFERENCE NUMBERS FOR CORRECT SEQUENCE

190043

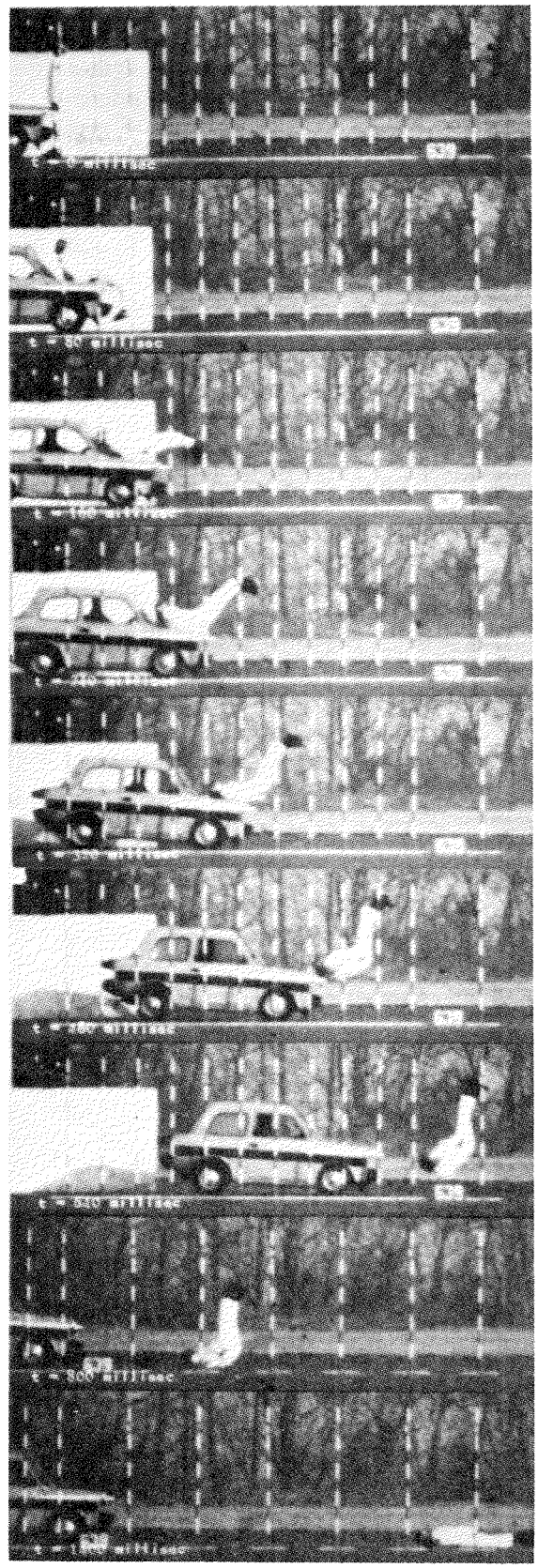
190044



1500 lb ESV HITS PEDESTRIAN AT 18 mph

190055

190056



1500 lb ESV HITS PEDESTRIAN AT 25 mph

Figure 18

The following basic concepts may be inferred:

- Pedestrian collision is made up of a succession of three impacts: first against vehicle front end, then against the hood, and, finally, against the ground.
- If speed is low, even the first impact alone may occur and, if the dummy falls on the ground, its feet are turned toward the car.
- At higher speeds the dummy is thrown up horizontally; and if the hood is short, it may eventually hit and possibly penetrate the windshield. After landing on the ground, the dummy's head is usually turned toward the car.
- At still higher speeds, the dummy spins in the air and drops down in different postures, depending on the angle of rotation.

A second set of tests was run, this time with the dummy struck laterally by the 2,000 pound car at speeds of 6, 12, 18 and 25 mph, and with brake application immediately after impact.

The second set of tests revealed:

- At 6 mph (Figure 21) – only the first impact occurs, the dummy sprawls its legs and falls on its back, with feet pointing toward the car.
- At 12 mph (Figure 21) – the dummy contacts the hood and falls horizontally.
- At 18 mph (Figure 22) – the dummy spins and then hits the ground headfirst.
- At 25 mph (Figure 22) – the spinning motion is still more evident.

Similar results were obtained in tests with the 1,500 and 2,000 pound cars. For the laterally struck dummy, it is possible to trace the basic and derived trajectories, as well as arriving at the same basic concepts outlined for the frontally struck dummy (Figure 20).

The frontal impact chart (Figure 19a) plots the dummy throwing distances versus car speed and indicates that practically no differences exist between the three types of cars. Moreover, with the laterally struck dummy, the results obtained are substantially identical (Figure 19b).

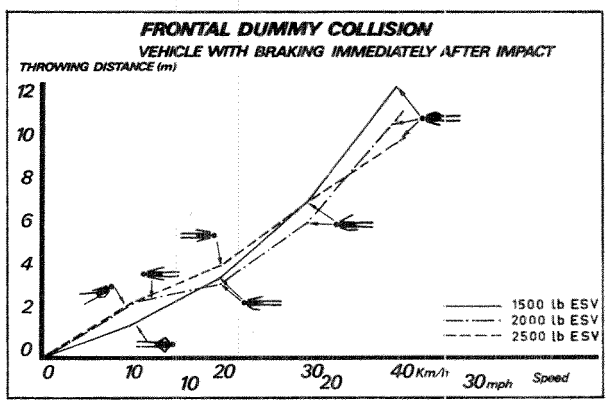


Figure 19(a)

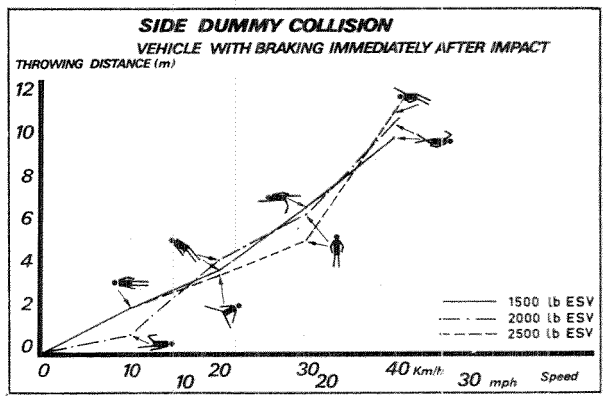


Figure 19(b)

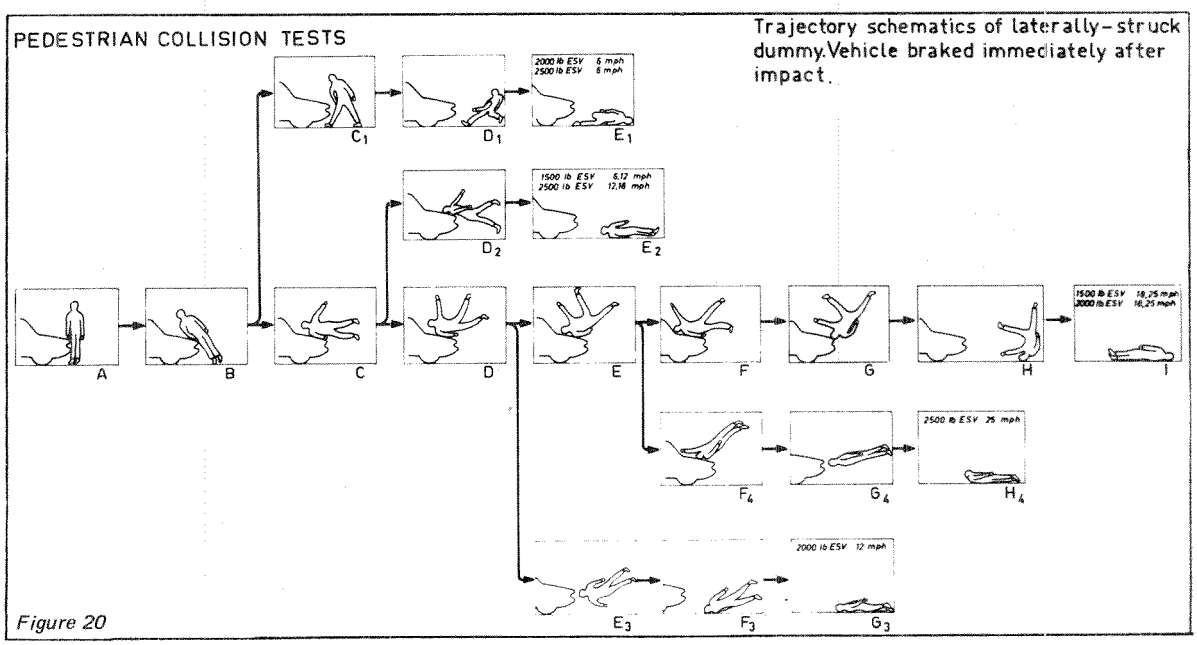
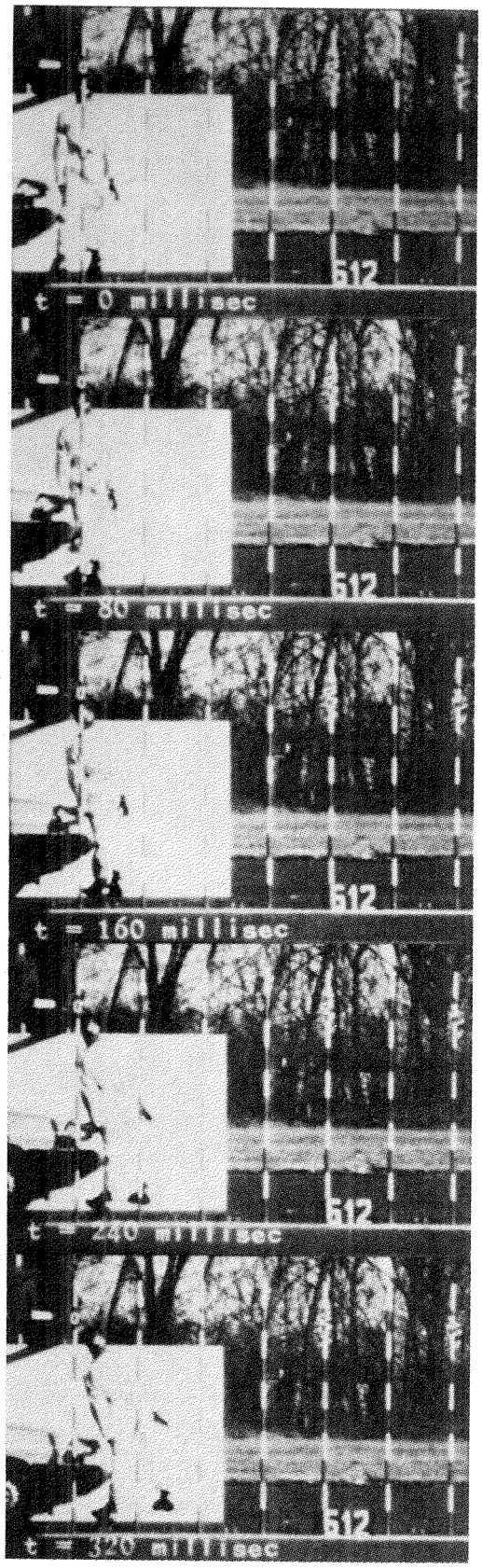


Figure 20

PHOTO REFERENCE NUMBERS FOR CORRECT SEQUENCE

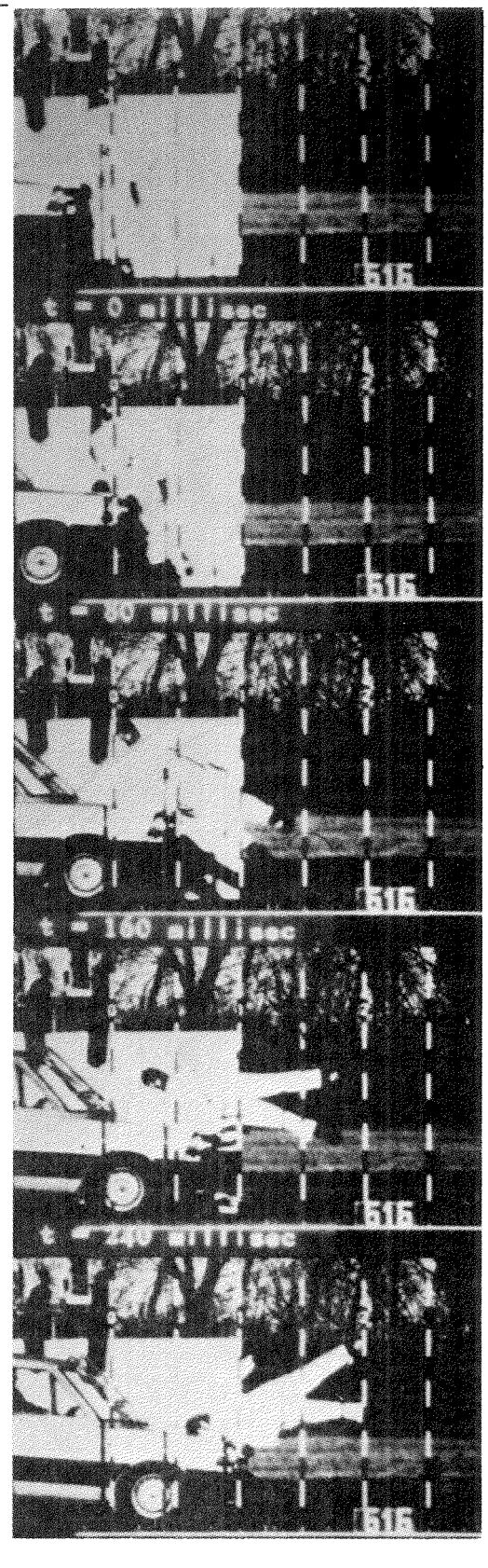
19003



2000 lb ESV HITS PEDESTRIAN AT 6 mph

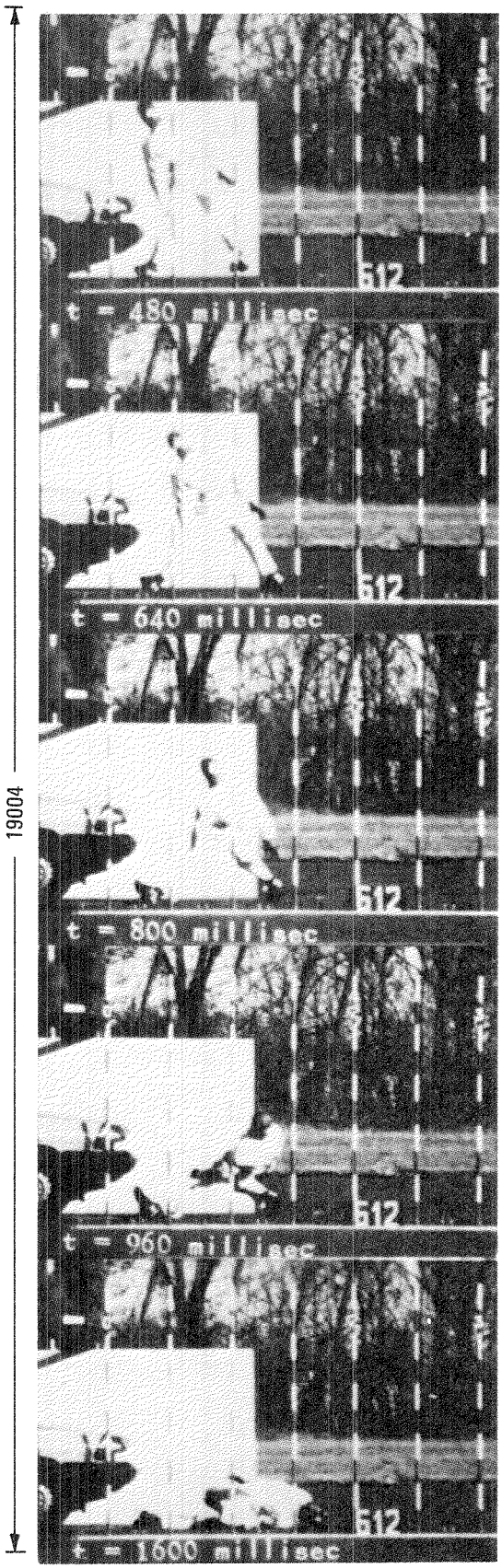
Figure 21(a)

190011



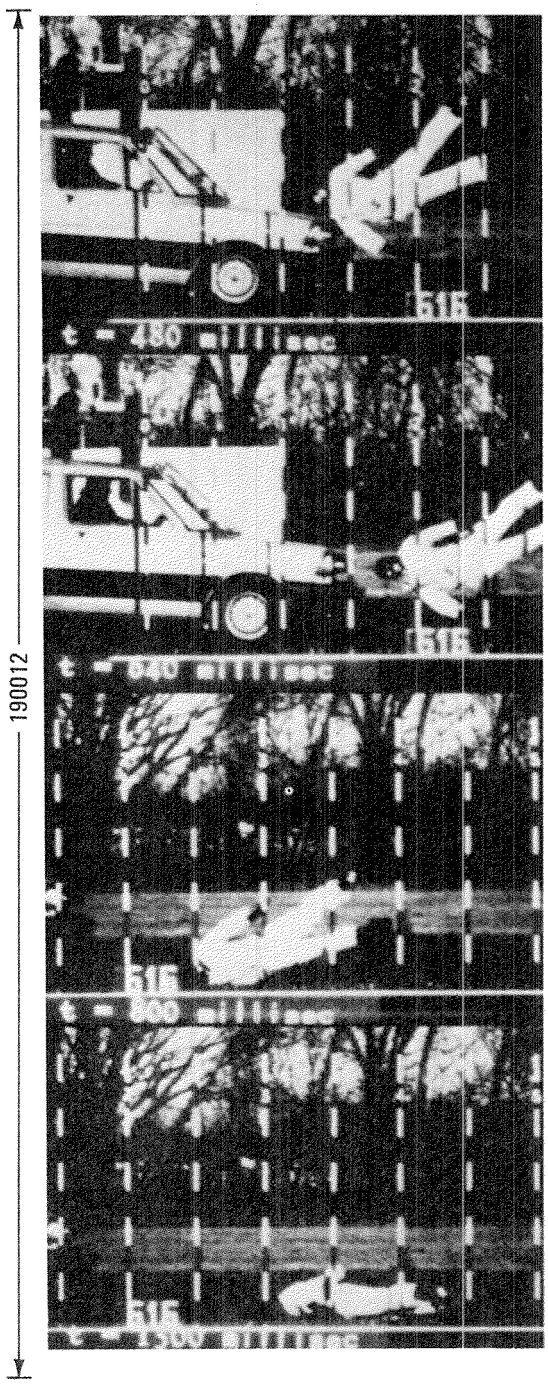
2000 lb ESV HITS PEDESTRIAN AT 12 mph

Figure 21(b)



2000 lb ESV HITS PEDESTRIAN AT 6 mph

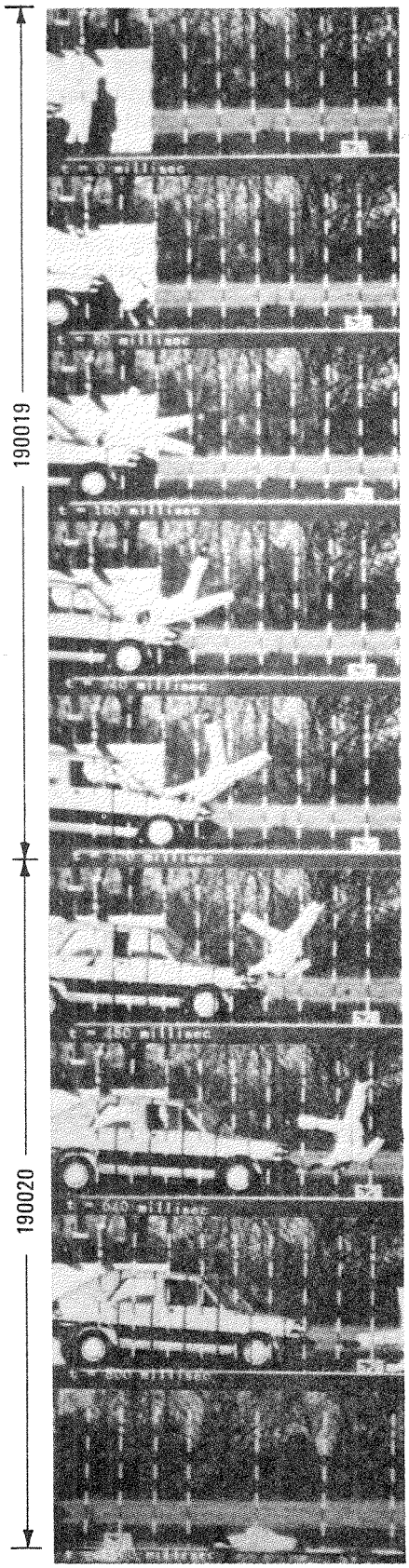
Figure 21(c)



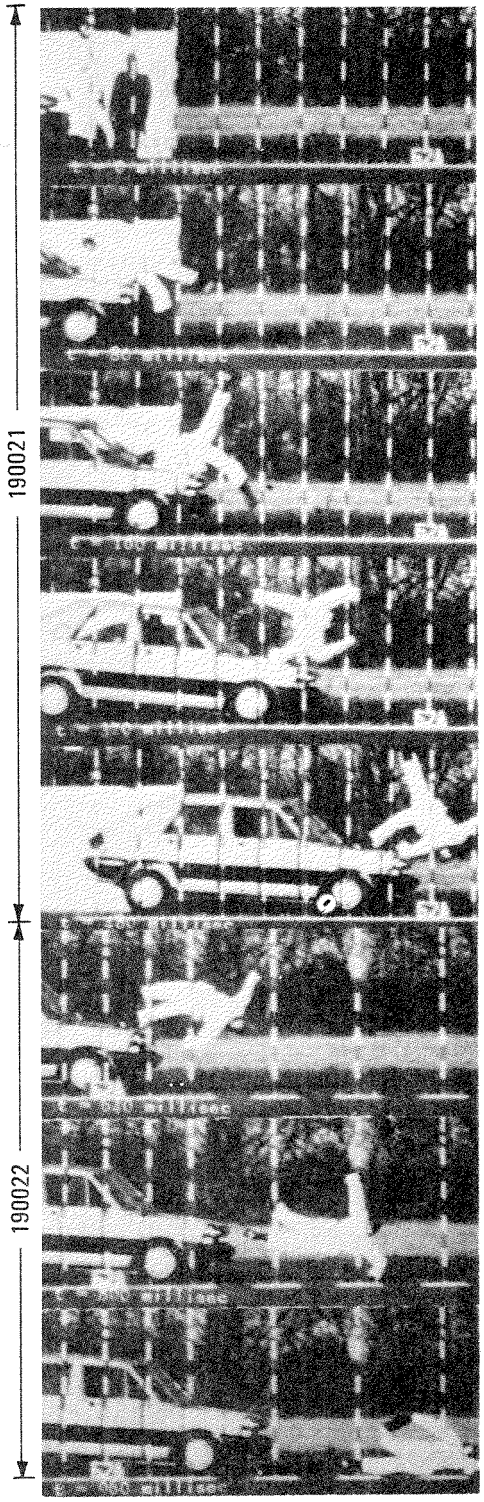
2000 lb ESV HITS PEDESTRIAN AT 12 mph

Figure 21(d)

PHOTO REFERENCE NUMBERS FOR CORRECT SEQUENCE



2000 lb ESV HITS PEDESTRIAN AT 18 mph



2000 lb ESV HITS PEDESTRIAN AT 25 mph

Figure 22

All the previously described tests were repeated with emergency braking (panic stop) two meters before impact. As a result, the front end dives down 11 cm and 15 cm for the 1,500 and 2,000 pound cars, respectively. The purpose was to see whether the different car attitude had any influence on the collision phenomena. No substantial difference was found: the only variable was the reduced distance to which the dummy was thrown. In no case was the dummy run over by the striking car.

In view of the results obtained, it was decided to check what would happen with standard production cars. The same tests run with the ESVs were repeated, this time with the 126, 128 and 124 models striking the dummy frontally and laterally. Dummy trajectories are substantially the same, although the striking car types are different; they are influenced essentially by impact velocity. Therefore, the following conclusions are drawn:

- Pedestrian collision is the succession of three impacts: first against the vehicle front end, then against the hood, and finally, against the ground.
- Pedestrian compatibility may be improved for the first and second impacts by making the front end and hood adequately deformable and "soft"; the third impact cannot be affected except by providing means to catch and hold the pedestrian on the car.

RESTRAINT SYSTEMS

For restraint system evaluation, the conclusions of the frontal impact tests up to 50 mph (Figure 23), presented at the Kyoto ESV Conference, were:

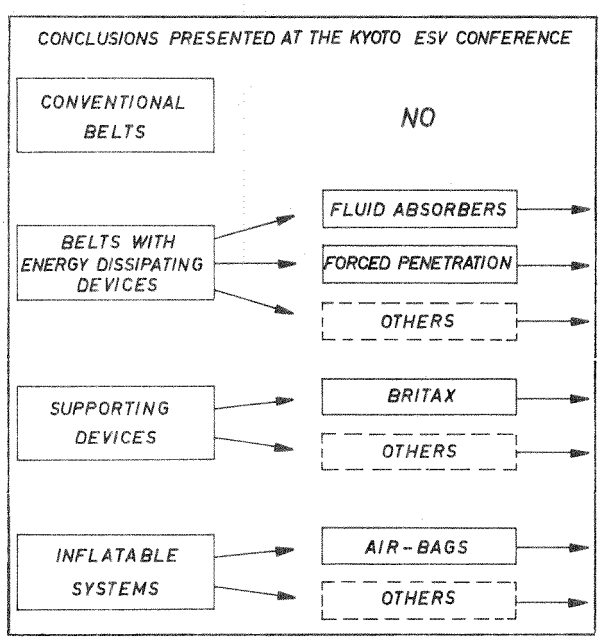


Figure 23

- Conventional belts: Not acceptable even at 30 mph.
- Dissipator belts: Promising – they are being optimized to meet our requirements.
- Supporting devices: Further investigation is needed.
- Air bags: Their use on small cars continues to be questionable.

Research was subsequently directed on 30 mph impacts only and focused on dissipator belts and supporting devices. Tests were conducted on dolly-mounted body mock-ups simulating the 128 Sedan interior (Figure 24) with the seat set in mid-position. In all tests the instrument panel was

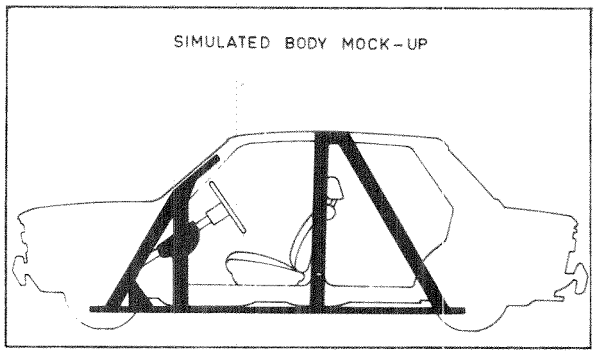


Figure 24

replaced by a polystyrene foam knee protector limiting the load on femurs. The dolly, launched on catapult, was decelerated through the deformation of tubes.

Deformation was 33.5 cm, and average deceleration was 30 g (Figure 25). Tube deformation (car crush) and pulse duration were less than on our standard production cars; in other words, testing

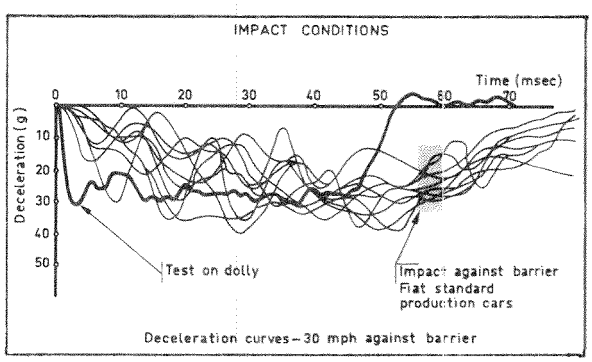


Figure 25

conditions were more severe than those of the usual barrier impact tests. The results shown (Figures 26 and 27) are averaged from the results obtained in three tests run under identical conditions.

PASSENGER															
RESTRAINT SYSTEM	Head				Thorax			Pelvis			Femurs		Belts		
	SI	HIC	Max. Res. Dec.	Δt 80 g	SI	Max. Res. Dec.	Δt 60 g	SI	Max. Res. Dec.	Δt 60 g	left	right	Thorax	Lap	
														Inboard	Outboard
	–	–	g	msec	–	g	msec	–	g	msec	kg	kg	kg	kg	kg
Conventional belts	963	–	59	–	328	66	1.5	336	58	–	147	307	1117	1403	603
Belts plus marketed dissipators	763	–	68	0.8	229	51	–	499	77	7.0	243	297	587	–	437
Belts plus FIAT dissipators	496	–	47	–	264	56	–	376	56	–	127	443	923	917	687
Accles Britax supporting device	262	–	47	–	261	45	–	405	59	–	650	650	–	–	–
FIAT mobile cross member	255	–	42	–	219	47	–	327	57	–	403	450	–	–	–
Above figures are averages of values obtained from 3 tests run under identical conditions.															

Figure 26

DRIVER															
RESTRAINT SYSTEM	Head				Thorax			Pelvis			Femurs		Belts		
	SI	HIC	Max. Res. Dec.	Δt 80 g	SI	Max. Res. Dec.	Δt 60 g	SI	Max. Res. Dec.	Δt 60 g	left	right	Thorax	Lap	
														Inboard	Outboard
	–	–	g	msec	–	g	msec	–	g	msec	kg	kg	kg	kg	kg
Conventional belts	1468	408	111	2.5	343	57	0.7	401	61	1.7	123	540	1247	1380	600
Belts plus marketed dissipators	1352	922	178	3.7	242	48	–	441	64	1.5	233	410	987	–	510
Belts plus FIAT dissipators	1002	412	128	3.5	290	53	0.3	518	72	6.3	150	627	843	723	653
Belts plus FIAT pre-loading dissipators	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–
Collapsible steering wheel and belts plus FIAT dissipators	758	287	68	–	312	54	–	562	74	7.5	150	675	975	525	910
Above figures are averages of values obtained from 3 tests run under identical conditions.															

Figure 27

Conventional Belts

Tests were conducted with conventional belts for both the passenger and the driver to obtain terms of comparison with other systems. Stresses are critical for the passenger and excessive for the driver; in both cases, load on the belts is very high.

Several types of energy dissipators – devices which when used in conjunction with belts having webbing stiffer than usual, permit occupant movement control and load limitiation, thereby reducing decelerations – were then investigated. The devices tested were installed in different ways – inserted in line with the belts and incorporated in retractors – and were based on different operation principles – friction or plastic deformation. The plastic deformation was obtained in two ways: extrusion by an ogive-shaped penetrator and crushing of tubes.

In-line Friction Dissipator Belts

Two versions of a marketed type were tested (Figure 28):

- Passenger belt – Two dissipators, one in the thorax and one in the inboard lap sections.
- Driver belt – One dissipator in the inboard lap section.

Better results were obtained than with conventional belts except for an increase in driver's head stress consequent to impact against steering wheel.

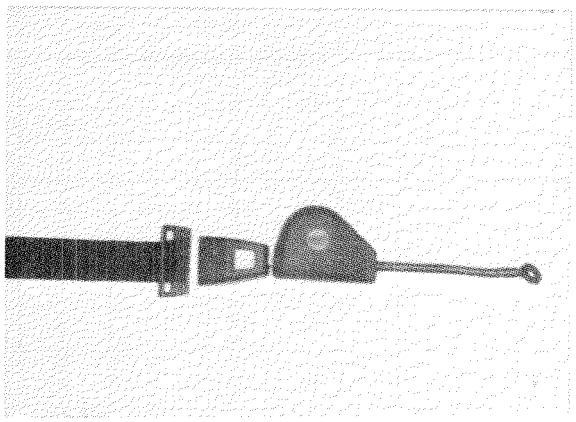


Figure 28. Stahl-Garant friction dissipator: lap belt inboard portion

In-line Deformation Dissipator Belts

The dissipator (Figure 29) works through extrusion of a plastic container by a metal ogive-shaped penetrator; this is the laboratory-developed type, presented at Kyoto, and now revised to save space.

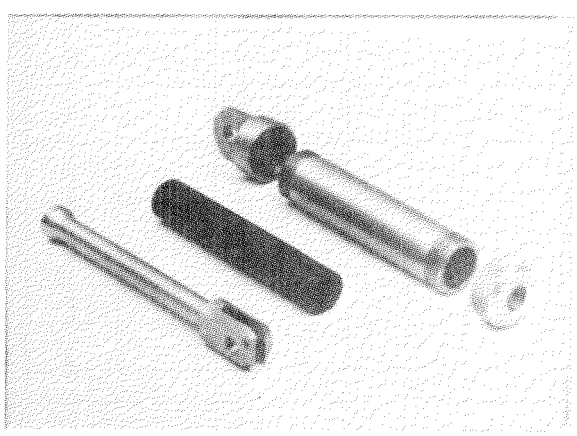


Figure 29. Deformation (extrusion) dissipator: components

Retractor-Incorporated Friction Dissipator Belts

In one laboratory device (Figure 30), a friction dissipator has been incorporated in a retractor. No tests on the the dummy were run with the systems shown in Figures 29 and 30. The investigation was limited to operational checks on the bench.

Retractor-Incorporated Deformation Dissipator Belts

Another dissipator (Figures 31 and 32) of the crushing-tube type was built in the laboratory and incorporated in a retractor; it was used in tests as passenger and driver restraint (Figures 33 and 34). Results indicated were:

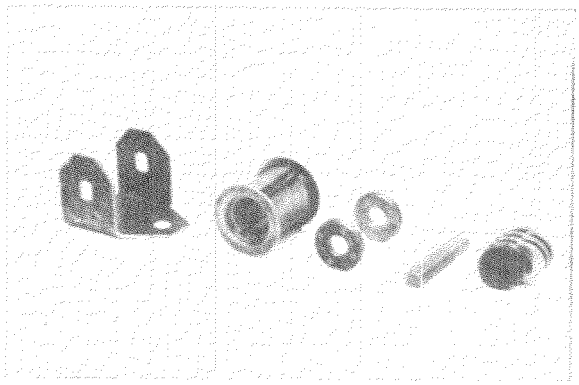


Figure 30. Friction dissipator: components (Simplified version, dissipator only)

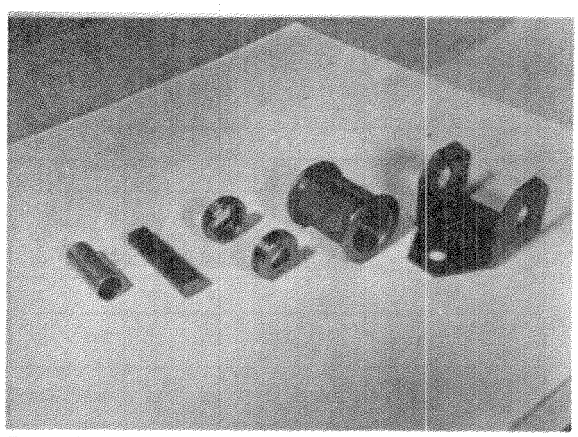


Figure 31. Crush-type dissipator: components

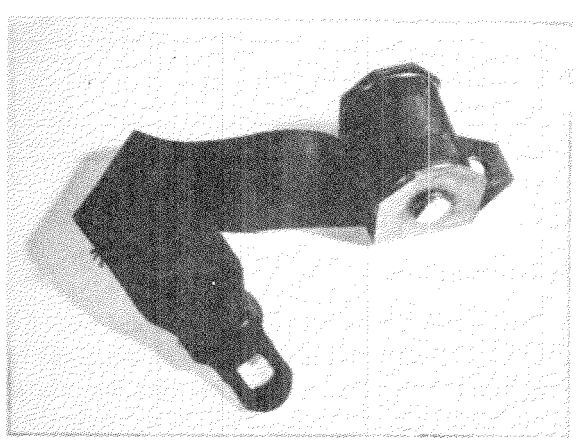


Figure 32. Crush-type dissipator: assembly

- Passenger – Increased forward displacement with respect to conventional belts, and lack of rebound.
- Driver – Severe head impact against steering wheel, consequent to increased displacement. On comparison with conventional belt tests, stresses were found to be considerably reduced except, obviously, for the driver's head.

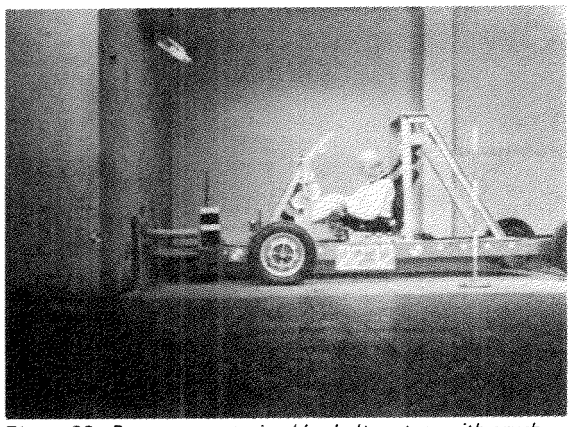


Figure 33. Passenger restrained by belt system with crush-type dissipator

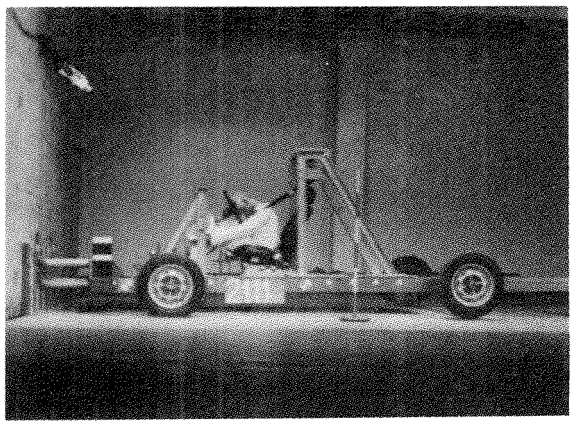


Figure 34. Driver restrained by belt system with crush-type dissipator

Dissipator/Pre-Loader Belts

To prevent head impact against the steering wheel, an attempt was made to put the belts in tension in the shortest possible time by using a preloading device. One such unit (Figure 35), operated by a hydraulic piston acting during car crushing, was constructed in the laboratory. Thanks to an inner load-limiting device, the hydraulic piston stretches the belt thorax section and simultaneously prevents a given limit load from being exceeded. With this system, the head impact is, in some cases, avoided (or is lightened), whereas, in some other cases, it is not. Results are not repeatable; consequently, the stress values are not quoted. This solution does not solve the problem.

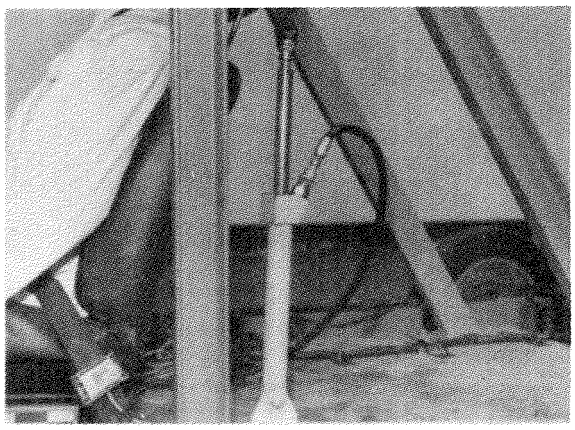


Figure 35. Pre-loading dissipator device mounted on a dolly

Tests were also conducted on supporting devices of the thoracic-plate swinging arm type, and mobile cross member type.

Swinging-Arm Supporting Device

The study on the Accles Britax device (Figure 36), presented at the ESV Conference in Kyoto, was continued. Dummy kinematics was less than satisfactory even though dummy stresses dropped to very low levels.

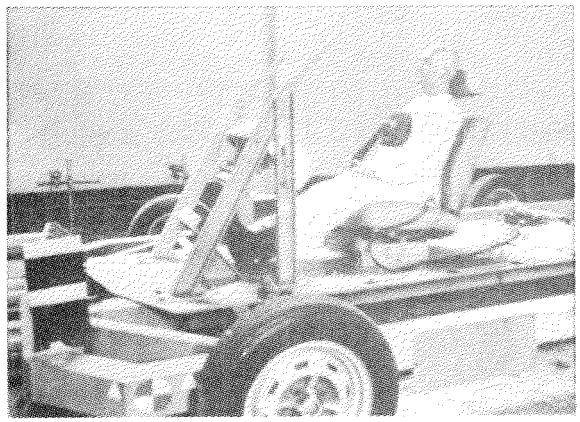


Figure 36. Accles Britax supporting device mounted on a dolly

Mobile Cross-Member Supporting Devices

Tests were run to evaluate an early laboratory-developed system, consisting of a mobile cross member (Figure 37) which, with the car in

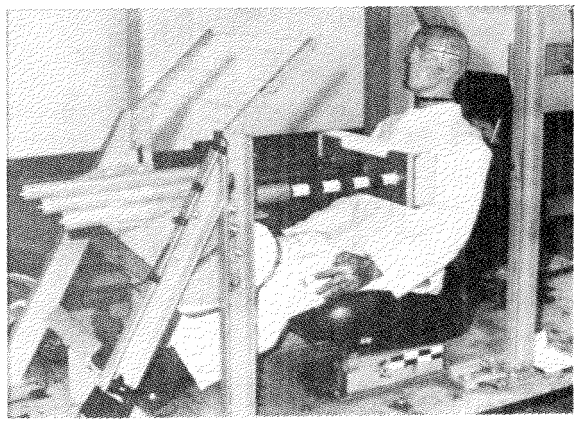


Figure 37. Mobile cross member supporting device mounted on a dolly

motion, is placed in contact with the occupant thorax and which, during impact, is pushed forward by the occupant with consequent deformation of energy dissipators. Occupant kinematics is good and dummy stresses are the lowest recorded during this research work. The device was experimentally installed on the car. If it should be required only for passenger use, few changes would be required (Figures 38 and 39). Conversely, modifications would be radical should

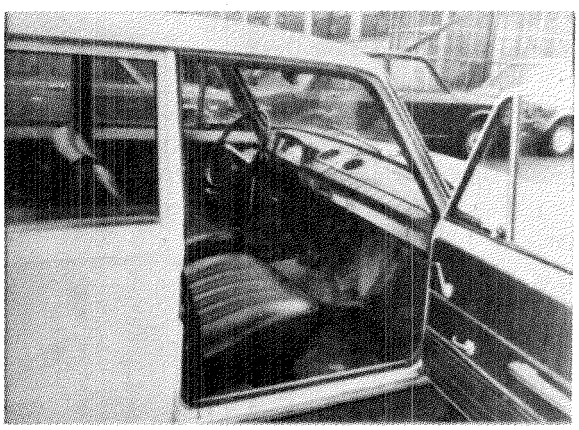


Figure 38

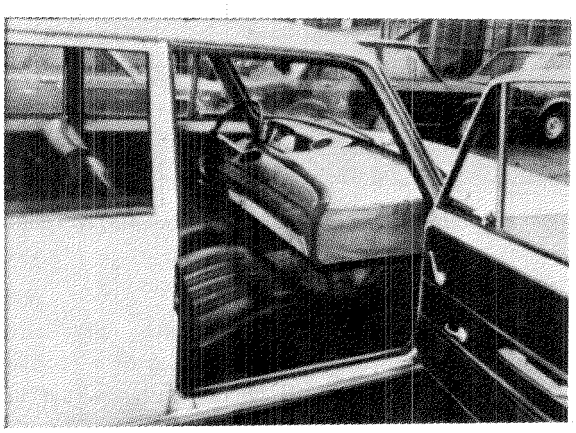


Figure 39

the device be required for both front occupants (Figures 40 and 41). With this solution, the steering wheel is replaced with a stick/lever control situated in the mid-floor between seats.

Presently, man already drives many vehicles not provided with a steering wheel, yet he would feel bewildered in a *car* without a steering wheel. It would be necessary to overcome this mental reluctance and devise a new mode of driving a car. The device could be easily made passive by connecting its control to the door. A similar restraint system could be developed for rear occupants (Figures 42 and 43).

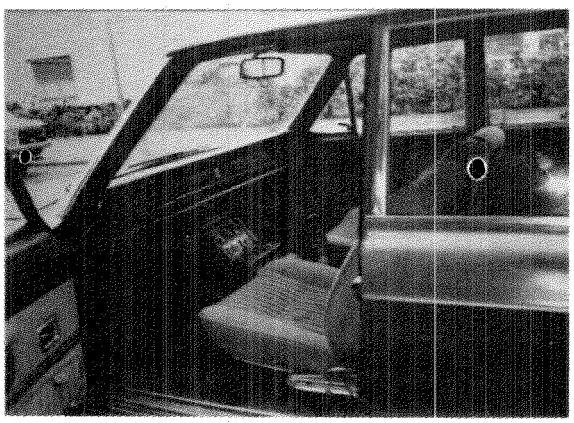


Figure 40

Figure 41

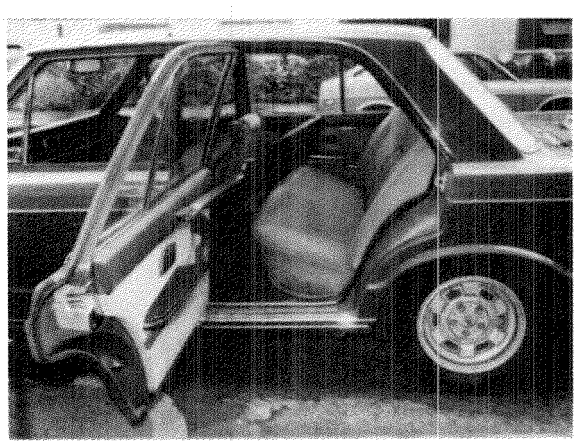


Figure 42

Collapsible or Retractile Steering Wheel

Tests were run to evaluate the suitability of using belts with energy dissipators in conjunction with a

retractile energy-absorbing steering wheel. It was found that the dummy – effectively restrained by the belts – strikes the steering wheel with an energy less than that required for dissipation by the wheel, but still sufficient for a whip effect which will induce high declerations on the head. The values measured are not reported, as they are practically the same as those obtained with the standard steering wheel.

A solution was then tried in which the steering wheel collapses during impact (Figures 44 and 45). By combining such a steering wheel with belts having

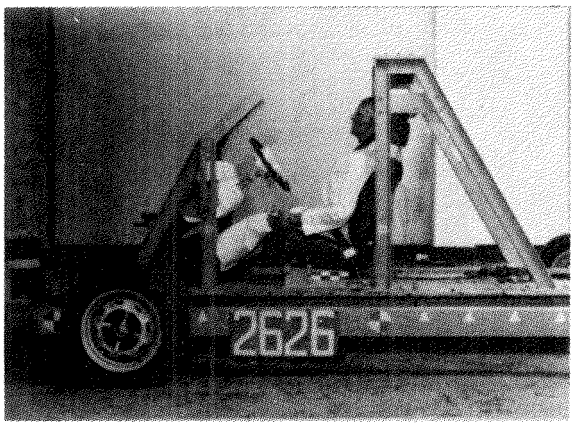


Figure 44

energy dissipators, a sound arrangement for the driver is obtained. Stresses on the head drop to acceptable limits.

Of all the restraint systems tested (Figure 46) under the above conditions, conventional belts are not acceptable. On the other hand, belts with energy dissipators are satisfactory and they can easily be provided. The swinging arm type of the supporting devices is not yet sufficiently perfected. The mobile cross member type gave the best results compared

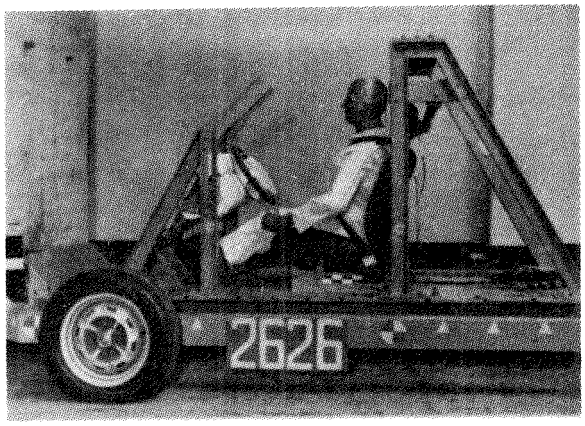


Figure 45

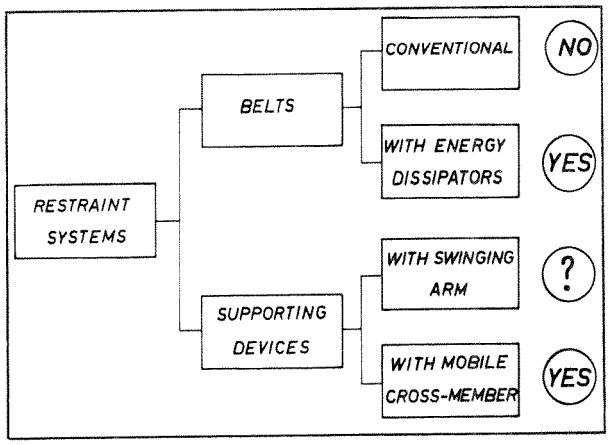


Figure 46

with all the other systems, although it would require radical changes in the car.

The study of different system designs for occupant restraint during frontal impacts confirmed the necessity for providing (as a prime requisite) in the passenger compartment sufficient additional room for the operation of such devices.

To meet targeted requirements laid down by the Kyoto conference – namely, occupant protection in a 30 mph front barrier impact – the most convenient and practical solution would be the use of belts with retractor-embodied energy dissipators, associated with a knee protector and, in the case of the driver, combined with a retractile steering wheel.

CONCLUSIONS

Predicated on DOT Specifications for the ESV – even though planned objectives of tests to assess crashworthiness and occupant protection have not yet been fully achieved – one may interpret test results for small and medium-small cars to demonstrate the unacceptabity of weight and cost increases.

GUSTAV ECKBERG
Head of the Vehicle Department
Swedish Road Traffic Safety Office

In Sweden, as in all other countries, the fight against road traffic accidents is going on continuously. We have endeavored to induce road users to behave in a more safety-minded way, to improve our roads to better suit the steadily increasing traffic, to make vehicles safer and to adjust the traffic laws to obtain a maximum of safety. I can say that we have made some progress in this field.

During the last few years the number of persons killed or injured in road traffic accidents has shown a slight tendency to decrease. If taken in relation to the increasing number of motor vehicles and growing traffic, this promising change becomes more evident. Thus, during the last fifty years the number of persons killed or injured per kilometer driven has been reduced to forty or fifty percent.

Many factors have contributed to this development, such as the introduction of speed limits for all roads, improved driver education, and by more severe requirements on and better control of motor vehicles. A factor of great importance is the compulsory installation of safety belts in passenger cars. Although the use of safety belts is not as frequent as it should be, statistics clearly indicate the protective effects of belts.

The Swedish government has, therefore, decided to introduce a new law on the compulsory use of safety belts. The law will probably come into force on the 1st of January, 1975. It is estimated that this measure will cause a reduction in the number of car occupants killed or injured by about 40%.

The Swedish contribution to the ESV program is, as you may recall, an investigation concerning nonlocking brakes called "Steerability During Emergency Braking." The work has proceeded as planned and Mr. Sten Rundkvist, who is the leader of this ESV project, will now give a summary of the final report.

STEN RUNDKVIST
Project Manager
Swedish ESV Project

The Swedish ESV project "Steerability during Emergency Braking" started in March 1972 with the following three objectives:

- To investigate the safety benefits of steerability during emergency braking
- To recommend test procedures
- To suggest performance standards

The project was a joint effort with representatives from the Swedish Government, represented by the Swedish Board for Technical Development (STU), Volvo, and Saab-Scania.

This presentation is an abstract of the final Report 1-01 "Steerability during Emergency Braking. Summary Report."

SAFETY BENEFITS

The investigation of the safety benefits of steerability during emergency braking is based on police reports and interviews about traffic accidents with locked wheels.

Locked wheels occurred in 10 to 15% of all traffic accidents involving passenger cars. If the vehicles had been steerable, 70% of the locked wheel accidents could have been avoided or reduced.

If consideration is taken of the driver's attempt to steer during those accidents, and if the vehicles had been steerable during emergency braking, 4 to 7.5% of all traffic accidents with passenger cars could have been avoided or reduced.

55 drivers were exposed to a situation in which a "dummy" car suddenly appeared in the driver's lane. The result indicates that the attempt to steer could be higher than the value used in the accident investigation.

One way to get steerability during emergency braking is to install an anti-lock system. Without an anti-lock system the vehicle continues straight ahead. But it is not enough to install an anti-lock system. The system must be adapted to the vehicle, otherwise the performance can be questionable.

TEST PROCEDURES

Different test methods were attempted, but in order to get comparable results under different conditions the test had to meet the following requirements:

- Steering, braking and stability performance during braking in a curve;
- Highest possible lateral acceleration;
- Well-defined conditions regarding braking and steering;
- A test independent of the driver's skill;
- Have good repeatability.

The high lateral acceleration was accomplished by using a steering wheel angle corresponding to the minimum free-rolling radius for the actual parameter combination (velocity, tire-road combination, vehicle type, loading condition). For a specific velocity and with the same tire-road combination, the minimum free-rolling radius is rather independent of vehicle type and loading condition.

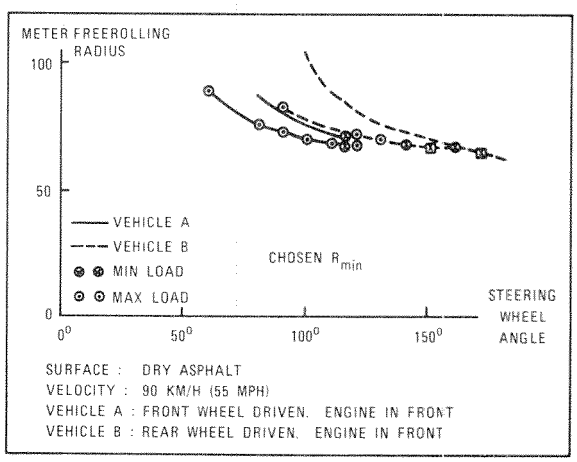


Figure 1

The other test requirements were met if the driver turned the steering wheel to a stop and kept it there without any compensating steering actions.

Those requirements led to a test method used during the systematic field testing and simulation consisting of braking in a curve, where the steering wheel is rapidly turned to a stop giving minimum free-rolling radius, and applying the brakes one second later.

However, this test method requires rather expensive measuring equipment, comprising a platform, stabilized for roll and pitch giving longitudinal deceleration, lateral acceleration and yaw velocity, and special amplifiers to feed the tape recorder.

To eliminate the expensive measuring equipment, the following test method is proposed:

- Braking in a curve with simultaneous brake application and turning the steering wheel to a stop giving minimum free-rolling radius under the actual parameter combination.
- Braking straight ahead with locked wheels to use as a reference.

The measurements are easy to perform and at braking in a curve the vehicle is driven on a marked, straight line, and the braking and steering commences when the correct velocity is attained.

After the vehicle has stopped, the following information is noted:

From the fifth wheel:

- Velocity at brake application
- Braking distance
- Maximum vehicle side slip angle
- Lateral deviation of the vehicle's center of gravity (measured perpendicular to the marked, straight line)

At braking straight ahead with locked wheels, the following information from the fifth wheel is noted:

- Velocity at brake applications
- Braking distance

During the systematic field tests, the following parameters were used:

- Four different vehicle anti-lock combinations
- Four different tire-road conditions
- Two or three different velocities
- Two loading conditions

The different vehicle anti-lock combinations give different steerability and brakeability values, and the tendency to instability on ice was different.

The four tire-road conditions were:

- Ice and summer tires
- Ice and studded winter tires
- Wetted sealed asphalt
- Dry asphalt

It was difficult to get comparable test results with summer tires on ice from one day to another. The reason was that the ice characteristics were not only dependent on temperature but also on moisture in the air, age of the ice and snowflakes driving with the wind. The longitudinal friction number (μ) was measured with a special measuring vehicle on the same ice with the same summer tire at two different occasions, and there was a big difference in the peak values.

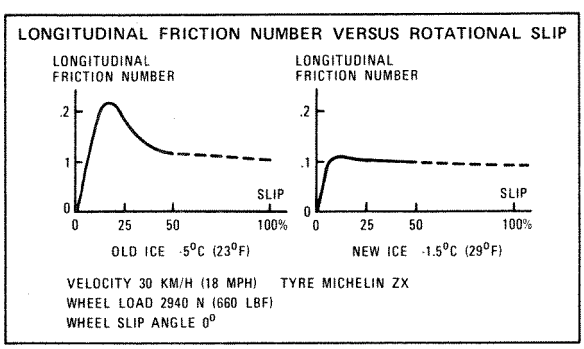


Figure 2

Studded winter tires on ice gave no such variation in the peak value why this combination is recommended for low-μ tests, which give the best indication of the vehicle's degree of stability.

Tests on wetted sealed asphalt gave no indication of instability, although tests on ice with the same vehicle anti-lock system showed a high instability tendency for some parameter combinations. This is probably due to the nature of the μ-value for the wetted sealed asphalt. For a high velocity the μ-value is very low, but at lower velocities the μ-value increases considerably (variable-μ surface). During braking the velocity goes down and the μ-value is increased which contributes to a stable performance.

Tests on wetted sealed asphalt did not give any more performance information than the tests on dry asphalt.

Tests on surfaces where the friction (μ-value) appreciably changes with the velocity, are not recommended for performance investigation according to the proposed test method.

Tests with different velocities gave only slight differences in the vehicle's behavior, when only one velocity for each surface should be sufficient.

The tendency for instability varied with the loading condition for different vehicles why tests are recommended both with maximum and minimum load.

The following test conditions are proposed:

- Minimum and maximum load
- Tests on low-μ surface at 50 km/h (30 mph) (studded winter tires on ice)
- Tests on high-μ surface at 90 km/h (55 mph) (dry asphalt)

PERFORMANCE STANDARDS

When the ESV project started two years ago, the intention was to study the relation between steerability and brakeability under different conditions. During the tests on ice it became apparent, however, that if the vehicle showed bad stability, the tests were of doubtful value. Accordingly, the stability had to be included in the study with highest priority.

Stability

Stability of a vehicle is here defined as the ability to resist external disturbances and keep the deviation to a minimum.

One way to measure the degree of stability is to measure the vehicle side slip angle (β) during braking

in the proposed curve test, where the steering wheel is kept fixed.

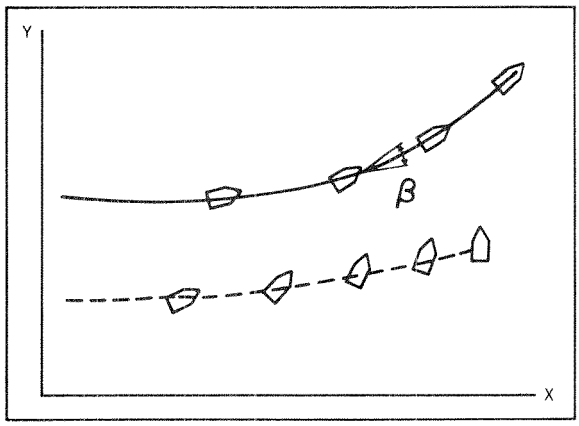


Figure 3

During a curve test the vehicle attitude varies according to the plotted curves. In the upper curve the vehicle side slip angle, β, is at a maximum and then decreases at the end of the braking. In the lower curve the vehicle side slip angle is continuously increasing, giving bad stability.

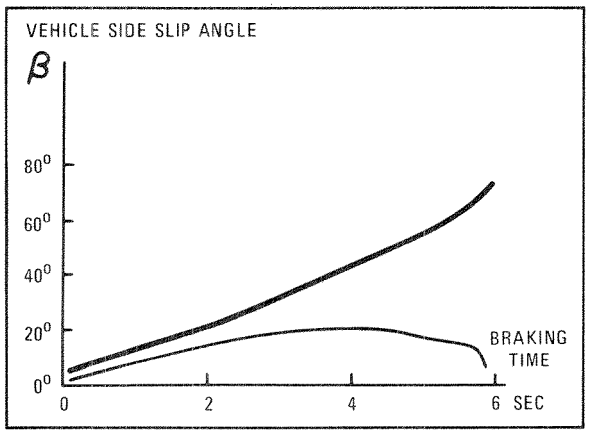


Figure 4

The results from the field tests indicate that there is a limit-value on the vehicle side slip angle, β, and when this value is exceeded, the vehicle side slip angle continues to increase resulting in increasing skidding. This limit-value is around 20°.

Practical field tests and simulations showed that the stability of a vehicle can be improved if the rotational slip percentage on the rear wheels can be lowered relative to the front wheels.

The rotational slip is the percentage difference between vehicle and wheel speed.

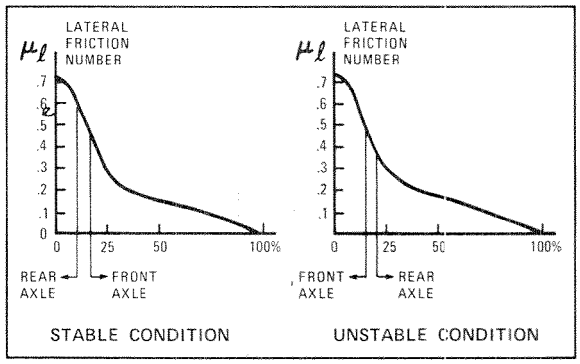


Figure 5

The figures show lateral μ-number for a certain wheel angle, where the average rotational slip value for the rear wheels in the stable condition is to the left of the value for the front wheels.

If the average value on the rear wheels is changed to a value to the right of the front wheels, it results in a less stable vehicle.

This can be shown with a histogram from a simulation. The X-axis indicates rotational slip percentage and the Y-axis indicates braking distance percentage.

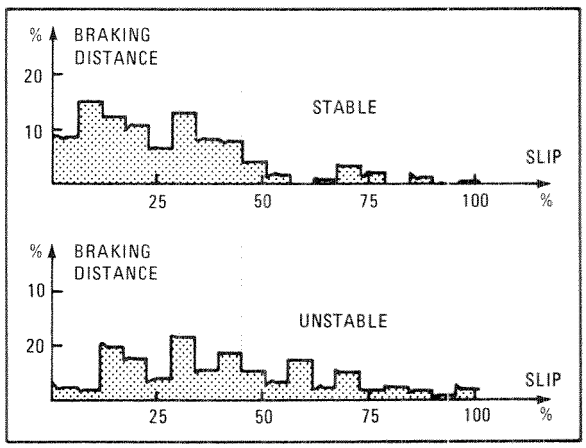


Figure 6

In the upper curve the rotational slip has been between zero and five percent under 10% of the braking distance. It can be seen that in the stable case the slip is concentrated on lower values giving stability. The slip on the front wheels is not changed.

To investigate if it was practically possible to solve this problem, some complementary tests were made during the winter of 1974 with a vehicle which, during the previous tests, had a tendency towards high instability. The anti-lock system was modified by the manufacturer on very short notice, and the vehicle

was stable when braking in a curve on ice with summer tires.

As different vehicles behave differently, the anti-lock system must be adapted to each vehicle type to attain best performance.

The vehicle side slip angle, β, can be measured with a potentiometer on a counterbalanced fifth wheel.

Steerability

Steerability is here defined as the ratio between lateral deviation from brake application to standstill and the braking distance.

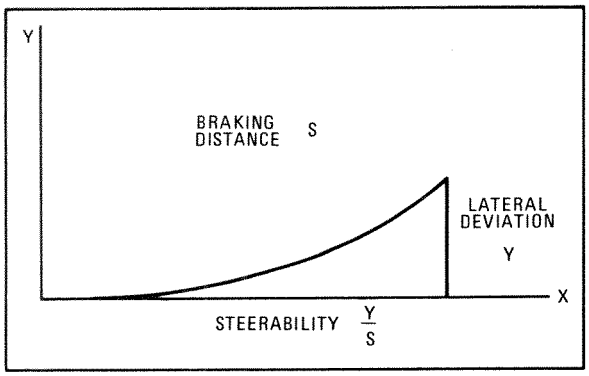


Figure 7

Both distances are easy to measure and small variations in velocity at brake application have minor influence. Performances at different tire-road conditions can also be compared.

Brakeability

Brakeability is based on the mean deceleration during braking in a curve and is calculated as

$$\bar{a}_{curve} = \frac{V_o^2}{25}$$

V_o = velocity at brake application

S = braking distance

In order to avoid measuring the tire-road characteristics with the complicated equipment, which must be used to obtain absolute values, the brakeability is here defined

$$\text{Brakeability} = \frac{\bar{a}_{curve}}{\bar{a}_{locked\ wheels}}$$

$\bar{a}_{locked\ wheels}$ = mean deceleration during braking straight ahead with locked wheels

The results from the field tests on dry asphalt indicate that there is a connection between steerability and brakeability.

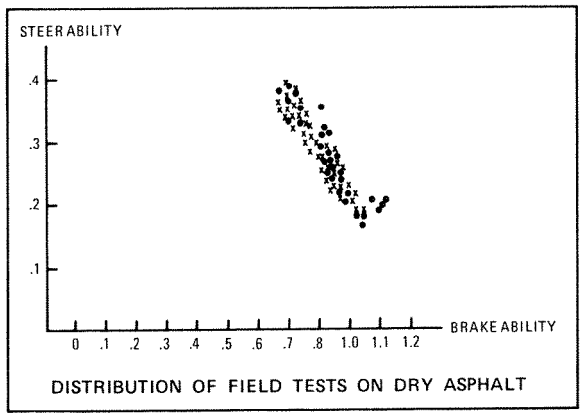


Figure 8

CONCLUSION

The results from the field tests and from the simulations indicate that for stability:

Maximum vehicle side slip angle should not exceed 20°.

The steerability and brakeability should be taken into consideration at the same time, and the performance should be within the "Recommended Area" for vehicles in the 1000-1500 kg (2200-3300 lb) class.

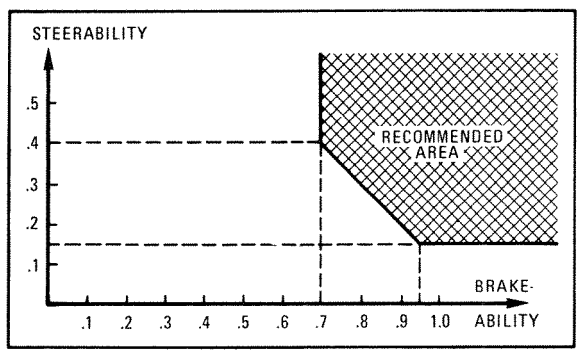


Figure 9

ICHIRO NOGUCHI
Deputy Director-General
Machinery and Information Industries Bureau
Ministry of International Trade and Industry

Presented by

NORIYOSNI UNO
Director of Engineering Division
Motor Vehicles Department
Road Transport Bureau
Ministry of Transport

It gives me a great pleasure to extend to you my greetings from Japan, together with a brief review of some aspects of Japan's traffic safety problems.

The number of motor cars now in use in Japan has reached the 25 million unit level. This means one car for every four persons, which is almost at the same level of Europe and America.

Riding on the wave of popular motorization, we were until recently quite elated over its advancement as a definite sign of a growing economy. However, as it is universally known, we are now much harassed by some of the bad effects arising from this motorization. The most serious of them all, no doubt, is the traffic accident.

The traffic fatalities of Japan reached their peak in 1971 and in the following three years a decline was noted. However, in 1973 the accidents reached 586,718 cases with 14,574 deaths. The tragic nature of a traffic accident is evident by the fact that casualties, including traffic accidents, account for the largest causes of death of a people under 34 years of age. Faced with this reality there are at least four measures which should be applied to reduce traffic accidents.

The first is the improvement of roads and road facilities. In Europe, we learn that the carriages and carts had been in use for transportation even in ancient days of Greek and Roman times. In Japan's history, however, for more than a thousand years, until 100 years ago, men had to travel on foot except the very selected few on a palanquin. Therefore, the roads in Japan were inadequate for modern motor vehicle transportation. The Japanese government in 1971 appropriated an outlay of close to 830 billion yen under a five-year plan to improve old roads and to set up traffic signals, guideboards and other safety facilities.

The second is the improvement of vehicles and the strengthening of their maintenance. Japan had adopted legal measures for these areas at the earliest date among the advanced nations. In 1951, the Road Transportation Vehicle Law was enacted. In this law technical provisions in detail related to the safety of vehicles were stipulated for the automobile manufacturers and their users. The ESV plan is, no doubt, based on a similar basic ideal. It is no wonder that Japan was ready to take an active part in the ESV project when it was first proposed by the United States Government.

The third is the educational and promotional campaign for observance of traffic safety measures by drivers and the public. In Japan, pedestrians account for 37 percent of deaths resulting from traffic accidents; of this amount, people over 60 years old and children under 15 years account for 65 percent. In this situation, the need for an educational and promotional campaign for traffic safety is keenly felt.

The fourth is the traffic control by the police. Ironically, the increase and decrease in traffic accident cases for the past ten years show a very close correlation with the traffic control by the police. This seems to indicate that a rigid control is most effective for traffic safety. With much reluctance we may have to acknowledge the fact that the best way to assure traffic safety is to strengthen the traffic control.

Needless to say, no satisfactory solution to traffic safety can be obtained if any one of the four measures is left out.

We now face a new situation wherein requirements for the vehicles become more and more severe with respect to improving fuel consumption, reducing harmful gas emission and abating noise. It is really no easy task to fully meet these requirements, while providing vehicles to the public at a fair and reasonable cost. In this respect, it is commendable that the project is being carried out to develop safety cars in a practical manner while recognizing the different traffic conditions in each country.

The Japanese government, fully acknowledging the significance of the ESV project, appropriated a

subsidy of one thousand and four hundred million yen (about 4.7 million dollars) on a three-year plan for Toyota Motor and Nissan Motor companies and Japan Automobile Research Institute Inc., while Honda Motor is building ESVs on its own initiative without a subsidy. Toyota and Nissan have both built ten units each for the final test as scheduled in the fall of 1973. Tests have been done following the ESV specifications by the Japan Automobile Research Institute. The results will be presented after my speech. Toyota and Nissan sent one unit each to the United States which are currently under test at the Dynamic Science Company, Phoenix, Arizona.

Through the development of these ESVs, the manufacturers have learned much in technical phases and it is not doubted that we can make a great contribution to the development of safety vehicles in the future. I would like to pay tribute to the builders of ESVs who did their best overcoming numerous difficulties in response to the request of the government. I would also like to take this opportunity to pay our highest respects to the United States Government for its foresightedness and initiative in proposing this significant project. At the the same time, I wish to express my deep appreciation to all personnel who devote themselves to this Conference in London.

THE TEST OF JAPANESE EXPERIMENTAL VEHICLES
(General Description)

YORSHIRO OKAMI
Japan Automobile Research Institute, Inc. (JARI)

1. INTRODUCTION

1.1 Pattern of Project

The Experimental Safety Vehicle (ESV) PROJECT in Japan dates back to the time of exchange of the memorandum between the Japanese Government (Ministry of International Trade and Industry and Ministry of Transportation) and the United States Government on the development of Experimental Safety Vehicles by the proposal of the United States Government (D.O.T.) as of 18 November, 1970.

The study and compilation works on the JAPANESE ESV SPECIFICATIONS was promptly started by the ESV Subcommittee (established within the Japan Automobile Manufacturers Association [JAMA] comprising the members of the JAMA and our Institute and the ESV SPECIFICATIONS which was based upon the specifications of the 4,000 lb class ESV sanctioned by the Government of United States and modified to 2,000 lb class in consideration of the Japanese conditions were finalized in May, 1971. The Japanese Government authorized these SPECIFICATIONS as the JAPANESE ESV SPECIFICATIONS and after the immediate inviting participation of the manufacturers in Japan to this PROJECT, the Government designated TOYOTA Motor Company Ltd. and NISSAN Motor Company, Ltd. as the formal participant enterprises and Honda Motor Company, Ltd. as the semi-formal participant enterprise. Since that time in respective enterprise, research and development works as well as manufacturing operations of the prototypes have been performed. Ten prototype ESVs each have been delivered through the Government to our Institute from NISSAN and TOYOTA between September and the end of November, 1973. Our Institute was assigned to perform the tests of the ESV, the final stage of the ESV PROJECT in Japan.

The Governmental request for performance of tests of the ESVs was issued to our Institute, as of 31 July, 1971. Our Institute received from the Government of Japan (Ministry of International Trade and Industry) allocation of subsidy totalling to about 696,700,000 yen (2,497,000 US dollars), (1,028,000 pounds) in 1971, 1972 and 1973 and such testing facilities as equipment, apparatuses measuring instruments, etc. have been reinforced. Since the fall of 1971, beginning with the preparatory tests, our Institute has been discharging our duty to this day.

1.2 Objectives of ESV Tests

The objectives of the ESV tests by the Japan Automobile Research Institute, Inc. (JARI) are to supply such informations that contribute to evaluation of the ESVs manufactured by TOYOTA and NISSAN and to promotion of automotive safety in the future by testing them by the JARI, an independent and neutral organization.

More specifically, the JARI is entrusted to observe the following two responsibilities:

(1) Execution of accident avoiding performance tests

(2) Execution of collision and roll-over tests

In this connection we should take the following into consideration:

We have conducted the performance test of both ESVs of TOYOTA and NISSAN at the request of the Government.

The evaluation of the test results is not described in this report.

1.3 Construction of ESV Test Reports

The ESV Test Reports are divisible into the Summary Reports and the Final Detailed Reports.

The present Summary Reports are composed of the following parts:

General Descriptions
Summary of TOYOTA and NISSAN ESV Tests

We shall submit the Final Detailed Reports containing full data to Japanese Government at a later date.

2. BRIEF DESCRIPTION OF ESV TESTS

2.1 ESV Test Project

We conducted the tests of ESVs in the compound of the JARI for the period of approximately seven months from September, 1973. The ESVs supplied for the present tests were manufactured by TOYOTA Motor Company Ltd. and NISSAN Motor Company Ltd., both being the formal participant enterprises in the Japanese ESV Project, including two different kinds of vehicles.

We conducted the destruction and non-destruction tests totalling to approximately 1,500 tests as represented in Table 1 on eight NISSAN and seven TOYOTA ESVs.

Tab. 1 Japanese ESV Test Program Scope

	NISSAN ESV	TOYOTA ESV
	TESTING VEHICLE	TESTING VEHICLE
Accident Avoidance Tests		
1. Visibility	NESV-1,4,6	TESV-4
2. Display and Control	NESV-1,6	TESV-1,4
3. Braking Performance	NESV-1	TESV-1
4. Steering	NESV-5	TESV-1
5. Handling	NESV-5	TESV-1
6. Overturning Immunity	NESV-1,5	TESV-1,10
7. Engine	NESV-1,	TESV-1,
8. Bumper Under ride/Over ride	NESV-1,	TESV-1,
Collision and Rollover Tests	NESV-1,6,7,8,9,10	TESV-6,7,8,9,10
TOTAL	8 VEHICLES	7 VEHICLES

2.2 ESV Test Methods

We conducted the ESV tests by the methods and conditions prescribed in the Japanese ESV SPECIFICATIONS (as of 21 May, 1971, Japan Automobile Manufacturers Association)

(However, test conditions and test methods for NISSAN ESV were partly modified as shown in Table 2)

Tab. 2 Distinction of test conditions and methods between both ESVs

ITEMS		TOYOTA ESV	NISSAN ESV
Visibility	Requirement for eyellipse for testing of forward vertical viewing angles	99% tile	95% tile
Collision	(1) test item : 80 km/h pole barrier impact (preferable)	yes	no
	(2) test condition : impact speed in rear-end impact	80 km/h	65 km/h

2.3 Testing Equipment and Apparatuses

Outlined specifications and distinctive features of the equipment and apparatuses employed for testing of the ESVs are as follows:

2.3.1 Series of Accident Avoidance – Visibility, Diaplay and Control – Tests

In the case of a series of the visibility tests, we employed a visibility test room (for test of front field of view, rear field of view, headlights, signal lights, windshield washers), wind generator (for test of wipers), low temperature chamber (for test of defrosters) and outdoor running track (for test of controlling devices).

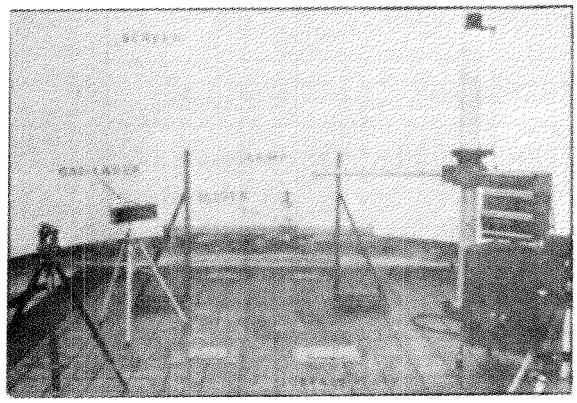


Fig. 1 "Field-of-View" Measuring Device

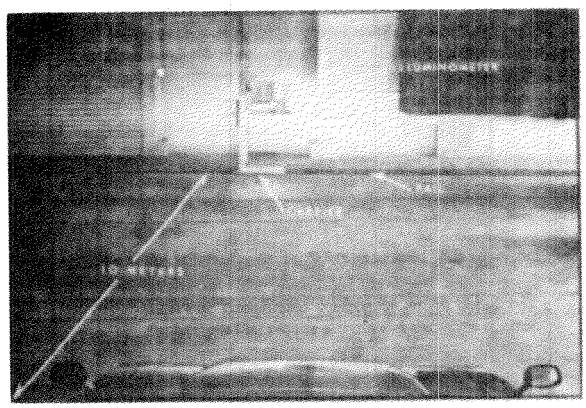


Fig. 2 Headlight Measuring Device

For the measurements of the field of view, we anchored a test vehicle on a surface plate and employed a variety of auxiliary jigs as illustrated in Fig. 1. The cylindrical screen is provided with a radius of 3 m, height of 3.3 m and effective surface of 180° in horizontal direction.

The 3-dimensional measuring device was used for measurements of field of view and in addition to them, it was used for measurements of body dimensions and of impact test vehicles.

Fig. 2 represents a headlight measuring device. The photometer slide is available for displacements between 3.5 U - 3.5 D and 30 R -.30 L, when it is used for the measuring range of 10 m.

For the wiper tests, we used the crosswind generator as illustrated in Fig. 6 and the test vehicle was located accurately in alignment with the wind.

The low-temperature chamber is a property of the Technical Research Laboratory of the Machine Promotion Association, Ltd. and it is provided with the overall dimensions of 8 m wide, 15 m long, 5 m high and capacity of -40°C ~ +50°C (±2.0°C)

For conducting the defroster tests, we erected a tent as well as a net for controlling the wind pressure inside of which a test vehicle was anchored.

2.3.2 Series of Accident Avoidance – Braking Performance, Steering, Handling, Overturning Immunity – Tests

1. Test Courses:

The features of various test courses employed for the tests and test items are summarized in Table 3 and skid resistance numbers of the surfaces of test courses in Table 4.

Tab. 3 JARI Accident Avoidance Test Facilities

TEST COURSE		TEST ITEMS
High Speed Track	Peripherial Length: 5500m Straight Course Width: 12m Radius of Banked Section: 400m Maximum Banking Inclination: 48° Maximum Velocity: 190km/h Paved with Concrete	Stopping Distance Brake Effectiveness Fading Braking Efficiency Crosswind Sensitivity
ESV Testing Ground	Area: 72200m^2 Available for Driving in from the High Speed Test Track at a High Velocity. Available for Circular Movements of 30R and 70R and for High Speed Running at 80km/h and 110km/h Available for a Steady Circular Running with 0.4G for 10sec. Paved with Asphalt.	Steady State Yaw Response Transient Yaw Response Returnability Steering Control Sensitivity Lateral Acceleration (Dry) Control at Breakaway J-Turn
Multi-purpose Test Tract	A 1000m Long Straight Course with a Section of 500m Length and 30m Width and a Section of 500m Length and 45 m Width. Paved with Asphalt.	Pavement Irregularity Sensitivity Slalom Bumper Under ride/ Over ride
Slippery Test Track	Overall Length: 450M Overall Width: 30m Width at the Center: 20m (Paved with Polished Concrete)	Braking Efficiency
Skid Pad	Radius: 50m Paved with Blown Asphalt Water Sprinkler is Installed at the Center	Lateral Acceleration (Wet)
Test Slope	Length of Gradient: 10m Width: 4m Gradient: 31% Paved with Concrete	Parking Brake
Crosswind Generator Equipment	Available for Three Steps of Wind Velocity (15m/sec, 22m/sec, 30m/sec) Five Units Are Installed.	Crosswind Sensitivity (Available Use for the Windshield Wiper Test)

Tab. 4 Skid Resistance Number of JARI Accident Avoidance Test Facilities

TABLE OF SKID RESISTANCE NUMBER* OF INDIVIDUAL TEST COURSE			
TEST COURSE	SURFACE CONDITION	SKID RESISTANCE NUMBER* **	REMARK
High Speed Track (Paved with Concrete)	DRY	78.8	* Conform to ASTM Standards ** Showing a Value Converted at the Road surface Temperature of 20℃ The Convertion Formula: $SN_{20}=SN_T-k(T-20)$ where T=Road Surface Temperature (℃) SN_T=Value at Road Surface Temperature T℃ k=Constant for Each Surface *** Value at a Velocity of 40km/h
	WET	57.5	
ESV Testing Ground (Paved with Asphalt)	DRY	78.5	
	WET	64.4	
Multpurpose Test Track (Paved with Asphalt)	DRY	75.5	
	WET	52.7	
Slippery Test Track (Paved with Polished Concrete)	WET	49.5	
Skid Pad (Paved with Asphalt)	WET	33.2***	

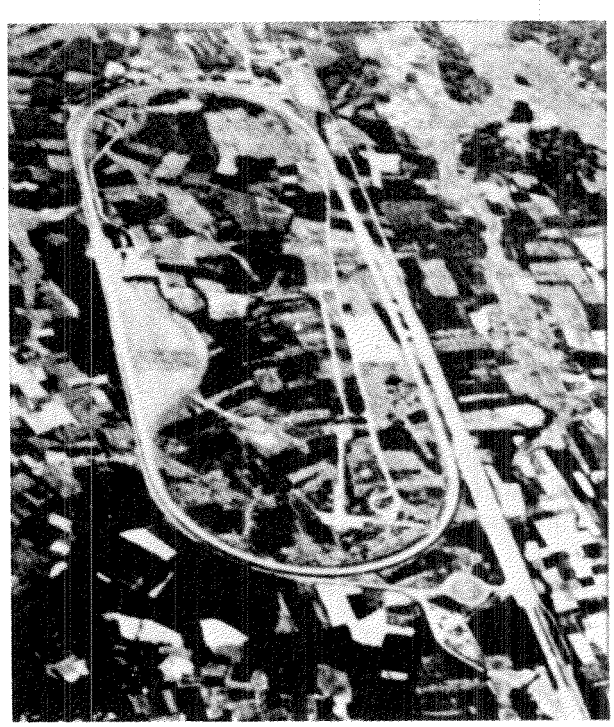


Fig. 3 Birds-eye View of JARI Test Course

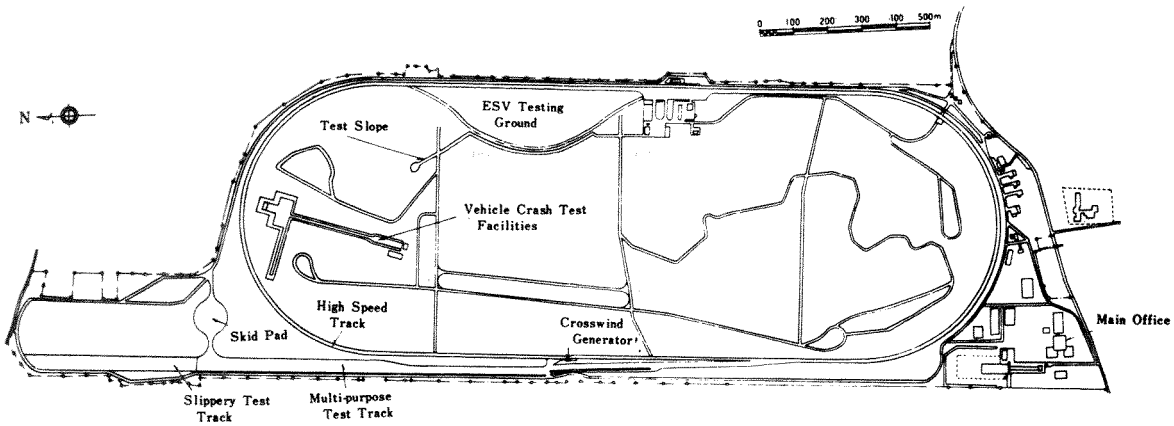


Fig. 4 Layout of JARI Test Course

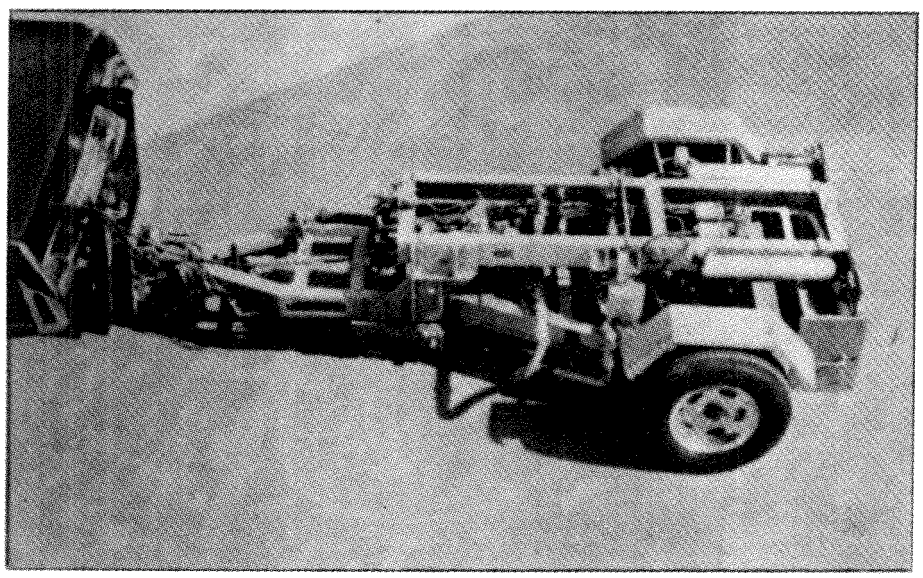


Fig. 5 Trailer-Type Test Vehicle for Measuring Tire Characteristics and Skid Resistance Number (SN)

2. Automatic Driving System for J-turn Tests

In consideration of safety and reproducibility in the J-turn test, we developed by the JARI's design an automatic driving system[1] and conducted this particular test. The automatic driving system is refered to a system comprising an automatic steering apparatus which operates a test vehicle along a guidance cable embedded along the test course, as operating its braking apparatus by the telemeter command, velocity controlling apparatus for driving the vehicle at a predetermined velocity and an emergency braking apparatus. The location detecting apparatus for starting the J-turn produces a J-turn trigger by combination of a ground element and on-board-vehicle element.

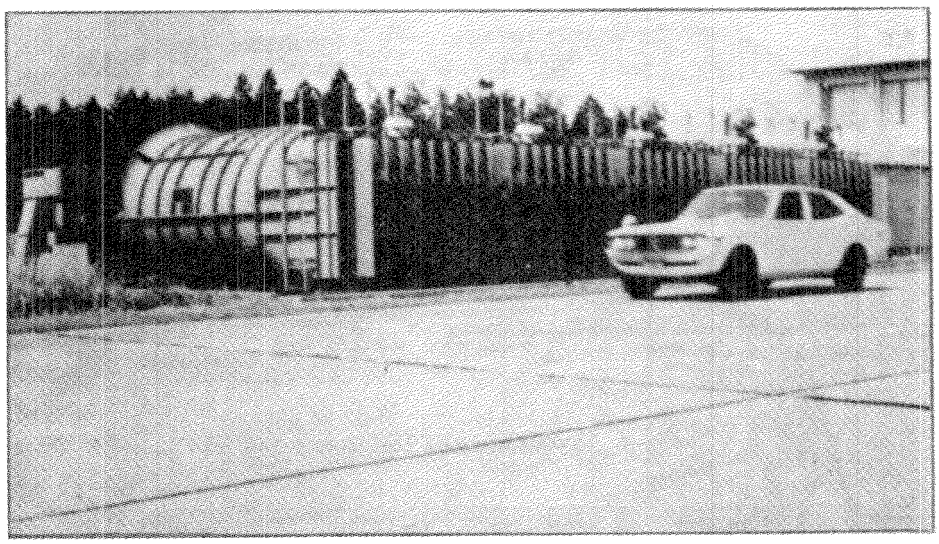


Fig. 6 Crosswind Generator Equipment

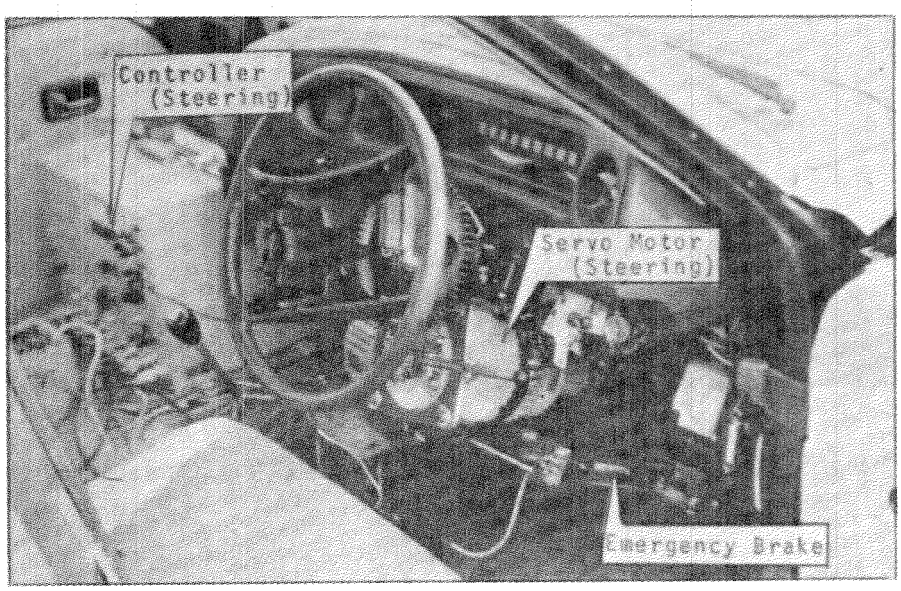


Fig. 7 Installations of Automatic Driving System for J-Turn Test

The J-turn tests by employment of the automatic driving system turned out to be excellent with the remarkable reproducibility as shown in Fig. 10.

NOTE (1) References are made to the following papers

1) "An automatic driving system of automobile by guidance cable," by Mr. T. Ito, SAE Paper 730127
2) "Application of automatic driving system to automobile running tests," by Mr. M. Furumata, Advanced paper of JSAE, Lecture meeting No. 732, 487 p ~ 492 p.

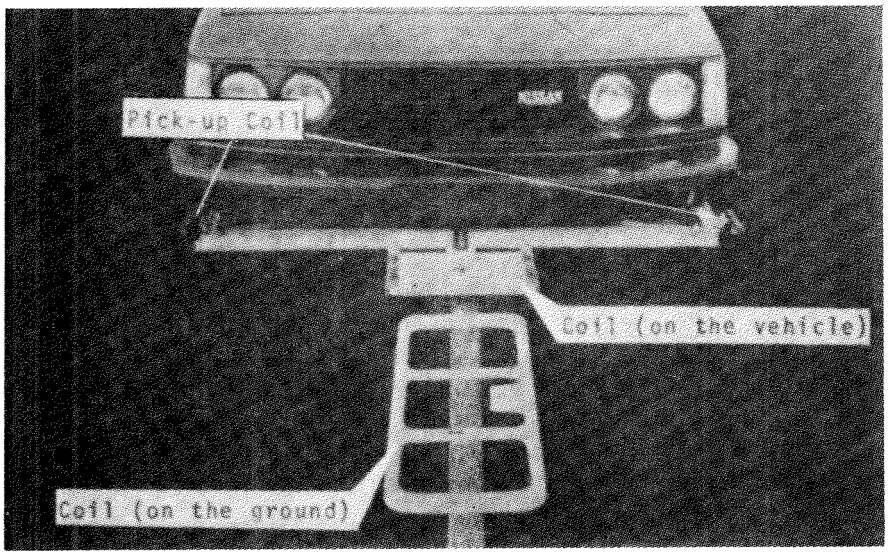


Fig. 8 Automatic Driving System for J-Turn Test (Induction Coil)

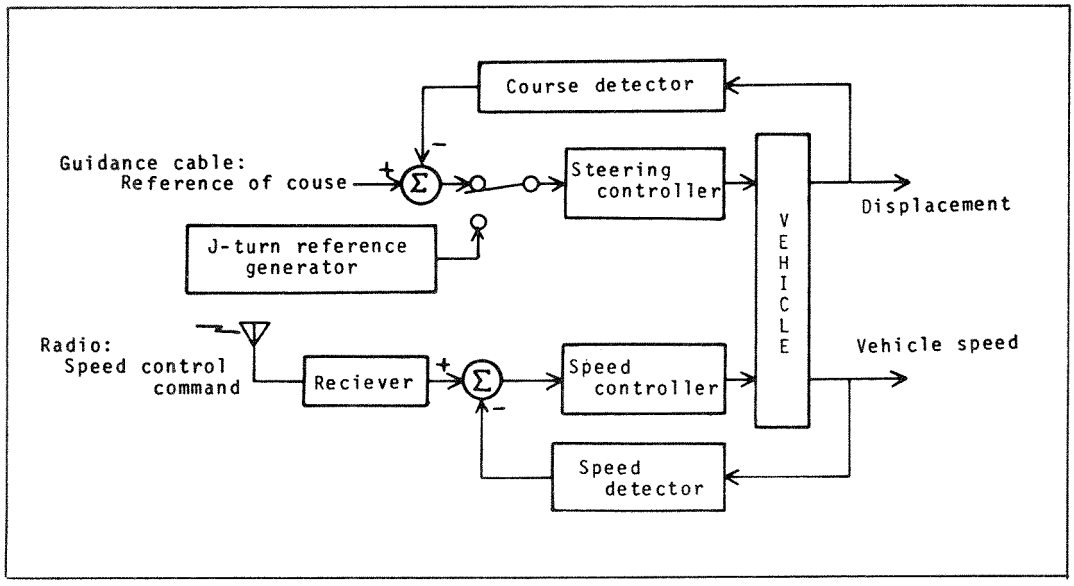


Fig. 9 Schematic of Automatic Driving System for J-Turn Test

3. Data Recording and Data Reduction System

With procedures as represented in Fig. 11, by means of the measuring apparatus and data recorders installed on the vehicle, all the measurements were recorded. Only tapes were brought back to the laboratory and they were processed by the data reduction system as shown in Fig. 12 and the required information was brought as the outputs through the X-Y plotter and line printer. An example of processed data is shown in Fig. 13.

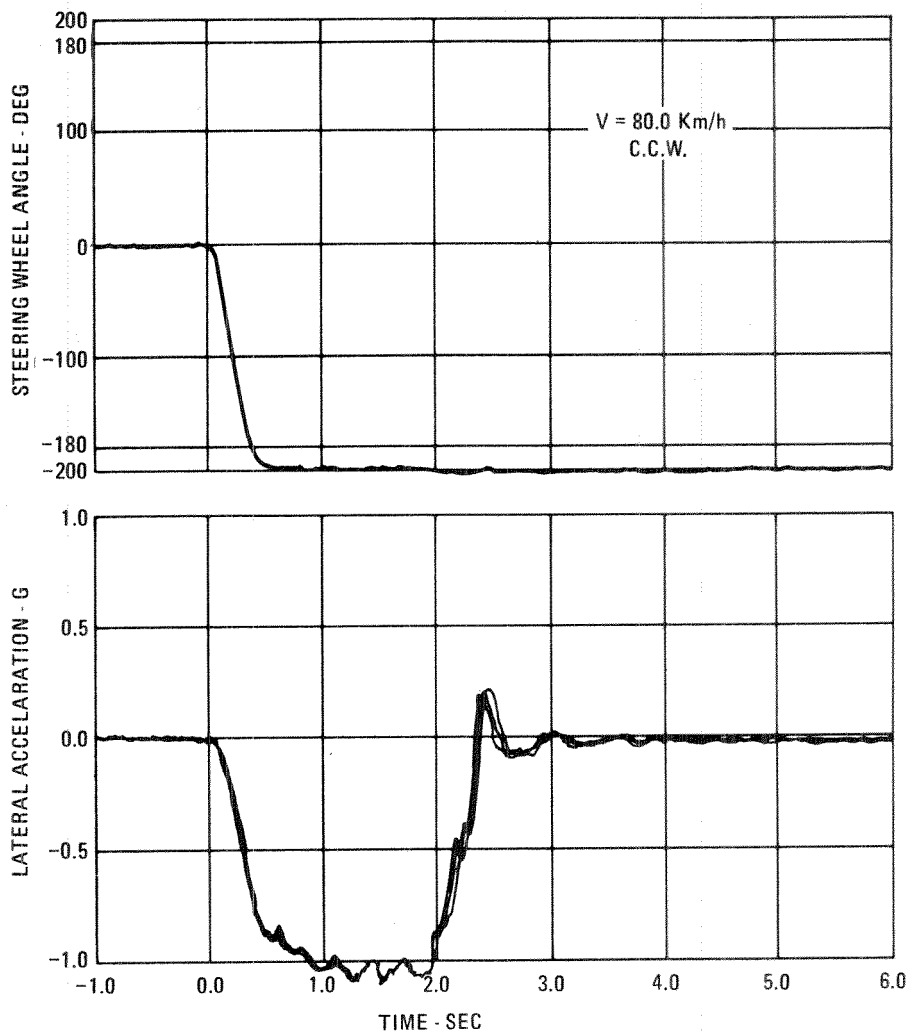


Fig. 10 An Example of J-Turn Results (Three Tests were Performed.)

4. Bumper Under ride/Over ride Measuring System

We measured the height of bumper by applying a new method. (Ref. Fig. 14). This particular method employs a laser projector equipped on the bumper in order that the He-Ne laser ray is projected to the road surface at an angle of 45° permitting the ray illuminated spot on the ground to sway to left and right side following the vehicle movements of bumper. The spot and a target anchored on board the vehicle are shot with a 35 mm motor driven still camera and the height of bumper is obtained by measuring the relative distance between the spot and the reference target.

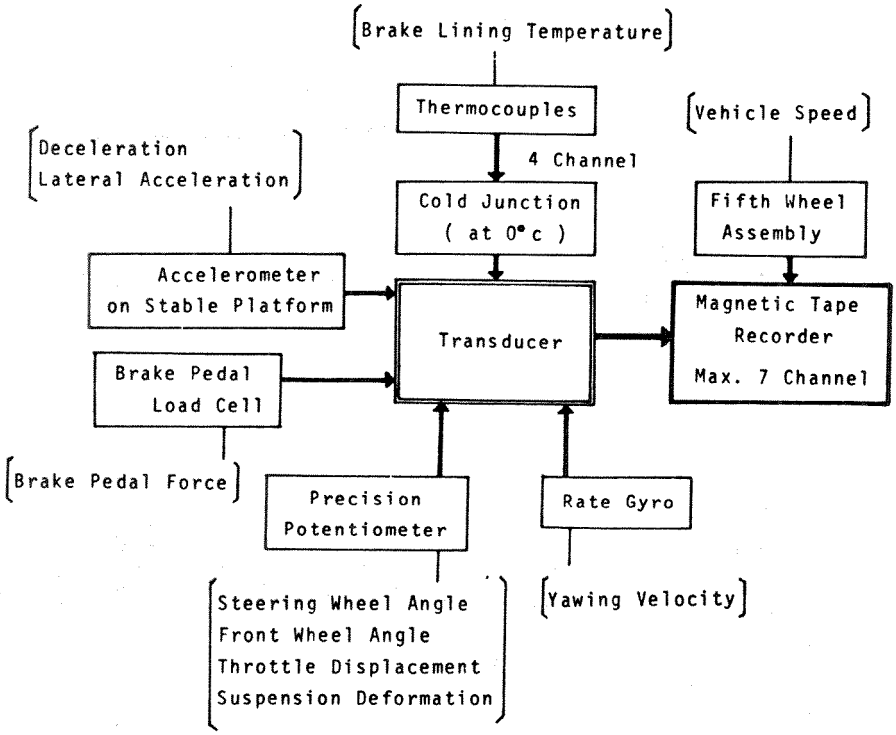


Fig. 11 Schematic of Accident Avoidance Data Acquisition System

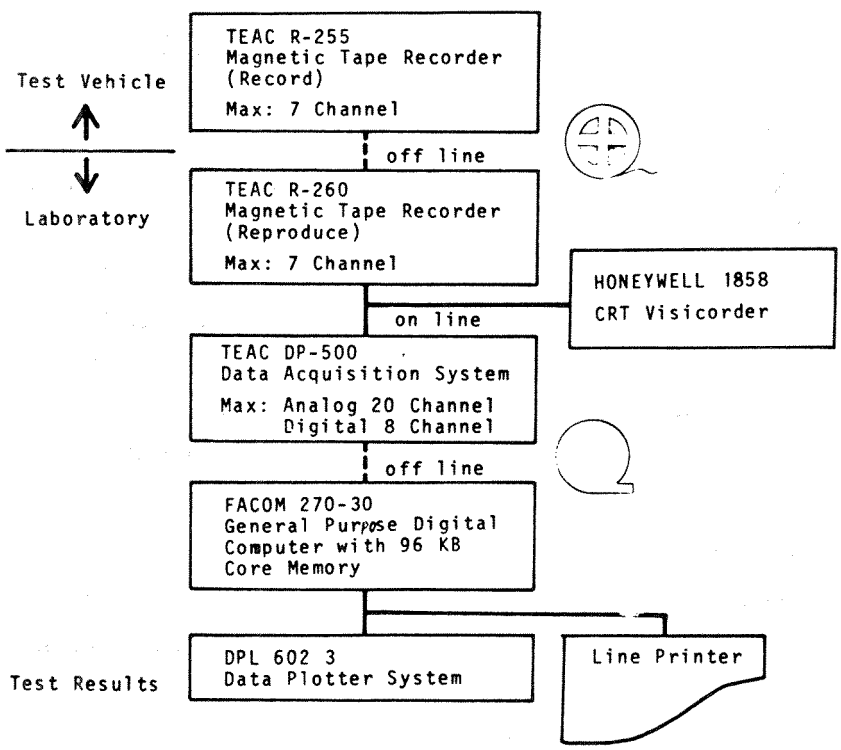


Fig. 12 Schematic of Accident Avoidance Data Reduction System

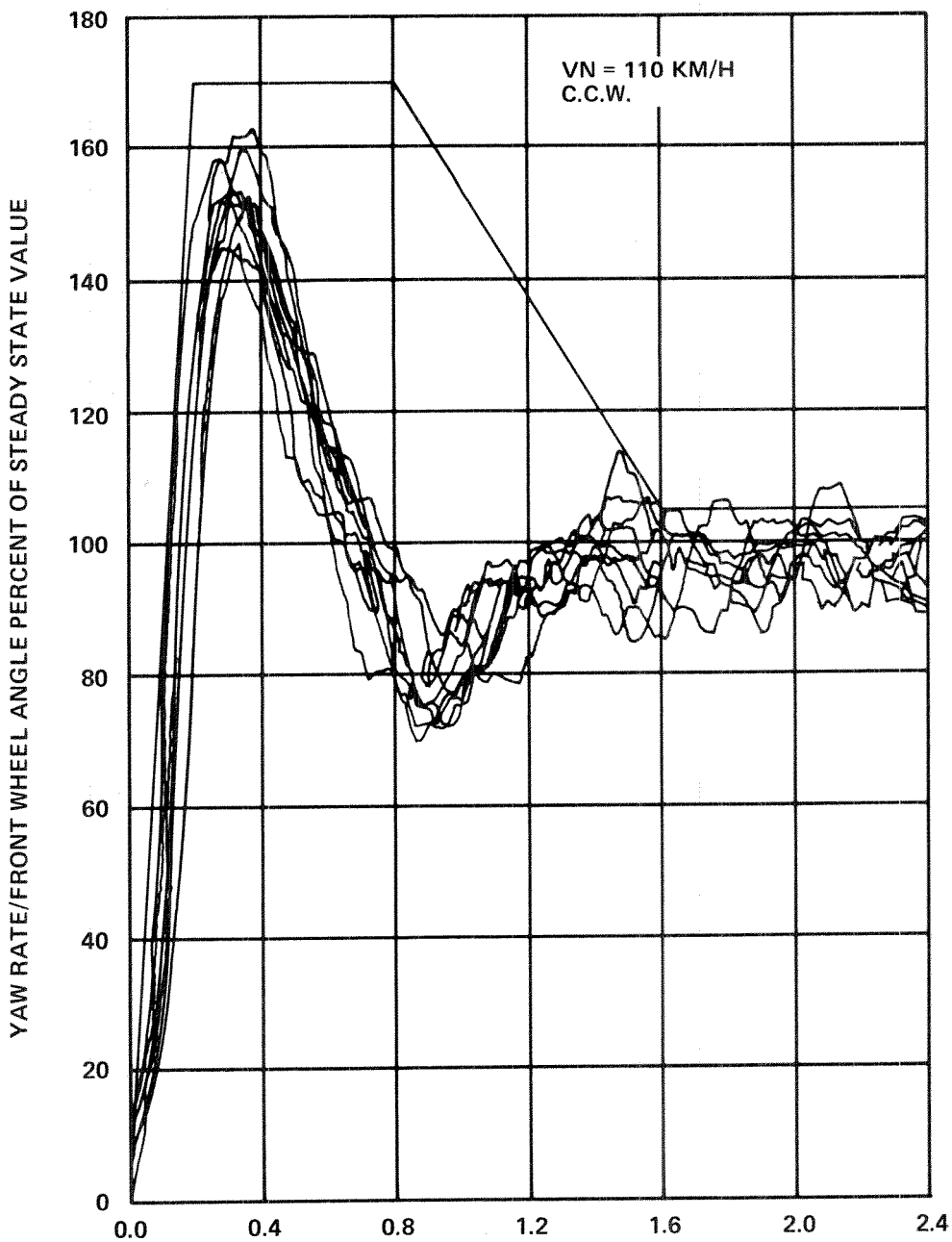


Fig. 13 An Example of Data Reduction
(Ten Transient Yaw Response Tests Performed.)

2.3.3. Collision and Rollover Tests

The crash injury reduction testing installations are divisible into the vehicle crash test course facilities and data acquisition and reduction system and the details of the installations are broken down as follows:

1. **Vehicle Crash Test Facilities**

Taking consideration of availability of highspeed photographic recording, the main tow road was constructed intersecting a northward direction through an angle

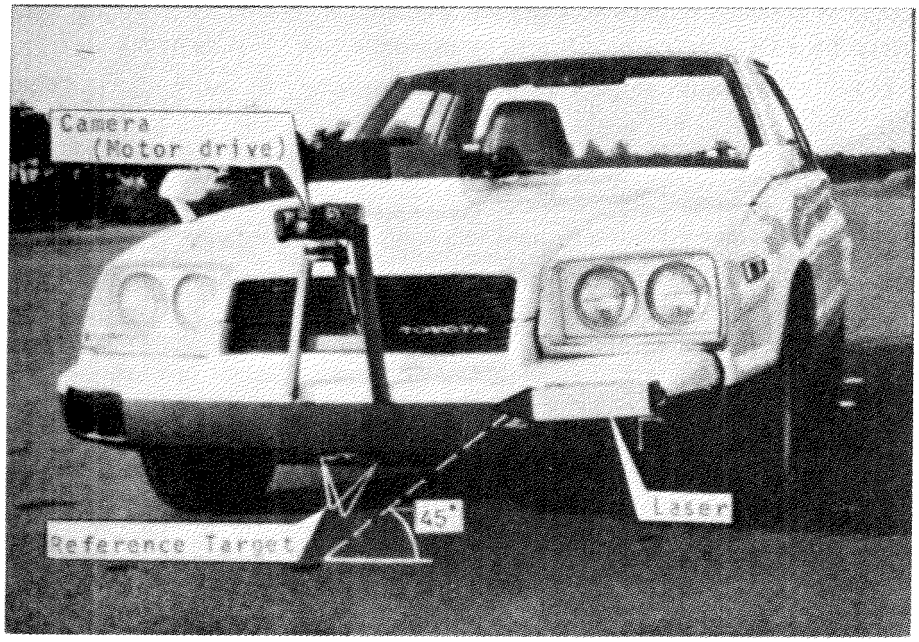


Fig. 14 Measuring System of Bumper Height

of 15° to the east. The testing course and laboratories are located as shown in the plot plan.

The outlined specifications of the vehicle crash test course facilities are given in the following:

1.1 Specifications of Vehicle Crash Test Facilities

(1) Main track 5 m wide, 350 m long, a single track with asphalt pavement (SN:78)

(2) Auxiliary track 5 m wide, 200 m long, a single track with asphalt pavement (SN: 78)

(3) Impact area: 32.5 m wide, 100 m long, a single area with asphalt pavement (SN: 78)

(4) Fixed barrier: 4.5 m wide, 2.5 m high, 5.0 thick weighing 250 ton

(5) Force transducer with a flat barrier measuring 360 ton and a pole barrier measuring 400 ton

(6) Photographic pit: 1 m wide, 6 m long, 2 m deep with a lighting system with 20 sets of 1.5 kW Halogen lamps

(7) Overhead crossing bridge: 2 m wide, 4 m long, 6.5 m high

(8) Buildings: Control room, power plant room and preparation room

1.2 Power Plant and Impact Velocity Control System

The power plant is of a double drum type as shown in Fig. 16 which is directly driven by a DC electric motor and incorporated an abort system.

(1) Maximum impact velocity: 36 m/sec (130 km/h) for a vehicle weight of 2,800 kg
21 m/sec (75 km/h) for a vehicle weight of 10,00 kg

(2) Accuracy of predetermined velocity: For 36 m/sec: ±1%
4.5 m/sec: ±3%

(3) Accelerating distance: 200m (with vehicle weight of 2,800 kg and velocity of 36 m/sec)

(4) Velocity adjustment and abort distance: 150 m

(5) Accelerating control method: Program control stepwide operating type

(6) Maximum driving force: 1125 kg

(7) Maximum braking force: 2800 kg for 1 G (In case of abort)

(8) Winch

(a) Type: Directly coupled type with a DC electric motor (Thyristor Leonard type)

(b) Driving output: 1,100 kW

(c) Driving torque: 1,550 kg·m

(d) Braking torque: 5,150 kg·m

(e) Drum: Feeding-out drum: 1 piece
Drawing-in drum: 1 piece
with a diameter of 1 m and a length of 2 m respectively

(9) Wire rope: Diameter: 12.5 mm
Length: 1,650 m

(10) Velocity control: By voltage control with a potentiometer

(11) Velocity sensing and indication:
Revolution of drum → Tacho-generator → Speedometer
Revolution of drum → Pulse generator → Digital counter → Digital printer

(12) Velocity control method:
Revolution of drum → Tacho-generator → Analogue circuit feedback

1.3 Tow system

(1) Towing and releasing method of test vehicle: A test vehicle is towed and released with a towing dolly.

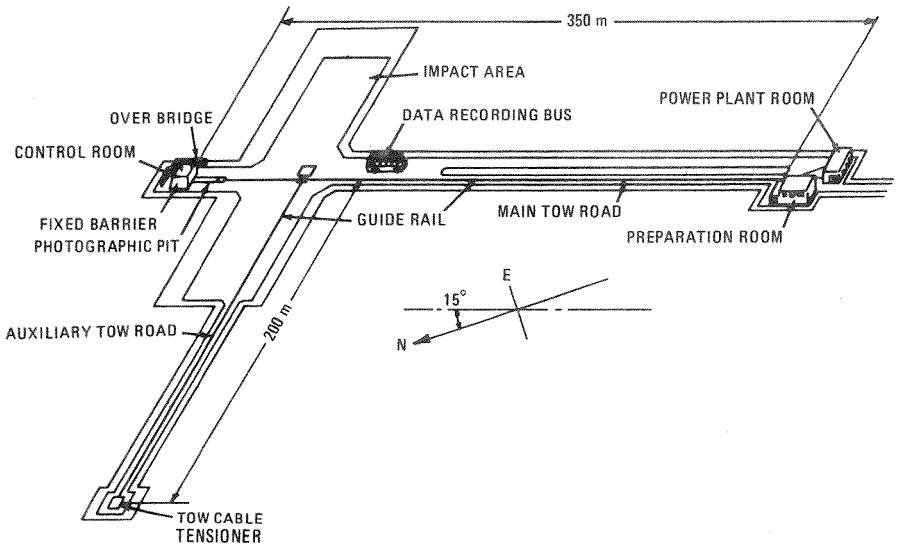


Fig. 15

Fig. 15 Vehicle Crash Test Facilities

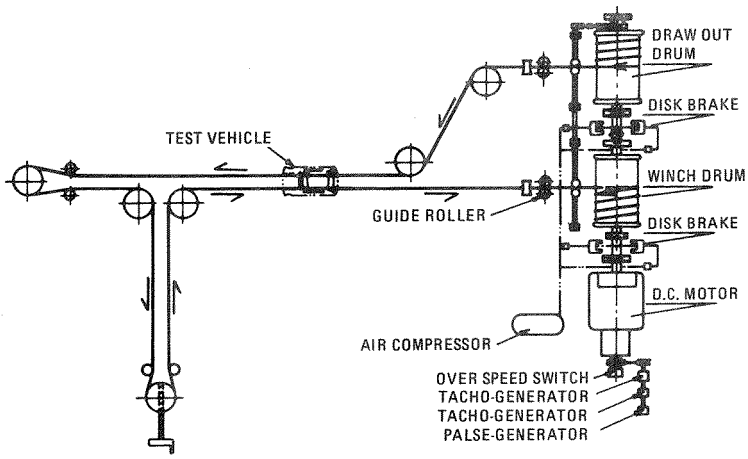


Fig. 16

Fig. 16 Vehicle Crash Test Facility (Tow System)

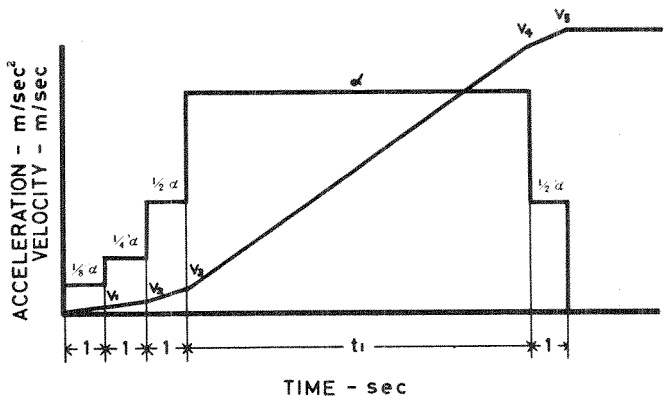
(2) Stopping method of dolly: A decelerating apparatus with a disc brake is employed.

(3) Safety device: This devise is composed of a mechanical locking unit, emergency stop switch, ready-to-start switch with key, interlocking devices and over-speed switch

(4) Abort method: Disconnecting by push-button system and external signal (DC electric motor generating and braking system with a pair of disc brake)

1.4 Accelerating method:

The accelerating method is of a 5-step start type as shown in Fig. 17, taking consideration of effects on the testing device (dummy) inslalled on board the test vehicle.



Symbols in Fig. 17 are

- α Towing acceleration
- t_1 Length of time during which the vehicle is towed with a full acceleration (sec)
- v_5 Target vehicle velocity

Fig. 17 Acceleration Mode of Collision Test Vehicles

2. Collision Test Data Acquisition and Reduction System

As the multi-channel experiments (ranging 50 to 100 channels), on-board and shock-durable electronic circuitry (25-54 kg) is adopted. On-board power is supplied by the remote cables from the ground station to reduce instruments' weight. Data signals are also transferred through the cables as channel-multiplexed and digitized signals called PCM (pulse code modulation). To ensure the reliability of the acquisition line, the double measurement system is adopted for the important part, for example dummy head.

Data reduction system, which consists of digital computing circuitry contains digital filters specified in SAE J 211a and other "optimum signal processing" softwares.

Brief specifications:

	Signal conditioners	PCM modulation
Capacity	64-96 ch.	80-120 ch.
Response	2,000 Hz	1,600, 800, 400 Hz programmable
Accuracy	S/N 45 dB	8 bits (±0.4%)

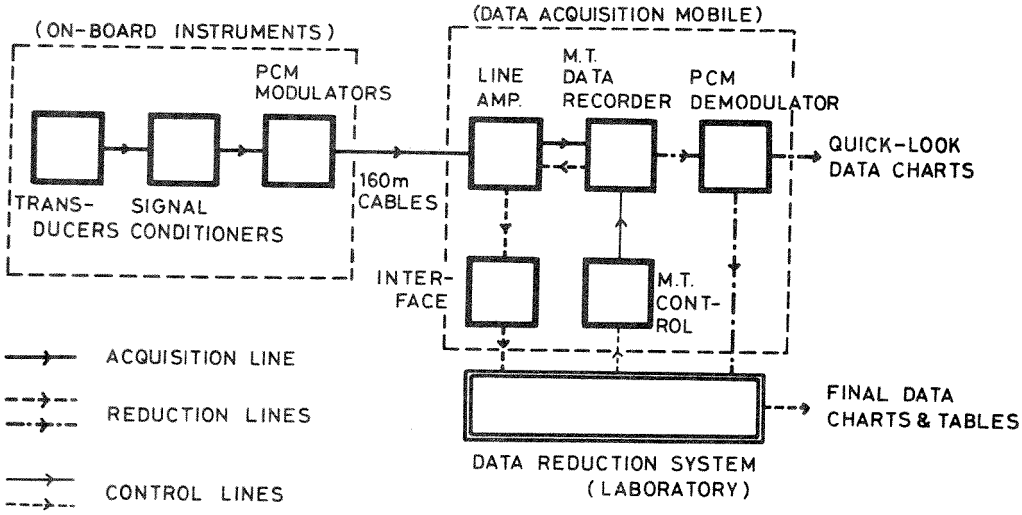


Fig. 18 Schematic of Collision Test Data Acquisition and Reduction System

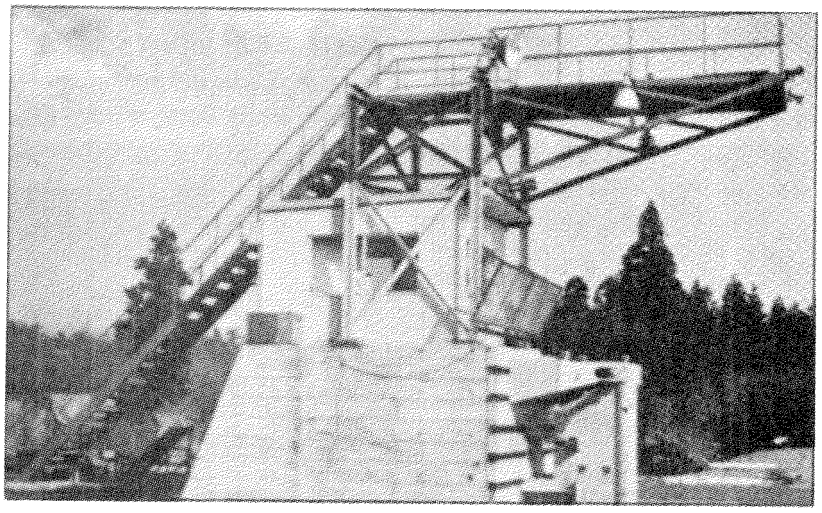


Fig. 19 Vehicle Crash Test Facility
(Fixed Barrier, Control Room and Over-Bridge)

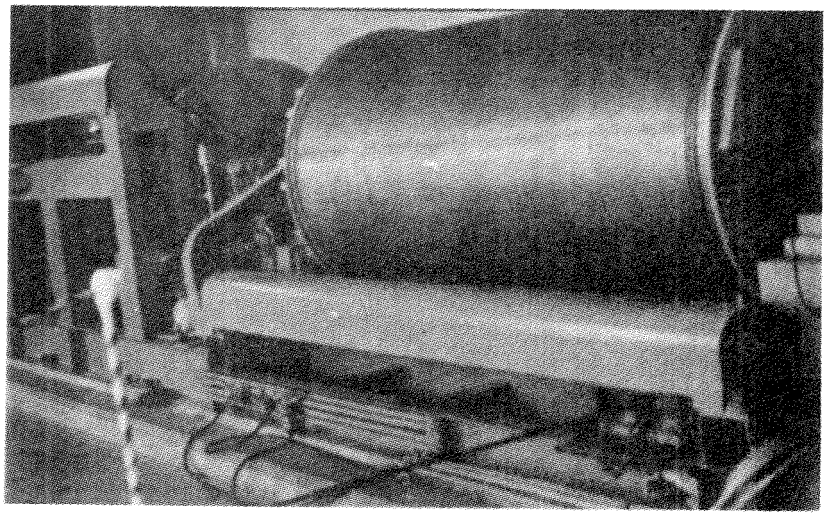


Fig. 20 Vehicle Crash Test Facility (Power Plant)

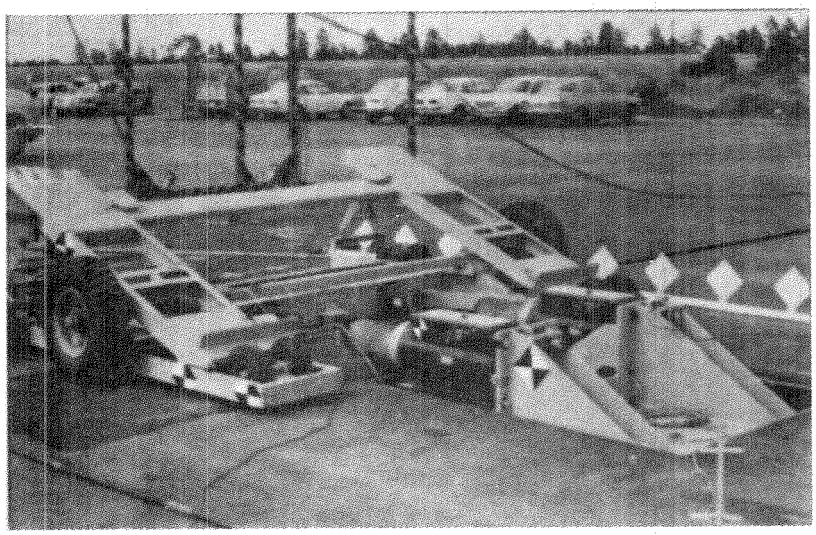


Fig. 21 Vehicle Crash Test Course Facility
(Dolly for Roll-over Test and Stopper)

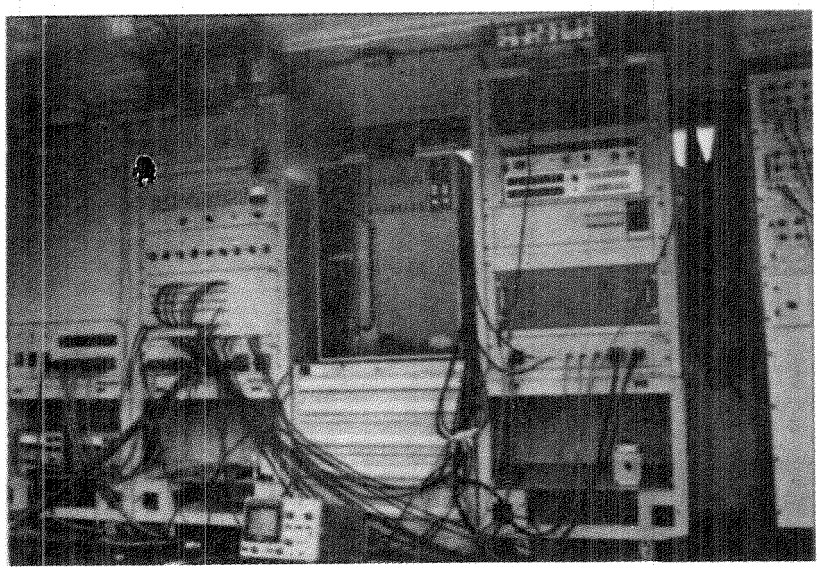


Fig. 22 Collision Test Date Acquisition System

3. CONCLUSION

The JARI has successfully completed as scheduled all the tests of the Japanese ESVs entrusted by the Government of Japan in accordance with the required items and the test results are as presented in other parts of this summary report.

We believe that these data are worthwhile for evaluation of the ESVs and at the same time that, they represent a corner stone upon which the automotive safety projects in the future of the Govenment and Industry will be established.

The ESV the tests of which we have been endeavoring is, when more precisely defined, the Technical Experimental Safety Vehicle and the strongest stress was laid on the pursuit of the optimum technical possibilities.

We are convinced that the second step should be taken in which the research and development works of a so-called Social Experimental Safety Vehicle are performed taking consideration of the social adaptability.

We interprete the Research Safety Vehicle proposed by U.S. Government (D.O.T.) as a vehicle which falls into this category and consequently, we expect the result of the research.

It is our sincere prayer that as the final goal, the Social Safety Vehicle may be brought to existence, accepted by the public and used extensively all over the world.

In conclusion, the JARI wish to express the heartfelt appreciation to the Government of Japan (Ministry of International Trade and Industry, Ministry of Transportation and Government Agencies responsible for research works) and the ESV manufacturers for their aidful assistances extended to the JARI in carrying out the ESV tests.

GENERAL REMARKS ON THE SPECIFICATIONS FOR JAPAN ESV

presented at

SECOND INTERNATIONAL TECHNICAL CONFERENCE ON EXPERIMEAL SAFETY VEHICLES

by

Mr. Yoshiro Okami

Japan Automobile Research Institute, Inc. (JARI)

To comply with the Japanese Government's request for drawing the Specifications for a Japanese Experimental Safety Vehicle, the ESV Subcommittee was established within Japan Automobile Manufacturers Association, Inc. (JAMA) in November, 1970. The Subcommittee, comprising of technical experts from the Government, subordinate manufacturers and Japan Automobile Research Institute, Inc., started operation on the basis of the U.S. specifications for the 4,000 lb. ESV, and completed an original draft in January, 1971.

Considration was later paid to the criticism of the U.S. National Highway Traffic Safety Administration, which had been levelled against the Japanee specifications. The Subcommittee decided on the final specifications in May, and JAMA submitted them to the Government.

Our specifications will, in general, meet the requirements, described in the U.S. Specifications. However, some differences will be noted because our ESV is a smaller type and traffic regulations in Japan are not the same as those in the U.S.A.

I will explain, from the Japanese point of view, the main differences.

1. The specifications for the Japanese ESV were drawn to enable them to also meet the Safety Regulations for Vehicles for Road Transportation (JAPAN). One example is that the requirement for an outside rearview mirror was added.

2. The automobile engineers in Japan have made every possible effort towards the reduction of curb weight, and we have considered it as an indication of technical progress. However, there arose the fundamental necessity of reinvestigating the weight of vehicles. The reason is that passengers should be protected in the collision to a fixed flat barrier at the impact speed of 80 kilometers per hour, and such requirement will lead to an increase in curb weight. It is doubtful whether the increase in curb weight will be able to be compensated by a decrease in the weight of other components, but this will doubtless be proved through the achievement of subsequent technical developments. In the first stage of the programme, we consider that curb weight is one of the subjects for the research into crashworthiness.

The Graph in Figure 1 shows a conceptual diagram of the relation between curb weight and the characteristics of crashworthiness with the number of occupants as a parameter. The following two ideas can be drawn from this:

1. How much will the minimum weight of ESV be for four passengers?
2. Under what condition will it be possible to make a 900 kilogram-weight ESV?

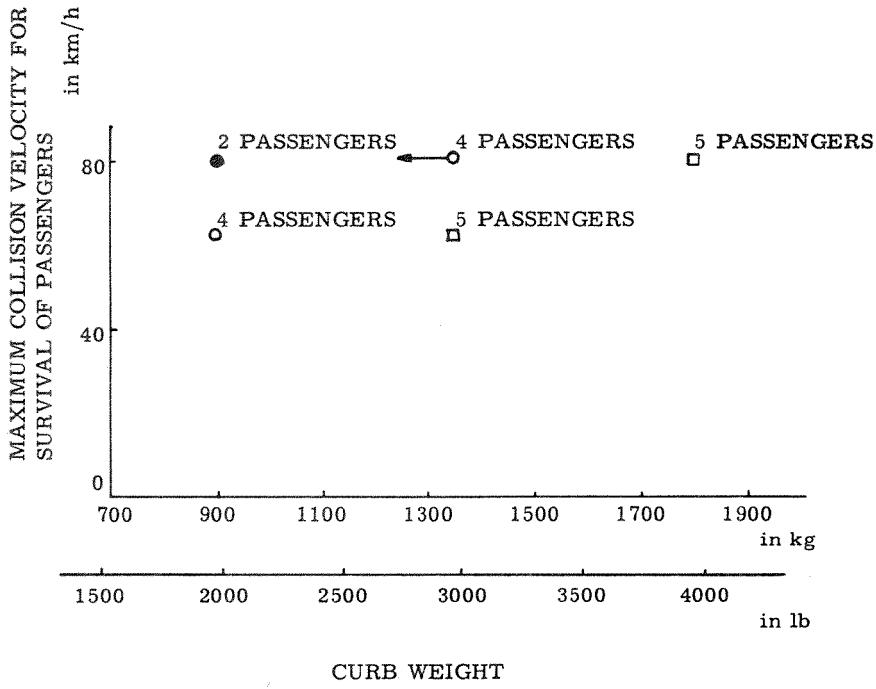


Fig. 1 — THE CONCEPTUAL DIAGRAM OF THE RELATION BETWEEN CURB WEIGHT, NUMBER OF OCCUPANTS AND CRASHWORTHINESS

We thought that in the present stage we should not conclude which type would be more pertinent to the future traffic system. We therefore adopted two specifications for the design based on the curb weight and the number of passengers (Fig. 2). The choice will be left to the participating manufacturers. Through the research into crashworthiness on the vehicles of

ITEM	SPECIFICATION	
NUMBER OF PASSENGERS	4 OR	2
CURB WEIGHT	1150 kg (2530 lbs)	900 kg (1980 lbs)
VEHICLE CAPACITY WEIGHT	322 kg (710 lbs)	161 kg (355 lbs)

Fig. 2 — THE BASIC SPECIFICATION OF JAPAN ESV

these different types, we believe that more technical information will become available, thereby assisting the achievement of our programme.

Nissan Motor Co. Ltd. will develop an ESV for four passengers, and Toyota Motor Co. Ltd. will develop an ESV of 900 kilogram weight designed for two passengers. Furthermore Honda Motor Co. Ltd. will develop a 750 kilogram weight ESV for four passengers, which is smaller in weight than that prescribed in the specifications. The details of these vehicles will be explained in the technical presentation by the individual Japanese manufacturers.

3. Our opinions were divided as to whether intrusion should be adopted as the criterion for judging the integrity of the occupant compartment. We have not sufficient information

on the relation between intrusion and human tolerance, and the injuries to the occupants will depend very much on the construction of the vehicle and on the method of the occupant restraint system. With these in mind, we are doubtful whether a uniform standard should be applied or not. We adopted a standard prescribing that "there shall be no passenger compartment intrusion greater than 125 millimeters at such places where the safety of occupants is involved."

We believe that this standard can be accepted as reasonable for a Japanese ESV, whereby the two approaches on curb weight and the number of passengers are adopted.

4. The Japanese specifications fail to describe the front bumper, which is sensitive at impact velocity. But the condition of collision is the same as that in the U.S.A. and also the requirement on occupant protection. The protection of occupants is related to such factors as vehicle configuration, body structure, occupant restraint system, interior design, etc. Therefore, in the original specifications, we decided not to specify for the front bumper, leaving room for flexibility in design. The progress of research may require further review of this problem.

5. We have tried to utilise the principles of the U.S. specifications on braking, steering and handling into a smaller type ESV, and found our results resembled the German specifications. As there is still room left for research in regard to the relation between steering and handling characteristics and the avoidance of accidents, we consider that all possible factors relating to accident avoidance should become the object of future study.

Our specifications will be increased depending on how our future research develops on these pending problems.

6. We can point out many differences in technical terms when expressed in English and also, as this programme will be extended to cover a new field of research, new terms will be introduced. Such differences could lead to confusion in the exchange of information and we earnestly hope that consideration will be given to the standardisation of technical terms.

THE TEST OF JAPANESE EXPERIMENTAL VEHICLES
(Summary of TOYOTA ESV Test)

YORSHIRO OKAMI
Japan Automobile Research Institute, Inc. (JARI)

1. Introduction

On 7 TOYOTA ESVs we conducted the accident avoidance tests totalling to 738 tests which covered 26 items and the collision and rollover tests totalling 6 tests which covered 6 items. Outlines of test results obtained for each individual test shall be given in Chapter 3 and 4 of this report.

Besides, outlined specification and distinctive features of the equipment and apparatuses employed for testing of ESVs shall be given in JARI ESV Report "The Test of Japanese Experimental Safety Vehicles (General Descriptions)."

Tab. 1 TOYOTA ESV Test Program Scope

	TOYOTA ESV	
	Testing Vehicle	Total Number of Tests
ACCIDENT AVOIDANCE TESTS		
1. Visibility	TESV-4	8
2. Display and Control	TESV-1,4	1
3. Braking Performance	TESV-1	196
4. Steering	TESV-1	220
5. Handling	TESV-1	250
6. Overturning Immunity	TESV-1, 10	30
7. Engine	TESV-1	23
8. Bumper Under ride/Over ride	TESV-1	10
COLLISION and ROLLOVER TESTS	TESV-6, 7, 8, 9, 10	6
Total	7 Vehicles	744

2. Brief Description of TOYOTA ESVs

The TOYOTA ESVs have been developed as two-door passenger cars giving accommodation to two passengers under the following basic principles.

With the basic attitude of exhaustive pursuit of engineering possibilities:

(1) Pertaining to the accident avoidance performance, attempts were made not only to meet the requirements of the Japanese ESV SPECIFICATIONS, but also to provide the prototype vehicles with the highest grade performance.

(2) Pertaining to the crashworthiness, attempts were made to incorporate the most extensive safety characteristics in addition to what is envisaged in the Japanese ESV SPECIFICATIONS.

Further, TOYOTA, upon designing and constructing the body, exercised the utmost care in utilizing to their possible extent the conventional techniques and at the same time to avail themselves with those materials that are available easily in the current market.

Outlined specifications and distinctive features of the TOYOTA ESVs are represented in Table 2.

Tab. 2 TOYOTA ESV Characteristics

Principal Dimensions of Vehicle Type: 2-door Accommodation: 2 Curb weight: 1,290 kg (Target 900 Kg) Engine Displacement: 1,600 cc Overall length: 4,300 mm Overall width: 1,800 mm Overall height: 1,380 mm	Wheelbase: 2,300 mm Tread: 1,500 mm (Front and rear) Output: 102 HP Transmission: Electric, remote control 3-speed type
Brake system:	Aiming at the shortest stopping distance and steadiest braking performance on all kinds of road surfaces, all the wheels are equipped with the disc brake, high pressure oil hydraulic boosted and anti-skidding apparatus. (ESC)
Headlamp:	The 4-mode type car velocity sensitive beam change-over system available for optimum distribution of intensity of illumination is equipped and additionally, a beam reserve system, i.e. a system of automatic switching to a new beam upon failure of a beam in use is installed.
Signal lamp:	A dual levelling system allowing discriminative use of luminous intensities for the daytime and night drive.
Electric wiring and control:	The SMN (Single-wire Multiplex Network)* system was developed and installed. *Note: The **SMN system** is referred to as a special computorized automatic system for fire prevention and safety improvement by simplifying the otherwise complicated controlling system occasioned by increase of such mechanisms that require electric and electronic control.
Warning instruments of defective functioning and operational units:	A warning instrument of defective functioning and operational units is installed on the center console for warning possible troubles totalling to approximately several ten troubles with respect to brake, lights, engine, etc.
Interior construction:	Safety pads are distributed in a generous manner all over inside the vehicle body. Simple and safe instruments and controls from the standpoint of the Human Engineering are conveniently located.

Tab. 2 (Continued)

Body construction:	For effective impact energy absorption and formation of a robust vehicle compartment, an energy-absorbing subframe (available for absorption of 67 % of impact energy generated by 80km/h front-end collision) plus the uniframe construction are adopted. This design principle limits amount of deformation by the passenger compartment intrusion less than 5″ for all the specified collision modes.
Bumper:	The front and rear bumpers are provided with the Silicon-rubber type shock-absorbers permitting restless energy absorption of impact of up to 15 km/h collision without accompaniment of damage on the vehicle body. Further, this arrangement permits absorption of 13% of total impact energy of the 80 km/h front collision. Additionally, the bumpers in front and rear are protected with soft urethane foam material covers to assure safety of pedestrians.
Passenger protecting device:	In addition to the energy-absorbing inside materials (for example, safety pads, etc.) the following safety equipment are installed:
Driver's seat	Energy-absorbing steering column, passive lap belt (of automatic type), 2-point safety belt *1 Gasbag *2
Assistant's seat	Passive lap belt Gasbag system

***1 Passive lap belt (Automatic fastening 2-point safety belt)**

This type of safety belt is referred to as a safety belt which is fastened on a wearer interlocking with opening and shutting of an automobile door. The belt in normal running condition allows the wearer free movements of the body, however, on the occasion of an impact or overturning, the belt restrains the body securely in accordance with the signal from the sensor.

***2 Gasbag system**

This system has a gasbag contained in the steering wheel for the driver's seat and a gasbag contained in the safety pad for the assistant's seat which detects a relative velocity and relative distance and by foreboding a collision, actuates through a radar-sensor at an above 30 km/h front-end collision.

Fig. 1 TOYOTA Experimental Safety Vehicle

3. Accident Avoidance Tests

We conducted the accident avoidance tests totalling to 738 times in the JARI compound as shown in Table 3 employing three different testing vehicles.

Outlines of test results obtained for each individual test shall be given in the following:

Tab. 3 Summary of TOYOTA ESV Accident Avoidance Tests

	Testing Vehicle	Total number of Tests
1. Visibility		
Head-lighting	TESV-4	1
Signaling System	"	1
Windshield Wipers	"	2
Washer	"	1
Defrost-Defog	"	1
Forward and Side Visibility	"	1
Rear Visibility	"	1
2. Display and Control	TESV-1, 4	1
3. Braking Performance		
Stopping Distance	TESV-1	40
Brake Effectiveness	"	80
Fading	"	1
Braking Efficiency	"	70
Parking Brake	"	5
4. Steering		
Yaw Response	TESV-1	200
Returnability	"	20
5. Handling		
Lateral Acceleration	TESV-1	120
Control at Breakaway	"	40
Crosswind Sensitivity	"	30
Pavement Irregularity Sensitivity	"	30
Steering Control Sensitivity	"	30
6. Overturning Immunity		
J-Turn	TESV-10	20
Slalom	TESV-1	10
7. Engine		
Emission	TESV-1	1
Vehicle Range	"	6
Passing Performance	"	10
Lateral Force Influence	"	6
8. Bumper Under ride/Over ride	TESV-1	10
Total	3 vehicles	738

3.1 Visibility

We performed tests on driver's field of view, clear glazing maintaining device and lighting system. For testing of forward vertical viewing angles, number of pillars, visibility affecting dimensions of pillar and rear visibility (fields of view of mirrors), we employed the screen projecting method in accordance with JASO 6824 and took measurements basing upon the

eyellipse recommended by SAE J 941c (Figure 2). Table 4 represents results with respect to the driver's fields of view.

We tested the windshield wipers on the wiping area and frequencies in accordance with the test procedures prescribed in SAE J 903b and the windshield washer on the system performance in accordance with the SAE J 942 and the defroster on the defrosting area in a low-temperature chamber of -18° C in accordance with the SAE J 902a. Table 5 represents the outstanding results of these tests.

In the measurements of headlamps, we obtained the illuminance values of beams on a vertical plane located 10 meters in the front of lamp projected from the vehicle installed in a dark room. We took measurements on individual lamp in the range of 15L-15R, 3U-3.5D and after summing up these individual measured values, we obtained the maximum luminous intensity. The operating voltage was 12.8V. We adopted the visual aiming method (SAE J 579a) for aiming. Table 6 represents the results with respect to lighting and signaling tests. The headlamp was provided with increased luminous intensity and equipped with 4-step beam switching apparatus in order to prevent glare hazardous to other drivers. The signaling system was the same as the conventional system and the dual intensity rear lighting system was adopted for better visibility.

3.2 Display and Controls

We assigned 29 drivers of different heights (146 ~ 192cm) to drive the vehicle for 30 minutes each in the day time and in the night time respectively and collected their subjective appraisals as to the consideration given by the designers of the vehicle on elements of human engineering, such as easiness of handling and operation, etc.

In these appraisals they were giving high credit to the excellent distribution of controls, etc. and they suggested for a little shortage of illumination of meters at night. Further they had a good impression on the wide field of view in the front and in the rear. Some tall drivers found their knees hitting against the steering wheel, because of a steep angle (22°) of the seat back and low position of the steering wheel.

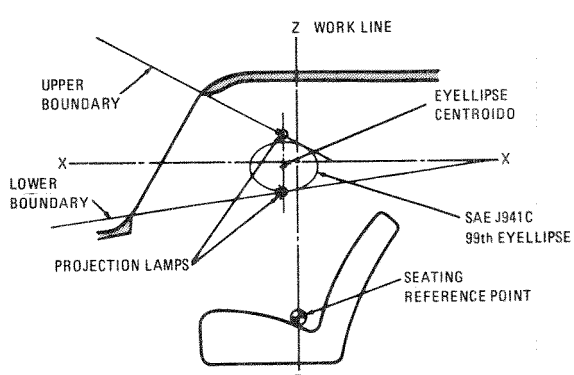


Fig. 2 Field-of-View Construction Lines and Projection Lamp Location

Tab. 4 Report of Driver's Field of View Tests

ESV specifications	Test results
1. Minimum vertical angles through windshield using SAE's 99th or 95th percentile eyellipse: Up = 13°, Down = 6°	Upward 99th 14.2° 95th 16.6° Downward 99th 6.6° 95th 7.0°
2. No more than 4 pillars in 270° arc from forward	4
3. Inside mirror (1) Minimum horizontal angle using the eyellipse centroid: 22° (2) In the upper field of view, the horizon shall be visible and in the lower field of view, the road surface within 15 meters behind the rear end of vehicle shall be visible.	29.6° Upward 0.5° Visible range 9.2 m
4. Outside mirror on the left side shall meet the requirements of Section 44 of Safety Regulations of Japan	Yes
5. Outside mirror on the right side (driver's side) (1) Minimum horizontal angle: 12° (2) Road surface visible 10 meters behind outside mirror	Horizon monocular 16.0° binocular 17.9° Visible range monocular 5.8 m binocular 5.8 m
6. Inside mirror and outside mirror on the right side (driver's side) adjustable from driver's seat	Yes

Tab. 5 Report of Clear Glazing Maintaining Device

ESV specifications	Test results
1. Windshield wiper Wiping area A: more than 80 % B: more than 94 % C: more than 99 %	A: 89 % B: 95 % C: 100 %
2. Washer 75 % of wiping area	100 %
3. Defroster Defrosting area after 30 minutes A: 80 % B: 95 % C: 100 %	A: 99 % B: 100 % C: 100 %

Tab. 6 Report of Tests of Lighting and Signaling

ESV specifications	Test results
1. The maximum luminous intensity of the head lamp shall not exceed 150,000 cd	116,000 cd
2. Separation of function for brake and turn	Yes
3. Side turn signal lamps	Yes
4. Lamp failure indicators	Yes
5. Passing signal light	Yes

3.3 Braking Performance

3.3.1 Stopping Distance

In the brake stopping distance tests, we conducted the tests by allowing the test vehicle to be braked from the initial velocity of 95 km/h with the brake pedal force within the requirements in the ESV SPECIFICATIONS.

The test vehicle did not deviate from the 3.6 m wide lane.

The tests upon failure of the booster, we conducted the tests by removing the drive belt of oil pump driven by the engine.

The tests upon a partial failure of the brake system, we conducted the tests by releasing individually the primary brake line and secondary brake line respectively from the master cylinder body. The test results are listed in Table 7.

3.3.2 Brake Effectiveness

In the brake effectiveness tests, we covered the following three component tests: tests of normally operating system, system with a booster failure and system with a partial failure. We performed 20 tests for each component test which can be broken down to 5 times of 4 tests with the target deceleration of 0.2 G, 0.4 G, 0.6 G and the highest decleration respectively the initial braking velocity was set at 95 km/h and the test driver performed the braking test maintaining the constant deceleration by watching the deceleration meter.

The ESV SPECIFICATIONS on the brake effectiveness calls for the range between (1) and (2) for (a) test of normally operating system, between (1) and (3) for (b) test of system with a booster failure and between (1) and (4) for (c) test of system with a partial failure, as shown in Fig. 3. The test results are shown in Fig. 3.

3.3.3 Fading

We performed this test in accordance with the testing procedure as prescribed in the SAE J843b, however, the load was limited to 112.4% of the full load. The results are listed in Table 8.

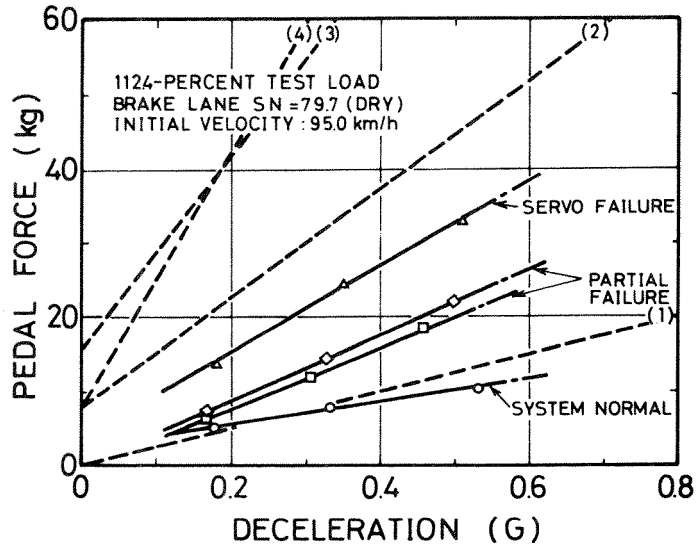


Fig. 3 Brake Effectiveness Test Results

3.3.4 Braking Efficiency

In the braking efficiency tests, we performed the tests on different road surfaces specified by the skid resistance number ranging from 33.2 to 78.8 (at 20°C) by measuring the stopping distances in locked wheel condition from the initial braking velocity of 95 km/h and the stopping distances in the condition of the skid control system operated from the same initial braking velocity.

The results of the measurements are represented in Table 9.

The definition of the braking efficiency set forth in the Japanese ESV SPECIFICATIONS is a ratio of the stopping distance in locked wheel condition to that in the condition operating the skid control system as expressed by the following calculation:

$$\text{Braking efficiency:} \quad E_\ell = \frac{D_\ell}{D_a} \times 100\ (\%)$$

where D_ℓ = stopping distance in locked-wheel condition (m)

D_a = stopping distance in the condition operating the skid control system (m)

NOTE:

(1) Toyota ESV is installed the skid control system for four wheels.

(2) In the Japanese ESV SPECIFICATIONS, the braking efficiency differ from the brake system efficiency of the United States. Incidentally, for this brake system efficiency, reference is made to Technical Session Paper "The Characteristics of Japanese ESV Tires."

3.3.5 Parking Brake

We performed the parking brake tests on a 31 % grade test slope.

The testing procedures were to drive the vehicle into the test slope aligning the vehicle center line parallel with the center line of the test slope and allowed it to stop by employing

the service brake in both uphill and downhill conditions. We limited the brake operating force to 60 kg, when we used the parking brake foot pedal and held the vehicle at a standstill for five minutes. We conducted this particular tests five times for each condition. The results are listed in Table 10.

Tab. 7 Stopping Distance Test Results

PARAMETER		ESV SPEC	TEST RESULTS
SYSTEM NORMAL		50 m MAX.	40.09 m
PARTIAL FAILURE	PRIMARY CIRCUIT FAILED	120 m MAX.	49.93 m
	SECONDARY CIRCUIT FAILED		57.90 m
SERVO FAILURE		105 m MAX.	44.16 m

Tab. 8 Fade Test Results

PARAMETER	ESV SPEC	TEST RESULTS
FADE	60 kg MAX.	10.8 kg
RECOVERY	60 kg MAX.	7.7 kg

Tab. 9 Braking Efficiency Test Results

TEST GROUND	SN (AT THE TEST) SURFACE CONDITION	TEST LOAD PERCENT(%)	STOPPING DISTANCE (m) ANTILOCK(D_a)	LOCK (D_l)	BRAKING EFFICIENCY ESV SPEC (%)	$E_l = D_l / D_a \times 100$ (%)
CONCRETE (HIGH SPEED TRACK)	80.0 (DRY)	112.4	40.09	45.25	≧ 80	112.8
	59.6 (WET)	112.4	43.92	62.14		141.5
POLISHED CONCRETE (i) (SLIPPERY TEST TRACK)	50.2 (WET)	112.4	58.41	84.45		144.6
(ii)	33.9 (WET)	112.4	71.87	98.90		137.6

Tab. 10 Parking Brake Test Results

PARAMETER	ESV SPEC	TEST RESULTS
UP HILL	60 kg MAX.	39.2 kg
DOWN HILL	60 kg MAX.	27.0 kg

3.4 Steering

3.4.1 Steady State Yaw Response

In the steady state yaw response tests, we allowed the test vehicle to maintain for at least three seconds the steady state circular running path at the specified velocity and lateral acceleration. The testing velocities were 40, 60, 80 and 110 km/h. We conducted 10 lefthand

and 10 righthand turning respectively for the lateral acceleration of 0.4G and 5 each lefthand and righthand turning tests respectively for that of 0.3G and 0.5G.

Additionally, for the TOYOTA ESV, we performed these tests with the front wheel angle defined as steering wheel angle/overall gear ratio and the rotating angle arround in king pin axis. The test results are shown in Fig. 4.

3.4.2 Transient Yaw Response

In the transient yaw response tests, we allowed the test vehicle to run on a straight line running path at the testing velocities of 40 and 110 km/h, providing the vehicle with the steering input at the rate of more than 500 deg/sec up to the predetermined angle and maintaining it to produce the lateral acceleration of 0.4G for a constant time. We conducted 10 lefthand and 10 righthand turning tests respectively for each vehicle velocity. The test results are shown in Fig. 5.

3.4.3 Returnability

In the returnability tests, we drove in the steady state the test vehicle at the testing velocities of 40 and 80 km/h along a circular running path sized to achieve the lateral acceleration of 0.4G and released completely the steering wheel at the predetermined point of the vehicle path. We conducted the 5 lefthand and 5 righthand turning tests respectively. The test results are shown in Fig. 6.

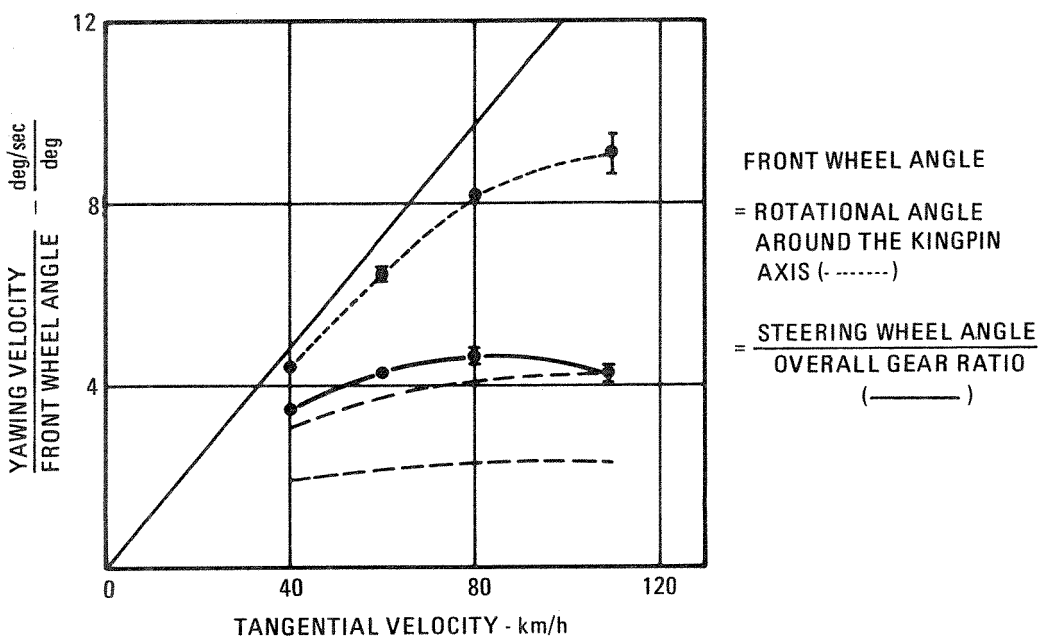


Fig. 4 Steady State Yaw Response Test Results

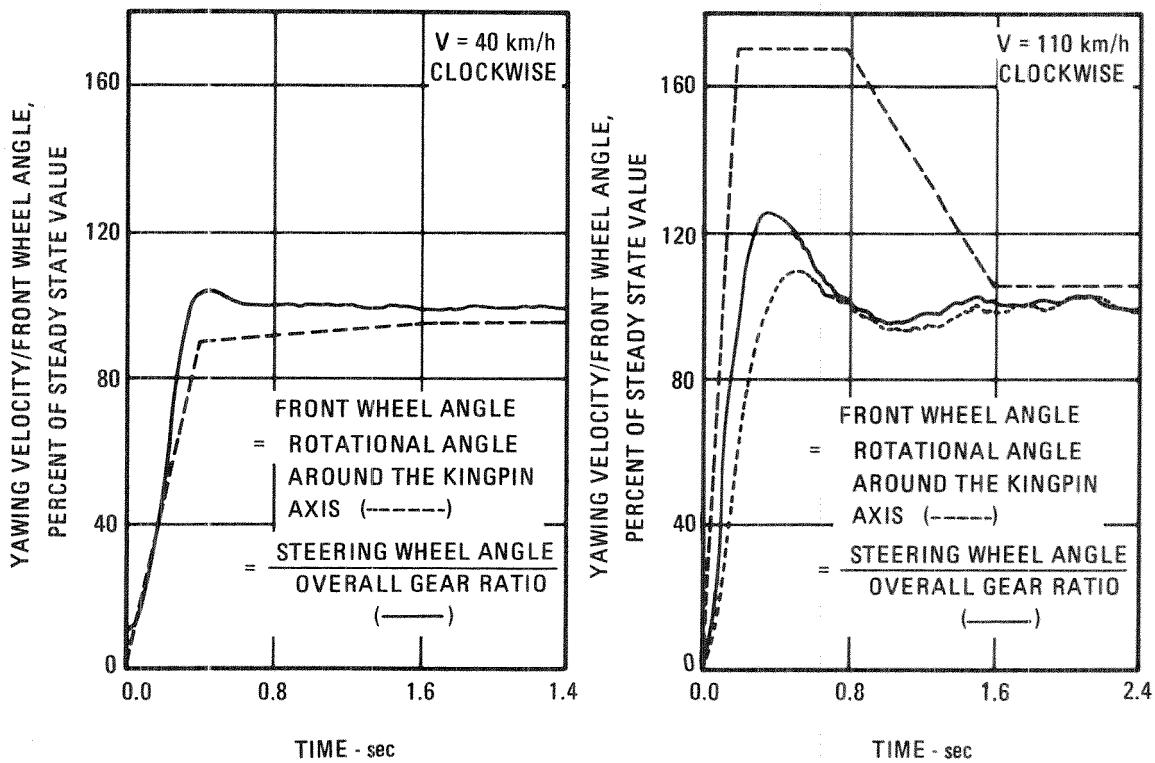


Fig. 5 Transient Yaw Response Test Results

3.5 Handling

3.5.1 Lateral Acceleration

In the lateral acceleration tests, we drove the test vehicle along a circular running path with the radius of 30 m, accelerating the velocity until the highest lateral acceleration is achieved. Handling of the steering wheel is divisible into two different controls; the fixed control and manual control in which the driver operates the steering wheel in a way he thinks it necessary to do the same. Further, as in JARI, we treat the value prescribed in the SPECIFICATIONS as the minimum requirement and we set an objective of our present test to exceed this particular requirement value, the results achieved did not necessarily turn out the limit optimum value. Fig. 7 represents the results of the test.

3.5.2 Control at Breakaway

In the control at breakaway tests, we allow the test vehicle to operate in the steady state along the running paths with the radius of 30 and 70 m at the velocity which produce the highest lateral acceleration(*) with the fixed control of steering wheel, accelerate the velocity gradually and started our test by closing the throttle at the point when the vehicle center line within the wheelbase moved radially outward by the distance of 3 m from the initial path.

* The value prescribed in the SPECIFICATIONS for the lateral acceleration test is 0.60 G.

Further, for indication of the point where the track breakaway from the initial path and returned to it, we employed a white-line indicator. Fig. 8 represents the results of the tests.

3.5.3 Crosswind Sensitivity

In the crosswind sensitivity tests, we drove the test vehicle into the straight line running path with the fixed steering wheel at the velocity of 50, 80 and 110 km/h respectively, letting the vehicle pass in front of a crosswind generator. The running track for two seconds after the vehicle began to run under the influence of the crosswind was measured with a jet track tracing apparatus. The test results are shown in Fig. 9.

3.5.4 Pavement Irregularity Sensitivity

In the pavement irregularity sensitivity tests, we drove the test vehicle along a running course provided with the predetermined surface irregularity at the velocities of 50, 80 and 110 km/h respectively. We measured the distance of deviation from the course after two seconds of contact of the test vehicle with a road bump. The test results are shown in Fig. 10.

3.5.5 Steering Control Sensitivity

In the steering control sensitivity tests, we operated the steering wheel in such a way that the yawing angular velocity changes gradually in the neighbourhood of 2 deg/sec at the velocities of 50, 80 and 110 km/h respectively and measured the values of steering torque which corresponds to the yawing angular velocity of ±2 deg/sec at this moment. Further, the TOYOTA ESVs involved are all equipped with the manual control system and not with the power steering system. The test results are shown in Fig. 11.

3.6 Overturning Immunity

3.6.1 J-Turn

In the J-turn tests, we drove the test vehicle with an automatic steering system for prevention of a hazardous accident. When the vehicle in operation at the velocities of 50, 80 and 110 km/h respectively on a straight line course arrives at the predetermined point on the course, a pair of trigger system composed of elements on board the vehicle and on the ground is actuated to steer the vehicle with more than 180 deg and 500 deg/sec and simultaneously the throttle is closed to initiate the test.

In the J-turn tests, the test vehicle did not overturn in any of the three lefthand and righthand turning tests respectively at the velocities of 50, 80 and 110 km/h.

3.6.2 Slalom

In the slalom course test, we conducted this test in allowing the test vehicle to run in slalom at a velocity of 80 km/h on the running course studded with 11 pylons arranged with the spacing of 30 m. In this test, the velocities of drive in time and mean value of velocities

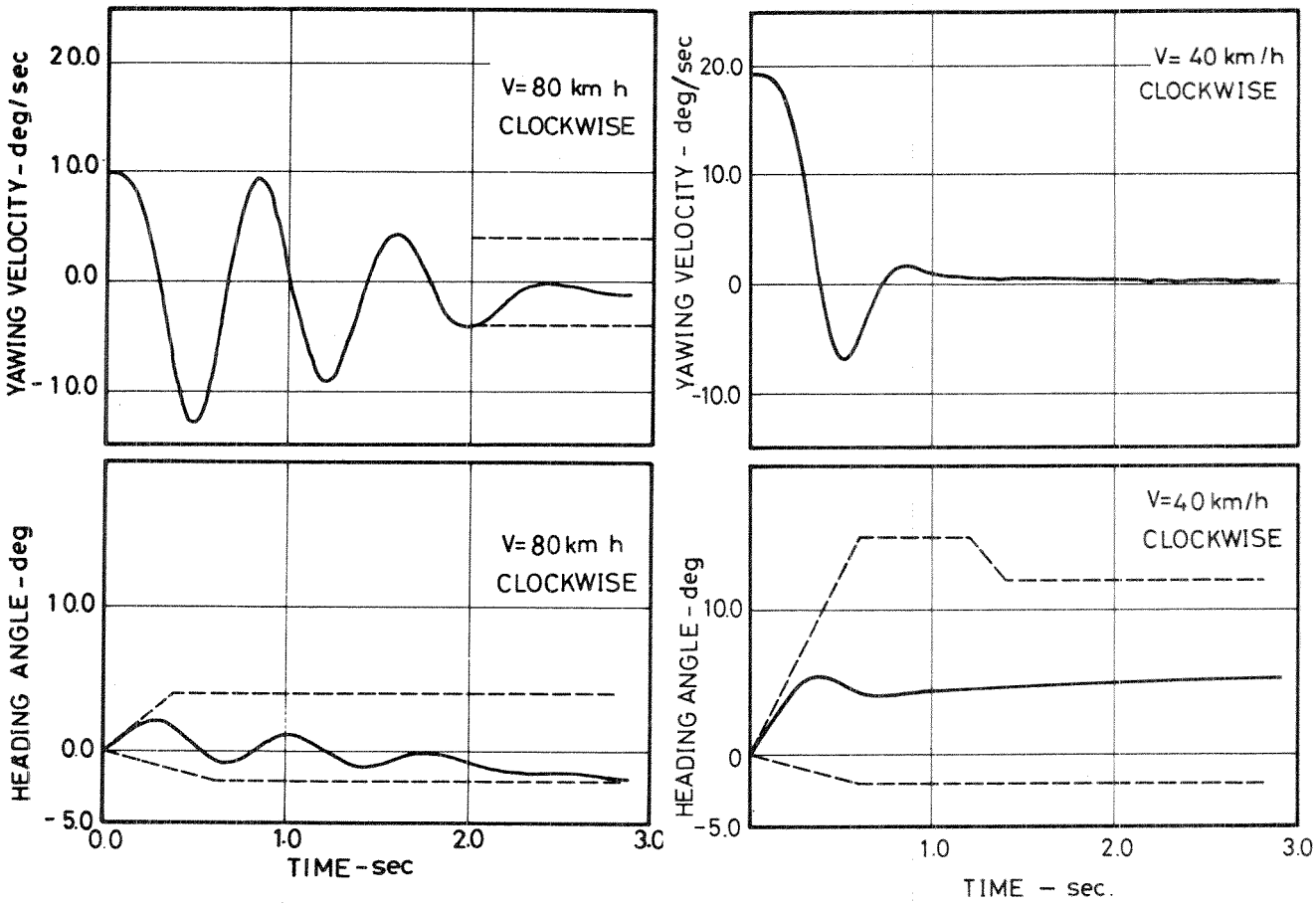


Fig. 6 Returnability Test Results

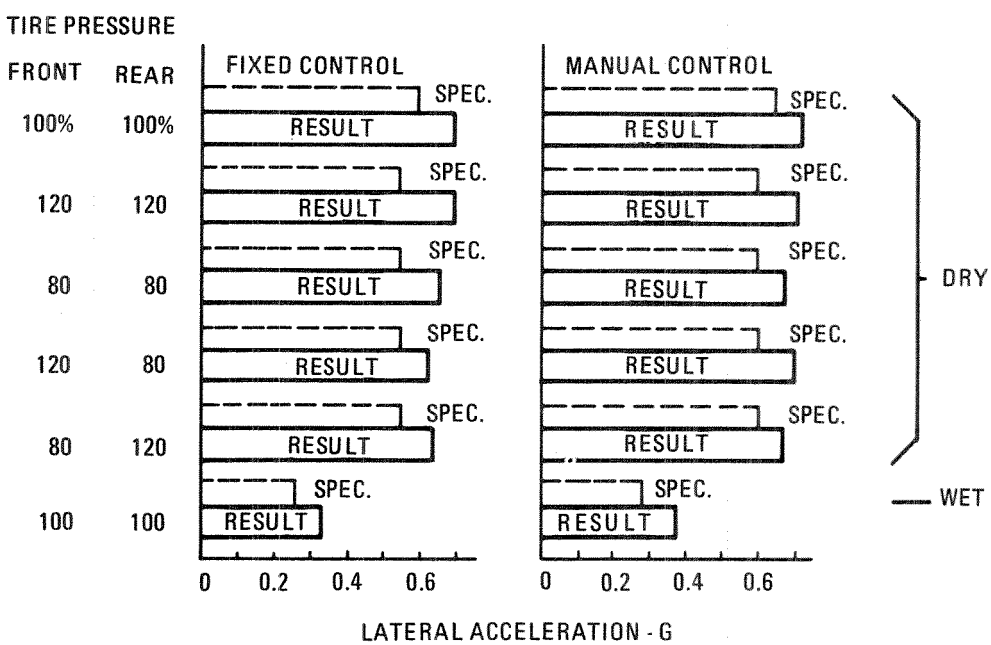


Fig. 7 Lateral Acceleration Test Results

running along the pyloned section are not supposed to be less than the specified values respectively.

In the slalom tests, the test vehicle did not overturn in any of ten driving tests.

3.7 Bumper Under ride/Over ride

The bumper under ride/over ride tests have been performed with some decelerations from 95 km/h and some accelerations from the low range. (TOYOTA ESV is installed the automatic transmission). The test results are shown in Fig. 12.

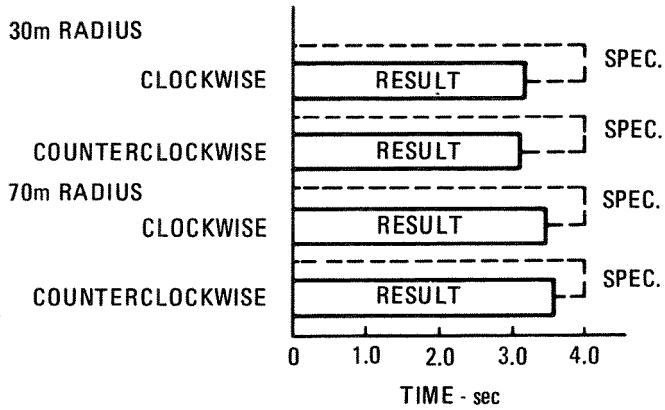


Fig. 8 Control at Breakaway Test Results

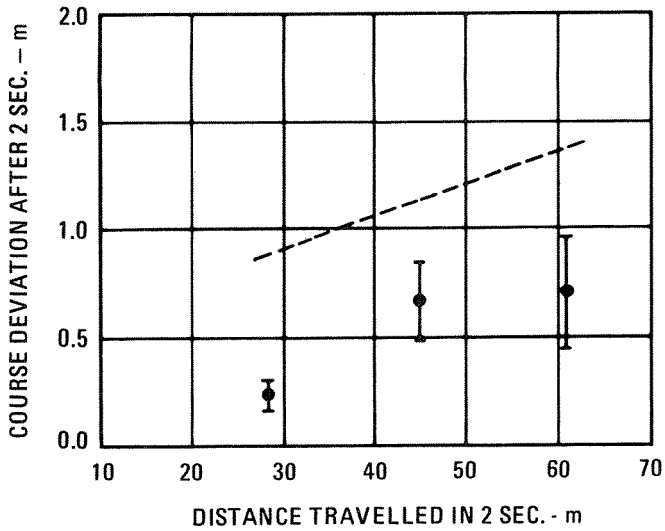


Fig. 9 Crosswind Sensitivity Test Results

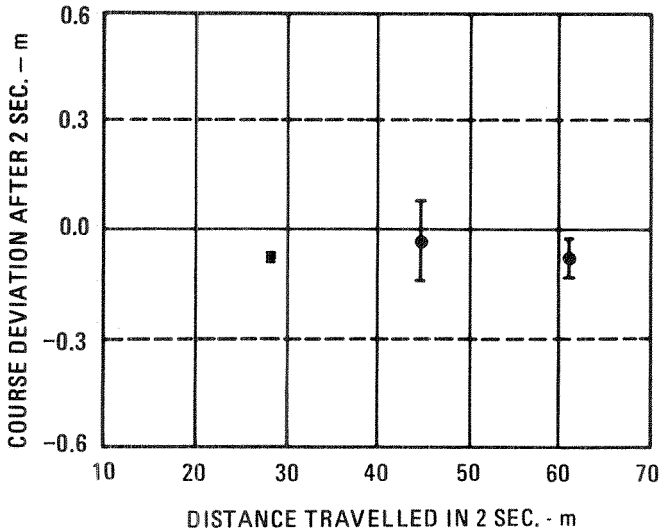


Fig. 10 Pavement Irregularity Sensitivity Test Results

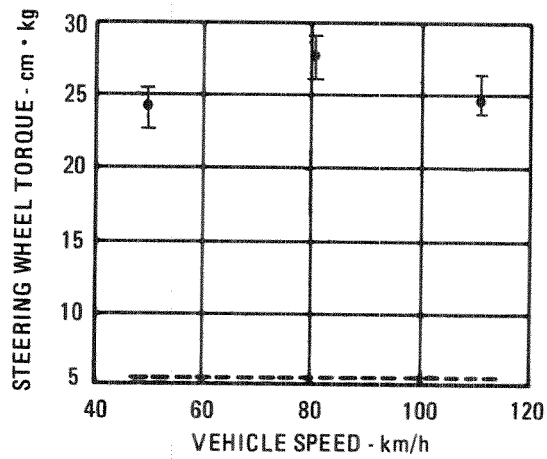


Fig. 11 Steering Control Sensitivity Test Results

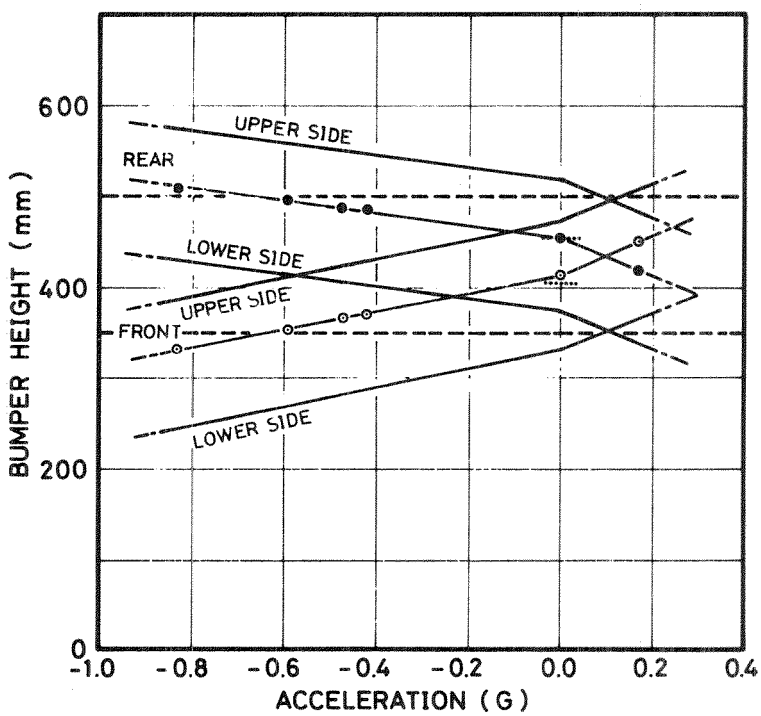


Fig. 12 Bumper Under ride/Over ride Test Results

4. Collision and Roll-over Tests

(1) We designated front end impact, rear end impact, side impact and roll-over, as four basic impact modes for differentiating effectiveness of the crash injury reduction of occupant.

(2) We performed the flat and pole barrier test for the front end impact, moving barrier tests (1820 kg) for the rear end impact, pole/side impact tests for the side impact and dolly test for the roll-over.

(3) The test items and the test conditions shall be given in the following;

Tab. 11 Summary of Collision and Rollover Test Items and Conditions

IMPACT MODE	TESTING VEHICLE	TARGET SPEED (km/h)	TEST SPEED (km/h)	NUMBER OF DUMMIES	TEST WEIGHT (kg)
FLAT BARRIER IMPACT	TESV- 6	$15^{+0}_{-1.0}$	16.0	2	1450
FLAT BARRIER IMPACT	TESV- 8	$80^{+0}_{-1.6}$	79.2	2	1450
POLE BARRIER IMPACT	TESV- 7	$80^{+0}_{-1.6}$	79.2	2	1450
POLE/SIDE IMPACT	TESV- 9	$25^{+0}_{-1.0}$	24.5	2	1450
REAR END IMPACT	TESV- 6	$80^{+0}_{-1.6}$	78.5	2	1460
ROLL-OVER	TESV-10	48	47.3	2	1455

(4) We employed exclusively the anthropomorphic test device VIP-50A manufactured by Alderson.

(5) Cut-off frequency is 250 Hz for the anthropomorphic test device (dummy) and in accordance with SAE J 211a for the other items.

(6) In this summary, we will report mainly the factors relating to the occupant injury protection during car crash, but we intend to present the full report containing the details of other characteristic to the Government of Japan later.

4.1 Test Result

No anthropomorphic test device was ejected from the vehicle passenger compartment in all tests above mentioned.

Outlines of test results obtained for each individual test shall be given in the following.

4.1.1 15 km/h Flat Barrier Impact Test

In this test, we installed 2 anthropomorphic test devices fastened with the lap belt system as the restraint device and conducted the subjected impact test.

The real testing velocity was in excess of the target value amounting to 16.4 km/h.

There was no body damage observed and all the component units related to the vehicle operation remained unscathed and no irregular running condition was witnessed in the

running test after the impact test. We took measurement of the bumper stroke at the moment of impact by means of a photographic measuring method and the value was 120 mm.

We used this vehicle as the test vehicle for the rear-end impact at a later date.

4.1.2 80 km/h Flat Barrier Impact Test

We performed the subjected test in a condition satisfying the specified test condition and with a testing velocity of 79.2 km/h,deviation of the target point of 10 mm and error of impact angle not exceeding 1 degree.

Fig. 13 represents a right side view of the vehicle after the 80 km/h flat barrier impact test.

Fig. 14 represents the time history of the resultant head acceleration and vehicle floor acceleration (longitudinal direction) during the 80 km/h flat barrier impact test.

Fig. 15 represents a time history of a resultant acceleration of the chest and Fig. 16 represents a bar chart showing the dummy femur load.

Data with respect to vehicle and to anthropomorphic test devices are shown in Table 12 and Table 13 respectively.

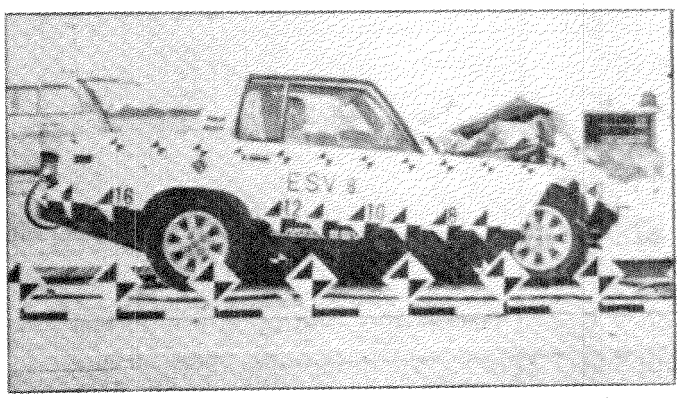


Fig. 13 Post-test View of TOYOTA ESV Following 80km/h Flat Barrier Impart

4.1.3 80 km/h Pole Barrier Impact Test

We conducted the subjected test in a condition satisfying fully the specified test condition and with a testing velocity of 79.2 km/h, the impact position error of 5 mm with respect to the target point and deviation of impact angle not exceeding 1 degree.

Fig. 17 represents an overhead view of the vehicle after the 80 km/h pole barrier impact test.

Fig. 18 represents the time history of the resultant head acceleration of dummies and vehicle floor acceleration (longitudinal direction) during in the course of test. A high value of the dummy head G located on the left front (passenger) seat is attributed to a imperfect action of the gas-bag system for the left front seat which caused the dummy head to hit against the front windshield glass.

Fig. 19 represents the time history of a resultant chest acceleration of the dummies and Fig. 20 represents a bar chart showing the dummy femur load.

Data with respect to vehicle are shown in Table 14 and data with respect to anthropomorphic test devices in Table 15.

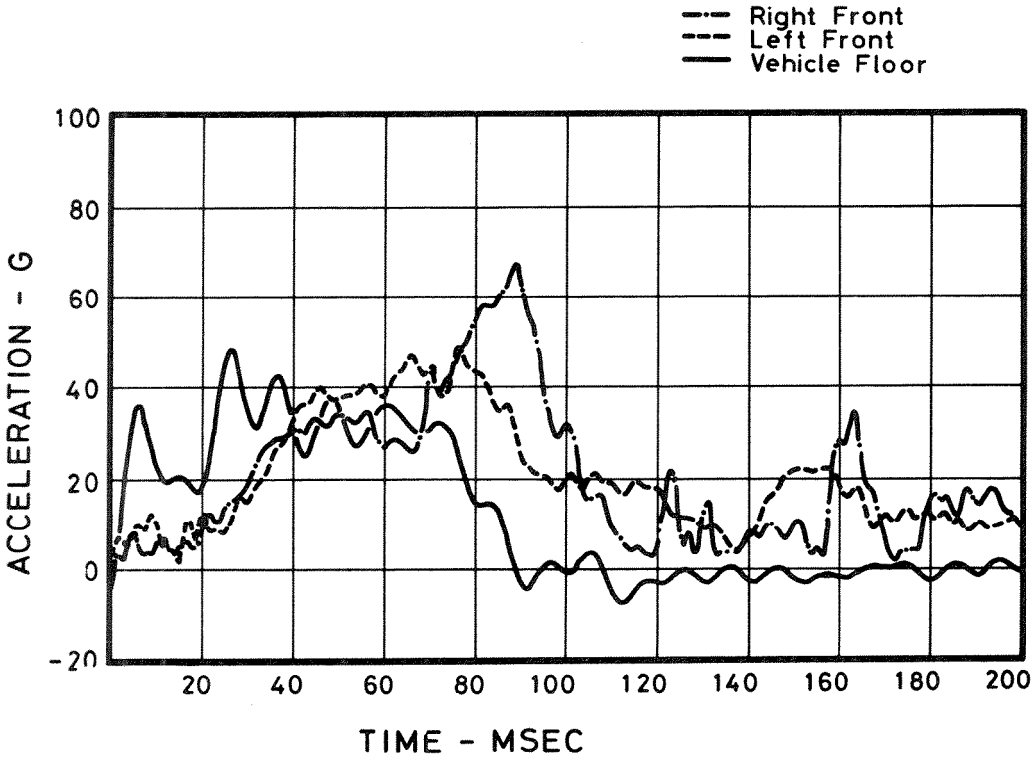


Fig. 14 Head G–80km/h Flat Barrier Impact

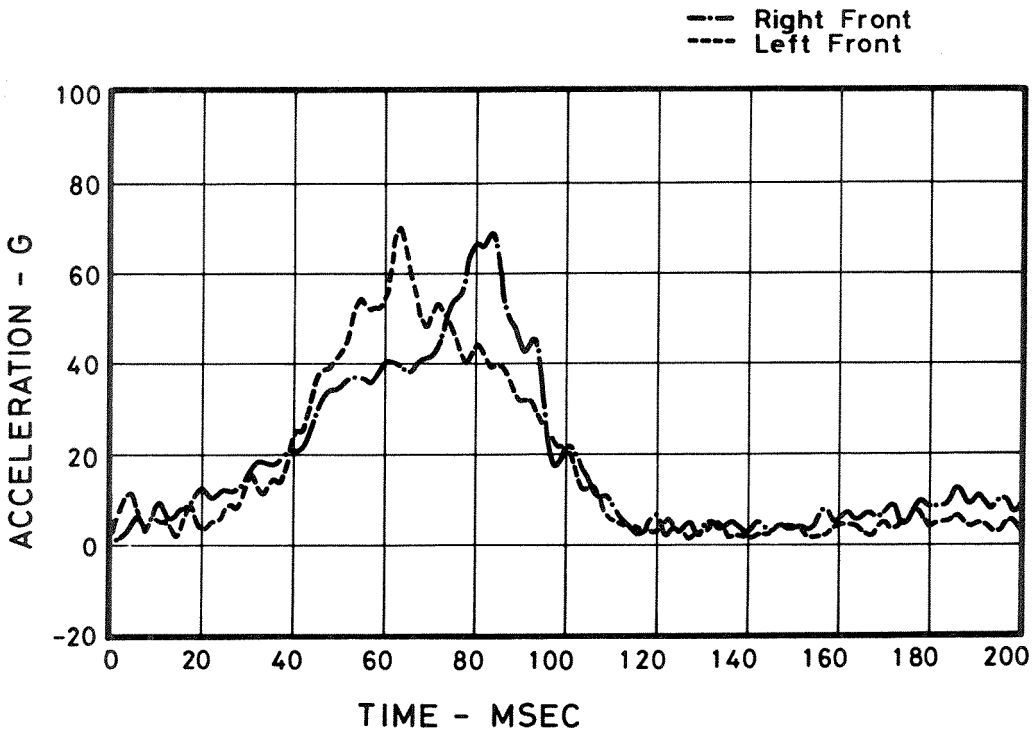


Fig. 15 Chest G–80km/h Flat Barrier Impact

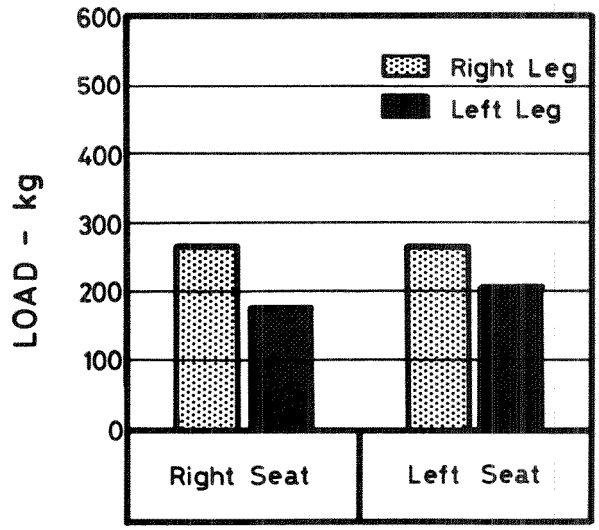


Fig. 16 Femur Loads–80km/h Flat Barrier Impact

(1) Data with respect to vehicle

Tab. 12 Vehicle Structural Response-80km/h Flat Barrier Impact

Peak Floor G (G)	48
Average Floor G ※ (G)	30
Residual Crush (mm)	585
Compartment Intrusion (mm)	65

The position of the measured maximum compartment intrusion was located at the firewall.

(2) Data with respect to test devices

Tab. 13 Dummy Dynamic Response-80km/h Flat Barrier Impact

		Right (driver)	Left
Head	Peak G (G)	67	49
	Time Over 80G (ms)	–	–
Chest G		69	70
Femur Peak Load	Left (kg)	180	210
	Right (kg)	270	270

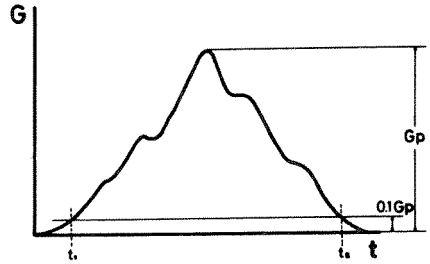
※ The average floor acceleration (longitudinal derection) $\bar{G}_f$ was calculated by the following expression

$$\bar{G}_f = \frac{1}{t_2 - t_1} \int_{t_1}^{t_2} G(t)\,dt$$

Here G (t) : time history of longitudinal acceleration of vehicle

Gp : Peak value of G (t)

t_1, t_2 : Points of intersection of G (t) and G = 0.1 Gp.



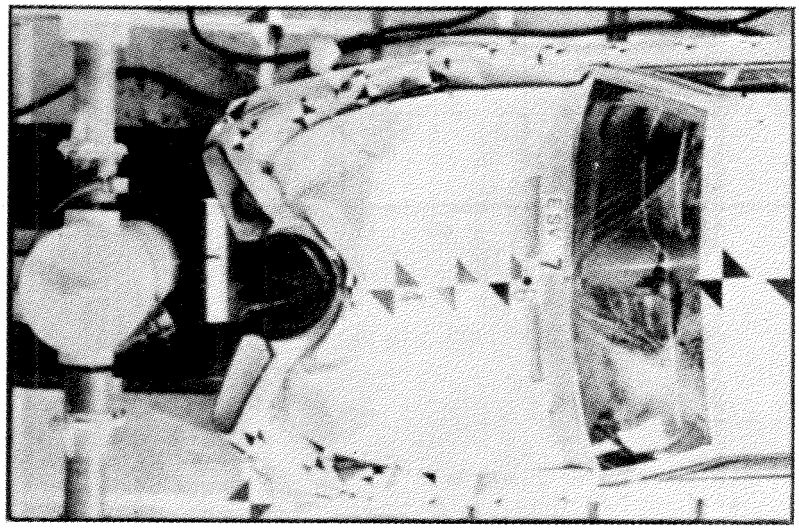


Fig. 17 Post-test View of TOYOTA ESV Following 80km/h Pole Barrier Impact

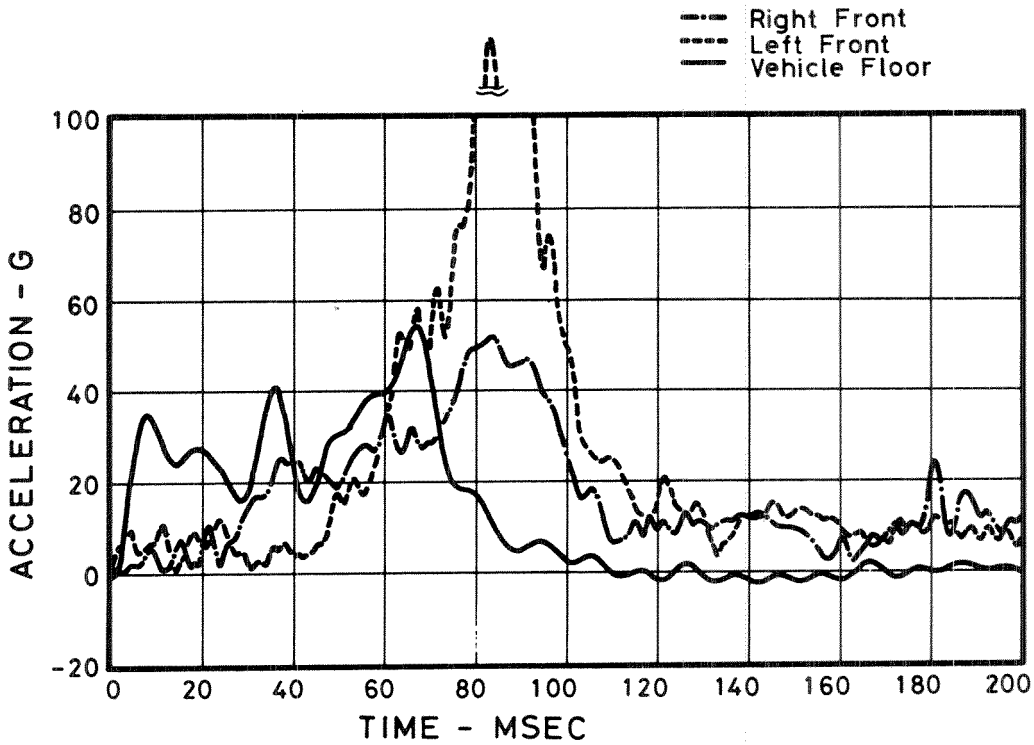


Fig. 18 Head G–80km/h Pole Barrier Impact

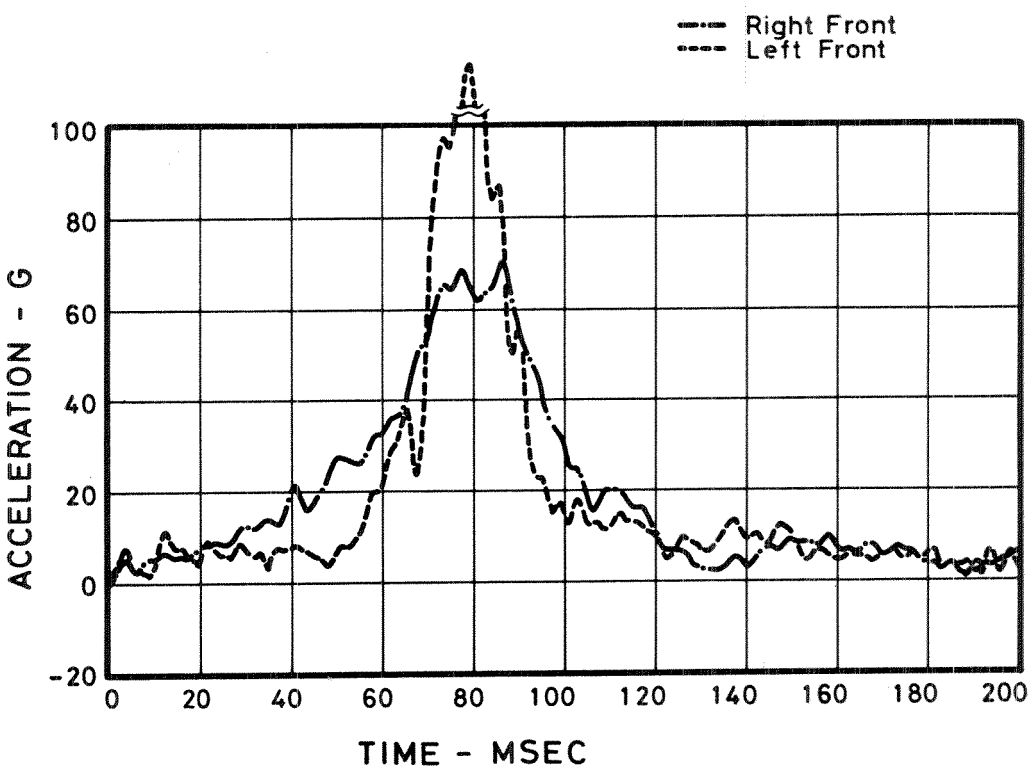


Fig. 19 Chest G–80km/h Pole Barrier Impact

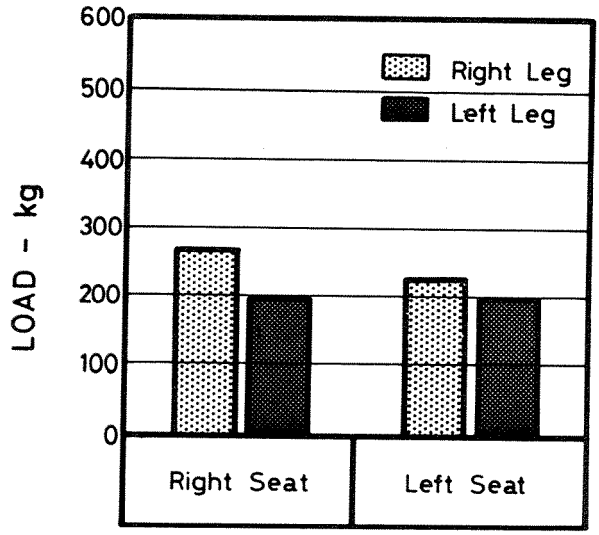


Fig. 20 Femur Loads–80km/h Pole Barrier Impact

(1) Data with respect to vehicle

Tab. 14 Vehicle Structural Response-80km/h Pole Barrier Impact

Peak Floor G (G)	54
Average Floor G (G)	28
Residual Crush (mm)	860
Compartment Intrusion (mm)	30

The position of the measured maximum compartment intrusion was located the firewall.

(2) Data with respect to test devices

Tab. 15 Dummy Dynamic Response-80km/h Pole Barrier Impact

		Right (driver)	Left
Head	Peak G (G)	52	255
	Time Over 80G (ms)	-	15
Chest G (G)		70	152
Femur Peak Load	Left (kg)	200	200
	Right (kg)	270	240

4.1.4 25 km/h Pole/Side Impact Test

We performed the subjected test with a test vehicle mounted on a dolly and with the impact target set on the righthand door maximum opening center. The test velocity was 24.5 km/h, deviation of impact position from the target point was 30 mm.

Fig. 21 represents an overhead view of the vehicle after 25 km/h Pole/Side Impact test.

Data with respect to vehicle and to anthropomorphic test devices are presented in Table 16 and Table 17 respectively.

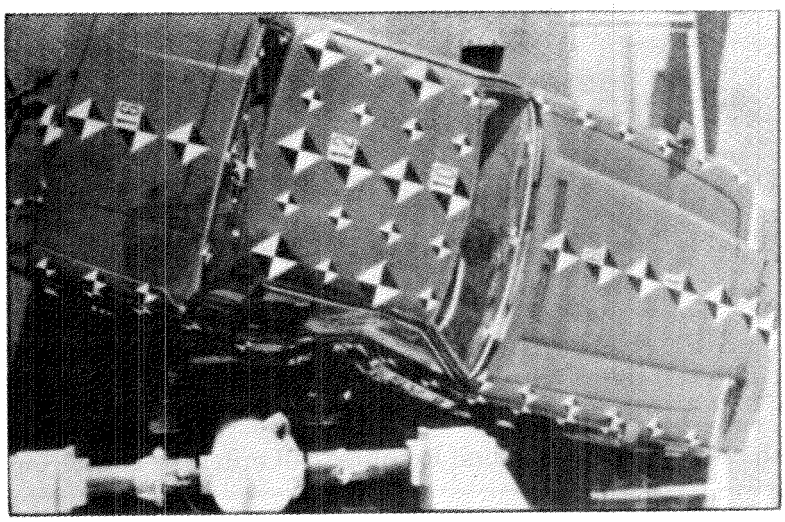


Fig. 21 Post-test View of TOYOTA ESV Following 25km/h Pole/Side Impact

(1) Data with respect to vehicle

Tab. 16 Vehicle Structural Response-25km/h Pole/Side Impact

Peak Floor G	(G)	19
Average Floor G	(G)	10
Residual Crush	(mm)	180
Compartment Intrusion	(mm)	55

(2) Data with respect to test devices

Tab. 17 Dummy Dynamic Response-25km/h Pole/Side Impact

			Right (driver)	Left
Head	Peak G	(G)	42	48
	Time Over 80 G	(ms)	—	—
Chest G		(G)	34	27
Femur Peak Load	Left	(kg)	80	20
	Right	(kg)	50	40

4.1.5 80 km/h Rear-end Impact Test

We performed the subjected test with the moving barrier (weighing 1820 kg) in accordance with the SAE J 972 with a test velocity of 78.5 km/h and with the deviation of impact position nearly 0 mm.

The braking performance of the moving barrier proved to be excellent and collisions between the test vehicle and the barrier never occurred more than one time.

Fig. 22 represents the right side view of the vehicle after 80 km/h rear end impact.

Data with respect to vehicle and to anthropomorphic test devices are presented in Table 18 and Table 19 respectively.

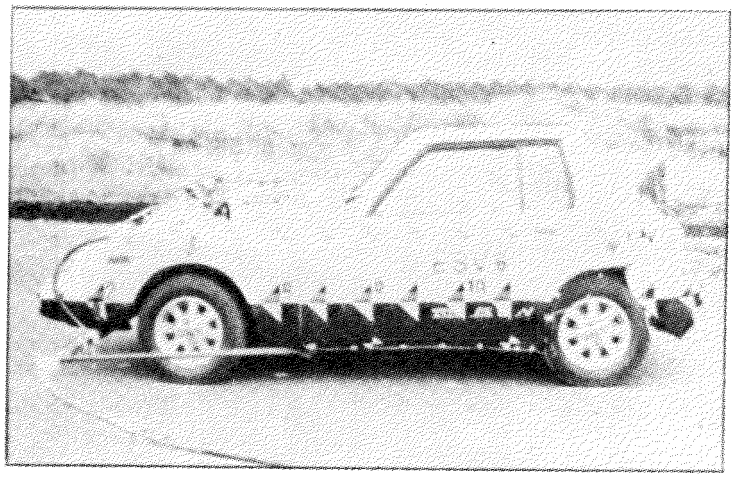


Fig. 22. Post-test View of TOYOTA ESV Following 80km/h Rear-end Impact

(1) Data with respect to vehicle

Tab. 18 Vehicle Structural Response-80km/h Rear-end Impact

Peak Floor G (G)	42
Average Floor G (G)	19
Residual Crush (mm)	480
Compartment Intrusion (mm)	40

(2) Data with respect to test devices

Tab. 19 Dummy Dynamic Response-80km/h Rear-end Impact

		Right (driver)	Left
Head	Peak G (G)	73	46
	Time Over 80 G (ms)	–	–
Chest G (G)		51	42
Femur Peak Load	Left (kg)	240	230
	Right (kg)	110	220

4.1.6 Roll-over Test

We performed the subjected test with a method employing the dolly in conformity with FMVSS No. 208, the testing speed and number of rollovers being 47.3 km/h and 3 respectively.

Data with respect to vehicle and to anthropomorphic test devices are shown in Table 20 and Table 21 respectively.

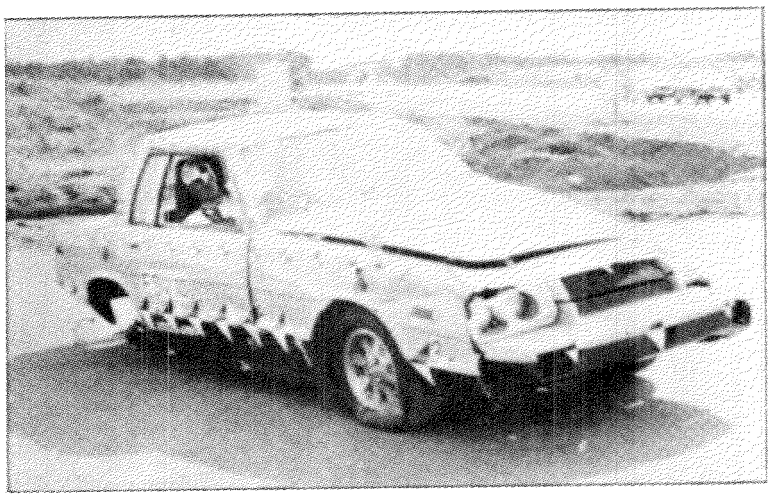


Fig. 23 Post-test View of TOYOTA ESV Following Rollover

(1) data with respect to vehicle

Tab. 20 Vehicle Structural Response-Rollover

Peak Floor G (G)	10
Compartment Intrusion (mm)	25

The maximum compartment intrusion occurred in the position of joint of the lefthand A-pillar with roof.

(2) Data with respect to the test devices

		Right (driver)	Left
Head	Peak G (G)	27	51
	Time Over 80 G (ms)	—	—
Chest G (G)		26	17
Femur Peak Load	Left (kg)	50	60
	Right (kg)	50	30

Tab. 21 Dummy Dynamic Response-Rollover

THE TEST OF JAPANESE EXPERIMENTAL VEHICLES
(Summary of NISSAN ESV Test)

YORSHIRO OKAMI
Japan Automobile Research Institute, Inc. (JARI)

1. Introduction

On 8 NISSAN ESVs we conducted the accident avoidance tests totalling to 810 tests which covered 26 items and the collision and rollover tests totalling 6 tests which covered 5 items. Outlines of test results obtained for each individual test shall be given in Chapter 3 and 4 of this report.

Besides, outlined specification and distinctive features of the equipment and apparatuses employed for testing of ESVs shall be given in JARI ESV Report "The Test of Japanese Experimental Safety Vehicles (General Descriptions)."

Table 1 NISSAN ESV Test Program Scope

	NISSAN ESV	
	Testing Vehicle	Total Number of Tests
ACCIDENT AVOIDANCE TESTS		
1. Visibility	NESV-1, 4, 6	8
2. Display and Control	NESV-1, 6	1
3. Braking Performance	NESV-1	266
4. Steering	NESV-5	220
5. Handling	NESV-5	250
6. Overturning Immunity	NESV-1, 5	30
7. Engine	NESV-1	23
8. Bumper Under ride/Over ride	NESV-1	12
COLLISION and ROLLOVER TESTS	NESV-1,6,7,8,9,10	6
Total	8 Vehicles	816

2. Brief Description of NISSAN ESVs

The NISSAN ESVs have been developed with a special emphasis placed on creation of such vehicles to which a high credit in terms of overall safety features could be given from the view point of current traffic system; this leading concept was realized as a 4-door family sedan accommodating four persons. This particular approach, accordingly, neither encouraged unreasonable extention of dimensions nor reinforcement of weight for building safe vehicles, but efforts were concentrated to achieve the performance essential to the ESVs within the limit of dimensions of the so-called small-sized cars which are most popular in the existing market of Japan.

Outlined specifications and distinctive features of the NISSAN ESVs are represented in Table 2.

Further, the NISSAN ESVs are composed of Two different type models, E1 system and E2 system. They are different from the proportional combination of the crashworthiness and occupant restraint. Disttinction between these two models are given in detail in Table 2.

Table 2 NISSAN ESV Characteristics

Principal Dimensions of Vehicle Type: 4-door sedan Wheelbase: 2,550 mm Accommodation: 4 Tread: 1,365 mm (Front) 1,375 mm (Rear) Curb weight: 1,250 kg (Target being 1,150 kg) Engine displacement: 1,770 cc; Transmission: Manual 4 speed type Overall length: 4,415 mm; Overall width: 1,640 mm Overall height: 1,385 mm (Excl. periscope)	
Brake System:	Aiming at the shortest stopping distance and steadiest braking performance on all kinds of road surfaces, the front wheel is equipped with the disc brake, the rear wheel with Alfin drum brake and antiskidding apparatus and all the wheels with vacuum booster.
Tire:	Double-construction safety tire is adopted which can stand approximately 500 km even under a flat-tire condition.
Visibility:	By far the wider field of view than the conventional vehicle is available with the objective of providing an extensive improvement on the illumination system, including ○ Newly developed double-sheet concave mirror type periscope providing rear view three times as wide as the current design. ○ Brighter three-beam headlamp ○ Trail lamp indicating accelerating operations of the own vehicle to the following vehicle. ○ Dual leveling lamp
Warning instruments of defective functioning and operational units:	Shortage of tire inflation pressure, defective anti-skidding apparatus, used up oil, defective airbag, etc. are indicated by a clustered warning apparatus to the driver.
Crashworthiness:	(1) Head-on collision To cope with the most difficult crashworthiness requirements to be met on the occation of 80 km/h front collision against a barrier, For E1 system vehicle, ○ a quick inflating solid-type airbag, ○ monocoque construction body and shock-absorber equipped bumpers, For E2 system vehicle, ○ a preloaded safety belt providing a immediate restraint to the passenger by the retraction of the seat belt directly after the impact of collision and ○ a body equipped with an energy-absorbing broach located in front of the body that functions by expanding out a tube at the time of collision are installed respectively. (2) Rear-end collision To assure safety of the passengers at a 65 km/h rear-end collision against a moving barrier,

Table 2 NISSAN ESV Characteristics (continued)

	○ high back-type rear-end collision protection seat and a body provided with energy-absorbing constructions are installed. (3) Lateral collision To assure safety of the passengers at a 50 km/h lateral collision, ○ pillar pads, door pads, ○ reinforced sills, ○ guard-bar incorporated doors and construction ○ a safety seat of lefthand and righthand side monoblock are installed.
Pedestrian Protection	To minimize injury of the pedestrian, ○ a pedestrian protecting bumper covered with a soft urethane sheath ○ a honey-comb sandwitch type energy-absorbing engine hood are installed.

Fig. 1 NISSAN Experimental Safety Vehicle (E1 System)

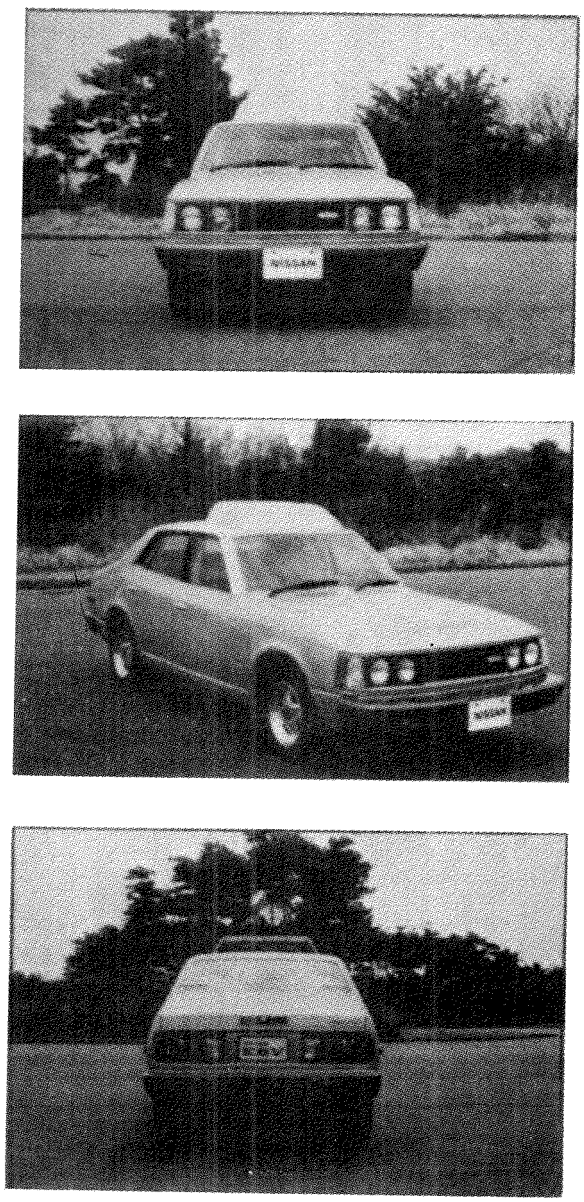


Fig. 2 NISSAN Experimental Safety Vehicle (E2 System)

3. Accident Avoidance Tests

We carried out the accident avoidance tests totalling to 810 times in the JARI compound as shown in Table 3 employing four different testing vehicles.
Outlines of test results obtained for each individual test shall be given in the following:

Table 3 Summary of NISSAN ESV Accident Avoidance Tests

Testing	Vehicle	Total number of tests
1. Visibility		
Head-lighting	NESV-6	1
Signaling System	NESV-4	1
Windshield Wipers	NESV-1, 6	2
Washer	NESV-1	1
Defrost-Defog	NESV-6	1
Forward and Side visibility	NESV-6	1
Rear Visibility	NESV-4	1
2. Display and Control	NESV-1, 6	1
3. Braking Performance		
Stopping distance	NESV-1	40
Brake Effectiveness	NESV-1	80
Fading	NESV-1	1
Braking Efficiency	NESV-1	140
Parking Brake	NESV-1	5
4. Steering		
Yaw Response	NESV-5	200
Returnability	NESV-5	20
5. Handling		
Lateral Acceleration	NESV-5	120
Control at Breakaway	NESV-5	40
Crosswind Sensitivity	NESV-5	30
Pavement Irregularity	NESV-5	30
Steering Control Sensitivity	NESV-5	30
6. Overturning Immunity		
J-Turn	NESV-1	20
Slalom	NESV-5	10
7. Engine		
Emission	NESV-1	1
Vehicle Range	NESV-1	6
Passing Performance	NESV-1	10
Lateral Force Influence	NESV-1	6
8. Bumper Underride Override	NESV-1	12
Total	4 Vehicles	810 times

3.1 Visibility

We performed tests on driver's field of view, clear glazing maintaining device and lighting system. For testing of forward vertical viewing angles, number of pillars, visibility affecting dismensions, of pillar and rear visibility (fields of view of mirrors), we employed the screen projecting method in accordance with JASO 6824 and took measurements basing upon the eyellipse recommended by SAE J 941c (Figure 3). Table 4 represents results with respect to the driver's fields of view. Some testing vehicles were attached a periscope for assuring the indirect rear visibility. Those vehicles were not equipped with rear view mirrors.

We tested the windshield wipers on the wiping area and frequencies in accordance with the test procedures prescribed in SAE J903b and the windshield washer on the system performance in accordance with the SAE J942 and the defroster on the defrosting area in a low-temperature chamber of -18°C in accordance with the SAE J902a. Table 5 represents the outstanding results of these tests.

Table 4 Report of Driver's Field of View Tests

ESV specifications	Test results
1. Minimum vertical angles through windshield using SAE's 99th or 95th percentile eyellipse: Up = 13°, Down = 6°	Upward 99th 12.1° 95th 14.8° Downward 99th 5.1° 95th 6.4°
2. No more than 4 pillars in 270° arc from forward	4
3. Inside mirror (1) Minimum horizontal angle using the eyellipse centroid: 22° (2) In the upper field of view, the horizon shall be visible and in the lower field of view, the road surface within 15 meters behind the rear end of vehicle shall be visible.	28.4° Upward 1.0° Visible range 13.3 m
4. Outside mirror on the left side shall meet the requirements of Section 44 of Safety Regulations of Japan	Yes
5. Outside mirror on the right side (driver's side) (1) Minimum horizontal angle: 12° (2) Road surface visible 10 meters behind outside mirror	Horizon monocular 9.6° binocular 13.5° Visible range monocular 8.6 m binocular 8.0 m
6. Inside mirror and outside mirror on the right side (driver's side) adjustable from driver's seat	Yes
7. Periscope (1) Horizontal viewing angle (2) Road surface visible range	Monocular 34.2° Monocular 15.6m

Table 5 Report of Tests of Clear Glazing Maintaining Device

ESV specifications	Test results
1. Windshield wiper Wiping area A: more than 80 % B: more than 94 % C: more than 99 %	 A: 92 % B: 95 % C: 100 %
2. Washer 75 % of wiping area	100 %
3. Defroster Defrosting area after 30 minutes A: 80 % B: 95 % C: 100 %	 A: 100 % B: 100 % C: 100 %

Table 6 Report of Tests of Lighting and Signaling

ESV specifications	Test results
1. The maximum luminous intensity of the head lamp shall not exceed 150,000 cd	143,000 cd
2. Separation of function for brake and turn	Yes
3. Side turn signal lamps	Yes
4. Lamp failure indicators	Yes
5. Passing signal light	Yes

In the measurements of headlamps, we obtained the illuminance values of beams on a vertical plane located 10 meters in front of lamp projected from the vehicle installed in a dark room. We took measurements on individual lamp in the range of 15L-15R, 3U-3.5D and after summing up these individual measured values, we obtained the maximum luminous intensity. The operating voltage was 12.8 V. We adopted the visual aiming method (SAE J579a) for aiming. Table 6 represents the results with respect to lighting and signaling tests. The headlamp was incorporated with three different illuminations of high beam, dimmer beam and city beam. The signaling system had similarity with the current system, however, the dual intensity rear lighting system was adopted for better visibility. The trail lamp was also mounted that indicates operation of the accelerator to the driver of the following vehicle.

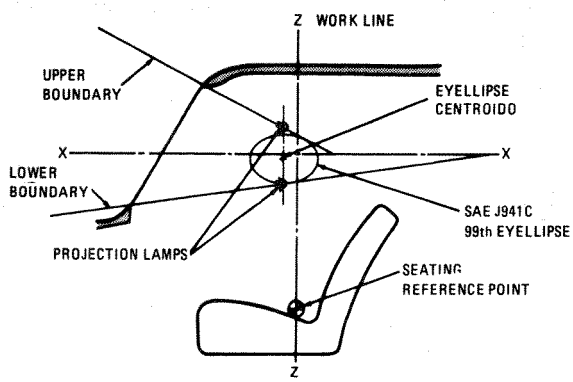
3.2 Display and Controls

We assigned 29 drivers of different heights (146 ~ 192 cm) to drive the vehicle for 30 minutes each in the day time and in the night time respectively and collected their subjective appraisals as to the consideration given by the designers of the vehicle on elements of human engineering, such as easiness of handling and operation, etc.

In these appraisals, they were giving high credit to the excellent driving position and speaking favorably of the position of steering wheel. Further, easiness and dependability of operation of accelerating and braking pedals were receiving favorable criticism. Instrument

panel got good marks for daytime driving, however, a little short of illumination was suggested for the nocturnal drive. The periscope was said to be ensuring an extensive rear view, however, some drivers pointed out some lateral aberration of image.

Fig. 3 Field-of-View Construction Lines and Projection Lamp Location



3.3 Braking Performance

3.3.1 Stopping Distance

In the brake stopping distance tests, we conducted the tests by allowing the test vehicle to be braked from the initial velocity of 95 km/h with the brake pedal force within the requirements in the ESV SPECIFICATIONS. The test vehicle did not deviate from the 3.6 m wide lane.

The tests upon failure of the booster, we conducted the tests by releasing the vacuum hose of booster into the atmosphere.

The tests upon a partial failure of the brake system, we conducted the tests by releasing individually the primary brake line and secondary brake line respectively from the master cylinder body.

The test results are listed in Table 7.

Table 7 Stopping Distance Test Results

PARAMETER		ESV SPEC	TEST RESULTS
SYSTEM	NORMAL	50 m MAX.	44.04 m
PARTIAL FAILURE	PRIMARY CIRCUIT FAILED	120 m MAX.	70.08 m
	SECONDARY CIRCUIT FAILED		70.21 m
SERVO	FAILURE	105 m MAX.	50.48 m

3.3.2 Brake Effectiveness

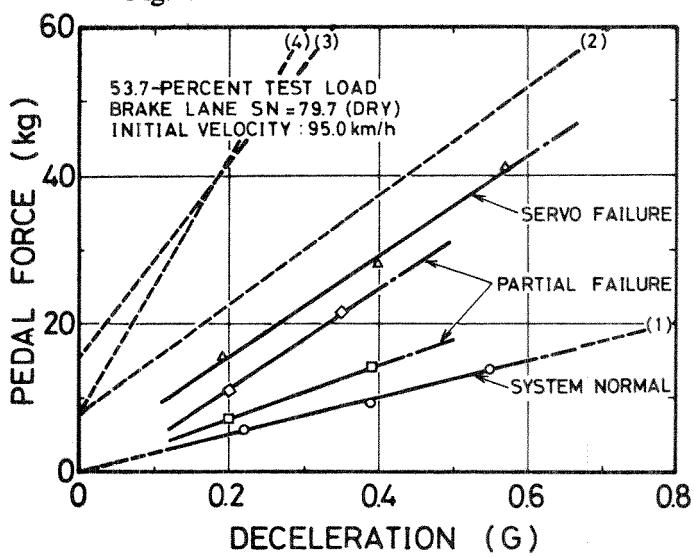
In the brake effectiveness tests, we covered the following three component tests: tests of normally operating system, system with a booster failure and system with a partial failure. We performed 20 tests for each component test which can be broken down to 5 times the 4 tests with the target deceleration of 0.2G, 0.4G, 0.6G and the highest deceleration

respectively. The initial braking velocity was set at 95 km/h and the test driver performed the braking test maintaining the constant deceleration by watching the deceleration meter.

The ESV SPECIFICATIONS on the brake effectiveness calls for the range between (1) and (2) for (a) test of normally operating system between (1) and (3) for (b) test of system with a booster failure and between (1) and (4) for (c) test of system with a partial failure, as shown in Fig. 4.

The test results are shown in Fig. 4.

Fig. 4 Brake Effectiveness Test Results



3.3.3 Fading

We performed this test in accordance with the testing procedure as prescribed in the SAEJ843b, however, the load was limited to 53.7 % of the full load.

The results are listed in Table 8.

Table 8 Fade Test Results

PARAMETER	ESV SPEC	TEST RESULTS
FADE	60 kg MAX.	12.7 kg
RECOVERY	60 kg MAX.	7.9 kg

3.3.4 Braking Efficiency

In the braking efficiency tests, we performed the tests on different road surfaces specified by the skid resistance numbers ranging from 33.2 to 78.8 (at 20°C) by measuring the stopping distances in locked wheel condition from the initial braking velocity of 95 km/h and the minimum available stopping distances in unlocked wheel condition from the same initial braking velocity.

The results of the measurements are represented in Table 9.

Table 9 Braking Efficiency Test Results

TEST GROUND	SN (AT THE TEST) SURFACE CONDITION	TEST LOAD PERCENT(%)	STOPPING DISTANCE (m) ANTILOCK (D_a)	STOPPING DISTANCE (m) LOCK (D_l)	BRAKING EFFICIENCY ESV SPEC (%)	BRAKING EFFICIENCY $E_l = D_l/D_a \times 100$ (%)
CONCRETE (HIGH SPEED TRACK)	77.4 (DRY)	53.7	44.04	45.48	≧ 80	103.3
	56.6 (WET)	53.7	50.79	65.65		129.3
POLISHED CONCRETE (i)	48.7 (WET)	53.7	72.03	79.10		109.8
(SLIPPERY TEST TRACK) (ii)	32.4 (WET)	53.7	120.34	106.08		88.2

The definition of the braking efficiency set forth in the Japanese ESV SPECIFICATION is a ratio of the stopping distance in locked wheel condition to that in unlocked wheel condition as expressed by the following calculation:

$$\text{Braking efficiency} \quad E_\ell = \frac{D_\ell}{D_a} \times 100\ (\%)$$

where D_ℓ = stopping distance in locked-wheel condition (m)
D_a = stopping distance in unlocked-wheel condition (m)

NOTE:

(1) NISSAN ESV is installed the antilocking system for rear wheels.

(2) In the Japanese ESV SPECIFICATIONS, the braking efficiency differ from the brake system efficiency of the United States. Incidentally, for this brake system efficiency, reference is made to Technical Session Paper "The Characteristics of Japanese ESV Tires."

3.3.5 Parking Brake

We performed the parking brake tests on a 31 % grade test slope. The testing procedures were to drive the vehicle into the test slope aligning the vehicle center line parallel with the center line of the test slope and allowed it to stop by employing the service brake in both uphill and downhill conditions. We limited the brake operating force to 40 kg, when we used the parking brake hand lever and held the vehicle at a standstill for five minutes. We conducted this particular test five times for each condition.

The results are listed in Table 10.

Table 10 Parking Brake Test Results

PARAMETER	ESV SPEC	TEST RESULTS
UP HILL	40 kg MAX.	35.7 kg
DOWN HILL	40 kg MAX.	26.8 kg

3.4 Steering

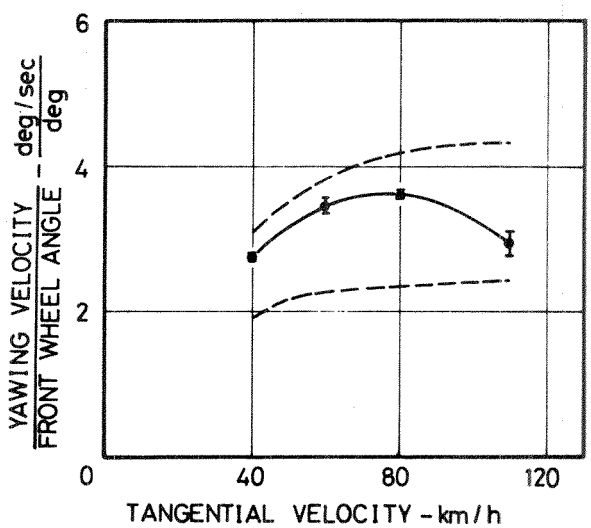
3.4.1 Steady State Yaw Response

In the steady state yaw response tests, we allowed the test vehicle to maintain for at least three seconds the steady state circular running path at the specified velocity and lateral acceleration. The testing velocities were 40, 60, 80 and 110 km/h. We conducted 10 lefthand and 10 righthand turning respectively for the lateral acceleration of 0.4 G and 5 each lefthand and righthand turning tests respectively for that of 0.3 G and 0.5 G.

Additionally, for the NISSAN ESV, we performed these tests with the front wheel angle defined as steering wheel angle/overall gear ratio.

The test results are shown in Fig. 5.

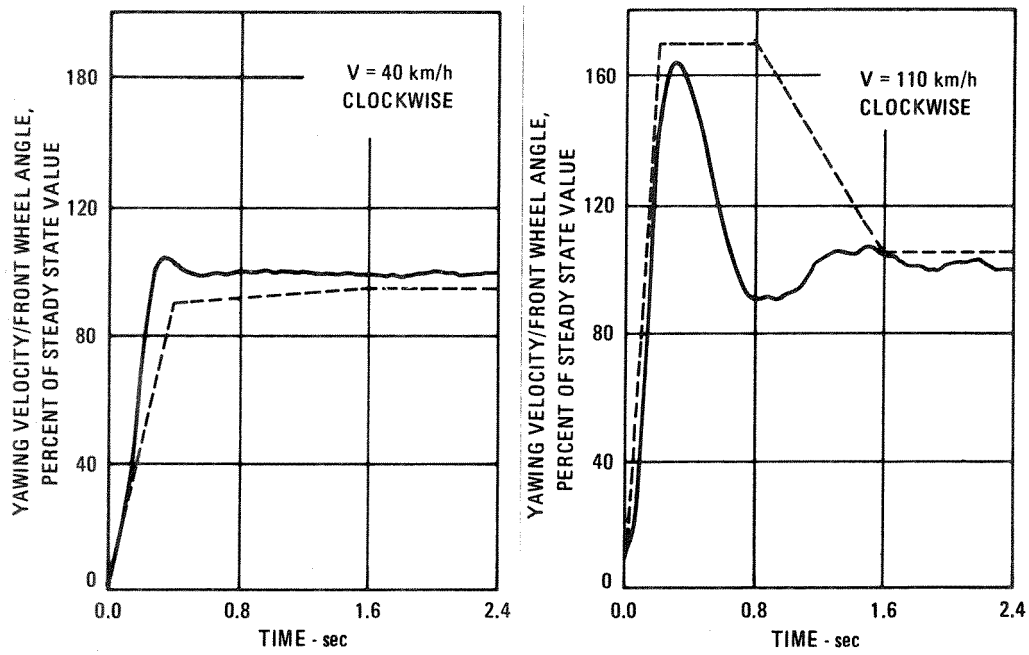
Fig. 5 Steady State Yaw Response Test Results



3.4.2 Transient Yaw Response

In the transient yaw response tests, we allowed the test vehicle to run on a straight line running path at the testing velocities of 40 and 110 km/h, providing the vehicle with the steering input at the rate of more than 500 deg/sec up to the predetermined angle and maintaining it to produce the lateral acceleration of 0.4G for a constant time. We conducted 10 lefhand and 10 righthand turning tests respectively for each vehicle velocity. The test results are shown in Fig. 6.

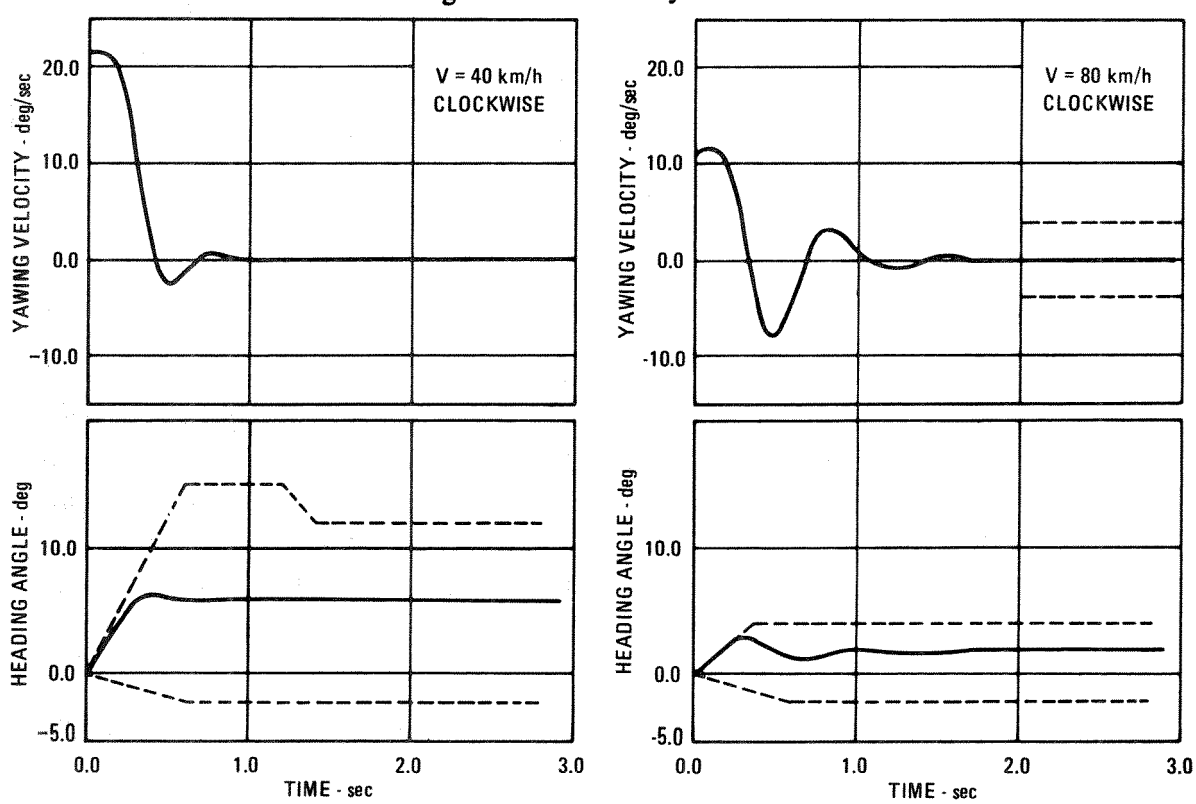
Fig. 6 Transient Yaw Response Test Results



3.4.3 Returnability

In the returnability tests, we drove in the steady state the test vehicle at the testing velocities of 40 and 80 km/h along a circular running path sized to achieve the lateral acceleration of 0.4G and released completely the steering wheel at the predetermined point of the vehicle path. We conducted the 5 lefthand and 5 righthand turning tests respectively. The test results are shown in Fig. 7.

Fig. 7 Returnability Test Results

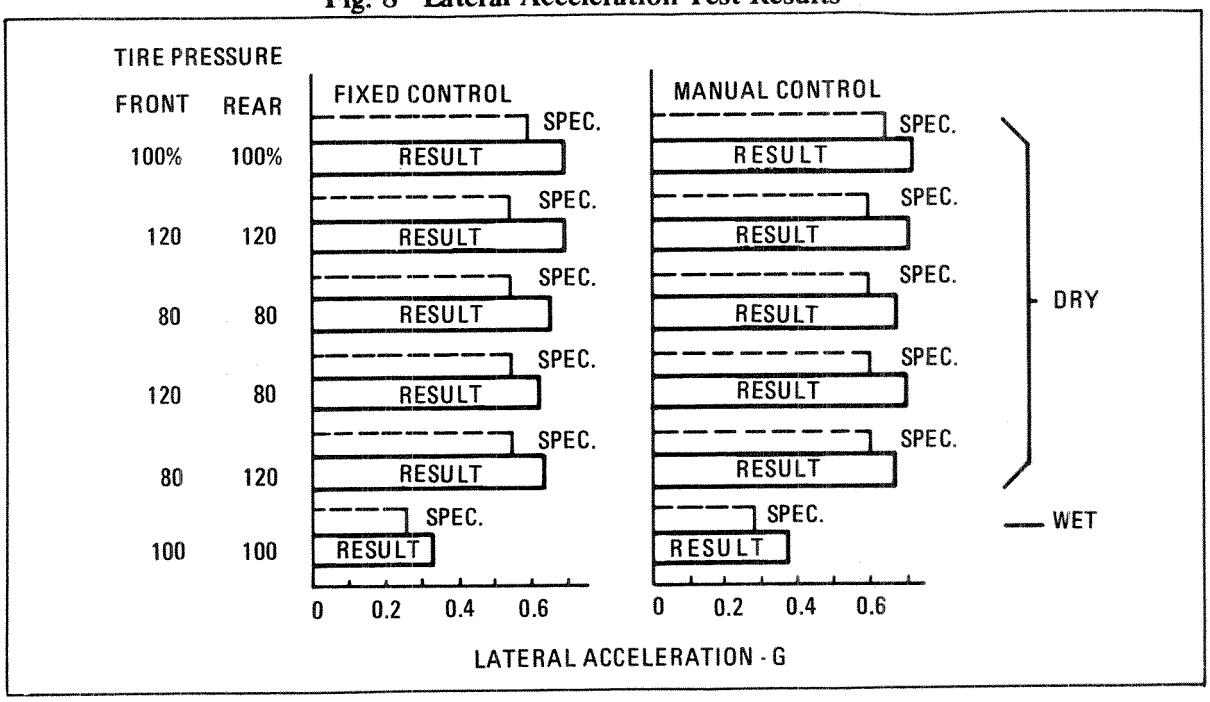


3.5 Handling

3.5.1 Lateral Acceleration

In the lateral acceleration tests, we drove the test vehicle along a circular running path with the radius of 30 m, accelerating the velocity until the highest lateral acceleration is achieved. Handling of the steering wheel is divisible into two different controls; the fixed control and manual control in which the driver operates the steering wheel in a way he thinks it necessary to do the same. Further, as in JARI, we treat the value prescribed in the SPECIFICATIONS as the minimum requirement and we set an objective of our present test to exceed this particular requirement value, the results achieved did not necessarily turn out the limit optimum value. Fig. 8 represents the results of the test.

Fig. 8 Lateral Acceleration Test Results

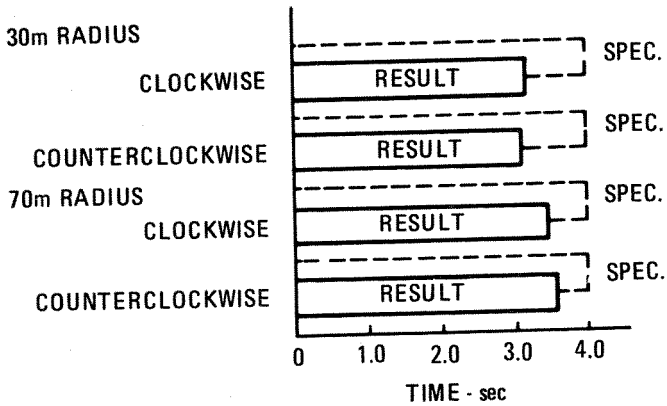


3.5.2 Control at Breakaway

In the control at breakaway tests, we allow the test vehicle to operate in the steady state along the running paths with the radius of 30 and 70 m at the velocity which produce the highest lateral acceleration(*) with the fixed control of steering wheel, accelerate the velocity gradually and started our test by closing the throttle at the point when the vehicle center line within the wheelbase moved radially outward by the distance of 3 m from the initial path. Further, for indication of the point where the track breakaway from the initial path and returned to it, we employed a white-line indicator. Fig. 9 represents the results of the tests.

* The value prescribed in the SPECIFICATIONS for the lateral acceleration test is 0.60 G.

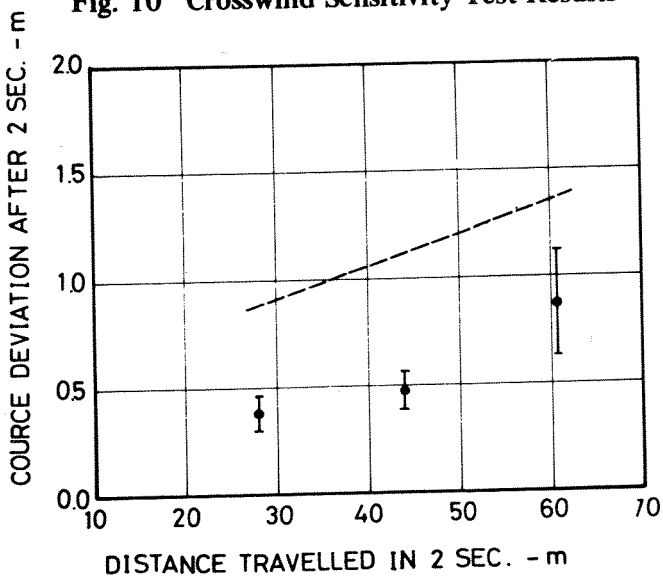
Fig. 9 Control at Breakway Test Results



3.5.3 Crosswind sensitivity

In the crosswind sensitivity tests, we drove the test vehicle into the straight line running path with the fixed steering wheel at the velocity of 50, 80 and 110 km/h respectively, letting the vehicle pass in front of a crosswind generator. The running track for two seconds after the vehicle began to run under the influence of the crosswind was measured with a jet track tracing apparatus. The test results are shown in Fig. 10.

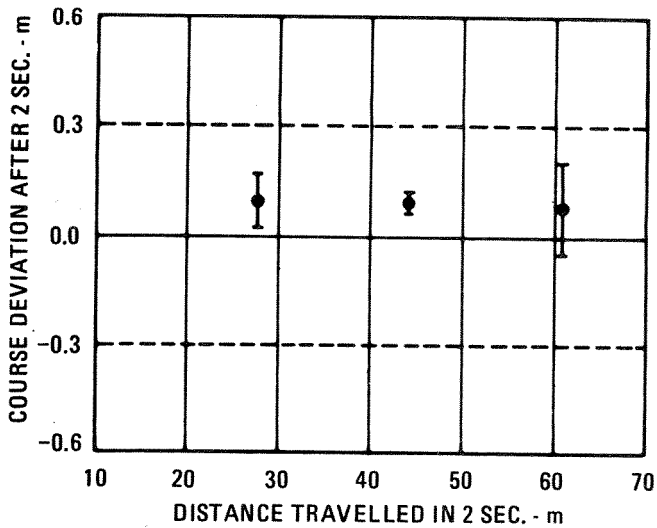
Fig. 10 Crosswind Sensitivity Test Results



3.5.4 Pavement Irregularity Sensitivity

In the pavement irregularity sensitivity tests, we drove the test vehicle along a running course provided with the predetermined surface irregularity at the velocities of 50, 80 and 110 km/h respectively. We measured the distance of deviation from the course after two seconds of contact of the test vehicle with a road bump. The test results are shown in Fig. 11.

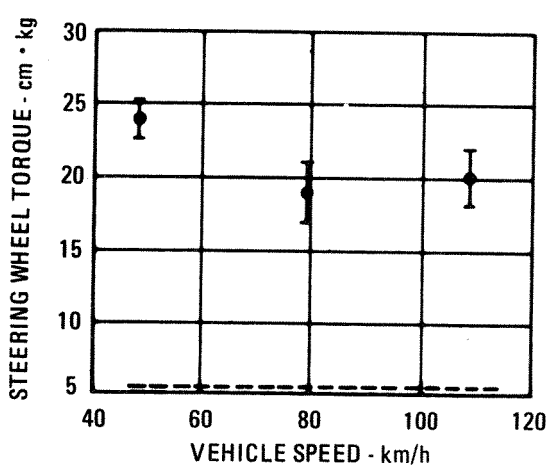
Fig. 11 Pavement Irregularity Sensitivity Test Results



3.5.5 Steering Control Sensitivity

In the steering control sensitivity tests, we operated the steering wheel in such a way that the yawing angular velocity changes gradually in the neighbourhood of 2 deg/sec at the velocities of 50, 80 and 110 km/h respectively and measured the values of steering torque which corresponded to the yawing angular velocity of ±2 deg/sec at this moment. Further, the NISSAN ESVs involved are all equipped with the manual control system and not with the power steering system. The test results are shown in Fig. 12.

Fig. 12 Steering Control Sensitivity Test Results



3.6 Overturning Immunity

3.6.1 J-Turn

In the J-turn tests, we drove the test vehicle with an automatic steering system for prevention of a hazardous accident. When the vehicle in operation at the velocities of 50, 80 and 110 km/h respectively on a straight line course arrives at the predetermined point on the course, a pair of trigger systems composed of elements on board the vehicle and on

the ground is actuated to steer the vehicle with more than 180 deg and 500 deg/sec and simultaneously the throttle is closed to initiate the test.

In the J-turn tests, the test vehicle did not overturn in any of the three lefthand and righthand turning tests respectively at the velocities of 50, 80 and 110 km/h.

3.6.2 Slalom

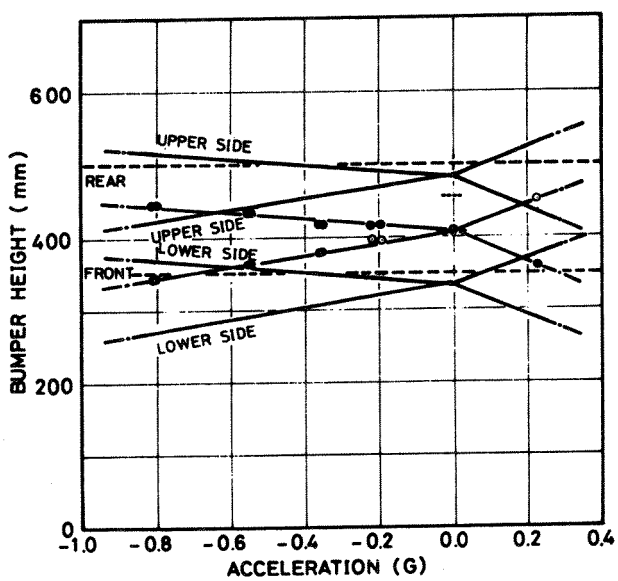
In the slalom course test, we conducted this test in allowing the test vehicle to run in slalom at a velocity of 80 km/h on the running course studded with 11 pylons arranged with the spacing of 30 m. In this test, the velocities of drive-in time and mean value of velocities running along the pyloned section are not supposed to be less than the specified values respectively.

In the slalom tests, the test vehicle did not overturn in any of ten driving tests.

3.7 Bumper Under ride/Over ride

The bumper under ride/override tests have been performed with some decelerations from 95 km/h and some accelerations from the low gear. (NISSAN ESV is installed the manual shift transmission.) The test results are shown in Fig. 13.

Fig. 13 Bumper Under ride/Over ride Test Results



4. Collision and Roll-over Tests

(1) We designated front end impact, rear end impact, side impact and roll-over, as four basic impact modes for differentiating effectiveness of the crash injury reduction of occupant.

(2) We performed the flat barrier tests for the front end impact, moving barrier test (1820 kg) for the rear end impact, pole/side impact test for the side impact and dolly test for the roll-over.

(3) The test items and the test conditions are shall be given in the following:

Table 11 Summary of Collision and Roll-over Test Items and Conditions

IMPACT MODE	TESTING VEHICLE	VEHICLE SYSTEM	TARGET SPEED (km/h)	TEST SPEED (km/h)	NUMBER OF DUMMIES	TEST WEIGHT (kg)
FLAT BARRIER IMPACT	NESV - 1	E_1	$15^{+0}_{-1.0}$	13.1	4	1520
FLAT BARRIER IMPACT	NESV-10	E_1	$80^{+0}_{-1.6}$	78.4	3	1495
FLAT BARRIER IMPACT	NESV - 7	E_2	$80^{+0}_{-1.6}$	79.2	4	1620
REAR END IMPACT	NESV - 9	E_1	$65^{+0}_{-1.3}$	64.3	3	1495
POLE/SIDE IMPACT	NESV - 8	E_1	$25^{+0}_{-1.0}$	22.5	3	1500
ROLL-OVER	NESV - 6	E_1	48	47.9	3	1500

(4) We employed exclusively the anthropomorphic test device VIP 50A manufactured by Alderson.

(5) Cut-off frequency is 250 Hz for the anthropomorphic test device (dummy) and in accordance with SAE J 211a for the other items.

(6) In this summary, we will report mainly the factors relating to the occupant injury protection during car crash, but we intend to present the full report containing the details of other characteristic to the Government of Japan later.

4.1 Test Results

During the course of all the above tests, we did not witnessed even a single case, in which the arthropomorphic test device (dummy) was ejected out of the vehicle.

The results of each individual test with its findings run as follows:

4.1.1 15km/h Flat Barrier Impact Test

In this test, we installed 4 anthropomorphic test devices fastened with lap belt system as the restraint devices and conducted the subjected impact test.

The measured testing speed was less than the target value recording 13.1km/h. There was a partial damage at lower part of radiator grille. All the component units related to the vehicle operation remained unscathed and no irregular running performance was witnessed in the running test after the impact test.

We took measurement of the bumper stroke at the moment of impact by means of a photographic measurement method and the value was 150 mm.

This test car was employed on avoidance test before this test.

4.1.2 80 km/h Flat Barrier Impact Test. (E1 system)

We performed the subjected test using E1 system in a condition satisfying the specified test conditions and a testing speed of 78.4 km/h, deviation of impact angle not exceeding 1 degree and error of impact position with respect to the target point of 60 mm. Fig. 14 represents a left side view of the post-test vehicle.

Fig. 14 Post-test View of NISSAN ESV (E1 System) Following 80km/h Flat Barrier Impact

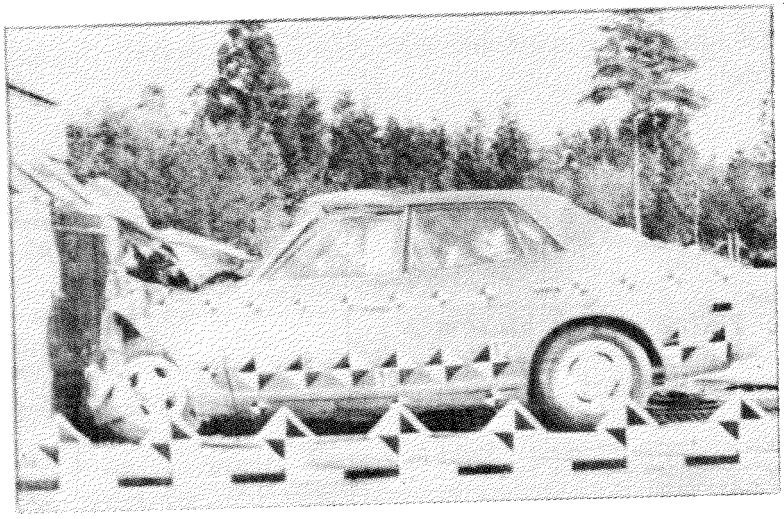


Fig. 15 represents the time history of the resultant head acceleration of dummies and vehicle floor acceleration (longitudinal direction)

Fig. 15 Head G–80km/h Flat Barrier Impact (E1 System)

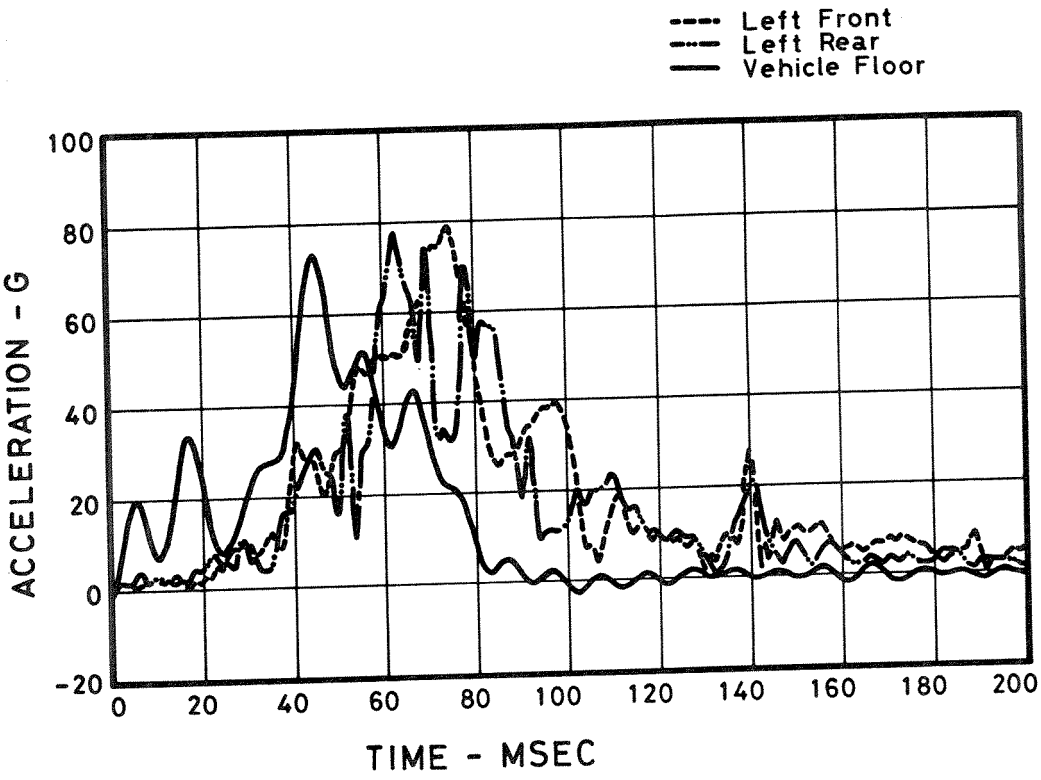


Fig. 16 represents the time history of the resultant chest acceleration of dummies and Fig. 17 represents a bar chart of femur load.

Concerning the test device located in the driver's seat, no data were available on the head acceleration, chest acceleration and femur load because of defective measuring processes.

Fig. 16 Chest G–80km/h Flat Barrier Impact (E1 System)

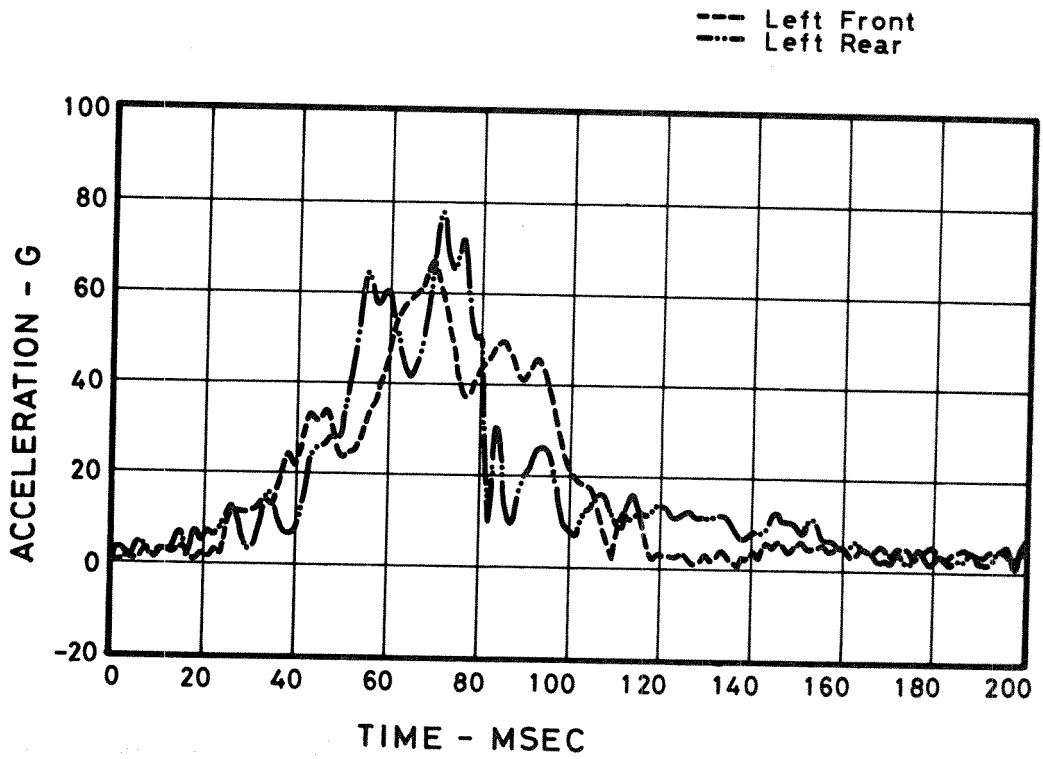
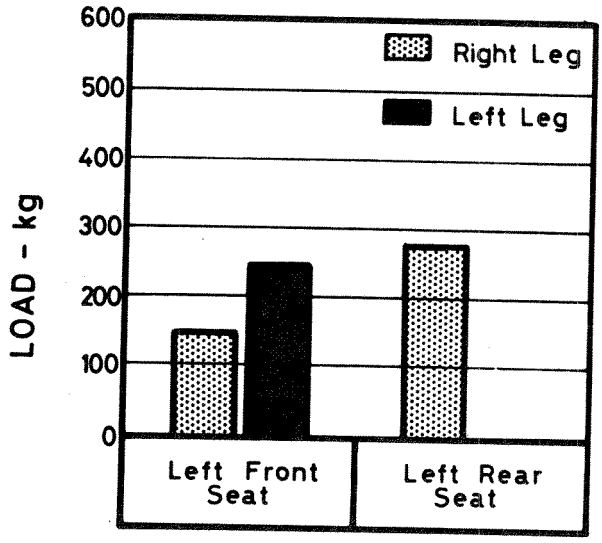


Fig. 17 Femur Loads-80km/h Flat Barrier Impact (E1 System)



Data with respect to vehicle are shown in table 12 and those to anthropomorphic test devices in table 13.

(1) Data with respect to vehicle

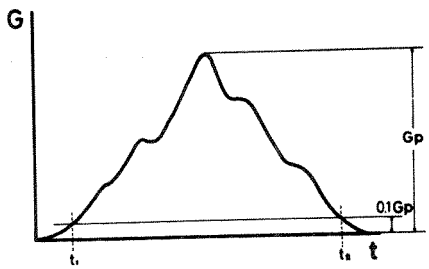
Table 12 Vehicle Structural Response-80km/h Flat Barrier Impact (E1 System)

Peak Floor G (G)	73
Average Floor G ※ (G)	30
Residual Crush (mm)	875
Compartment Intrusion (mm)	30

The position of the measured maximum compartment intrusion was located at the fire wall.

※ The average floor $\bar{G}_f$ (longitudinal direction) was calculated by the following expression.

$$\bar{G}_f = \frac{1}{t_2 - t_1} \int_{t_1}^{t_2} G(t)\, dt$$



Here

G (t) : time history of longitudinal acceleration of vehicle

Gp : Peak value of G (t)

t_1, t_2 : Points of intersection of G (t) and G = 0.1 Gp.

(2) Data with respect to test devices

Table 13 Dummy Dyanamic Response-80/km/h Flat Barrier Impact (E1 System)

		Right Front (Driver)	Left Front	Left Rear
Head	Peak G (G)		78	77
	Time Over 80 G (ms)		—	—
Chest G (G)			67	77
Femur Peak Load	Left (kg)		250	
	Right (kg)		150	280

4.1.3 80 km/h Flat Barrier Impact Test (E2 system)

We performed the subjected test in a condition satisfying the specified test condition and with a testing speed of 79.3 km/h, deviatiin of impact angle not exceeding 1. degree and error of impact position with respect to the target point of 10 mm.

Fig. 18 represents a left side view of the vehicle after the 80 km/h flat barrier impact test.

Fig. 18 Post-test View of NISSAN ESV (E2 System) Following 80km/h Flat Barrier Impact

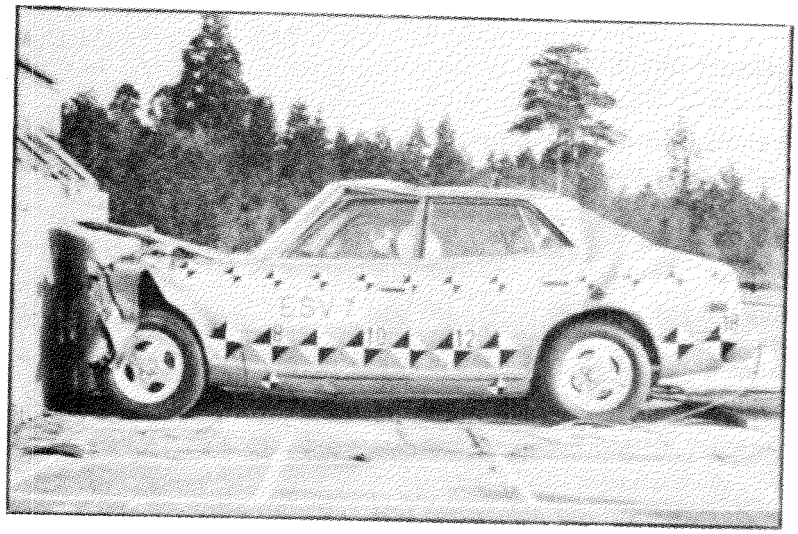
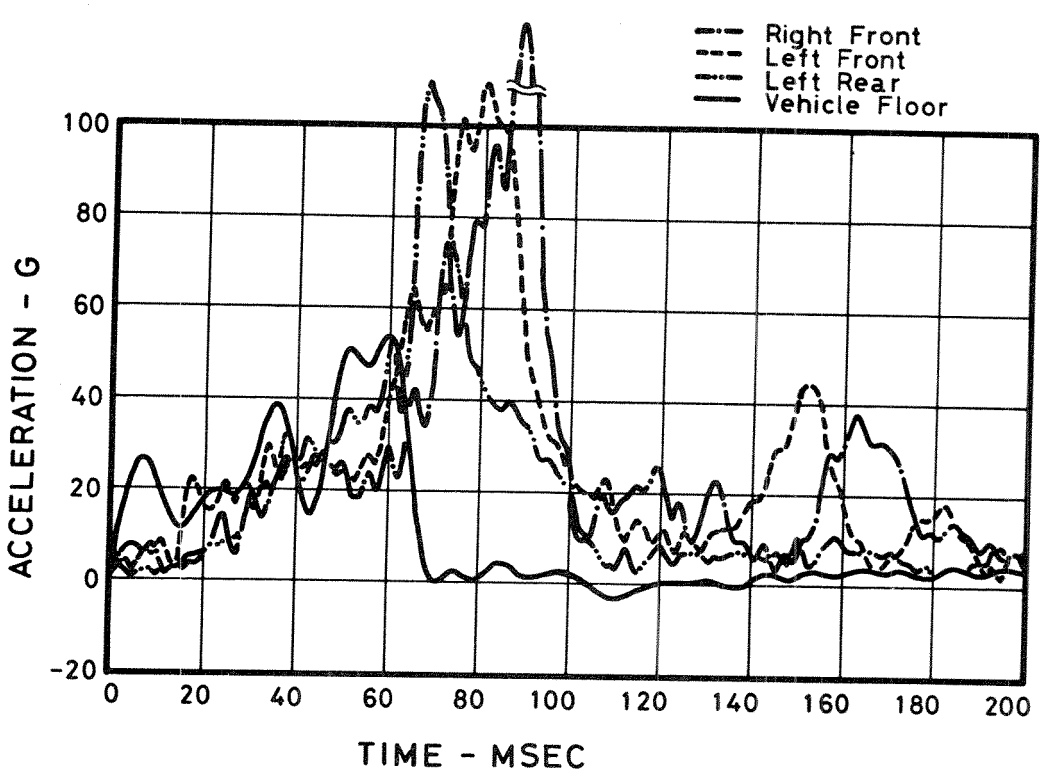


Fig. 19 represents the time history of the resultant head acceleration of dummies and vehicle floor acceleration (longitudinal direction).
As represented in the illustration, the resultant head acceleration ranged from 106 G to 143 G and time duration over 80 G ranging from 7 to 14 millisecond.

This vehicle was equipped with only the lap and shoulder belt system for the passenger restraint system.

Fig. 19 Head G–80km/h Flat Barrier Impact (E2 System)



In this case any dummy head did not cause the second collision against any part of the compartment interiror, nevertheless such high acceleration as over 100 G was recorded. From the fact that the chin hitted against the chest during impact, we consider that the seat belt performance and the neck characteristic of the test device have an important bearing.

Fig. 20 represents the time history of the resultant chest acceleration of dummies and Fig. 21 represents a bar chart showing the dummy femur load.

Fig. 20 Chest G-80km/h Flat Barrier Impact (E2 System)

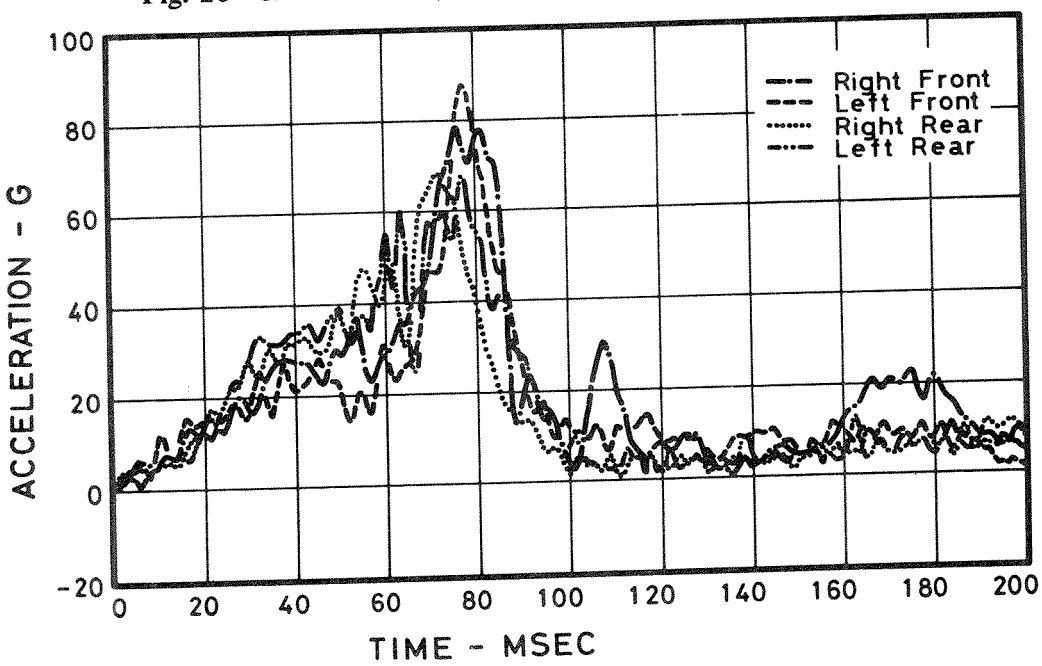
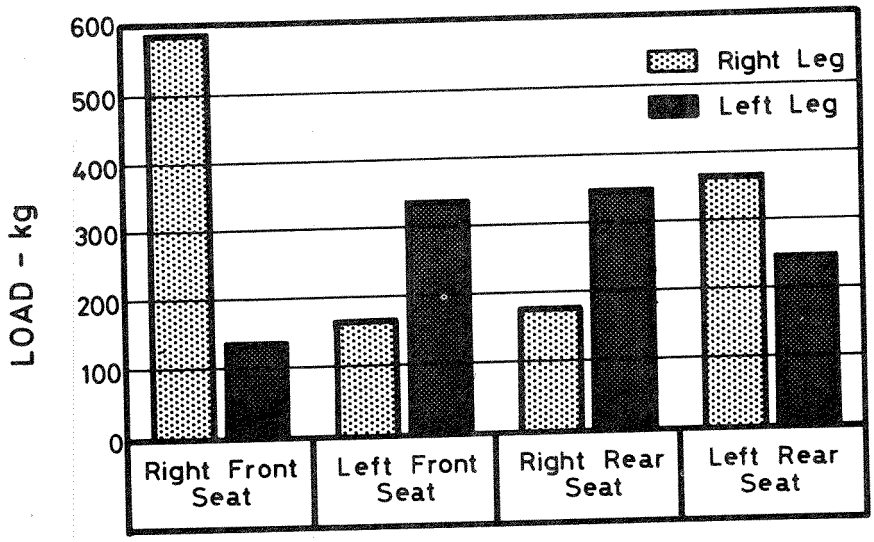


Fig. 21 Femur Loads-80km/h Flat Barrier Impact (E2 System)



Data with respect to vehicle are shown in table 14 and to anthropomorphic test devices in table 15.

(1) Data with respect to vehicle

Table 14 Vehicle Structural Response-80km/h Flat Barrier Impact (E2 System)

Peak Floor G (G)	53
Average Floor G (G)	28
Residual Crush (mm)	860
Compartment Intrusion (mm)	20

The maximum compartment intrusion occurred in the position at the upper edge of the instrument panel.

(2) Data with respect to test devices

Table 15 Dummy Dyanamic Response-80km/h Flat Barrier Impact (E2 System)

		Right Front (driver)	Left Front	Right Rear	Left Rear
Head	Peak G (G)	143	109	106	109
	Time Over 80 G (ms)	12	14	8	7
Chest G (G)		78	87	67	67
Femur Peak Load	Left (kg)	130	340	350	250
	Right (kg)	590	170	180	370

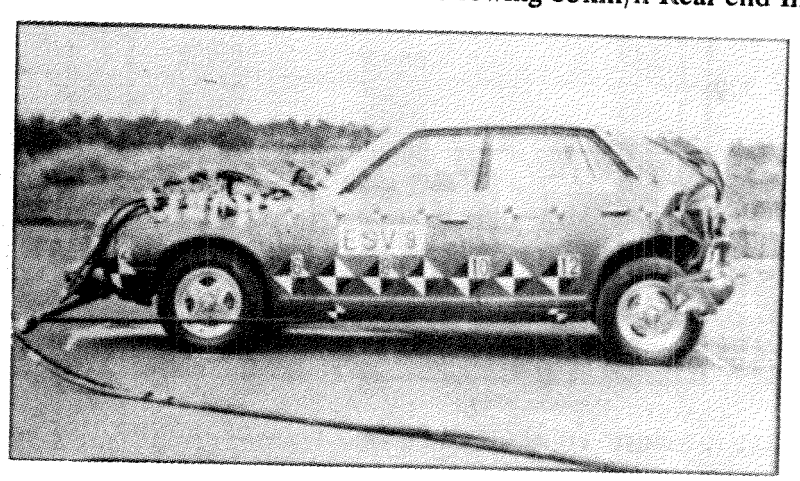
4.1.4 65 km/h Rear End Impact Test

We performed the subjected test with the moving barrier (weight 1820 kg) in accordance with the SAE J972 and with a testing speed of 64.4 km/h and with the deviation of impact position 20 mm.

The braking performance of the moving barrier turned out satisfactory also; collision between the testing vehicle and moving barrier never occurred more than one time.

Fig. 22 represents a vehicle after the 65 km/h rear end impact test.

Fig. 22 Post-test View of NISSAN ESV Following 65km/h Rear-end Impact



Data with respect to vehicle are shown in table 16 and to anthropomorphic test devices in Table 17.

(1) Data with respect to vehicle

Table 16 Vehicle Structural Response-65km/h Rear-end Impact (E1 System)

Peak Floor G	(G)	
Average Floor G	(G)	
Residual Crush	(mm)	490
Compartment Intrusion	(mm)	25

(2) Data with respect to test devices

Table 17 Dummy Dyanamic Response-65km/h Rear-end Impact (E1 System)

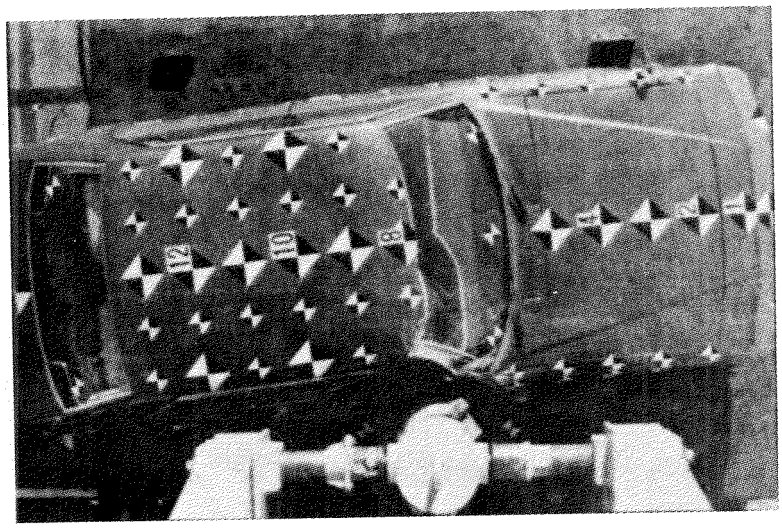
		Right Front (driver)	Left Front	Right Rear
Head	Peak G (G)	50	57	43
	Time Over 80G (ms)	–	–	–
Chest G	(G)	25	45	41
Femur Peak Load	Left (kg)	130	170	200
	Right (kg)	140	220	160

4.1.5 25 km/h Pole/Side Impact Test

We performed the subjected test with a test vehicle installed on the dolly and with the impact target of the righthand door maximum opening center. The deviation of impact position with respect to the target point point was 20 mm.

The test speed decreased to 22.5 km/h, because of strong wind during testing. Fig. 23 represents the condition of vehicle after the test.

Fig. 23 Post-test View of NISSAN ESV Following 25km/h Pole/Side Impact



Data with respect to vehicle are shown in table 18 and to anthropomorphic test devices in Table 19.

(1) Data with respect to vehicle

Table 18 Vehicle Structural Response-25km/h Pole/Side Impact (E1 System)

Peak Floor G (G)	16
Average Floor G (G)	10
Residual Crush (mm)	215
Compartment Intrusion (mm)	150

The position, where the maximum intrusion into the compartment was shown, was the center of the maximum opened section of the door with the value around the head of test dummy being 55 mm.

(2) Data with respect to test devices

Table 19 Dummy Dyanamic Response-25km/h Pole/Side Impact (E1 System)

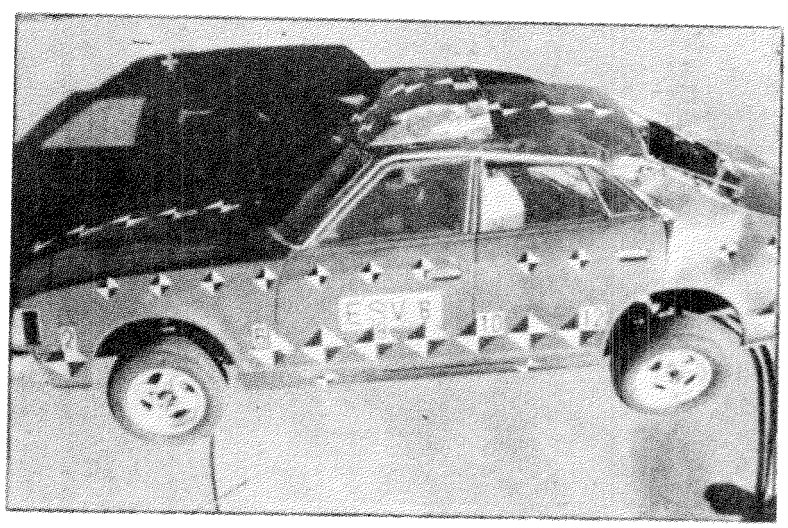
		Right Front (driver)	Left Front	Right Rear
Head	Peak G (G)	47	19	54
	Time Over 80G (ms)	–	–	–
Chest G (G)		23	22	21
Femur Peak Load	Left (kg)	60	40	30
	Right (kg)	30	50	30

4.1.6 Roll-over Test

We performed the subjected test with a method employing the dolly in conformity with FMVSS No. 208, the testing speed and number of roll-overs being 47.9 km/h and 3¼ respectively.

Fig. 24 represents a view from above of the final pose of the vehicle as it overtuned side ways.

Fig. 24 Post-test View of NISSAN ESV Following Roll-over



Data with respect to vehicle are shown in table 20 and to anthropomorphic test devices in table 21.

(1) Data with respect to vehicle

Table 20 Vehicle Structural Response-Roll-over (E1 System)

Peak Floor G (G)	23
Compartment Intrusion (mm)	85

The maximum compartment intrusion occurred in the position of lefthand C-pillar.

(2) Data with respect to test devices

Table 21 Dummy Dyanamic Response-Roll-over (E1 System)

		Right Front (driver)	Left Front	Right Rear
Head	Peak G (G)	23	26	35
	Time Over 80G (ms)	—	—	—
Chest G (G)		19	15	18
Femur Peak Load	Left (kg)	80	30	40
	Right (kg)	40	50	250

SECTION 3

TECHNICAL SEMINARS

PART ONE ACCIDENT ANALYSIS

Themes: Data Collection; Use of Data for Car Safety; Examples of Particular Results

M. HALPERN-HERLA
Introduction

DONALD F. MELA
United States Overview

M. HALPERN-HERLA
European Overview

Mr. NAGAKUNA
Japanese Overview – Traffic Accidents in Japan

United States Technical Papers

JAMES O'DAY
A Proposed New National System for Collecting Multipurpose Accident Data: SIR

BRIAN O'NEILL, HANS JOKSCH, and WILLIAM HADDON, Jr.
Emperical Relationships Between Car Size, Car Weight, and Crash Injuries in Car-to-Car Crashes

JOHN W. GARRETT and DONALD L. HENDRICKS
Factor Influencing the Performance of the Energy Absorbing Steering Column in Accidents

DONALD F. MELA
NHTSA's Evaluation of Air Cushion Restraint System Effectiveness (ACRS)

European Experimental Vehicles Committee Technical Papers

F. HARTEMANN and C. TARRIERE
Synthesis of Statistical Data on Traffic Accidents in France, West Germany, Italy and United Kingdom

J. C. BLUET
The Road Safety Checklist – A Periodical Inquiry Regarding Traffic

DR. M. DANNER and K. LANGWIEDER
The Frequency of Corresponding Vehicle Damage in Crash Tests and Actual Accidents

BARBARA E. SABEY and I. D. NEILSON
Accident Investigation as an Aid to Planning the Future of Car Safety

E. GRATTAN and J. A. HOBBS
Some Patterns and Causes of Injury in Car Occupants

PART TWO HUMAN FACTORS

Themes: Human Tolerance Levels to Injury; Use of Dummies and Similar Devices; Impact Testing and Accident Injuries

B. ALDMAN
Introduction

United States Technical Papers

ARTHUR E. HIRSCH
NHTSA Programs in Biomechanics

J. A. TENNANT, R. H. JENSEN, and R. A. POTTER
GM-ATD 502 Anthropomorphic Dummy – Development and Evaluation

LOUIS C. LUNDSTROM, R. A. WILSON, and GEORGE R. SMITH
Relating Air Cushion Performance to Human Factor and Tolerance Levels

European Experimental Vehicles Committee Technical Papers

J. G. WALL and R. W. LOWNE
Human Injury Tolerance Level Determination from Accident Data using the OPAT Dummy

C. TARRIER, G. STCHERBATCHEFF, P. DUCLOS, and A. FAYON
Pedestrian Protection

D. CESARI and M. RAMET
Biomechanical Study of Side Impact Accidents

PETER WARNER
The Development of the OPAT DUMMY

C. M. HASLEGRAVE and M. A. MACAULAY
Performance Measures of the OPAT DUMMY

V. J. JEHU
Towards Pedestrian Safety

AUTO RESTRAINT SYSTEMS, LTD.
A Review of Development of Passive Restraint Systems

JOHN PICKARD
Reduced Laceration from a New Laminated Windshield

Japanese Technical Papers

YOSHIO SERIZAWA
Evaluation of Seat Belt System and Dummy Characteristics

PART THREE VEHICLE STRUCTURAL PROPERTIES

Themes: Design for Occupant Protection; Protecting Other Road Users; Energy Management

EDWARD M. CHANDLER
United States Overview

United States Technical Papers

A. B. KELLEY
RSV, Crash Hazards and Public Support

WILLIAM RUPP
Front Energy Management Parametric Variation Study

SOL DAVIS
Analysis of Test Results for AMF/FIAT ESV Head-on Collision

JEROME M. KOSSAR
Big and Little Car Compatibility

JONATHAN T. LANGE
Flexible Exterior Automotive Trim

European Experimental Vehicles Committee Technical Papers

ULRICH SEIFFERT
European Overview – Compatibility on the Road

P. VENTRE
Compatibility Between Vehicles in Frontal and Semi-Frontal Collisions

J. ELSHOLZ
Relationship Between Vehicle Front-End Deformation and Efficiency of Safety Belts During Frontal Impact

EDMUND HELLRIEGEL and AXEL RAUTHMANN
The Ford Automotive Safety Belt System

E. S. KIEFER
Structural and Occupant Protection Systems of the Opel Safety Vehicle

DR. L. ROSTI-ROSSINI
Some Considerations of Body Structure Crushability in Relation to Impact Speed

GAETANO DANESE
A Research Study for an Energy Absorbing Sliding Seat

P. WATSON
Some Methods of Absorbing the Energy of Motor-Vehicles and Their Occupants

PETER M. FINCH
Vehicle Compatibility in Car-to-Car Side Impacts and Pedestrian-to-Car Frontal Impacts

W. C. EMMERSON and J. E. FOWLER
The Application of Computer Simulations in Vehicle Safety

K. DEUTSCH, M. G. J. ELSON, and R. W. MOORE
The Role of Extrusion Devices in Energy Absorption for Safety

DR. D. A. NEWTON
The Dunlop MK 11 Composite Energy Absorbing Bumper System

R. S. MILLMAN and M. J. CHAPPELL
Energy Absorption by Variable Shear Strength Duplex Materials

JOHN PICKARD
Reduced Laceration from a New Laminated Windshield

W. EGGLESTONE and G. D. SUTHURST
Occupant Protection in Frontal Impacts: A Static, Passive, Restraint System

D. BURLAND
Occupant Protection in Rear Impact

K. STONE
Occupant Protection During Vehicle Rollover

G. PERSICKE, F. I. PLANT, and J. R. CHILD
A Practical Approach to the Protection of Motor Vehicles by the Absorbtion of Impact Energy

I. M. THOMSON
Development and Manufacture of Automobile Bumpers

Japanese Technical Papers

MASAHIKO SAKIMURA
Reviews on Accuracies and Results of ESV Collision Tests

KAZUD HIGUCHI
Characteristics of Body Energy Absorption and Restraint System

PART FOUR ACCIDENT AVOIDANCE

Themes: Estimation of Effectiveness of Safety Measures; Engineering Systems; Interaction of Driver and Vehicle

DR. G. POCCI
Introduction

United States Technical Papers

FRANCIS A. DILORENZO
United States Overview

GORDON G. HAYES
Determination of Motor Vehicle Characteristics Affecting Driver Handling Performance

RICHARD H. KLEIN
Driver/Vehicle Response Research

European Experimental Vehicles Committee Technical Papers

KURT ENKE
The Relation Between Vehicle Handling and Accident Avoidance

DR. RÜDIGER WEBER
Measurement of Transient Slip Angles of Tires – A Contribution of Evaluating Safety-Relevant Driving Conditions

DR. HANSJÖRG MANGER
Safety and Motor Vehicle Equipment

DETLEF BANHOLZER
Improving Directional Stability Underbraking

F. SURACE
Further Research on the Driver/Vehicle System Study of Braking in a Turn

PAOLO SOARDO
The Stop Lamp: Some Optimum Visibility Considerations

T. HOLMES and T. FRENCH
Safety Aspects of Denovo Run Flat Tires

G. JONES
Factors Leading to Loss of Control – A Guide from Accident Investigations

H. A. J. PRENTICE
Drivers' Limitations

M. A. JACOBSEN
The Punctured Pneumatic Car Tire from the User's Viewpoint

Japanese Technical Papers

HIDEO SAKAI
Characteristics of Japanese ESV Tires

SHOICHI SANO
Steering and Handling Characteristics of a Vehicle When Fail-Safe Tire is Deflated

DR. G. POCCI
Closing Remarks

PART FIVE OVERALL CAR REQUIREMENTS

Themes: Cost Benefit for Safety Measures; Interaction of Safety; Environmental and Energy Factors

United States Technical Papers

WILLIAM C. STEBER
The Evaluation of Safety, Energy, and Environmental Factors in the Automotive Transportation System

JOHN J. BROGAN
An Overview of the U. S. Government Program to Evaluate Alternative Powerplants to the Conventional Internal Combustion Engines for Automobiles

JOHN J. EDWARDS
Research Safety Vehicle Crash Effectiveness Methodology

European Experimental Vehicles Committee Technical Papers

DR. ERNST FIALA
Benefit/Cost Analysis as a Basis for Decisions in the Automotive Industry

N. MUHLRAD
Cost-Effective Study

P. VENTRE
Study on Cost-Efficiency Ratio Based on "Basic Research Vehicles"

DR. JOHN FITCHIE
An Overview of TRRL Programs

Japanese Technical Papers

YOSHIO SERIZAWA
A Feasibility Study on Small Safety Vehicles

TEIJI IIDA
Cost-Benefit Analysis of Bumper Systems for Small Cars

Discussion – Concluding Remarks

PART SIX APPLICATION OF RESEARCH RESULTS TO PRODUCTION CARS AND ESV ACTIVITIES

Themes: Application of Research Results to Standards; Future Activities

United States Technical Papers

DR. JAMES B. GREGORY
The Research Safety Vehicle Program and U. S. Motor Vehicle Safety Standards

European Experimental Vehicles Committee Technical Papers

A. OSSELET
Application to Regulations of Studies Made Pursuant to the ESV Programs

DR. G. BRENKEN
Contribution of the German Motor Industry

R. J. A. SHARP
Translating the Results of the Experimental Safety Vehicle Program into Production Cars

DR. POCCI
Application of Results to Standards – Future Activities

Japanese Technical Papers

NORIYOSHI UNO
Application of Results to Standards – Future Activities of Japan

Discussion

SECTION 3

TECHNICAL SEMINARS

PART ONE – ACCIDENT ANALYSIS

Chairman: M. HALPERN-HERLA
ONSER
France

Themes: Data Collection;
Use of Data for Car Safety;
Examples of Particular Results

Introduction by the Chairman:

After some preliminary remarks on my part we shall have the general point of view on the problem as presented by the United States, Japan and Western Europe. We shall then hear individual technical reports which have been prepared for the seminar. Today's meeting will cover the question of the collection of statistics on vehicle accidents. The presentations will concentrate on three areas: collecting the basic information, using the information for studies on vehicle safety, and finally some examples of the results obtained from such studies. May I draw your attention to the fact that the examples of the results do not apply solely to this meeting, because they could also apply to seminars 2, 4 and 5. We shall begin with an overview from the United States by Mr. Mela.

DONALD F. MELA
Office of Statistics and Analysis
National Highway Traffic
Safety Administration

I would like to speak very briefly about how the problem of accident collection and analysis looks to us and to give you a brief overview of the material that will be covered by the three speakers during our portion of the technical presentations.

The major problems in accident analysis have not changed very much during the last few years in the United States, although our approach is gradually shifting. The major problem appears to us still to be the problem of obtaining adequate data rather than the problem of how to use it or to analyze it once it is obtained. That is not to say that those latter problems are not real ones, but the most difficult part seems to be getting the data we need. In this connection the problem, or the part that appears to us to require the highest priority effort, is the problem of getting a representative set of data, that is, having an adequate sampling basis. There are many bodies of data with very, very large numbers of cases and one could assume that the random error associated with any statistics that one might compute from them is small but the problem is that in most cases we don't know exactly what the population is. One finds large bodies of data collected by state police, by insurance companies and so on, but if one wishes to generalize and to state what the actual population is and how it was selected, there is a real problem. It should be pointed out that in many cases there is no need to estimate very precisely the frequency of occurrence of a particular type of accident or a particular situation, if one knows that there is a proper sample. Customarily in the U.S., and I'm sure in other countries, a sample of one to two thousand very well selected individuals is adequate to estimate the frequency of occurrence of opinions or to try to determine how election trends are running. And, if properly done, such samples can come within one or two percent error.

Because we in the National Highway Traffic Safety Administration have recognized this problem, we have been searching for a way to get the proper kind of national system and national sample. Mr. O'Day, who will speak later, was the principal investigator on a contract which started as an attempt to use properly the data from our multidisciplinary accident investigation teams. The task was expanded to try to find a better way for our data collection.

A second problem that we see is the problem of the nonsampling errors. When we have an adequate sample and it represents the accident population, we still have the problems associated with correct reporting of injuries and with the uniformity of definitions. In a situation where we are dealing with many government levels, we have the problem of promulgating and getting agreement and general usage. For even when the standard is set, it frequently occurs that considerable oversight is needed to ensure that different people will not be making different interpretations.

The problems associated with data collection and the systematic biases that occur in reporting are also of considerable concern. For example, in the case of safety belts there is considerable error in determining whether or not the belt was worn, and a little arithmetic will show you that rather small percentage errors in classification lead to large errors in estimates of the injury reducing effect of safety belts.

A final area where we are hoping for more effort is in the subject of measurement scales. Right now we do not have scales for injury severity or for accident severity that can be used in the same sense as real numbers, that is, as yardsticks. This is a major problem that we have to deal with in doing benefit-cost analyses and we are attempting to get research done in this area.

As I mentioned, Mr. O'Day is going to talk to you about a proposal which his institute has made that is not an official position of our agency. However, NHTSA has asked him to look further into the actual, practical implementation of the type of system he has suggested.

Mr. Garrett of the Calspan Corporation will discuss a case study in which there was a combination of accident statistics, examination of individual cases as clinical cases followed by some theoretical work and

finally by laboratory testing. The combination of these findings is essential in many cases but there are not very many organizations that are capable of bringing all these elements together.

Finally, Mr. O'Neill of the Insurance Institute for Highway Safety will talk about a statistical study that involves two-car collisions and the study of the effects of weight and size. This problem has been touched on before in the ESV conferences. It has become increasingly important in the U.S. because the shift toward smaller cars began several years ago and has been accelerated by the fuel shortage – the energy crisis.

Mr. O'Neill will discuss results based on accident data collected in the State of North Carolina.

If time permits, I will show some additional charts based on similar investigations in New York state which produce qualitiatively very similar results. The data is used primarily in the estimation of the magnitude of weight effects. It bears on the relative importance of the problem of aggressivity and the corresponding problem of defensiveness.

M. HALPERN-HERLA
ONSER
European Experimental Vehicles Committee

I shall now exercise my privilege as chairman of this session to present the European point of view – the West European point of view of the problem of accident data related to vehicle safety.

A European committee was set up for experimental vehicles, the purpose of which was to create a common effort on the part of the various countries producing cars in western Europe. Within the framework of this committee we have a group which deals with questions which are on our agenda today. This working group has as its aim: (1) to survey the accident data which are available today in Europe; (2) to examine the use made of them for accident studies, particularly to improve vehicle safety; and (3) to propose recommendations to improve the present system of information and study.

This group came to the unanimous conclusion that although there are sufficient data in quantity and quality to begin making proposals on vehicle safety, there are a number of improvements which should be made if one wishes to continue along the road of progress in the field of safety. This group made a number of recommendations which I propose to give you now.

First of all, on accident data and safety in general, this group thinks that this information should be communicated freely and should be made compatible as far as possible among all the countries. Secondly, the group believes that accident research must be undertaken in close cooperation between the governments and the car manufacturing industry and must be considered a preliminary to most of the modifications which are to be made to vehicle regulations.

Thirdly, if one is to know the importance of the various problems posed by security, one should have available a system for collecting information on casualty accidents. On the national level, police statistics should deal with simple objective information and give the essential information on the vehicle itself. This level should be supplemented by an intermediate level which would give representative data setting out in greater detail specific problems which are considered to be priorities. The third level of data collection would consist of a detailed examination of accidents by multidisciplinary teams dealing with samples which must necessarily be small and not necessarily representative.

A fourth recommendation is to integrate the data collection on accidents with data where one sets out the exposure to risks. For instance, the exposure classification is undertaken according to the types of passengers and vehicles. This would enable one to compare the specific populations to a reference population. One should make use of the classical techniques for collecting data.

A fifth recommendation is that proposals to improve vehicle safety should be examined whenever possible from the point of view of their cost and likely effectiveness. When a potential measure seems to be expensive, considerable effort should be made to estimate experimentally its cost before it is finally adopted.

A sixth recommendation called for decisions taken on vehicle regulation to be the subject of an a posteriori evaluation. However, in the light of the difficulties which often occur in this type of assessment, it is recommended that the programs be undertaken so that one can estimate the efficacy of the effectiveness of the measures taken.

Another recommendation called for studies of human tolerances and the performance criteria of the vehicle. These studies should be undertaken by associating the observations on real accidents and those noted on reconstituted accidents in control conditions.

A seventh recommendation called for more safety studies of accident avoidance, while recognizing that they do present certain difficulties.

Finally, the last recommendation called for cooperation and coordination of these studies which are particularly difficult between technical research centers and should be greatly encouraged at the international level.

This, then, is a summary – a brief summary of the point of view within the EEVC on the collection of data on vehicle safety.

TRAFFIC ACCIDENTS IN JAPAN

Mr. NAGAKUNA
Traffic Bureau of the National
Police Agency

GENERAL CONDITION OF ROAD TRAFFIC

The number of motor vehicles (excluding mopeds of 125 cc or less) in use in Japan is on the increase year by year. With an increase of 2,630,000 units in 1973, at the year end, their total reached 26,180,000 units, or one vehicle for every 4.2 persons. The holders of driver's licenses numbered 30,780,000 at the end of 1973, accounting for 1 to 2.6 persons over 16 years old who are eligible for driver's licenses.

The total length of roads at the end of March, 1973, was 1,050,000 km, of which 25.2 percent are paved.

Figure 1 shows the tendency of increase of motor vehicles in use, the number of holders of driver's licenses and the length of paved roads from 1954 to 1973. It clearly indicates a rapid progress in motorization. Such a rapid progress in the motor vehicle traffic, however, has brought about a number of unfavorable effects on the human environment – traffic accidents, traffic congestion and traffic hazards of various forms.

However, in traffic accidents (personal accidents) an epochal change in the casualty was noted, there was a decrease both in the number of deaths and injuries for three consecutive years since 1971.

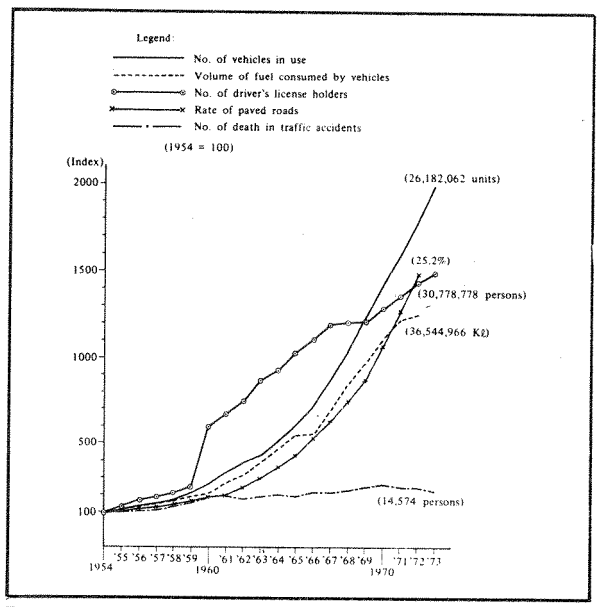


Figure 1. Trends in the Number of Motor Vehicles in Use and Number of Driver's License Holders (1954-1973)

Number of Vehicles in Use Has Doubled Every Five Years

When this increase is traced by the type of vehicle, the number of passenger cars in 1973 showed an increase of 11.7 times the number of passenger cars in 1963. This accounts for 56 percent of the total number of all types of motor vehicles, which is a 2.5 times increase in its composition rate as compared with that of 1963 at 22 percent.

However, when the number of motor vehicles in use in Japan is compared with that in Europe and America, the composition rate of passenger cars is far below the level obtained in Western nations. In other words, the rate of trucks in use is far higher in Japan than in other nations. This is a factor which characterizes the motor vehicle traffic and, at the same time, the type of accidents in Japan.

Class / Country	Passenger car			Truck		Bus		Total
	Number of units	Persons per car	Rate	Number of units	Rate	Number of units	Rate	
Canada	7,400,000	3.0	78.7	1,957,000	20.8	42,400	0.5	9,379,400
France	13,920,000	3.7	87.4	1,951,000	12.3	49,000	0.3	15,920,000
W. Germany	16,323,997	3.8	92.8	1,219,804	7.0	53,763	0.3	17,597,564
Italy	12,484,313	4.4	92.4	995,591	7.4	36,150	0.3	13,516,054
Sweden	2,456,940	3.3	93.8	146,003	5.6	15,030	0.6	2,617,973
U.S.A.	96,397,000	2.2	82.0	20,801,294	17.7	407,706	0.3	117,606,000
Japan	12,531,151	8.5	56.6	9,423,010	42.5	202,819	0.9	22,156,980

Note: 1) Source of figures: Number of vehicles and population from reports of Japan Automobile manufacturers Association, Inc. and the U.N. Statistics (1972), monthly reports.
2) Vehicles for special use and motorcycles are excluded.

Figure 2. Comparison of Motor Vehicles in Use (1972)

Drivers by Sex: Male – 63%; Female – 15%

The number of licensed drivers is increasing year after year. This is largely affected by the increase in the owner-driver cars. A notable fact is that the increase rate in the total driver's licenses issued is higher for the female than for the male.

	Male	Rate compared with preceding year (%)	Female	Rate compared with preceding year (%)
1971	22,699,349 (81.1)	+ 4.7	5,301,018 (18.9)	+ 11.2
1972	23,675,142 (80.3)	+ 4.3	5,799,501 (19.7)	+ 9.4
1973	24,477,063 (79.5)	+ 3.4	6,301,715 (20.5)	+ 8.7

Note: Figures in ( ) indicate percentage of share in the total number of holders of driver's license.

Figure 3. Trends in Number of Driver's License Holders by Sex (National Police Agency Statistics)

At the end of 1973, among the eligible male or female group (above 16 years) in the population, the driver's license issued for the male accounted for 62.8% and 15.4% for the female.

Persons from 25 through 29 years of age are the highest age group in driver's license possession ratio: males – 84%; females – 30%.

Reconstruction of Roads in Progress

At the end of March, 1973, paved roads represented 25.2 percent of Japan's roads, 4.9 times the 1963 figure. Compared to the Western nations, this rate is very low as yet, but the length of roads per unit motor vehicle is comparable and the length of roads per square km is remarkably higher in Japan. However, 74 percent of the total length of roads are narrow and all of them are under 4.5 meters in width.

Classification / Country	Extension of roads (1000 km)	Paved road rate (%)	Extension of roads per 1 km² of total national area (M)	Extension of road per unit vehicle (M)
France	789	82.5	1,442	52.5
W. Germany	445	79.9	1,795	25.2
Italy	284	95.2	942	21.0
U. K.	360	99.7	1,475	24.6
U.S.A.	6,049	45.0	646	50.9
Japan	1,049	25.2	2,782	43.9

Note: 1. Source from International Road Federation, 1972 edition statistics.
2. Figures for Japan are as of the end of March 1973.

Figure 4. Comparison of Road Conditions

GENERAL CONDITIONS OF TRAFFIC ACCIDENTS

Trends in Traffic Accidents

The casualty rate due to traffic accidents, which had been on the increase, reached its peak in 1970 and has followed a downward curve since then.

The worst year in history was 1970 when 16,765 dead and 981,096 injured in traffic accidents were recorded. In spite of the fact that the number of vehicles in use has made a notable increase every year since then, the traffic casualty rate, both in dead and injured, showed a decrease for three consecutive years. The factors contributing to the decrease may be traced to the following: improvement of traffic safety facilities, strengthened traffic law enforcement, improved management system of drivers, and understanding and efforts for traffic safety made by the nation at all levels.

Even with this favorable turn, however, the traffic casualty rate for 1973 reached over 800,000, and the prevention of traffic accidents remains as a great question.

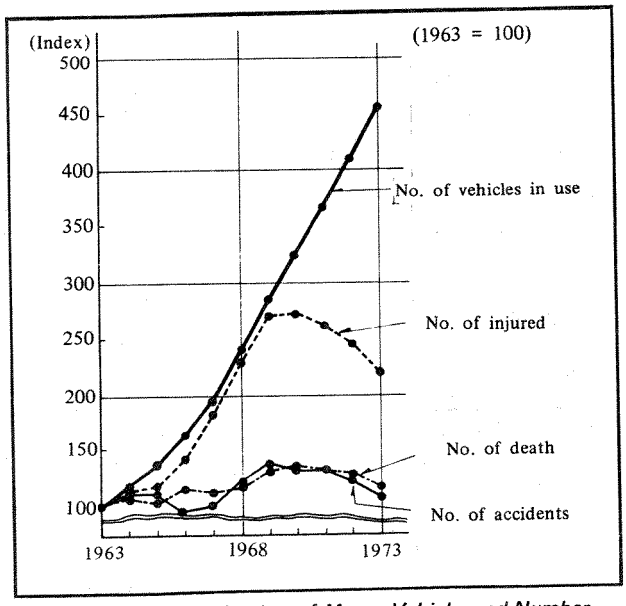


Figure 5. Trends in Number of Motor Vehicles and Number of Traffic Casualties

Traffic Accidents in 1973 – One Death Every Thirty-Six Minutes

In 1973, traffic accidents in Japan involving human casualties reached 586,713 cases with 14,574 deaths and 789,948 injuries, amounting to one death every 36 minutes and one injury every 40 seconds.

	For the year	Monthly average	Daily average	Remarks
No. of accidents	586,713	48,893	1,607	1 case in every 53.8 seconds
No. of death	14,574	1,215	40	1 dead every 36 minutes
No. of injured	789,948	65,829	2,164	1 injured in every 39.9 seconds

Note: The number of deaths relates those who died within 24 hours counting from the moment at accidents.

Figure 6. Occurrence of Traffic Accidents (National Police Agency Statistics, 1973)

The traffic accidents in 1973 showed a decrease of 1,344 deaths (8.4%) from the preceding year. It was the second largest decrease on record, next to 1962. Both personal injury cases and injuries have decreased by 72,570 cases (11.0%) and 99,250 persons (11.2%), respectively. These decreases established the highest record in the degree of decrease in history.

ANALYSIS OF TRAFFIC ACCIDENTS (ESPECIALLY FATAL ACCIDENTS)

Outline

Half of the Recorded Total Deaths: Pedestrians and Bicycle Riders. When a comparison between 1973 and 1972 is made, a slight decrease is noted

during 1973 in deaths of pedestrians and those riding bicycles. Of the total deaths, 37 percent are accounted for by pedestrians and 12 percent by those riding bicycles. These two categories amount to the high rate of about half of the total deaths.

Class \ Year / No. of death	1970		1971		1972		1973	
	No. dead	Rate	No. dead	Rate	No. dead	Rate	No. dead	Rate
While walking	5,939	35.4	5,761	35.4	5,689	35.7	5,376	36.9
While riding bycycle	1,940	11.6	1,741	10.7	1,756	11.0	1,700	11.7
While riding motorcycle	2,941	17.5	2,933	18.0	2,586	16.3	2,330	16.0
While riding motor vehicle	5,612	33.5	5,538	34.0	5,657	35.6	5,079	34.8
Others	333	2.0	305	1.9	230	1.4	89	0.6
Total	16,765	100.0	16,278	100.0	15,918	100.0	14,574	100.0

Figure 7. Trends of Deaths Caused by Accidents, Classified by Condition (National Police Agency Statistics)

Figure 8a relates pedestrian deaths and accompanying rate to total deaths due to traffic accidents in foreign countries. In comparing the death rate of pedestrians to the total death rate, the United Kingdom shows the highest rate at 39%, followed by Japan and Norway. The United States shows the lowest rate at 17.9%.

The number of pedestrian deaths in cities of foreign nations is shown in Figure 8b. The rate of pedestrian deaths to total deaths is higher in the world's larger cities. Pedestrian death rate due to traffic accidents per 100,000 population is classified by age group and shows the highest rate is held by children and the aged.

Country \ Class / Year		No. of death by traffic accidents		A/B x 100
		Total death (A)	Pedestrians' death (B)	
Austria	1970	2,238	710	31.7
Belgium	1970	1,544	361	23.4
France	1969	14,664	3,117	21.3
W. Germany	1970	19,193	6,056	31.6
Italy	1970	10,208	2,651	26.0
Netherlands	1970	3,181	609	19.1
Norway	1970	560	194	34.6
U. K.	1970	7,499	2,925	39.0
U.S.A.	1968	55,500	9,950	17.9
Japan	1973	14,574	5,376	36.9

Note: Source from Report of U.N. Economic Commission for Europe

Figure 8a. Pedestrian Deaths in Foreign Countries

	U.S.A.		U.K.	France	Sweden		Japan
	Washington D.C.	New York	London	Paris	Stockholm	Goteborg	Tokyo
	person	person	person	person	person	person	person
Total no. of death (A)	73	992	723	175	56	24	551
No. of (B) Pedestrians' death	31	526	418	86	25	13	238
$\frac{A}{B}$ x 100	42%	53%	58%	49%	45%	54%	43%

Note: Source Survey by International Traffic Symposium (sponsored by Asahi Shimbun, September 1973)

Figure 8b. Comparison of Pedestrian Deaths Among Leading Cities of the World (1972)

The age level composition in the total number of 5,376 deaths in Japan in 1973 is shown in Figure 9. Almost two-thirds of pedestrian deaths were children and the aged. Of the 1,700 killed while riding bicycles, 57% was accounted for by children and the aged.

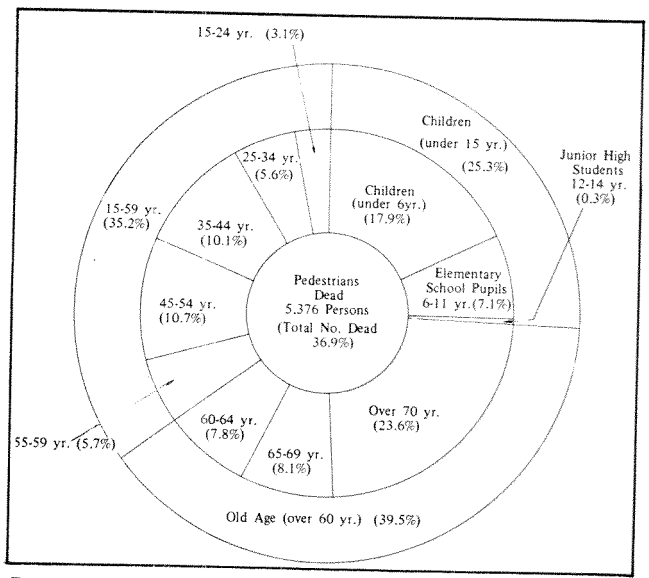


Figure 9. Pedestrian Deaths by Age-level Composition (National Police Agency Statistics)

More Accidents by "Running Weapon" Type. Fatal accidents are divided into two types: the "running weapon" which includes (1) man versus vehicle (motor vehicle and bicycle with a motor) and (2) bicycle versus vehicle; and the "running coffin" which includes vehicle versus vehicle and the vehicle by itself. There were 6,702 "running weapon" cases recorded (48.6% of total deaths) against 6,488 "running coffin" cases (47.1% of total deaths), showing a ratio of 51:49.

Other Fatality Causes. Owner-driven motor vehicles accounted for about 70 percent of accidents involving deaths. About 91 percent of the total recorded accidents was caused by drivers of vehicles (passenger cars and mopeds). In 6,205 accidents (45% of the total fatal accidents) passenger car drivers were responsible with owner-driven cars accounting for 93 percent.

Truck drivers caused 4,608 accidents (33.5% of the total fatal accidents, of which owner-driven vehicles account for 80%).

When fatal accidents by drivers of these two types of owner-driven vehicles are combined, they account for about 70 percent of the total fatal accidents.

Death Rate Is Highest at Midnight. Over 70 percent of personal accidents takes place between 6 a.m. and 6 p.m., while fatal accidents most frequently happen between 4 p.m. and 8 p.m. The highest fatality rate is recorded around midnight when traffic congestion is at its lowest.

Type of accidents		1973	Number	Rate (%)	
Weapon Type	Man vs. Vehicle	While crossing on Pedestrians' crossing	653	4.7	48.6
		While crossing outside Pedestrians' crossing	2,660	19.3	
		Others	1,805	13.1	
		Sub-total	5,118	37.1	
	Bicycle vs. Vehicle	Bumped into each other	357	2.6	
		While turning right	259	1.9	
		Others	968	7.0	
		Sub-total	1,584	11.5	
Coffin Type	Vehicle vs. Vehicle	Head on collision	1,849	13.4	47.1
		Rear-end collision	460	3.3	
		Bumped & side collision	1,024	7.4	
		Others	592	4.3	
		Sub-total	3,925	28.4	
	Vehicle by self	Overturn, fallen off-road	1,360	9.9	
		Collision with stalled vehicle	313	2.3	
		Collision with other objects	890	6.5	
		Sub-total	2,563	18.7	
Railroad Crossing		Man	201	1.4	3.4
		Light vehicle without motor	87	0.6	
		Mopeds of 125 cc or less	40	0.3	
		Vehicle (stalled, fallen)	9	0.1	
		Vehicles (others)	135	1.0	
		Sub-total	472	3.4	
Others			129	0.9	
Total			13,791	100	

Note: Vehicles include motor vehicles & bicycle with a motor.

Figure 10. Death Accident Cases, Classified by Types (National Police Agency Statistics)

Persons Involved in Accidents and Traffic Accidents

Age and Fatal Accidents. The 17-year-old male is most freqently involved in fatal accidents. Figure 14 indicates the number of male deaths in 1973 due to traffic accidents, classified by age and conditions, per 100,000 males.

The highest death rates are found for infants (3-5 years), teen-agers (16-20 years), and the aged (over 70 years). The 17-year-old males have the highest death rate (54.2 per 100,000 males), fifteen times higher than the lowest (3.6) male death rate held by the 14-year-old male. The 17-year-old male has the highest death rate among those riding motorcycles

alone or with a companion rider; and the 16 to 17-year-old male has the highest death rate among those riding bicycles with a motor.

The highest death rate for the 75-year-old-and-over group occurs while they are walking or bicycling.

Male death rate while driving suddenly rises from the 18-year-old male (the qualifying age for receiving a driver's license) to peak at the 20-year-old male group driving a vehicle. The female death rate per 100,000 is highest for the infant (2-3 years) and the aged (70 years and over) while walking.

Main offender	No. of Fatal accident	Percent
Buses	164	1.2
Passenger Cars for Business Use	259	1.9
Passenger Cars for Private Use	5,782	42.0
Passenger Cars for Business Use	903	6.5
Passenger Cars for Private Use	3,705	27.0
Motorcycles	804	5.8
Mopeds (125 cc or less)	969	7.0
Bicycles	322	2.3
Other vehicles	75	0.5
Pedestrians	612	4.4
Not Reported	196	1.4
Total	13,791	100

Figure 11. Fatal Accidents, Classified by Main Offender (National Police Agency Statistics, 1973)

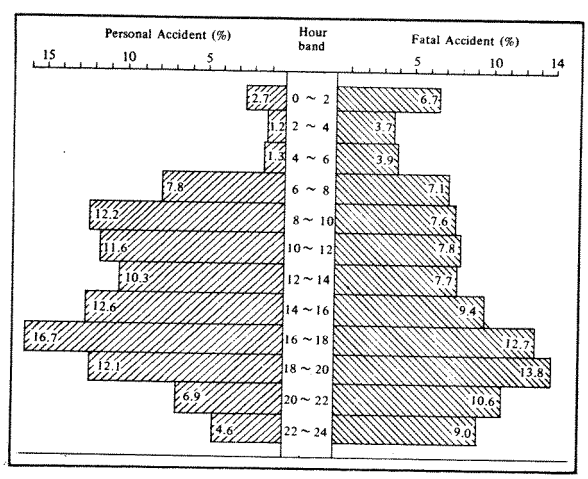


Figure 12. Traffic Accident by the Hour (National Police Agency Statistics, 1973)

Driving Experience and Accident Rate. Figure 15 demonstrates that 16 to 19-year-olds hold the highest accident rate by age grouping, followed by the 20 to 24-year-old group and the 25 to 29-year-old group. The accident rate declines proportionally as the age advances.

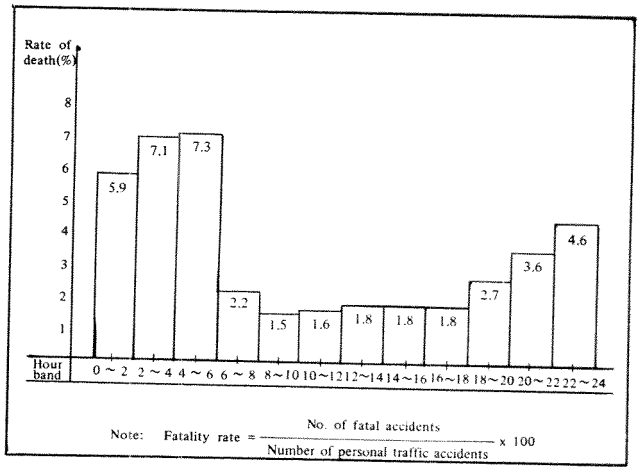


Figure 13. Fatality Rate by the Hour (National Police Agency Statistics, 1973)

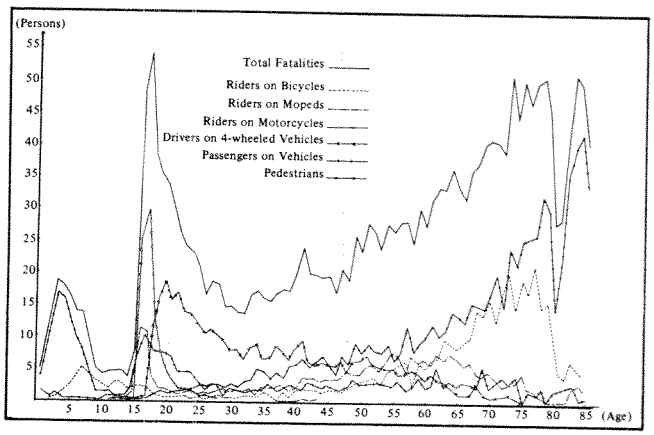


Figure 14. Number of Deaths (Male) by Traffic Accidents Per 100,000 Persons — By Condition and Age (1973)

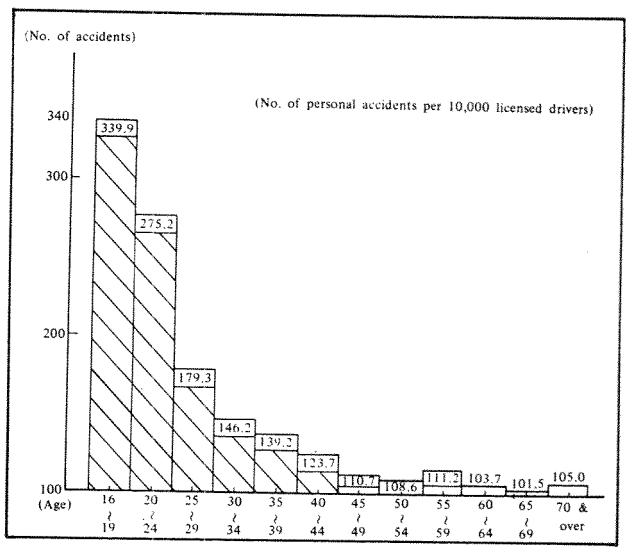


Figure 15. Accident Rate by Age of Drivers Causing Accidents (National Police Agency Statistics, 1973)

Figure 16 reports a survey conducted in October, 1971, and indicates that 28 per 10,000 licensed drivers are beginners who have had traffic accidents within one year after acquiring a license. In order to prevent the occurrence of traffic accidents caused by beginners, a measure was adopted in October, 1972, requiring beginning drivers to label their vehicles with the word "beginner" for a period of one year following the issuance of a driver's license.

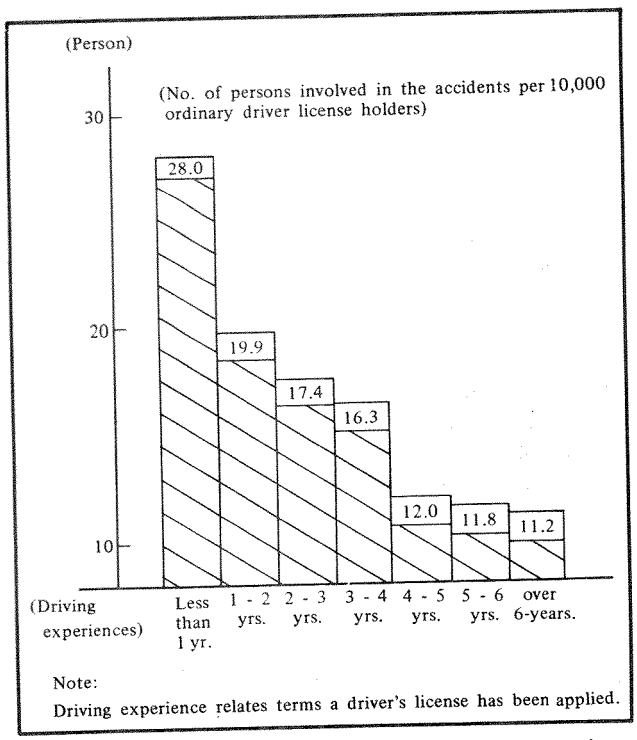


Figure 16. Accident Rate by Driving Experience of Licensed Drivers (National Police Agency Statistics, 1971)

Causes of Accidents. The highest number of fatal accidents are caused by drivers violating the speed laws. The next highest number of fatal accidents are caused by drivers who are driving under the influence of liquor and drivers who violate the no-passing regulations (Figure 17).

The highest rate of pedestrian fatalities are caused by pedestrians jumping into traffic and by pedestrians crossing the road in front or the rear of a car.

Except for railroad crossing accidents, fatal accidents are highest in the violation of maximum speed limits (12.1 cases per 100 personal accidents); driving under the influence of liquor (5.6 cases per 100 personal accidents); and violation of no-passing regulations (3.7 cases per 100 personal accidents).

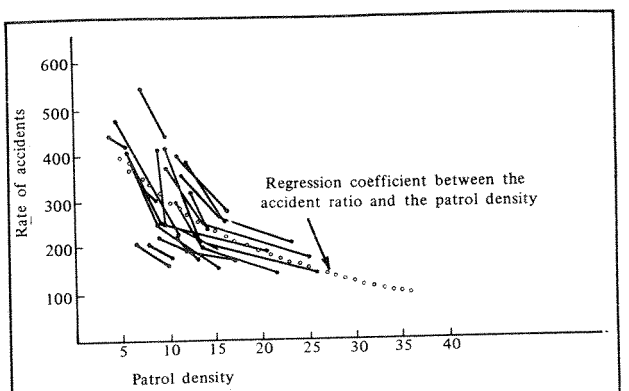
Figure 18 shows the effect of traffic law enforcement on accident control, resulting from the mobile patrol activities conducted by white patrol

Causes	Class	Number of fatal accidents (A)	Number of personal accidents (B)	Fatality rate ($\frac{A}{B}$ x 100)
Drivers of vehicles	Violation of maximum speed limit	1,564	12,910	12.1
	Driving under influence of liquor	1,187	21,124	5.6
	Violation of no-passing rule	705	19,039	3.7
	Violation of slow-down or stop signal at crossing	475	46,703	1.0
	Violation of protection of pedestrians	439	22,742	1.9
	Violation of traffic zone	325	8,941	3.6
	Driving with no license	281	–	–
	Failure to confirm safety factors and temporarily stop at railway crossings	274	897	30.5
	Violation of signals	215	13,375	1.6
	Violation of right turn	222	20,415	1.1
	Improper back drive	220	7,450	2.9
	Violation of left turn	201	10,759	1.9
	Violation of slow-down at fixed places other than crossing	129	5,521	2.3
	Improper driving operation	74	24,060	0.3
	Improper distance between cars	56	29,101	0.2
	Driving irremaintained cars	24	1,310	1.8
Pedestrians	Jump-into traffic	172	18,297	0.9
	Crossing in front and rear of the car	160	6,128	2.6
	Loitering while get drunk	26	512	5.1
	Infants walking without help	18	309	5.8
	Total	6,767	269,593	

Notes: 1. Data relate comparison between fatal accidents due to main violations numbering 6,767 out of total fatal accidents 13,791 and personal accidents.

2. Fatality rate = $\frac{\text{No. of fatal accidents}}{\text{Number of personal accidents}} \times 100$

Figure 17. Occurrence Conditions by Main Causes of Accidents (National Police Agency Statistics, 1973)



Notes:

1. The graph relates the patrol density and accident ratio on specified roads (21 routes) during the term from 1969 through 1971.
2. Patrol density is a value which indicates the density of mobile patrol force on the route, calculated as the following:

$$\frac{\text{No. of daily patrols}}{\sqrt{\frac{\text{Daily traffic volume}}{10{,}000}}}$$

3. Rate of accidents calculated as the following:

$$\frac{\text{No. of traffic accidents per year}}{\text{Running km per year}} \times 100{,}000{,}000$$

4. Accident ratio . . . (Y), Patrol density . . . (X)

$Y = AX^B$ (A, B: fixed numbers)

$Y = 881.6X^{-0.518}$

Figure 18. Relation Between Patrol Density and Accident Rate (National Police Agency Material)

cars (motorcycles). They are shown as actual data. When the mobile patrol density is doubled from 8 to 16, the accident rate is reduced from 300 to 200 – about 30 percent.

Trends in the relation between the number of law enforcement and the occurrence conditions of traffic accidents are noted in Figure 19, showing the effect of traffic law enforcement on accident control.

Road Environment and Traffic Accidents

Traffic accident hazards have been identified as "tunnels, crooked and bending parts, etc." and "other nonintersection parts of roads."

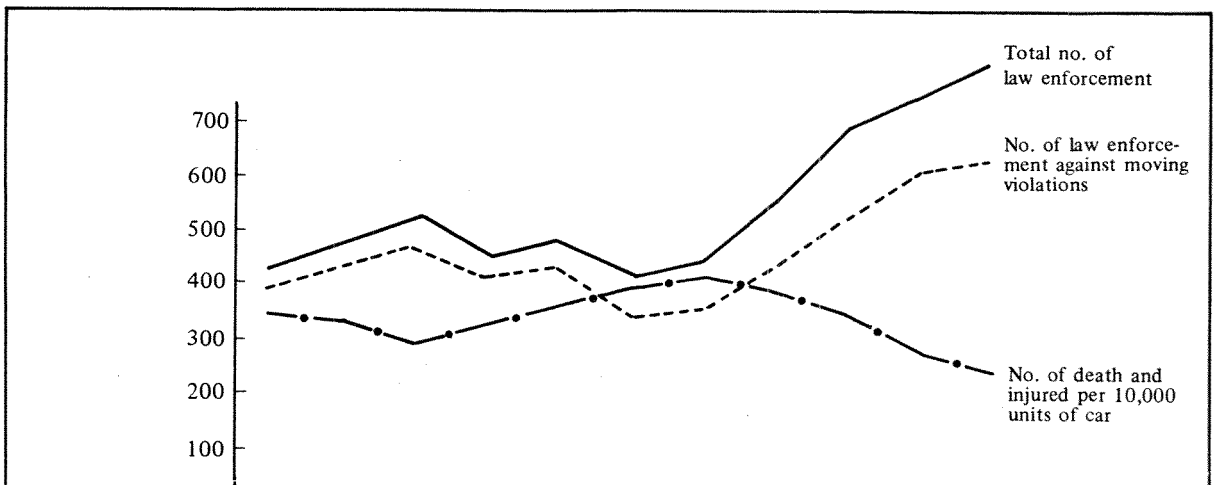
Two areas where fatal accidents take place are: built-up areas (areas where houses are closely built) and nonbuilt-up areas. A built-up area is subdivided into city sections and others. Built-up areas account for 43.5 percent of fatal accidents and nonbuilt-up areas account for 56.5 percent of fatal accidents. The city section occupies 60 percent of fatal accidents in built-up areas (Figure 20).

When comparing road condition to fatalities, the highest rates are at intersections in the built-up areas; "nonintersections" in the nonbuilt-up areas; and "nearby intersections" in the city section of built-up areas. Comparing the number of fatal accidents per 100 personal accidents with road conditions, the hazards in nonbuilt-up areas are 2.7 times higher than in built-up areas. Also, fatalities are highest at railroad crossings, followed by "tunnels, crooked parts" and "other nonintersection parts of the road." The death rate for pedestrian crossings is 2.6 cases per 100

Formation of road \ Conditions		Builtup area: City section %	Builtup area: Others %	Non-builtup area %	Total	Death rate
Crossing	Within intersection	8.4	4.2	9.5	22.1	1.4
	On pedestrian crossing of intersection	2.4	0.7	0.7	3.8	2.6
	Nearby intersection	3.3	1.3	2.2	6.8	1.6
	Sub-total	14.1	6.2	12.4	32.7	1.5
Single roads	On pedestrian crossing	0.3	0.1	0.2	0.6	2.3
	Nearby pedestrian crossing	0.3	0.1	0.3	0.7	2.2
	Tunnel, crooked roads, near turning corners and sloping roads	0.9	1.3	10.2	12.4	4.3
	Other non-intersection part	9.4	9.1	31.2	49.7	3.0
	Sub-total	10.9	10.6	41.9	63.4	3.2
Rail road crossing		0.9	0.6	1.9	3.4	37.9
Others		0.1	0.1	0.3	0.5	2.7
Total		26.0	17.5	56.5	100.0	2.4
Death Rate		1.3	2.2	4.2	2.4	
		1.6				

Note: Death rate is the number of fatal accidents per 100 personal accidents.

Figure 20. Occurrence Situation Related Death Accidents by Road Shape (National Police Agency Statistics, 1973)



Year	1963	1964	1965	1966	1967	1968	1969	1970	1971	1972	1973
Total no. of law enforcement (unit=10,000)	429	471	512	468	478	404	420	539	680	732	807
No. of law enforcement against moving violations (unit=10,000)	390	425	456	411	416	344	346	418	506	550	609
No. of death and injured per 10,000 cars (unit=persons)	349.7	328.1	299.7	318.4	352.3	390.3	404.8	373.0	334.4	287.9	238.5

Figure 19. Relationship Between Traffic Law Enforcement and Occurrence of Traffic Accidents – Number of Deaths and Injured (National Police Agency Statistics, 1963-1973)

personal accidents and 2.3 cases at the nonintersection parts of the road.

Expressway Accidents. Since expressways are limited to the exclusive use of motor vehicles and have regulated intersections, the personal accident rate is about 1/8 of the ordinary roads per 100,000,000 units/km. However, because of the higher speed limits, the casualty and death rate per accident is higher than for the ordinary roads.

Class	Expressways	Ordinary roads
Number of personal accidents by 100,000,000 units per Km (case)	28	225
Number of death & injury per accident (person)	2.0	1.4
Death rate $\left(\frac{\text{No. of death}}{\text{No. of death \& injury}}\right) \times 100$ (%)	3.2	1.8

Note: Expressways include Tomei, Meishin and Chuodo and the value is based on these three only.

Figure 21. Accident Rate on the Expressway (National Police Agency Statistics, 1973)

On expressways the single car accidents (vehicle alone accidents) account for 40 percent of the accidents, approximately 4.2 times the rate of personal accidents occurring on ordinary roads. (Vehicle alone accidents account for 9.3 percent of the total accidents on the ordinary roads, excepting accidents with pedestrians and at railroad crossings.)

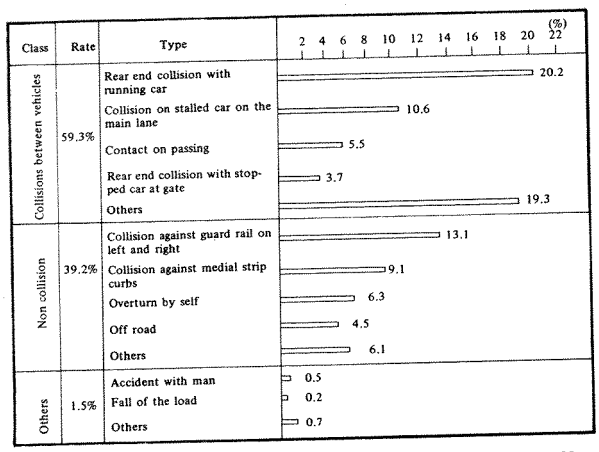
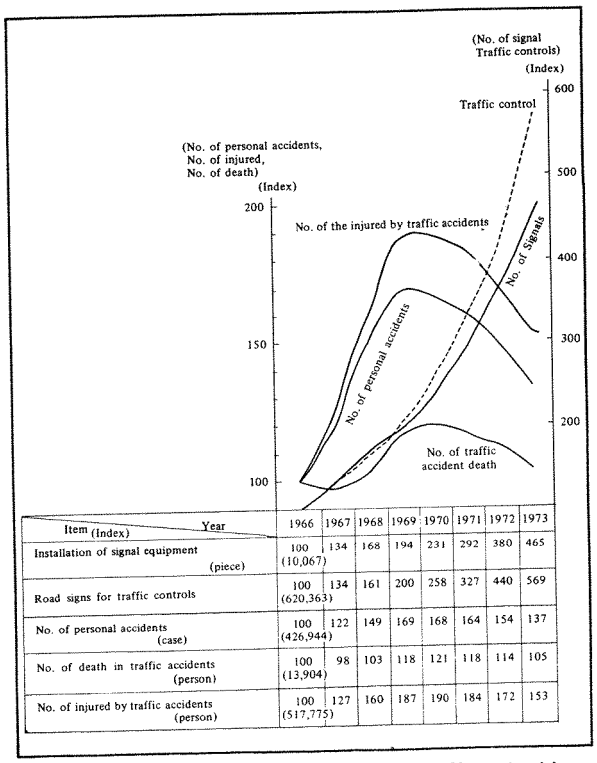
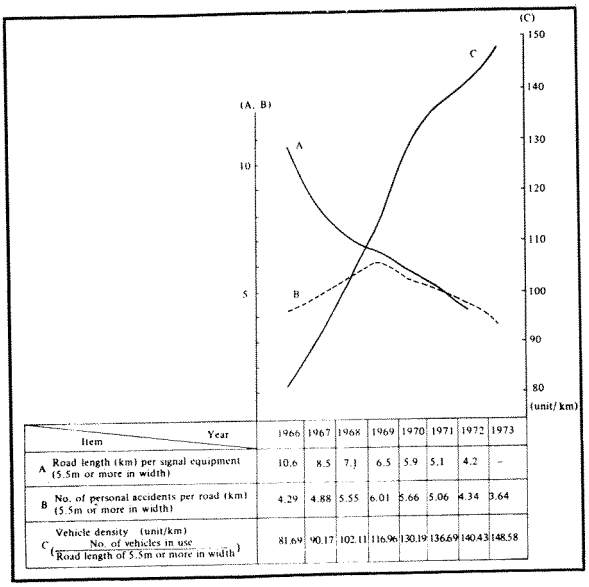


Figure 22. Occurrence Situation Related to Traffic Accidents on Expressways, Classified by Types (National Police Agency Statistics, 1973)



Item (Index) / Year	1966	1967	1968	1969	1970	1971	1972	1973
Installation of signal equipment (piece)	100 (10,067)	134	168	194	231	292	380	465
Road signs for traffic controls	100 (620,363)	134	161	200	258	327	440	569
No. of personal accidents (case)	100 (426,944)	122	149	169	168	164	154	137
No. of death in traffic accidents (person)	100 (13,904)	98	103	118	121	118	114	105
No. of injured by traffic accidents (person)	100 (517,775)	127	160	187	190	184	172	153

Figure 23. Signal Equipment and Traffic Accidents (National Police Agency Statistics)



Item / Year	1966	1967	1968	1969	1970	1971	1972	1973
A Road length (km) per signal equipment (5.5m or more in width)	10.6	8.5	7.1	6.5	5.9	5.1	4.2	–
B No. of personal accidents per road (km) (5.5m or more in width)	4.29	4.88	5.55	6.01	5.66	5.06	4.34	3.64
C Vehicle density (unit/km) $\left(\frac{\text{No. of vehicles in use}}{\text{Road length of 5.5m or more in width}}\right)$	81.69	90.17	102.11	116.96	130.19	136.69	140.43	148.58

Figure 24. Road Length Per Signal Equipment; Number of Traffic Accident Occurrences Per Road km (National Police Agency Statistics)

Improvement of Traffic Safety Facilities Versus Traffic Accidents. As a result of a rapid growth in motorization, there has been a sharp increase in the number of traffic accidents. The installation of traffic safety facilities – such as sidewalks, overhead bridges for pedestrians, bicycle roads, signal equipment, road signs, etc. – has failed to keep pace with those advancing aspects related to motorization.

Therefore, the Metropolitan and Prefectural Public Safety Commissions and the Road Administrators jointly formulated two 3-year plans for establishing traffic safety facilities. Starting in 1966, the First Plan ran through 1968. The Second Plan started in 1969 and ran through 1971. Presently, a new 5-year plan (1971-1975) is in progress on a much larger scale, based on the Act for the Emergency Measures for Improvement of Activity for Traffic Safety Facility. Due to these traffic control and traffic accident prevention measures, in spite of the fact that motor vehicle use has increased yearly, the traffic accidents which reached their peak in 1969-1970 have begun to fall in number since then.

Effects of Signal Equipment on Accident Prevention. By the end of March, 1974, 46,000 signal equipment units were installed. Figure 25 shows a comparative study on traffic accidents before and after the signal equipments were installed.

	Effects of accident prevention / Signal equipment newly installed, improved	No.of places installed (unit)	Installation: Before improvement: No.of cases	No.of death	No.of injured	After improvement: No.of cases	No.of death	No.of injured
Pedestrians' accidents	Signal equipment installed	6,485	3,892	137	3,956	1,492	24	1,526
	Lighting apparatus added for pedestrians	1,302	669	10	702	259	1	306
	Coordinated signal system	1,676	938	27	974	707	13	726
Vehicles' accidents	Signal equipment installed	6,485	11,967	147	14,368	6,613	44	8,229
	Lighting apparatus added for vehicles	1,202	2,973	18	3,296	1,740	7	2,060
	Coordinated signal system	1,676	5,294	47	5,583	4,042	24	4,233

Number of accidents reduced: No. of cases	Reduced rate (%)	No. of death	Reduced rate (%)	No. of injured	Reduced rate (%)	No. of accidents reduced per year at 1 place annually (per unit): No. of cases	No. of death	No. of injured
2,400	61.7	113	82.5	2,430	61.4	0.740	0.035	0.749
410	61.3	9	90.0	396	56.4	0.315	0.014	0.608
231	24.6	14	51.9	248	25.5	0.276	0.018	0.296
5,354	44.7	103	70.1	6,139	42.7	1.651	0.032	1.893
1,233	41.5	11	61.1	1,236	37.5	2.052	0.018	2.057
1,252	23.6	23	48.9	1,350	24.2	1.494	0.027	1.611

Notes: 1. The figures indicate the results obtained from a survey on the newly installed or improved signal equipments during 1972.
2. The period of survey of accidents covered 6 months each before and after the installation and improvement were made.

Figure 25. Effects of Consolidated Signal Equipment

The installation of this equipment resulted in a decrease of 61.7 percent in the number of pedestrian accidents and 82.5 percent in the number of deaths; 44.7 percent in the number of vehicle accidents and 70.1 percent in the number of deaths.

Where the signal equipment has been switched to the "coordinated signal system," the number of accidents involving pedestrians has been decreased by 24.6 percent and the number of deaths by 51.9 percent; the number of vehicle accidents by 23.6 percent and the number of deaths by 48.9 percent. This undoubtedly indicates a significant contribution by the coordinated signal system equipment not only to smoother traffic flows but also to traffic safety.

Vehicles and Traffic Accidents

Fatal Accident Rate by Type of Vehicle. By type of vehicle in number of fatal accidents per 10,000 vehicles, the bus shows the highest rate, followed by motorcycles, large and ordinary sized trucks and light buses. By number of fatal accidents per running distance, large and ordinary sized trucks, light buses, ordinary passenger cars, and buses are almost equally high. Light cars (360 cc and under) show a low rate in the fatal accidents per 10,000 and per running distance. It may be said that the smaller the vehicle, the lower the rate of fatal accidents.

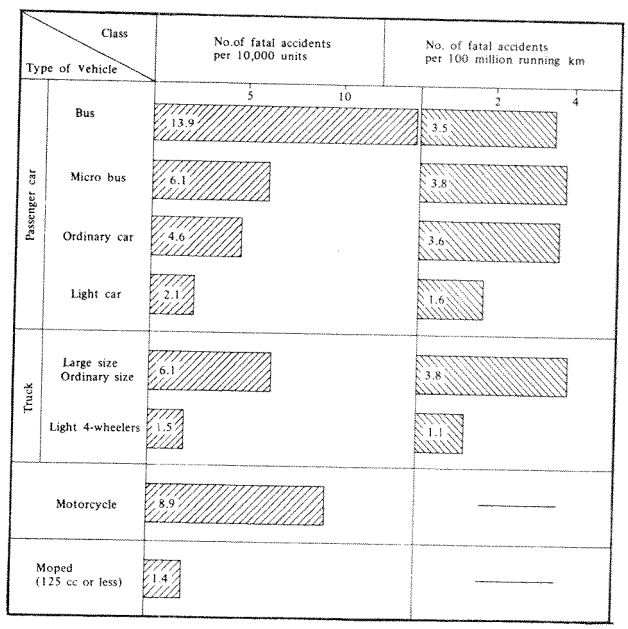


Figure 26. Fatal Accident Rate, Classified by Vehicle Types (National Police Agency Statistics, 1973)

Speed and Traffic Accidents. About 70 percent of the offenders involved in personal accidents were traveling at a speed of 40 km/h immediately before an accident, compared to 99.3 percent who were traveling at a speed of 80 km/h or less immediately before an accident, revealing that the lower rate of speed accounts for only about 30 percent of total fatal accidents, compared to the higher rate of speed.

Because the power of impact is proportional to the square of the speed, only one fatal accident case in 100 personal accidents was recorded at the 20-30 km/h speed; but the fatal accident rate suddenly rises when the speed exceeds 40 km/h, so that almost all the vehicle occupants face possible death when an accident occurs at speeds exceeding 80 kmh.

Offender's speed just before accident \ Composition of personal accident	Rate (%)	Rate excluding unknown
Under 20 km/h	24.4	27.1
20 – 30 km/h	17.3	46.3
30 – 40 km/h	21.7	70.4
40 – 50 km/h	13.8	85.7
50 – 60 km/h	8.4	95.0
60 – 70 km/h	2.6	97.9
70 – 80 km/h	1.3	99.3
80 – 100 km/h	0.5	99.9
100 km/h and over	0.1	100.0
Unknown	9.8	–
Total	100.0	

Figure 27. Speed Just Before Accident (National Police Agency Statistics, 1973)

Offender's speed just before accident	No. of personal accidents (A)	No. of fatal accidents (B)	Fatal rate ($\frac{B}{A}$ x 100)
Under 20 km/h	1,297	12	0.9
20 – 30 km/h	908	10	1.1
30 – 40 km/h	1,365	30	2.2
40 – 50 km/h	779	41	5.3
50 – 60 km/h	391	25	6.4
60 – 70 km/h	113	19	16.8
70 – 80 km/h	57	13	22.8
80 – 100 km/h	20	8	40.0
100 km/h and over	6	2	33.3
Unknown	53	2	–
Total	4,989	162	3.2

Figure 28. Speed Just Before Accidents and Fatal Rate (by Toyama Prefectural Police Headquarters, 1973)

Regions of Injury in Fatal Accidents (The Need for Seat Belts). During 1973 in Japan, the regions of injury sustained in traffic accidents (excluding motorcycles) which caused death were 45 percent for head injuries and 10 percent for chest injuries. The United Kingdom report at the Fourth ESV International Conference stated that it is possible to reduce head injury by 56 percent and chest injury by 60 percent by wearing seat belts. At this rate, Japan could reduce motor vehicle fatalities (excluding motorcycles) by one-third with the use of seat belts – over 1,600 fatalities – representing an 11 percent decrease in the total fatalities for one year.

In Japan motor vehicle occupants are required to wear a seat belt on expressways and motorways. Because of the proven merit of the seat belt, a study is now underway determining the advisability of compulsory full-time use of the seat belt.

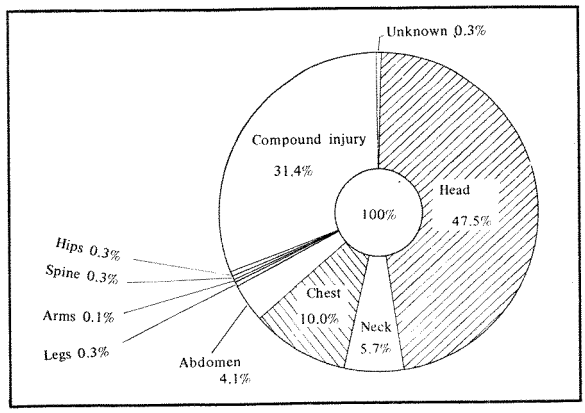


Figure 30. Fatalities While Riding in Vehicles (Except 2-Wheeled Vehicles), Classified by Main Regions

(unit=%)

Condition for death \ Region of main injury	Head	Neck	Chest	Abdomen	Hips	Spine	Arms	Legs	Compound	Unknown	Total
Passenger cars	11.4	1.3	2.1	0.8	0.05	0.1	0.02	0.04	7.0	0.1	22.8
Trucks	5.4	0.7	1.4	0.7	0.04	0.03	0.01	0.1	3.8	0.03	12.0
Motorcycles	10.1	0.7	0.8	0.5	0.1	0.01	0.03	0.1	3.9	0.01	16.1
Bicycles	7.3	0.4	0.4	0.3	0.1	0.03	–	0.03	3.3	0.01	12.0
Pedestrians	21.0	1.3	1.3	1.3	0.7	0.1	0.01	0.3	11.0	0.1	36.9
Others	0.1	0.01	0.02	0.01	–	–	–	–	0.03	0.01	0.1
Total	55.1	4.4	6.0	3.6	0.9	0.2	0.1	0.6	29.0	0.2	100.0

Figure 29. Main Region of Injury Caused Death (National Police Agency, 1973)

A PROPOSED NEW NATIONAL SYSTEM FOR COLLECTING MULTIPURPOSE ACCIDENT DATA: SIR

JAMES O'DAY
Head, Systems Analysis
Highway Safety Research Institute
The University of Michigan

INTRODUCTION AND SUMMARY

About 100 million vehicles operate on the roadways of the 50 United States. Each calendar year, about one-quarter of those vehicles is involved in some kind of traffic accident. Summary data on those accidents are published by the National Safety Council and at times by federal agencies, and this information is widely distributed. It might appear to Europeans that the U.S. has a long-established, precise, and consistent accident reporting system. However, the fact is that the United States has no national system for collecting traffic accident data. Because it does not, aggregated data obtained from various federal, state, and local sources cannot provide reliable and generally accepted answers to questions about causes of accidents and injuries, the value of particular countermeasures, or the value of particular safety standards for cars, highways, or drivers.

Beginning in about 1968 the U.S. government began establishing a small number of in-depth or multidisciplinary accident investigation teams, developing investigative techniques, and collecting information about a small number of collisions. This effort was more sporadic than systematic, and while the data has been complied, it has seen little actual use. In the latter half of 1972 and early 1973, under contract with the National Highway Traffic Safety Administration, the University of Michigan's Highway Safety Research Institute examined several existing accident investigation programs and attempted to answer two very basic questions:

(1) What can we do and what can we not do with the existing in-depth accident data?
(2) How can we create a national system that overcomes the principal deficiencies of the present data at a feasible cost?

The output from the current NHTSA multidisciplinary accident investigation program was found to have been useful in many applications. However, it does not represent a sample from a well-defined accident population, and has therefore reduced to conjecture any inferences about the national accident population. Further, the program in the past has been incapable of supporting "one-time" special studies designed to obtain quick and reliable answers needed by safety program managers.

Sufficient data of the requisite precision and representativeness could be economically collected by 30 skilled investigative teams operating in 30 carefully defined zones of the United States under direction and control of a single federal or federally supported agency. Such a data collection system would cost about $6 million per year. The sooner it is put into effect, the sooner government and industry will begin getting conclusive answers to cause-and-effect questions and definitive answers to cost-benefit questions.

PROBLEMS WITH PRESENT DATA

The data now available on traffic accidents have been collected principally through two processes: police reports and multidisciplinary research studies. Police agencies routinely create a vast amount of accident data. But, because the reports are so undetailed, and because reporting criteria vary so widely among police jurisdictions, this information is not very useful for analytic purposes; it cannot be combined to create any valid representation of national or regional accident populations. Such combinations of police-reported data are neither precise enough nor consistent enough to support accurate and reliable conclusions regarding cause-and-effect mechanisms in the national population of traffic accidents.

Accident data collected by multidisciplinary teams over the past several years is much more detailed and precise than police-reported data. Yet the data in these collections of case studies have some of the same limitations as police-reported data. Most of these studies have been "special," in the sense that the investigators were studying particular types of accidents in a particular geographic area. The crashes reported on cannot be considered a representative sample of either the variety of accidents occurring in that area during that period, or representative of the frequency of that type of accident for that area and that time period. The data do not permit analysts to define those samples of accidents in terms of how they relate to the total universe or population of all accidents that were occurring in those geographic areas during those time periods. European highway safety analysts are having the same problems with inconsistent and incomparable data. As G. M. Mackay reported last year at the International Conference on Experimental Safety Vehicles, the Peugeot-Renault investigative team has been concentrating mainly on expressway collisions, the Fiat and Volkswagen teams had been studying single-car collisions with rigid objects, and, in general, data collected by all five of the European teams discussed were biased fairly heavily toward the severe end of the accident spectrum.

In the few projects in which data have been collected in such a manner as to truly represent some small region of the U.S., those data are used as a base for drawing inferences about the national accident population. However, the validity of those inferences depends upon the assumption that the accident population in the given small region has the same characteristics as the national accident population. While such inferences are indeed drawn, the authors of such projections must be considered engaged in a hazardous occupation when their critics converge on their data.

A full census of traffic accidents--i.e., a detailed report on each accident, consistently prepared by a trained investigator and compiled quickly on a large computer--would solve the problem of

accurately representing the national accident experience. Police agencies do report on this number of accidents each year, but their reports lack the detail and consistency required. And the cost of sending fully trained teams to all accidents would be exorbitant.

An alternative to a census (and, as we shall see, a more practical alternative) is to have enough skilled investigators studying enough crashes--on a continuing basis and using consistent procedures--so that analysts can construct and maintain a data file that is actually an accurate and reliable sample--a representative model--of all U.S. traffic accidents of all types as they occur during specified periods of time.

This "model" or "microcosm" of the total universe of U.S. traffic accidents would not have to contain hundreds of thousands of accident cases. As few as 1,250 case reports per month, or 15,000 per year, would suffice to build and maintain a data file that would accurately and reliably represent many important characteristics of the nation's traffic accidents. Once this carefully defined model of all accidents were set up, analysts would be able to observe modest changes in national phenomena very soon after they occurred--for example, changes in fatality and injury rates as a result of seat belt and air bag usages, or as a result of design changes involving frames, hoods, windshields, instrument panels, steering columns, fuel systems, and other components.

It is important that these changes be detected and analyzed, because this is the final evidence for government and industry to learn just how well any particular changes in car design, highway design, or driver licensing practices are accomplishing what they were intended to accomplish. Many people want to know <u>how well</u> a given change is performing--whether it is preventing accidents, or how many accident injuries or fatalities it is eliminating. To know this requires having relatively unbiased comparative data which represent the number and types of accidents and injuries occurring. This is what cannot be accurately determined from data collected in current programs.

USES OF PRESENT DATA

Although current data are inadequate for purposes of modeling national phenomena accurately, they provide a reasonably useful means of establishing various relationships among characteristics of the accidents studied by multidisciplinary teams. For example, relations have been established between impact speed and vehicle damage, between vehicle damage and occupant injuries, and so forth. By establishing those relationships in the multidisciplinary data and then applying those findings to police-reported data, analysts can arrive at rough and proximate conclusions concerning national losses resulting from various kinds of accidents.

Consider, for example, the problem of fuel leakage in collisions. How serious is this problem? To establish its magnitude, one must know what kinds of collisions result in such leakages, in what percentage of those collisions do the gas tanks or lines leak, and how frequently each of those kinds of collisions occur. HSRI conducted such a study and produced the findings briefly summarized in the following table:

TABLE 1.

PERCENTAGE OCCURRENCE OF FUEL LEAKAGE IN VARIOUS TYPES OF COLLISIONS (DERIVED FROM MDAI DATA RECORDS)

Impact	Extent of Vehicle Damage: MINIMAL	MODERATE	EXTENSIVE
FRONT	1%	5%	12%
SIDE	3%	9%	25%
REAR	15%	36%	60%

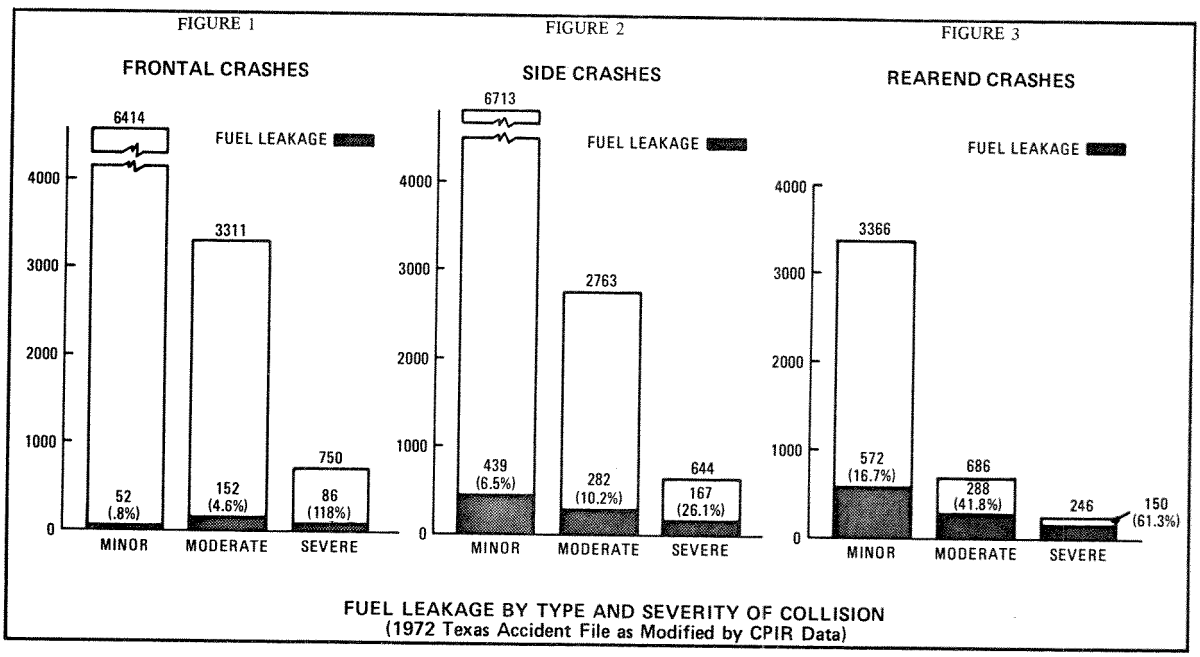
TABLE 2.

FUEL LEAKAGE AS PERCENT OF ALL COLLISIONS (DERIVED FROM TEXAS POLICE ACCIDENT REPORTS AND MDAI DATA RECORDS)

Impact	Extent of Vehicle Damage: MINIMAL	MODERATE	EXTENSIVE
FRONT	0.26%	0.74%	0.36%
SIDE	0.81%	0.99%	0.62%
REAR	2.00%	1.00%	0.30%

The finding that fuel leakage most often occurs as a result of rear-end crashes inflicting extensive damage to the vehicle merely confirms what one might assume from knowledge that gas tanks are located near the rear end of most vehicles. However, relatively few such crashes occur. To establish the magnitude of this problem in terms of all the actual numbers of accident-produced fuel leakages in all U.S. traffic accidents, these percentages were applied to police-reported data on the frequencies of those various types and severities of accidents. That produced these findings: (TABLE 2)

Those findings are represented in histograms in the following Figures 1, 2, and 3.



FUEL LEAKAGE BY TYPE AND SEVERITY OF COLLISION
(1972 Texas Accident File as Modified by CPIR Data)

The findings show that rear-end collisions inflicting only minimal damage to the vehicle produce twice as many cases of fuel leakage as comparable crashes resulting in moderate damage, and almost seven times as many cases of fuel leakage as comparable crashes resulting in extensive damage to the vehicle.

That is an example of one kind of analysis that can be conducted with current data. Some other uses of the data can be illustrated by one more example.

Consider this question: If crash investigators know enough about the severity of a crash, how accurately can they state the severity of the injuries sustained by the front-seat occupants? For the presently used Injury and Vehicle Damage Scales, a rough linear relationship exists; that is, in general, the more severe the crash, the greater the likelihood of severe injuries. However, until recently, analysts have made little progress in developing a means of predicting one from the other very accurately. To develop a better means of relating crash severity to injury severity, HSRI researchers analyzed multidisciplinary data to isolate crash characteristics that related to injury severity in some way. These four factors were found to be relevant:

(1) Vehicle Damage Index (VDI)
(2) Impact velocity
(3) Whether or not the windshield bond separated
(4) Whether or not the occupant was ejected

The case files used for these analyses contain numeric values assigned from scales in two standard reporting instruments used by multidisciplinary investigators. One, the Abbreviated Injury Scale (AIS), identifies the severity of injuries by a scale ranging from 0 to 9. The other, the Vehicle Damage Index (VDI), identifies the extent of vehicle damage by a scale ranging from 0 to 9.

By conducting a series of "retroactive experiments" involving various subgroups of actual crash data in these multidisciplinary files, it is possible to arrange these four relevant factors within a quantitative multiple regression model. The best-fit model for this set of data is:

$$\begin{aligned}\text{Predicted AIS} = & -.21 + .34\ (\text{VDI extent code}) \\ & + .29\ (\text{impact velocity squared})^{*} \\ & + 3.00\ (\text{if occupant was ejected}) \\ & + .74\ (\text{if windshield bond separated})\end{aligned}$$

This model is used, then, to compute the expected injury to occupants of crashed vehicles (presented as an average value of the AIS codes assigned) as a function of the four factors.

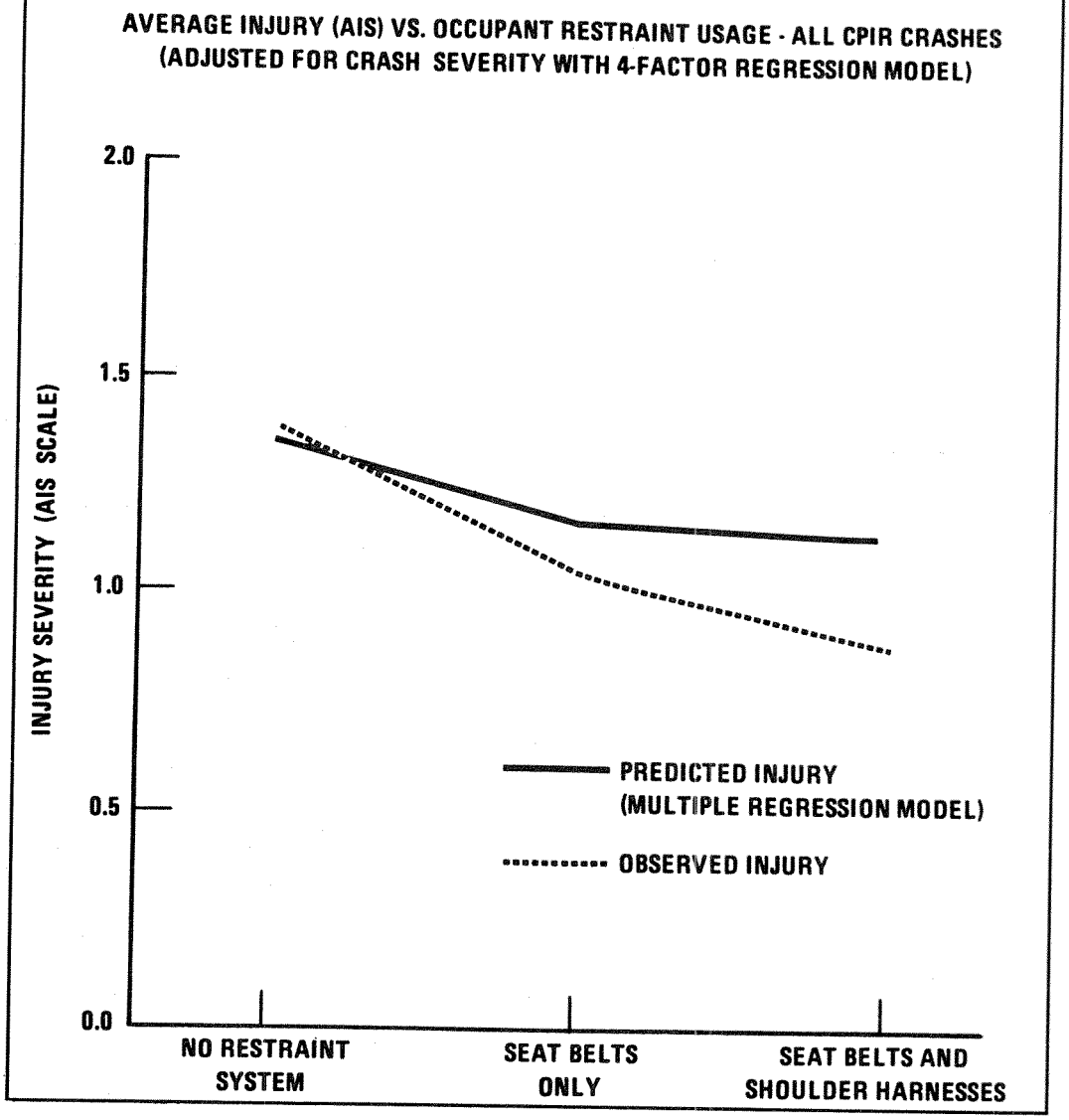
If the crash-injury process were perfectly understood (and the necessary data recorded), one might be able to develop a model that would predict exactly what injury would result from a given crash. In the existing data, the Vehicle Damage Index alone predicts injury in a crash reasonably well; the R^2 (the correlation coefficient squared) is 0.42. The model described above yields a multiple correlation coefficient of 0.53, and thus is a better estimator of injury extent than the VDI alone.

This multiple regression model has been applied to subgroups of crash cases to study the relative effect of various factors (such as restraint systems) on injury reduction. Carlson, for example, has

*The velocity in this term is numerically a relative velocity in miles per hour divided by the square root of 1000. For example, for a frontally damaged car in an intersection collision, V=that vehicle's impact velocity divided by 1000. The constant was chosen as a matter of convenience to keep the coefficients in a limited range.

reported (see "Can an Automotive Designer Find Happiness with AIS," by William L. Carlson, SAE Paper 740300, presented at the Automotive Engineering Congress, March 1, 1974) that the mean injury to occupants with seat belts and shoulder harnesses is about 0.35 AIS units less than that for persons without restraint, as shown in Figure 4. Note that this result is now independent of (i.e., adjusted for) variations in crash severity.

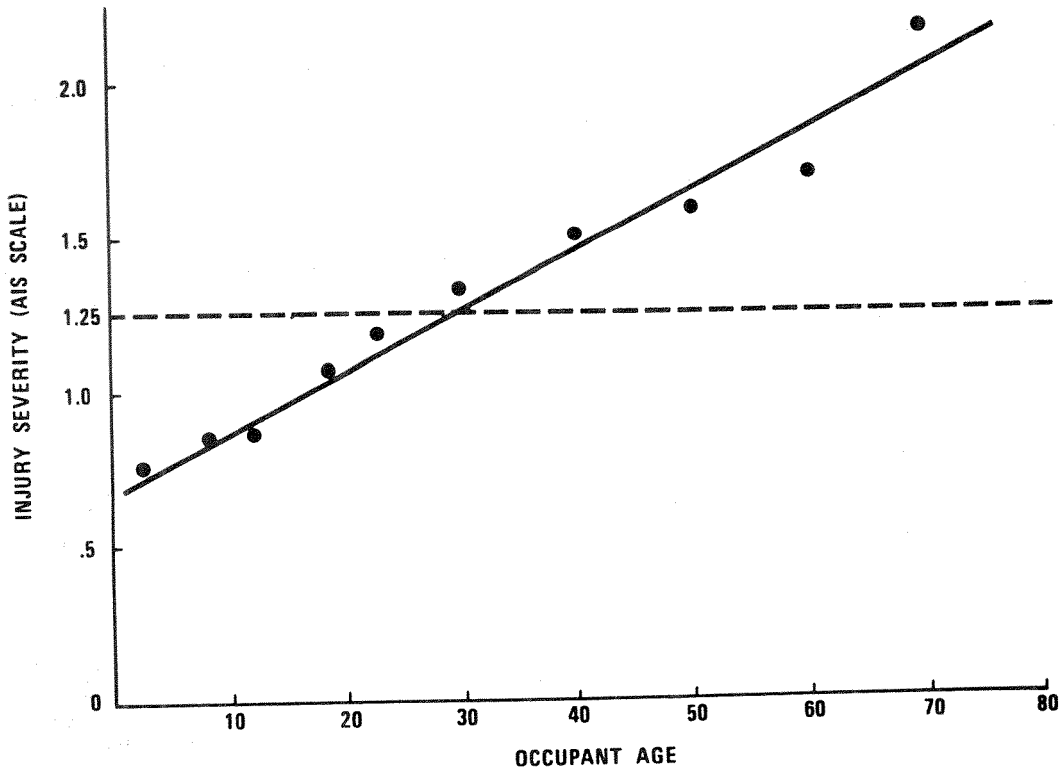
FIGURE 4



By itself a change of 0.35 AIS units is not very meaningful. But Carlson also reported the change in average AIS as a function of occupant age (see Figure 5), showing that 0.35 AIS units is equivalent to a change in occupant age of about 22 years (from, say, age 40 to age 18). This comparison is perhaps more understandable--i.e., the wearing of a full restraint system will make you 22 years younger--at least as far as your susceptibility to injury is concerned.

FIGURE 5

AVERAGE INJURY (AIS) VS. OCCUPANT AGE - ALL CPIR CRASHES (ADJUSTED FOR CRASH SEVERITY WITH 4-FACTOR REGRESSION MODEL)



Conclusions produced through studies of this kind still do not possess the precision which would be obtainable from similar analyses of data more rigorously defined and collected as part of a national sampling plan of traffic accident phenomena. Nonetheless, such studies of current data illustrate that the multidisciplinary case files do serve a variety of useful purposes.

A PROPOSED THREE-PROGRAM SYSTEM OF DATA COLLECTION: SIR

As the changes resulting from ESV programs are introduced, vehicles with these characteristics are going to have collisions. The collection of accident information for a small number of collisions will continue to provide anecdotal information about the likely value of component changes, but information of statistical value will have to consist of detailed damage and injury information along with data on crash configuration, occupant characteristics, and vehicle characteristics. We will see the fielding of designs which have undergone extensive analysis, laboratory tests, and controlled crash tests. But the final values will be best demonstrated with

performance on the highway. A national system of data collection must be efficient and flexible enough to provide information suited to the multiple needs of government and industry. We have recommended three coordinate kinds of data collection programs:

(1) A Sampling program, to collect data for an accurate and sensitive model of national phenomena.
(2) An In-depth program, to collect multidisciplinary data useful for identifying new problems.
(3) A Rapid-response study program to collect data needed for answering pressing questions.

Each of these activities would be performed by small groups of professional accident investigators operating in carefully defined geographic regions. To ensure a high quality and consistency of reported data, the investigators would have to be trained professionals and their activities would have to be continuously monitored. Wherever feasible, the investigative team in each geographic area would conduct all three activities in the SIR system--Sampling data collection, In-depth studies, and Rapid-response short studies of particular questions. However, it would not be essential that all teams conduct In-depth studies.

The Sampling Program

This program would entail positioning 30 teams of skilled investigators in 30 different areas of the country. Each team would investigate approximately 500 accidents per year. These 15,000 cases, reported in medium detail, would provide sufficient data to construct and maintain a probability sampling model of U.S. traffic accident phenomena.

The cases to be investigated in each jurisdiction would need to be selected in ways that satisfy the requirements of probability sampling. Essentially these requirements are that the probability of occurrence of the type of case selected be known, and that, for example, each nth case of that type be investigated. This does not mean that the same percentage of all types of cases must be investigated, but merely that the cases investigated bear a known relation to the incidence of other cases of that type occurring in the jurisdiction during that time period. The cases selected must also be related to population figures for the accident investigation jurisdiction. In the U.S. this can be accomplished by defining the investigative jurisdictions in terms of Standard Metropolitan Statistical Areas (SMSA's) now used by the Bureau of the Census and other federal agencies, and then defining a representative subsector of that SMSA as the investigative jurisdiction for the team operating within that SMSA.

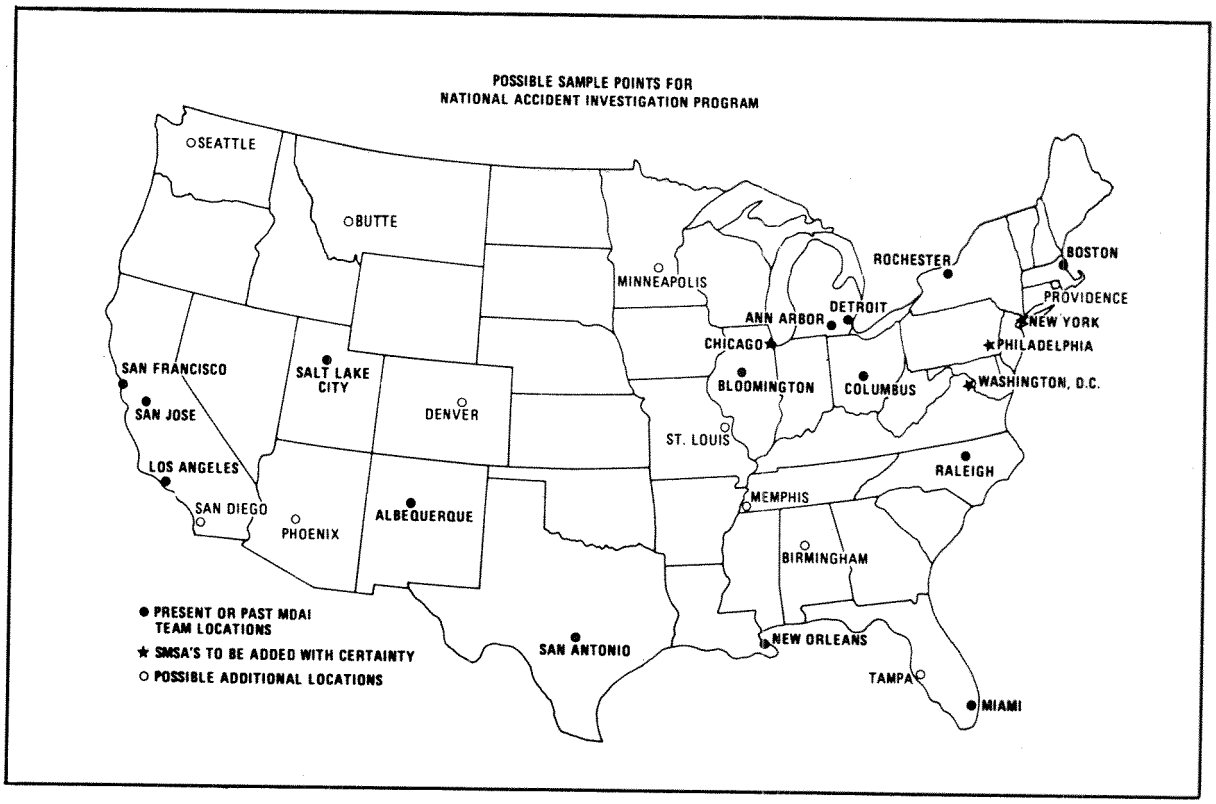
Probably the most difficult statistical problem in this design is the establishment of a precise and repeatable local sampling procedure. In order to evaluate such changes as restraint systems, interior protection, or the effect of controlled crush designs, it is useful to define an accident population on the basis of some crash dynamic criterion, and then to compare injuries within such a population. One approach to this has been to look at all "towaway" vehicles, with the expectation that this defines a level of crash severity. Other possibilities would be to control on severity as measured by the vehicle damage index, and to compare injuries as has been discussed above, using some sort of regression model. The

towaway criterion seems straightforward, and it is not until one tries to use this that problems and uncertainties arise. In our own experience, we find that the definition of towaway is different in different police jurisdictions. For example, we find a small number of cars which were towed from the scene because the driver was too drunk to drive. Others are towed in one jurisdiction (e.g., from a freeway) because of a flat tire, when they would be driven from the scene on a less busy highway. We have seen cars towed from a ditch, but then driven from the scene. Thus if we want to have comparable populations of towaways from several jurisdictions, some very careful definition is required.

Given this definition, and the need for a certain quantity of collisions for statistically sound measurement, a field implementation of sampling of the population is in order. It is important that the sampling not be done as a matter of convenience--i.e., recording data on the vehicles that are easiest to get to and dropping those that are not--but rather on a definable basis which truly represents the target population. In one program which we are currently conducting we are sampling by using the last digits on the license plate, which are assigned in a random fashion. There are other possibilities for defensible sampling plans. But, however they are implemented, one must be able to reconstruct the desired population in a mathematical sense for analysis.

In a national sampling program the investigative teams must rely on local agencies for identification of accidents for investigation and the local reporting agencies serving the 30 investigative teams would need to use comparable definitions. The Sampling model can be only as precise as its data inputs. Thus the problem of establishing definitions and reviewing reporting practices of local agencies not directly controlled by the investigative teams or program director requires considerable thought and planning, probably pilot-testing, and certainly monitoring of operations by the controlling agency.

Analyses of the locations of present and past multidisciplinary teams with respect to present U.S. Standard Metropolitan Statistical Areas have shown that 15 teams have been located close enough to where they should ideally be to satisfy the populational requirements of an accurate Sampling program. However, an additional 15 teams would need to be set up in other SMSA's throughout the U.S. to satisfy the overall requirements of the Sampling model. These new teams would have to be recruited, trained, and set up, though not necessarily all at the same time. The following map shows the SMSA's, locations of present or past teams, and where new teams would need to be established.



The In-depth Program

This program would entail having each of the 30 teams conduct detailed investigations of about 20 accident cases per year. As with present in-depth case studies, the investigators would conduct detailed interviews with occupants and witnesses, make detailed measurements of vehicle damage, take numerous photos of the vehicles and scenes, obtain full medical reports, state their opinions of causative factors, and recommend countermeasures.

No attempt would be made to make In-depth cases statistically representative of either the local or national accident population. The investigative teams would strive to obtain reports on a wide range of types of accidents, so that, over time, the In-depth records would contain data on almost every conceivable type of accident.

The files of these In-depth cases would be considered a reference library. They would permit researchers to read about several cases having common characteristics of one kind or another, so that they might identify emerging problems and generate hypotheses to be tested by analyses of data compiled in other programs. In contrast with the case investigations pursued in the Sampling and Rapid-response programs, teams would have considerable freedom in selecting In-depth cases to be investigated, and would tend to select cases related to particular research interests of team members. This is a strong inducement to scientific personnel whose findings would constitute a continuing influence on the substantive and procedural aspects of the other two programs in the SIR system. Although all 30 teams would be free to pursue In-depth case studies, there is no overriding systematic or programmatic requirement that all teams do so, or, if they do, that each team complete some specified minimal number of investigations per month or per year.

Past MDAI programs have frequently emphasized particular segments of the accident population--Rochester University has studied neck injuries, Stanford Research Institute has studied vehicle defects as accident causation factors, and several teams have supported local Alcohol Safety Action programs. This procedure is at once useful (in the sense that qualified experts are able to comment on accidents in their fields of knowledge), and harmful (in the sense that the statistical value of their combined data is limited). But if these cases are viewed as individual reports, they can lead to identification of new problems.

The Rapid-Response Study Program

This program of special investigations would involve having each of the 30 teams investigate some moderate number of accidents* each year. The type of accident to be investigated in each study would be specified by the central agency. Each special study would provide data rapidly for analyses of problems identified by the NHTSA, but only after a search of existing files had determined that the question cannot be answered conclusively from data already on hand.

The central agency would, for each Rapid-Response study, provide the field teams with the criteria to be used in selecting cases, the variables for which data are to be collected, and the format to be used in reporting on each case as soon as it is collected. These directions could be teletyped to the investigators, and the teams would teletype their reports, case by case, from their site directly to the computer at the central agency. The central agency would analyze these data frequently to determine when enough cases had been inputted to answer the study question, at which point the field teams would be notified to terminate their investigations for that study. The basic purpose of this Rapid-Response Study Program is to provide a rapid mechanism for obtaining quick answers to important questions. For most special studies, the directives could be transmitted, the cases investigated and reported, the data analyzed, and the study question answered within four to six weeks.

Examples of problems suitable for the Rapid-Response study program include current questions relating to steering columns, air bags, bumper systems, roadside obstacles, unlicensed drivers, recreational vehicles, shoulder harnesses, warning buzzers, school bus accidents, and so forth. With steering columns, for instance, one objective would be to find out how effective energy-absorbing columns are in reducing injury severity in frontal collisions. Collisions involving new-model cars equipped with an E-A column would be matched (by the central agency) on the basis of collision severity (impact speed and VDI) with collisions involving older cars unequipped with an E-A column. The main data item sought would be the extent of injury to the driver, and the sample would likely be subdivided further on some control variables such as the use of restraint systems.

*The number of investigations will depend on the type of case desired.

Similarly, for a study of the effectiveness of child restraint seats in reducing injuries, a Rapid-Response study would collect data on accident cases involving child passengers up to five years old. The data needed would be the number of occupants in the car; the VDI; the age and sex of the driver and the child or children; the seated position of the child; whether it was restrained by a lap belt, child seat, or nothing; and AIS data on the child and driver. Such a study would establish not only the protective benefits of child restraint seats but the incidence of their usage.

COMPARABLE FEDERAL INFORMATION SYSTEMS

The basic idea of a national system for collection of traffic accident data has several parallels in programs now conducted by various federal agencies. All of these programs were designed to solve the problem of how to obtain national data accurate, reliable, and consistent enough to support conclusions concerning the phenomena of interest, and to detect changes in those phenomena quickly and reliably. One of the oldest of these programs is the Current Population Survey (CPS) conducted by the Bureau of Labor Statistics. One of the best-known items of information the CPS develops is the national unemployment rate. This is computed from a probability-sample survey of 50,000 U.S. households each month. The results are available within two weeks after the interviews are completed.

Another well-designed sampling program is the Consumer Price Index. The CPI is constructed from survey data relating to 52 categories of expenditures for food, housing, apparel, upkeep, transportation, health, and recreation.

A newer example of federally applied survey techniques is the National Crime Survey recently field-tested by the Census Bureau and the Law Enforcement Assistance Administration. This program is designed to measure the nation's crime rate through interviews of 60,000 families and 10,000 businesses each month. The program yields monthly crime statistics which will identify problem areas and lead to well-considered allocations of federal, state, and local resources.

The principle common to those three examples of national survey programs--and to a national sampling system for collection of accident data--is that the surveys have well-defined objectives that determine what data must be collected, by whom, where, how, and how often. The National Crime Survey Program, for example, was begun only after it was concluded that data collected and reported by local, autonomous jurisdictions throughout the U.S. contain far too many biases and inconsistencies to be used as the basis for any accurate national index of changes in the frequencies of various types of crimes. A parallel situation exists with traffic accident data. Local agencies lack the trained manpower, standard forms and procedures, and sufficient incentives to provide adequate accident data.

The system described here is intended to provide, for the traffic accident problem, the same sort of national survey capability. The three essential activities of the system are the highly structured sampling of about 15,000 collisions each year, the in-depth reporting by expert analysts of a small number of cases, and a series of short-term data-taking operations to solve pressing problems.

Based on the costs of previous MDAI operations, we have estimated that the entire cost of these activities would be of the order of 6

million dollars per year, of which approximately half would be devoted to the strict sampling activity. The remainder would be divided between the in-depth and special studies as necessary.

Considered strictly on the basis of dollars per item of data collected, the proposed SIR system is somewhat more expensive than past programs. However, data collected according to a careful plan will permit valid and reliable inferences to be drawn to the national population of accidents. Past programs have not met the overall information needs of the NHTSA and the automotive industry; the SIR programs can meet those needs.

EMPIRICAL RELATIONSHIPS BETWEEN CAR SIZE, CAR WEIGHT AND CRASH INJURIES IN CAR-TO-CAR CRASHES

BRIAN O'NEILL
Insurance Institute for Highway Safety

HANS JOKSCH
The Center for the Environment and Man, Inc.

WILLIAM HADDON, JR.
Insurance Institute for Highway Safety

ABSTRACT

Relationships between car size, car weight, and severity of occupant crash injuries in car-to-car crashes are developed using a simple mathematical model to analyze real-world crash data.

For vehicles using the same roads, these relationships suggest a crashworthiness design concept for intervehicular crashes that regards increases in vehicle *size* as primarily *protective,* and *increases* in vehicle weight as primarily *hostile,* indicating the desirability of relatively sizeable but not heavy vehicles.

INTRODUCTION

The present highway environment is populated by vehicles differing widely in their sizes, shapes, weights and performance characteristics. Because of these differences, the vehicles in which humans travel and frequently crash are often incompatible with each other in terms of occupant crash protection.

The proportion of small (sub-compact, compact and import) cars in the U.S. population of all vehicles has been increasing for several years. In 1969, approximately 17 percent of the new vehicles registered were small cars (1).* In 1973, this proportion increased to approximately 27 percent (2), and sales reports for 1974 indicate that this trend is continuing. At the same time, the percentage of new trucks registered has also increased from approximately 17 percent to over 25 percent (1, 2). These increasing numbers of vehicles with weights at the extremes of the vehicle population will accentuate the problems of crash incompatibility.

Research studies have consistently shown that the occupants of smaller cars, once they become involved in crashes, tend to sustain more severe injuries than the occupants of larger cars.

A 1968 New York State study (3) based on more than 400,000 vehicles in reported crashes found a "strong association" between "the weight of a car and the percent of accidents in which there was a fatality or serious injury in that type of car." Results of that study showed a large, exponential increase in the percent of serious or fatal injuries with decreasing vehicle weight.

Campbell (4) concluded that occupants of "larger cars ... show generally less injury than average," while occupants of "small cars ... show generally more than average injury" when involved in crashes. Although the Campbell study demonstrated a general relationship between vehicle size and severity of injury, it also showed that there were considerable variations between individual vehicle types.

The problems in intervehicular crashes between incompatible cars were demonstrated by filmed Insurance Institute for Highway Safety (IIHS) tests in 1971 in which four separate pairs of vehicles, in each case one large and one small car produced by the same manufacturer, were crashed head-on. Each vehicle in each crash was travelling at about 48 mph (8). In each case the passenger compartment of the smaller car was demolished, while that of the larger remained intact and virtually unchanged.

O'Day, et al., (5) showed that "once involved in an accident, the chance of injury ... increases at the rate of about 2.5% for each decrease of 100 pounds in vehicle weight." In the O'Day, et al. study it was found that the percent of cars in which there was a minimum of one injury was approximately 30 percent higher in head-on crashes between two "small" ($\leqslant$3100 lbs. each) cars than for two "large" cars ($\geqslant$3300 lbs. each). When all crashes between two vehicles of the same weight were combined, the percent of cars in which there was a minimum of one injury did not differ substantially for crashes of "small" and "large" cars. It should be noted, however, that only two vehicle weight classes were used and the "small" cars included "the lighter half of the intermediates," cars not normally considered small. Furthermore, *all* injuries, including minor injuries, were considered; differences in the percent of more severe injuries were not reported.

Two very recent studies from the University of North Carolina have added to the growing evidence that light car occupants face much greater risk of injury and death in crashes than those faced by occupants of heavy cars. In one of these studies, Campbell and Reinfurt (6) presented results showing that the risk of severe injuries and death was approximately twice as great when two lighter cars (<2,700 lbs.) crash than when two heavier cars

*Numbers in parentheses refer to references found at end of this paper.

(≥4,100 lbs.) crash. In the other study (7) the results showed that belted drivers in sub-compact cars were seriously or fatally injured as frequently as unbelted drivers in full size cars.

There is conflicting evidence concerning the relative frequency with which small and large cars are involved in crashes. Highway Loss Data Institute (HLDI) results have shown that, in general, small cars have higher insurance collision coverage claim frequencies than full size cars (9, 10). In addition, Casassa, et al. (11), in a survey of unrepaired automobile crash damage, reported that the percentage of passenger cars that each year sustained crash damage that was left unrepaired was higher for small cars than for large cars. Milic (12), however, in a study of reported crashes in New York State observed that "Many of the cars with the lowest accident rates are among the lightest and cheapest . . ., and many with the highest rates are heavier, more expensive cars." This could be due to the greater likelihood of reporting minor crashes involving newer and more expensive cars, what the author calls the "Luxury Effect." It is important to note that despite the apparent lower crash frequency for the lightest cars found in that study, serious and fatal crash injuries per registered vehicle were still higher for these lighter cars than for full size cars.

Thus, there is considerable evidence that the decreasing size of the cars in the vehicle population may tend to generate more severe and more frequent losses.

These various studies, however, provide no basis to separate the effects due to vehicle size differences and the effects due to vehicle mass differences. In fact, in many of the studies the terms size and weight are used interchangeably. This is understandable since, in general, there is a relationship between the physical dimensions and the weights of vehicles; weights tend to increase in direct proportion to size. There are, however, exceptions to this general relationship. There are, for example, models with relatively small physical dimensions whose weights are similar to those of larger vehicles. To the contrary, increasing use of lightweight materials may lead to lighter cars without corresponding decreases in size.

In this paper, some elementary relationships between car size, car weight and severity of occupant crash injuries in car-to-car crashes are explored and developed using reasonable, simplifying approximations.

METHOD

The following simple model showing a relationship between car size, car weight and driver crash injuries was postulated:

$$X_{ij} = \alpha + \beta i + \gamma(i-j) \text{ for } i \geqslant j \qquad (1)$$

and

$$X_{ij} = \alpha + \beta i + \partial(j-i) \text{ for } j > i \qquad (2)$$

where

X_{ij} = predicted percentage of driver crash injuries in cars of group i colliding with cars of group j,

α = a constant,

β = a coefficient representing the vehicle size effect,

γ = a coefficient representing the benefits of an intervehicular mass difference,

∂ = a coefficient representing the penalties of an intervehicular mass difference,

and

i = vehicle size and weight groups.

Equation (1) represents the percentage of driver crash injuries in cars of group i colliding with cars of group j, where group i cars have greater (or equal) mass than group j cars. Thus, equation (1) represents the situation where the mass differentials should tend to benefit the occupants of cars from group i by reducing the percentage of injuries.

Equation (2) represents the percentage of driver crash injuries in cars of group i colliding with cars of group j, where group i cars have less mass than group j cars. Thus, equation (2) represents the situation where the mass differentials should tend to penalize the occupants of cars from group i by increasing the percentage in injuries.

The form of the model was suggested by elementary theory which indicates that both vehicle size and vehicle weight can be expected to be related to the occurrence of occupant injuries. Two separate intervehicular mass difference coefficients were used because elementary Newtonian physics suggests that when two unequal masses collide the increment in deceleration, due to the mass difference, experienced by the lighter mass is greater than the corresponding reduction in deceleration experienced by the heavier mass.

Several alternative models were investigated, including multiplicative models, non-linear models, and models using mass ratios; in general, the fits of the data to these alternative models did not improve over the model specified by equations (1) and (2).

RESULTS

Campbell and Reinfurt (6) presented results based on 23,070 real-world crashes showing the percentage of serious and fatal crash injuries to unbelted drivers

of 1966 through 1970 model cars in three configurations of car-to-car crashes; front-into-front, front-into-side, and front-into-rear. For each crash configuration the results were presented in a 6 x 6 matrix in which the six rows and columns represented different car weight groups. The six weight groups were:

1	<2700 lbs.
2	2700 - 3099 lbs.
3	3100 - 3299 lbs.
4	3300 - 3699 lbs.
5	3700 - 4099 lbs.
6	≥4100 lbs.

The results showed the percentage of serious and fatal driver injuries in crashes in which the specific weight categories for both cars were known.

The injury percentage as presented in the Campbell and Reinfurt study were smoothed to reduce the random fluctuations. For purposes of the present analysis, the raw, unsmoothed percentages were obtained. These are presented in Table 1, for the

TABLE I

PERCENTAGES OF SERIOUSLY AND FATALLY INJURED DRIVERS IN FRONT-TO-FRONT CRASHES BY VEHICLE WEIGHT (CAMPBELL AND REINFURT 1974)

FRONT–STRIKING CAR	1	2	3	4	5	6
6 (≥4100 lbs)	4.6 / 86 / 11.6	6.1 / 98 / 13.3	4.0 / 149 / 10.7	5.1 / 156 / 5.8	2.5 / 162 / 3.7	5.8 / 87 / 10.4
5 (3700 - 4099 lbs)	4.6 / 151 / 13.2	3.8 / 183 / 10.4	5.3 / 225 / 8.4	5.0 / 278 / 6.1	4.0 / 249 / 4.4	6.5 / 153 / 5.2
4 (3300 - 3699 lbs)	5.2 / 134 / 11.9	6.2 / 162 / 8.6	7.6 / 212 / 9.4	5.9 / 236 / 5.5	6.6 / 241 / 6.6	4.9 / 142 / 2.1
3 (3100 - 3299 lbs)	5.4 / 111 / 11.7	5.8 / 156 / 11.5	9.2 / 174 / 10.4	9.4 / 213 / 9.9	5.6 / 214 / 6.1	8.5 / 121 / 5.0
2 (2700 - 3099 lbs)	7.8 / 102 / 12.8	5.9 / 152 / 4.0	8.0 / 175 / 8.6	7.0 / 172 / 5.2	8.7 / 184 / 6.5	8.2 / 122 / 3.3
1 (<2700 lbs)	12.5 / 80 / 8.8	11.6 / 121 / 6.6	6.4 / 126 / 3.2	12.4 / 145 / 3.4	9.3 / 140 / 1.4	10.6 / 94 / 5.3
CAR GROUP	1	2	3	4	5	6

FRONT–STRUCK CAR

For matrix cell at the intersection of row i and column j:

% injuries in group i cars colliding with group j cars	% of injuries in group j cars colliding with group i cars
No. of collisions	

front-to-front crashes (13). These data were used to estimate the parameters in the following model.

It should be noted that the interpretation of β as a vehicle size coefficient is an assumption based on theory which suggests that vehicle size is an important parameter. The values of β should be interpreted as applying to the size classes that correspond to the six weight groups for 1966 through 1970 model automobiles. It should be noted, however, that it is not possible to directly separate size and weight effects from the Campbell and Reinfurt data.

Using their data, the parameters of the model, α, β, γ, and ∂, were estimated by a single least squares multiple regression. The fit of the front-to-front crash data to the model was good; the variance in the original data was reduced 56 percent by the fitted model. In this symmetrical crash impact case there were 72 fitted data points because the matrix had two observed values for each weight group crash combination. The following parameter estimates were obtained:

$$\hat{\alpha} = +9.97$$

$$\hat{\beta} = -0.83$$

$$\hat{\gamma} = -0.32$$

$$\hat{\partial} = +0.46$$

The largest absolute value for the three estimated coefficients was $\hat{\beta}$ (-0.83) suggesting that the effect of vehicle size is more important than vehicle mass differences. As the basic theory predicted, however, the estimated mass difference coefficients, $\hat{\gamma}$ (-0.32) and $\hat{\partial}$ (+0.46), suggest that penalties associated with a mass difference are greater than the benefits.

Similar models were fitted to the unsmoothed data for the front-to-side and front-to-rear crashes, Tables 2 and 3 (13). In each of these crash configurations two sets of parameters were estimated because the crash configurations are not symmetrical as they are in the front-to-front case, and the percentages of driver injuries vary substantially with crash configuration. Therefore, separate parameters are necessary to represent the percent of injuries in the striking and struck cars in the non-symmetric crash configurations. Each set of parameters was therefore estimated from 36 observed percentages rather than 72 as previously.

In the front-to-side configuration the following parameter estimates were obtained for the front-striking cars:

$$\hat{\alpha} = +6.86$$

$$\hat{\beta} = -0.74$$

$$\hat{\gamma} = -0.04$$

$$\hat{\partial} = +0.39$$

The fit in this case was good; the variance of the original data was reduced 49 percent by the fitted model. As before, the largest absolute value for the three estimated coefficients was $\hat{\beta}$ (-0.74), representing the effect of vehicle size. Again, as the basic theory predicted, the estimated coefficients suggest that penalties (+0.39 associated with a mass difference are greater than the benefits (-0.04). The very small estimated value obtained for the mass difference benefit coefficient, $\hat{\gamma}$ (-0.04), suggests that in this crash configuration only small benefits are obtained by a mass difference. The size of the constant, $\hat{\alpha}$ (+6.86), indicates that this crash configuration – front-to-side – in general produces fewer injuries to the drivers of the front-striking cars than in the front-to-front configuration.

For the front-to-rear crash configuration the following parameter estimates were obtained for the front-striking cars:

$$\hat{\alpha} = +3.38$$

$$\hat{\beta} = -0.46$$

$$\hat{\gamma} = +0.02$$

$$\hat{\partial} = +0.28$$

Again, the fit was good; the variance of the original data was reduced 51 percent by the fitted model. As in all of the previous cases, the largest absolute value for the three estimated coefficients was $\hat{\beta}$ (-0.46). In this case, the weight benefit coefficient, $\hat{\gamma}$ (+0.02), had a very small positive value suggesting that there is no substantial benefit associated with a mass difference for front-striking cars in front-to-rear crashes. Again, however, the weight penalty coefficient, $\hat{\partial}$ (+0.28), suggests that the penalties of a mass difference substantially outweigh the benefits. The size of the estimated constant, $\hat{\alpha}$ (+3.38), indicates that this crash configuration produces fewer injuries to the drivers of the front-striking cars than the other two configurations.

For the side-struck cars in the front-to-side configuration the following parameter estimates were obtained:

$$\hat{\alpha} = +9.86$$

$$\hat{\beta} = -0.95$$

$$\hat{\gamma} = -0.58$$

$$\hat{\partial} = +0.32$$

The fit in this case was good; the variance of the original data was reduced 51 percent by the fitted

TABLE II

PERCENTAGES OF SERIOUSLY AND FATALLY INJURED DRIVERS IN FRONT-TO-SIDE CRASHES BY VEHICLE WEIGHT (CAMPBELL AND REINFURT 1974)

FRONT–STRIKING CAR	1	2	3	4	5	6
6 (≥4100 lbs)	1.4 / 9.9 / 71	2.1 / 8.5 / 92	3.7 / 7.4 / 136	3.1 / 9.2 / 130	4.3 / 7.9 / 140	2.4 / 3.5 / .85
5 (3700 - 4099 lbs)	6.0 / 9.0 / 134	1.2 / 7.3 / 164	3.0 / 4.0 / 202	2.4 / 5.7 / 210	3.8 / 5.5 / 236	5.8 / 8.0 / 137
4 (3300 - 3699 lbs)	2.4 / 13.6 / 125	2.4 / 6.1 / 163	5.5 / 6.1 / 181	5.0 / 5.0 / 182	3.5 / 4.0 / 198	0.8 / 2.3 / 132
3 (3100 - 3299 lbs)	4.6 / 16.7 / 108	2.1 / 7.0 / 142	4.8 / 5.4 / 168	3.7 / 3.7 / 188	5.3 / 5.3 / 171	5.0 / 2.5 / 119
2 (2700 - 3099 lbs)	3.4 / 9.0 / 89	2.4 / 4.8 / 126	4.7 / 3.9 / 128	4.3 / 5.8 / 139	6.9 / 1.7 / 175	7.4 / 3.7 / 108
1 (<2700 lbs)	10.5 / 14.5 / 76	9.5 / 9.5 / 95	5.8 / 1.9 / 103	8.9 / 1.6 / 123	8.9 / 3.2 / 123	8.2 / 2.4 / 85
CAR GROUP	1	2	3	4	5	6

SIDE–STRUCK CARS

For matrix cell at the intersection of row i and column j:

% injuries in group i cars striking group j cars	% of injuries in group j cars struck by group i cars
No. of Collisions	

model. The largest absolute value for the three estimated coefficients was again $\hat{\beta}$ (-0.95), representing the effect of vehicle size. In this case, the estimated coefficients suggest that the benefits associated with a mass difference (-0.58) are greater than the penalties (+0.32). Whether this would still have been found to be the case if impacts on the driver's and right sides of the struck cars could have been treated separately cannot be determined.

For the rear-struck cars in front-to-rear crashes, the following parameter estimates were obtained:

$$\hat{a} = +1.90$$

$$\hat{\beta} = -0.20$$

$$\hat{\gamma} = -0.06$$

$$\hat{\partial} = -0.09$$

The fit in this case was relatively poor; only 34 percent of the variance of the original data was accounted for by the fit.

The preceding results suggest that, with the exception of the rear-struck cars in front-to-rear

TABLE III

PERCENTAGES OF SERIOUSLY AND FATALLY INJURED DRIVERS IN FRONT-TO-REAR CRASHES BY VEHICLE WEIGHT (CAMPBELL AND REINFURT 1974)

FRONT–STRIKING CAR	1	2	3	4	5	6
6 (≥4100 lbs)	1.1 / 0.3 / 180	0.4 / 2.5 / 243	1.8 / 1.1 / 277	0.9 / 0.9 / 333	1.4 / 0.8 / 361	1.1 / 0.4 / 269
5 (3700 - 4099 lbs)	1.0 / 1.0 / 296	1.1 / 1.4 / 351	1.5 / 1.0 / 477	1.8 / 0.6 / 491	0.6 / 0.6 / 612	1 5 / 1.0 / 402
4 (3300 - 3699 lbs)	0.3 / 1.9 / 309	1.3 / 1.3 / 318	1.5 / 0.8 / 399	2.3 / 0.9 / 430	2.3 / 0.8 / 515	1.3 / 0.6 / 311
3 (3100 - 3299 lbs)	2.2 / 1.1 / 268	1.8 / 1.6 / 390	1.7 / 2.1 / 421	0.7 / 1.2 / 435	2.2 / 1.0 / 508	1.4 / 0.3 / 297
2 (2700 - 3099 lbs)	1.3 / 2.2 / 225	1.8 / 0.7 / 285	4.1 / 2.3 / 345	2.7 / 1.4 / 368	3.1 / 0.2 / 418	1.2 / 0.4 / 248
1 (<2700 lbs)	2.0 / 1.5 / 202	4.6 / 1.2 / 241	6.2 / 0.7 / 291	4.8 / 0.6 / 335	3.2 / 1.0 / 312	6.5 / 0.5 / 215
CAR GROUP	1	2	3	4	5	6

REAR–STRUCK CARS

For matrix cell at the intersection of row i and column j:

% injuries in group i cars striking group j cars	% of injuries in group j cars struck by group i cars
No. of Collisions	

crashes, much of the variance in the observed percent of driver injuries occurring when two vehicles from specific weight classes collide can be explained by the model defined by equations (1) and (2). The parameter estimates for the various crash configurations all strongly suggest that the size of the cars which the driver is in has more influence on the occurrence of serious and fatal driver injury than the difference in mass between the two colliding cars. Furthermore, in most cases it appears that the penalties of a mass difference, in terms of increases in the occurrence of serious and fatal injuries, are greater than the benefits (decreases in the occurrence of serious and fatal injuries).

It is interesting to note that in each crash configuration the ratio of $\hat{\alpha}$ to $\hat{\beta}$ was approximately 10:1, suggesting that increases in vehicle size corresponding to the differences between two successive weight groups used in this study reduce the serious and fatal driver crash injuries by approximately 10 percent.

The preceding results suggest the desirability of vehicle populations in which the range of vehicle weights are small. However, it appears that the size of a vehicle has much more influence on the likelihood of injuries to its occupants, the larger the vehicle – even among vehicles of equal weight – the lower both the theoretical and actual occurrence of injury.

DISCUSSION

It has long been believed by some that, in order to have compatibility in terms of occupant crash protection, vehicles operating on the same roads should be as nearly as practicable the same weight. The results of this study lend support to this conclusion. They also suggest, however, that the size of vehicles is an even more important determinant of the likelihood of occupant injuries. Although, in general, there are positive relationships between the sizes and weights of vehicles, there are sufficient exceptions to suggest that better crashworthiness could be obtained by building relatively larger, sizeable but not correspondingly relatively heavier vehicles. It is important to remember that the kinetic energy of a moving object is equal to one-half the product of its mass and velocity squared. Thus, the amount of energy to be absorbed or dissipated in a collision increases in direct proportion to the weight of the car. Therefore, if increases in vehicle size can be obtained without substantial relative increases in their weight, occupants could be afforded greater crash protection without adding to the kinetic energy that must be absorbed or dissipated in a crash.

There appears to be no evidence, and no theoretical reason to believe, that the size of vehicle, unless associated with increased vehicle mass, produces any penalties to the occupants of other cars in intervehicular collisions. The results of this study, however, suggest that there are penalties (in terms of increased occurrence of occupant injuries associated with differences in vehicle mass) to the occupants of the lighter car and that the penalties are, in general, greater than the benefits to the occupants of the heavier car.

For vehicles using the same roads these relationships suggest a crashworthiness design concept for intervehicular crashes that regards increases in vehicle *size* as primarily *protective,* and *increases* in vehicle weight as primarily *hostile,* indicating the desirability of relatively sizeable but not heavy vehicles.

REFERENCES

1. H. A. Stark, ed. *Ward's 1973 Automotive Yearbook,* 35th ed. Detroit, Michigan, Ward's Communications, Inc. 1973.
2. *Automotive News; 1974 Almanac Issue.* Detroit, Michigan, Marketing Services, Inc., April 24, 1974. pp. 4, 23.
3. 1968 New York state study, prepared for the Department of Transportation (Submission for the Record by U.S. Department of Transportation to U.S. Subcommittee on Antitrust and Monopoly, March 10, 1970.)
4. B. J. Campbell. Driver Injury in Automobile Accidents Involving Certain Car Models. Chapel Hill, North Carolina, University of North Carolina, Highway Safety Research Center, July, 1970.
5. J. O'Day, D. H. Golomb, and P. Cooley, HIT Lab Reports, Vol. 3, No. 9, Ann Arbor, Michigan, University of Michigan, May, 1973.
6. B. J. Campbell and D. W. Reinfurt. Relationship between Driver Crash Injury and Passenger Car Weight. Chapel Hill, North Carolina, University of North Carolina, Highway Safety Research Center, November, 1973.
7. B. J. Campbell. Safety, Small Cars and the Gasoline Shortage, *Highway Safety Highlights,* Vol. 7, No. 10, January, 1974. Chapel Hill, North Carolina, University of North Carolina Highway Safety Research Center.
8. "Tests Show Small-Car Dangers." *Status Report,* Vol. 6 No. 21, November 16, 1971. Washington, D.C., Insurance Institute for Highway Safety.
9. "Automobile Insurance Losses, Collision Coverages. Initial Results for 1973 Models Compared with 1972 Models." Washington, D.C., Highway Loss Data Institute, Research Report HLDI R73-1, September, 1973.
10. "Automobile Insurance Losses, Collision Coverages. Variations by Make and Series, 1973 Models." Washington, D.C., Highway Loss Data Institute, Research Report HLDI R73-2, March, 1974.
11. J. Casassa, II, B. O'Neill, I. Miller and S. Stone, A Survey of Unrepaired Automobile Crash Damage. I. 1968-1972 Models. Washington, D.C., Insurance Institute for Highway Safety report, September, 1973.
12. New York State Department of Motor Vehicles. An Analysis of Accidents in New York State by Make of Vehicle. Report prepared for the U.S. Department of Transportation, National Highway Traffic Safety Administration. Washington, D.C., June, 1972.
13. B. J. Campbell and D. W. Reinfurt. Highway Safety Research Center, University of North Carolina, Chapel Hill, North Carolina. Personal communications, 1974, to B. O'Neill, Insurance Institute for Highway Safety, Washington, D.C.

FACTORS INFLUENCING THE PERFORMANCE OF THE ENERGY ABSORBING STEERING COLUMN IN ACCIDENTS

JOHN W. GARRETT and DONALD L. HENDRICKS
Calspan Corporation

INTRODUCTION

The steering assembly was first identified as a major source of driver injury in the earliest studies of automobile crash injuries. Findings concerning the steering assembly were included in testimony presented by John Moore of the Cornell University Automotive Crash Injury Research (ACIR) Program at the 1956 Hearings before a Subcommittee of the Committee on Interstate and Foreign Commerce, on Investigation of Highway Traffic Accidents. The data revealed that the steering assembly was the leading cause of any injury among car occupants and that it ranked second only to ejection as a cause of dangerous or fatal injuries (injury rated using the ACIR injury index, from which the AIS later was developed). A 1961 ACIR Report[1] confirmed this finding for injury accidents that occurred during the period from 1953 to 1959. Similar results were obtained by Huelke and Gikas.[2]

Early efforts by the automotive industry to reduce injury associated with the steering assembly concentrated on design modifications to the wheel itself. Protruding hubs were eliminated and "deep dish" wheel designs of varying depths and configurations were introduced on 1956 car models. Invasion of the driver's area by the steering column in severe accidents continued to occur however, and the energy absorbing steering column was introduced in 1967 models and was used in all 1968 model cars in the United States. Although design varies, these columns generally have an energy absorbing device (EAD) in the column, a shear capsule to prevent intrusion into the driver area and a flexible coupling at the base of the column.

A number of studies of the energy absorbing steering column suggested some apparent improvement in injury associated with the new column, but most reports were based on a small volume of data and statistical confirmation was lacking. A 1972 report by Anderson[3] of Calspan indicated that there was only a limited decrease in overall driver injury associated with the energy absorbing columns and that the decrease appeared in the more severe accidents. He also reported that the overall reduction was due to a decrease in head injuries caused by the steering assembly while thorax injuries from this source showed little or no improvement. A paper by Nelson and Wilson[5] showed a considerable data scatter when EAD compression was plotted against estimated closure speed suggesting that performance was not entirely consistent.

These two studies led the authors of this report to examine case histories from Calspan's in-depth investigations to reconcile these findings with earlier papers. It soon became evident that the effectiveness of the energy absorbing column appeared to be minimal under a certain range of accident conditions while under other conditions its performance was optimal. Anderson had noted these early results in his report. It should be noted that the detection of a trend through statistical analysis followed by more detailed analysis of case histories is a particularly useful technique which has been used successfully in other studies of subjects such as door latches, windshields and restraint systems at Calspan.

Test data available at Calspan also were examined to glean from this source whatever additional performance details might be available. Movie film from approximately 50 crash tests involving various types of frontal impacts were studied. Results of a series of bench tests of steering columns also were reviewed. Thus, all sources of data available to us were reviewed and analyzed in an effort to better understand and describe the performance of the steering assembly.

An attempt was then made to determine if the performance of the EAD column could be predicted in terms of a number of specific accident variables such as area of impact, direction of force, vehicle rotation and a number of others. Using these accident variables, a prediction of column performance under certain theoretical conditions was made, and case histories were than examined in order to validate the predicted performance. It was thought that, if this could be accomplished, further improvement in the design of the EAD system might be possible. The study results are presented in this paper.

LITERATURE REVIEW

In 1968, Garrett[4], using 219 ACIR cases, indicated that there appeared to be a reduction in injury associated with the new steering assemblies but that more data were needed for confirmation because of the small data volume. Nelson and Wilson[5], also in 1968, selected 257 cases involving EA column GM

cars in which vehicle speed was reported, for study. The results indicated that there was a considerable data scatter when EAD compression was plotted against estimated closure speed indicating that performance was not completely consistent. They concluded that the column was saving lives and reducing injuries.

In the same year, Huelke and Chewning[6] reported that the EA column had reduced driver thorax injury, but that head injuries continued to occur. Skeels,[7] Patrick,[8] and Gadd and Patrick[9] also reported generally similar results. In 1970, Nahum, Siegel and Brooks[10] reported that, for drivers, "the major cause of injury in the 1960-66 (model) group, the wheel column, decreased 43% in the later model year group". They attributed this improvement to the energy absorbing column. In 1971, Levine and Campbell[11] reported significant reductions in serious injury associated with the EA column in the medium speed frontal and run-off-road accidents. The magnitude of the reduction was approximately 30 percent, and was found to be statistically significant. In the same year, O'Day and Creswell[12] using data from the multi-disciplinary teams at the Universities of Michigan and California also reported a significant decrease in overall driver injury associated with the energy absorbing column.

In 1972, a report by Anderson[3] (mentioned earlier) indicated that there was only a slight decrease in the occurrence of overall injury associated with the EA column. It appeared that the decrease was associated with the higher speed accidents. Examination of data for thorax injuries revealed that there was essentially no reduction in the incidence of thorax injury associated with the steering assembly. The data indicated that the decrease in overall injury was due largely to a reduction in head injuries caused by the steering assembly. DuWaldt[13] in a study published in 1973, reported on bench tests of 1968 steering columns that were conducted to determine the effects of angular impacts on EA steering column performance. Results indicated that the EA columns were less effective when angular impacts were imposed than when the impact was imposed to the wheel without any angular component, that is, applied parallel to the end of the column. A 1973 paper by Searle[14] reviewed and summarized some of the aforementioned data as well as some test results and European accident data. He concluded that "for the unrestrained driver it would appear that the conventional energy absorbing steering column produces little benefit."

A 1973 New York State Department of Motor Vehicles study[15] of head-on collisions between two vehicles found essentially no beneficial effect associated with the EA column cars.

A 1973 report by Griffin[16] reviews and summarizes a number of the reports cited above. Also included were data from a study of North Carolina accidents by McLean[17] which appears to show a substantial reduction in serious and fatal injuries (police-reported K and A injury). Griffin concludes from his review that the energy absorbing steering column "may reduce deaths and serious injuries to drivers by 5 or 10 percent."

THE ENERGY ABSORBING STEERING COLUMN

A generic EA column, from the Collision Performance and Injury Report form is shown in Figure 1. The system consists of an energy absorbing device in the steering column which can absorb impact forces from either the occupant or from forward impact. A key element in the system is the shear capsule which is essentially an "anti-intrusion" device. It is attached at the juncture of the panel and column and limits rearward movement of the column at the panel while permitting the occupant to move forward and compress the EAD. There is a flexible coupling at the lower end of the column and there may also be a telescoping unit in the engine compartment, or a steering wheel EAD. It is clear from the sketch that the optimum impact mode for the occupant would be to contact at a right angle to the wheel directly upon the column (arrow). A lap belted occupant most nearly meets this condition. The ideal condition, of course, would be to wear the lap and shoulder restraint and minimize contact entirely.

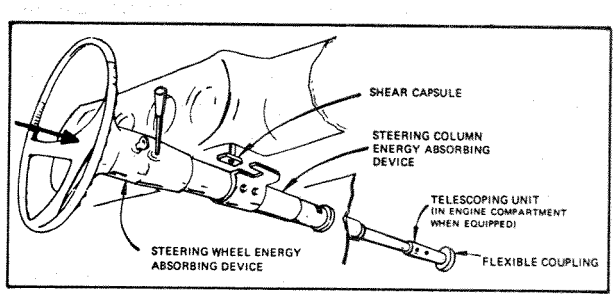


Figure 1. Generic Steering Column

It should be noted that a very severe impact to the front of the column compresses the EAD but may also cause the column to rotate upward, pivoting about the shear capsule. In fact, when deformation of the front of the car moves the forward end of the column in any direction the column pivots about the shear capsule mounting so that the upper end moves in the opposite direction. Thus, if the forward end of the column moves to the right, the upper end is

moved to the left rotating about the shear capsule. This is an essential point to keep in mind throughout this paper.

The various types of EA devices, used in the vehicles included in this study, are shown in Figure 2. The Plymouth Barracuda and Dodge Challenger have steering wheel EA devices. Since there were insufficient numbers of these vehicles in the data sample, they were excluded from the study.

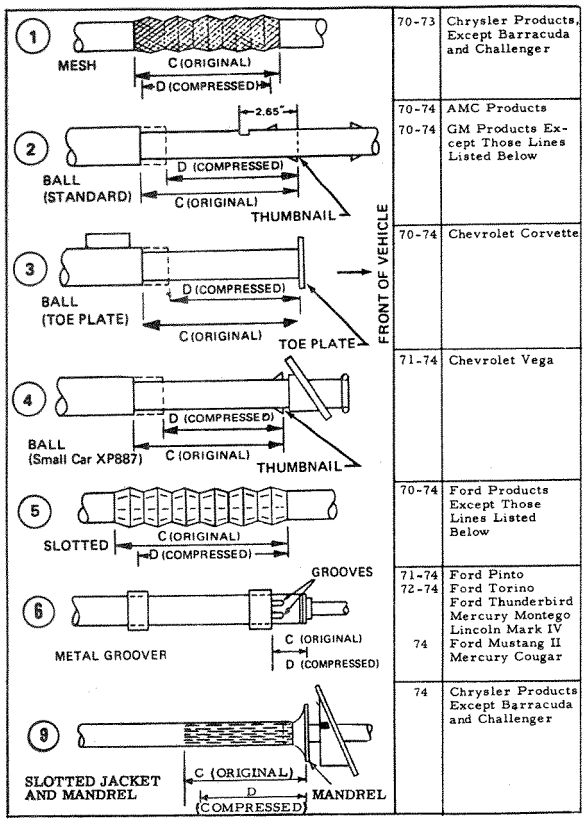


Figure 2. Sketches of Steering Column Energy Absorbing Devices

STUDY DATA AND APPROACH

The objective of this study is to determine those factors that influence the performance of the energy absorbing column in accidents. This paper is divided into three sections based upon the type of data analyzed or reviewed:

- Test Results:
  - Laboratory Tests
  - Compliance Tests
- Development and Validation of Predictive Process
- Statistical Summary of Case Data

In the last two sections of the paper, case data collected by the Calspan Tri-Level Accident Study were analyzed. Available for study was a total of 1,290 Calspan in-depth investigation cases. (All of these cases involve recent model cars in which at least one occupant received an injury that required hospital treatment.) Selected for analysis were those cases involving a frontal impact where the direction of force was 11 to 1 o'clock. These cases were selected because these are the conditions under which the driver generally contacts the steering wheel. There were 549 cases involving cars that met the above criteria.

Study data were collected during the period from November 1, 1970 to December 31, 1973, and include only those cars manufactured by U.S. firms. The year of manufacture ranges from 1970-1974.

TEST RESULTS

Laboratory Test Results

One phase of the Calspan Crashworthiness program supported by NHTSA included tests of the energy absorbing steering column. This work by F. DuWaldt[13] was in progress in 1972 when the present study was undertaken. Data from his study therefore were reviewed to determine if they corroborated our early findings.

Laboratory tests included static column crush tests and body block impact tests to illustrate angularity effects. The results showed a strong dependence on angle of contact.

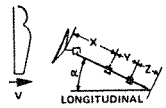
A static crush test was performed to determine the characteristics of the force/deflection curve of a 1968 Ford EA column. Body block tests were than performed to determine if imposing bending moments on the steering shaft caused increased friction between the telescoping elements during collapse, producing higher force levels and reduced stroke. The results and a sketch of the test arrangement are shown in Figure 3. Nominal test speed was 18 mph.

Data in Figure 3 indicate that the energy absorbing columns tested were less effective when longitudinal angular impacts were imposed with either a small lateral component or no lateral component.

Compliance Test Results

In order to ascertain typical response patterns of the steering assembly during impact sequences, a series of approximately 50 crash test films and the recorded data from these tests were reviewed. The tests involved a range of impact types (i.e., vehicle-to-vehicle impacts including front, side, and rear configurations; vehicle to SAE pole impacts; and vehicle to SAE barrier impacts) and a range of impact speeds. In most of the films it was apparent that the

SUPPORT AND OVERHANG DIMENSIONS



X INCHES	Y INCHES	Z INCHES	FIXITY FREE	FIXITY *
**	**	*		✓

* RESTRAINED BY LOAD CELL AT STEERING SHAFT COUPLING

**STANDARD 1968 FORD DIMENSIONS

RUN IDENTIFICATION	LONGITUDINAL DEGREES	LATERAL DEGREES	ESTIMATED VELOCITY FT/SEC	STEERING WHEEL YOKE ALIGNMENT	JACKET COLLAPSE DISTANCE INCHES
BLT-1	0	0	26	HORIZONTAL	5.0
BLT-2	0	15	26	HORIZONTAL	5.5
BLT-3	26	15	25	HORIZONTAL	0.6
BLT-4	26	0	27	HORIZONTAL	3.2
BLT-5	26	0	26	VERTICAL	1.3

Figure 3. Bench Test of the Influence of Impact Angularity on Steering Column EA Effectiveness

steering column and wheel moved rapidly in the vertical plane several times (oscillated) during the impact sequence. This movement, or oscillation, was observed throughout the range of impact types, however, the occurrence and amplitude of oscillation varied with impact speed. Oscillation was minimal in low and high speed impacts. In general, amplitude of movement increased in proceeding from low to moderate speed impacts and then decreased again in proceeding to high speed impacts.

In many of the test vehicles, a dummy was placed in the driver's seat. There did not appear to be any consistent pattern with respect to steering wheel vertical location when dummy contact occurred, i.e., wheel position ranged from high to low, when contacted. The precise position was not consistent in terms of impact speed or other available data. In general, it appeared that dummy contact with the wheel, while at its lowest position, was the optimum condition. Contact with the wheel then was more nearly in a plane perpendicular to the column and the force exerted on the wheel by the dummy was distributed more equally over the wheel.

Compliance test films and associated data were most useful in attempting to identify typical response patterns of steering assemblies. Many of the test vehicles were equipped with high speed movie cameras specifically to record column and wheel movement and a scratch tube linkage to measure this movement. An example of these test data appears in photographs 1 to 6, from a compliance test crash of a 1971 Plymouth Fury II, 4-door sedan with an SAE barrier. The test was conducted to check for compliance with FMVSS 204 (Steering Control Rearward Displacement), FMVSS 212 (Windshield Mounting), and FMVSS 301 (Fuel Tanks, Fuel Tank Filler Pipes, and Fuel Tank Connections). Impact

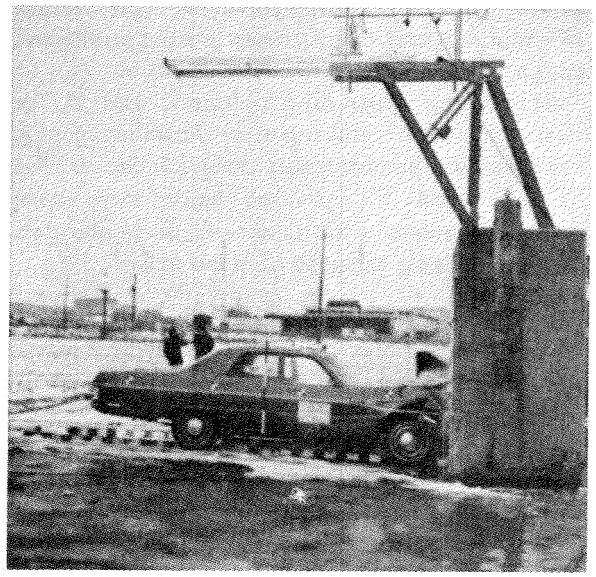


Photo 1

speed was measured at 29.6 mph. Photograph 1 shows the test vehicle during the impact sequence. Photographs 2 and 6 document movement of the steering assembly during the impact and are arranged to depict wheel movements in the same sequence as they occurred. The pre-impact or steady-state position of the wheel is shown in photograph 2. Maximum rearward displacement of the assembly (1.62") is shown in photograph 3. Maximum downward displacement (1.37") is shown in photograph 4 and maximum upward displacement (0.53") is shown in photograph 5. The total amplitude of this oscillation is 1.9". There also was a second wheel oscillation of smaller amplitude. Photograph 6 shows the post impact rest position of

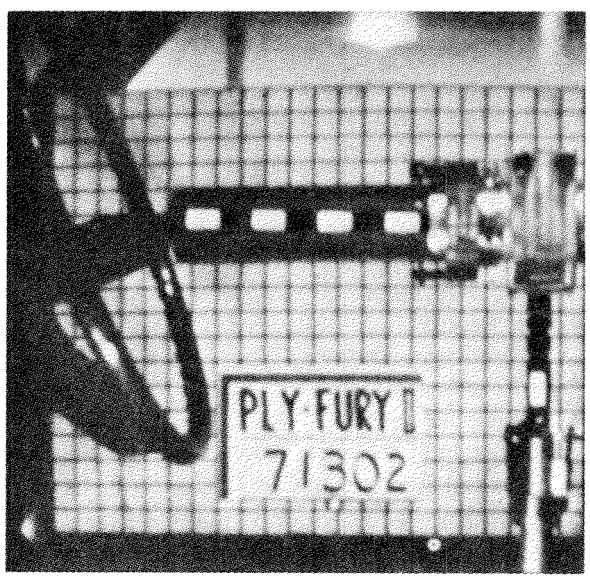


Photo 2

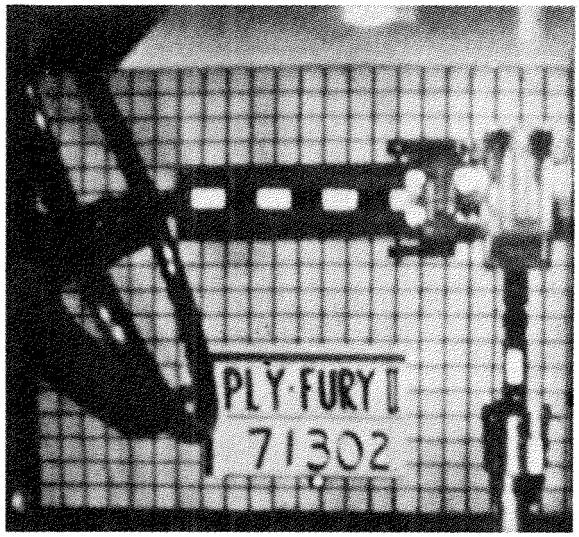


Photo 3

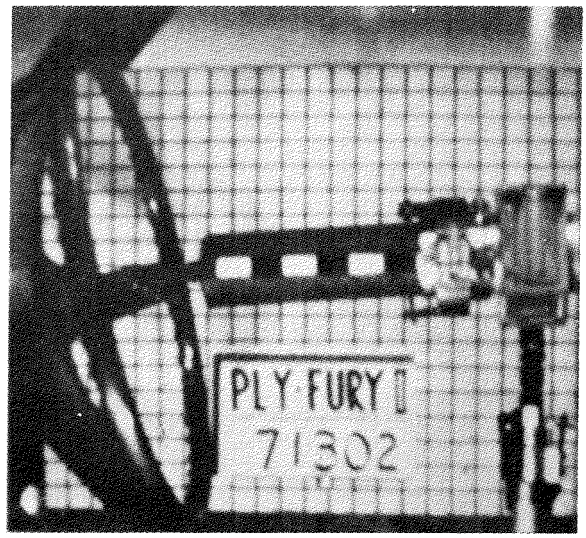


Photo 4

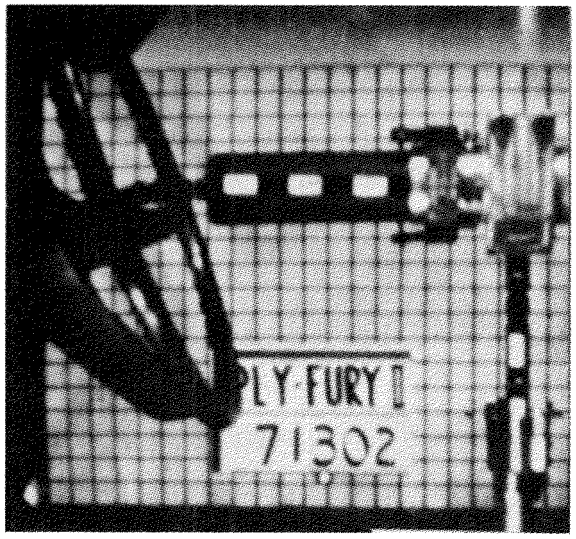


Photo 5

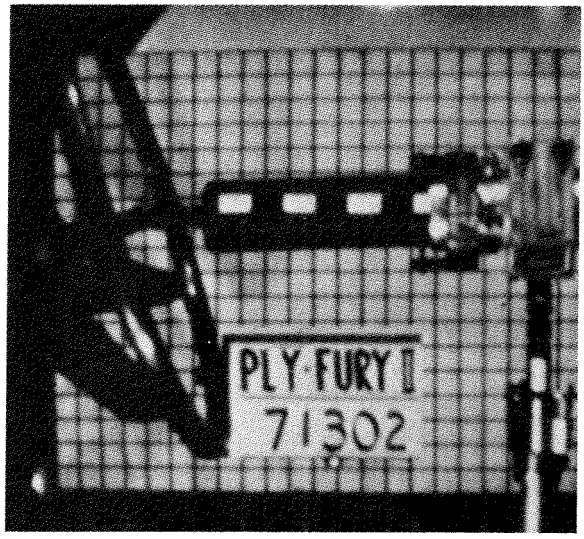


Photo 6

the wheel. (Note that this position is lower than the pre-crash wheel position.)

In the crash test described above, the steering assembly oscillated twice and then returned to a steady state condition. As noted previously, the maximum amplitude of movement was 1.9". In other tests larger and smaller oscillations (duration and amplitude) were observed.

Case History

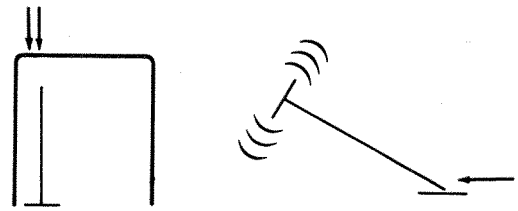
71-283A. It is difficult to predict or to demonstrate the effect of steering column oscillation in a particular accident sequence. As noted previously, the wheel may be at various points in its oscillation path (from high to low) when it is contacted. The negative oscillation mode is generally easier to detect than the positive mode. In case 71-283A (photographs 7 and 8), the driver probably contacted the wheel as it was in a negative oscillation mode as evidenced by the distribution of the load over the steering wheel spoke.

Development and Validation of Predictive Process

A total of 549 Calspan Level 3 cases were available for this phase of the analysis. These cases represent all of the frontal impacts in the study area in which at least one occupant was treated at a hospital.

In examining these data, it appeared that certain factors played a basic role in column performance while others appeared to modify that performance in specific situations. An attempt therefore was made to analyze the factors involved, in order to determine if

Case No.	VDI	Maximum Crush	EAD Collapse	Shear Capsule	Column Rotation
71-283A	12–FYEN	13"	4.5"	3.1"	-4.0°



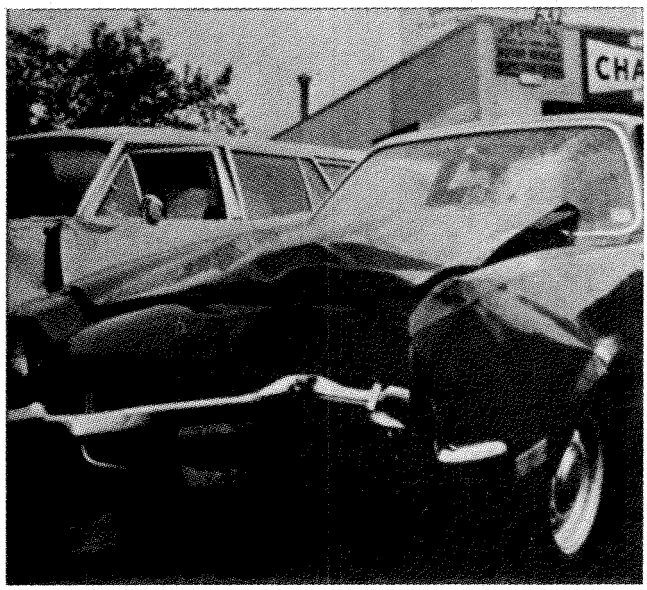


Photo 7

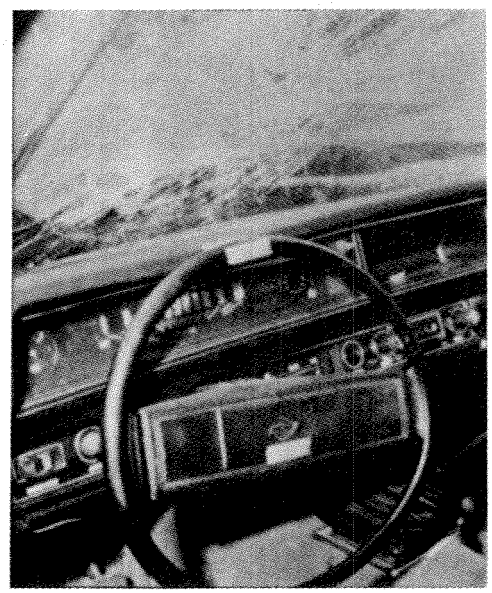


Photo 8

one could predict column performance. It was determined that this could be done if sufficient accident details were available. Factors necessary for this evaluation are discussed in the following section. To simplify discussion, a 12 o'clock direction of force is assumed in the initial review of factors until direction of force is discussed later in the section.

Area of Impact and Impact Severity (Figure 4)

In minor severity cases (VDI 1) the area of impact has little influence on the column and there is generally no EAD collapse. As severity increases, however, the area of impact becomes increasingly important in terms of column behavior. When the impact involves the extreme left or right front of the car (4A and E) there is little or no influence on the steering column because these become, in effect, sidewipe accidents (see Case History 72-208A). When the impact is to the area directly in front of the steering column (4G) there is little or no lateral movement, and vertical rotation depends upon the severity and height of the impact (discussed below – Case Histories 71-314A, 71-138A, 71-18A, 71-266A, 72-185A).

When a center front impact involves a narrow object such as a utility pole (4H), deformation of vehicle structure tends to move the forward end of the column laterally to the right and the upper end rotates to the left. If the object is wider, or penetrates close to the column, it often limits or prevents lateral movement of the column (Case History 72-58A).

When an impact to the right third or half of the front occurs (4I), the forward end of the column moves laterally to the right as the front end of the car is displaced in that direction. The upper end of the column therefore moves to the left (Case History 72-307A).

These movements of the column influence the manner in which the driver contacts the wheel. As the column movement becomes more extreme, the driver contacts the wheel at an increasingly acute angle and is less likely to obtain maximum compression as friction between the telescoping elements increases and "binding" occurs. The case histories which follow

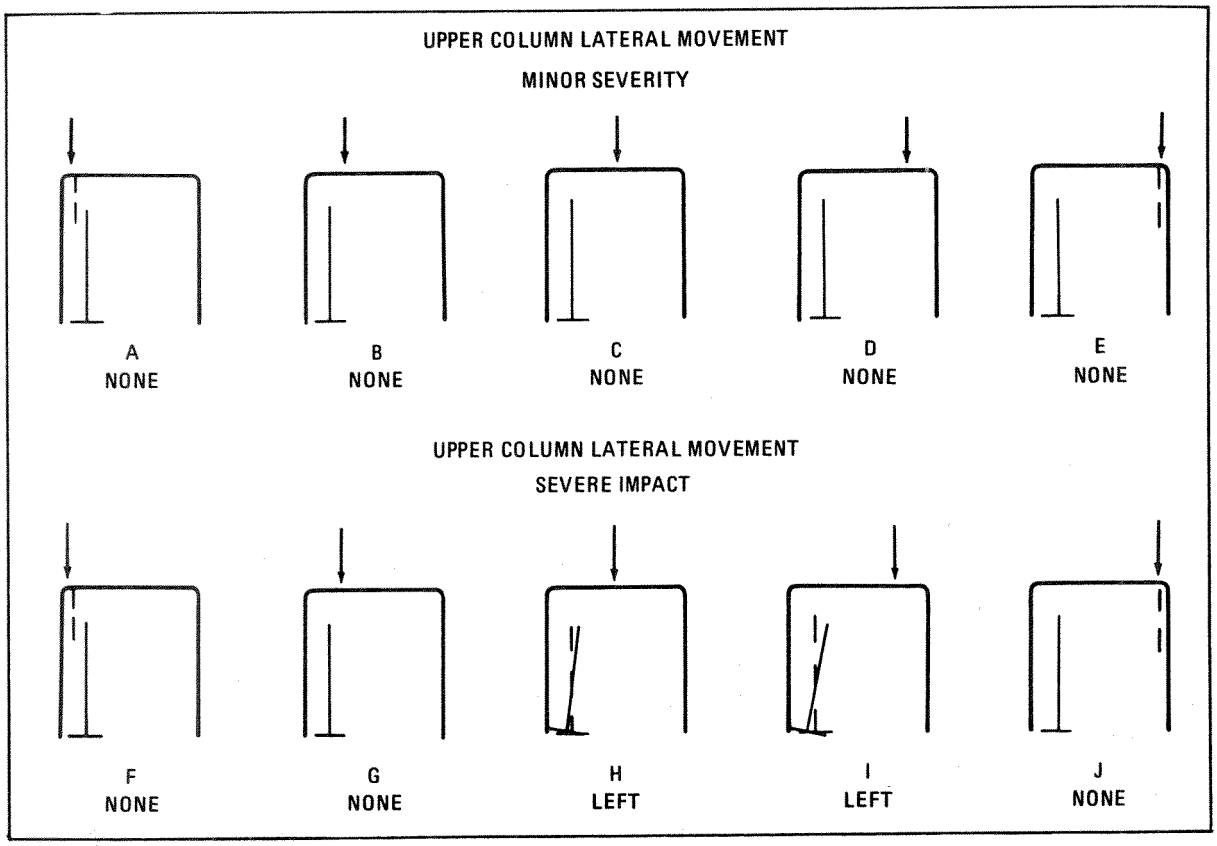
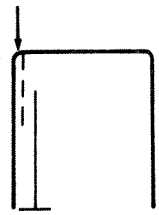


Figure 4. Impact Area and Accident Severity

confirm the predicted behavior described in this section.

Case Histories

72-208A. Case No. 72-208A (photographs 9 and 10) illustrates an initial force application at the left front corner with subsequent rearward crushing of sheet metal. EAD performance was not affected by this crush because the extent of inboard damage did not influence column movement.



Case No.	Max. Crush	EAD Collapse	Shear Capsule	Column Rotation
72-208A	36"	0.0		+2.6°

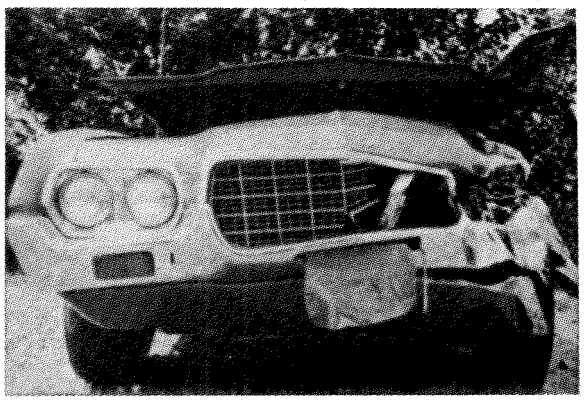


Photo 9

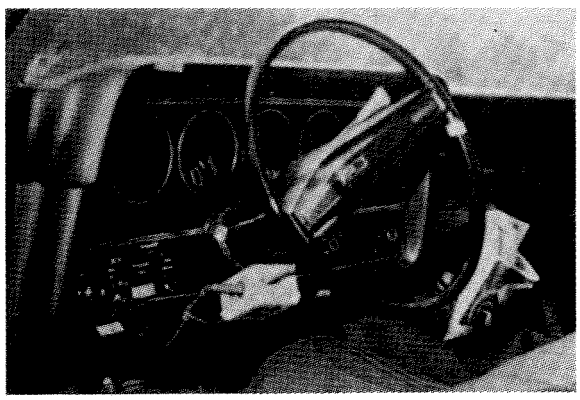
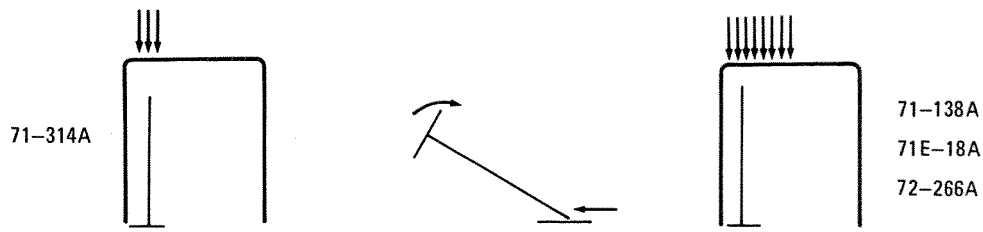


Photo 10

Case No.	VDI	Maximum Crush	EAD Compression	Shear Capsule	Column Rotation
71-314A	12–FYEN	23″	6.7″	3.0″	-10.0°*
71-138A	12–FYEW	25″	4.0″	1.5″	+2.0°
71-18A	12–FYEW	36″	3.6″	4.4″	+10.1°
72-266A	12–FYEW	68″	2.8″	1.8″	+12.7°

*Rotation was positive. See discussion of Case 314A.



71-314A, 71-138A, 71-18A, 71-266A, 72-185A. Impacts directly in front of the steering column, or distributed across entire front, generally produce little or no lateral movement of the column. However, as impact severity increases, the gearbox moves rearward rotating the upper end of the column vertically (upward). (The height of impact also effects vertical rotation.) A series of cases involving impact induced column rotation is tabulated below. Note that, as crush increases, column rotation also increases and EAD compression decreases.

Photographs 11 to 13 are from Case 71-314A. Although negative column rotation was measured by the investigator, there is evidence that positive rotation occurred (photograph 12). Photograph 13 reveals a separation between the base of the wheel and the collar below it on the same side as the rim loading. Generally, when a driver contacts the lower wheel rim, it is forced forward and a separation appears on the opposite side of the wheel (photograph 14). In Case 71-314A, it appears that the load was applied to the rim edge essentially parallel to the wheel. This drove the lower rim upward, forcing the upper rim forward, resulting in separation at the lower (contact) side of the wheel.

An edge loading pattern can also occur if the driver is in a rising trajectory as he moves forward into the wheel. Such loading may result in severe injury due to the localization of force on a narrow area of the thorax or abdomen (the wheel rim does

Photo 11

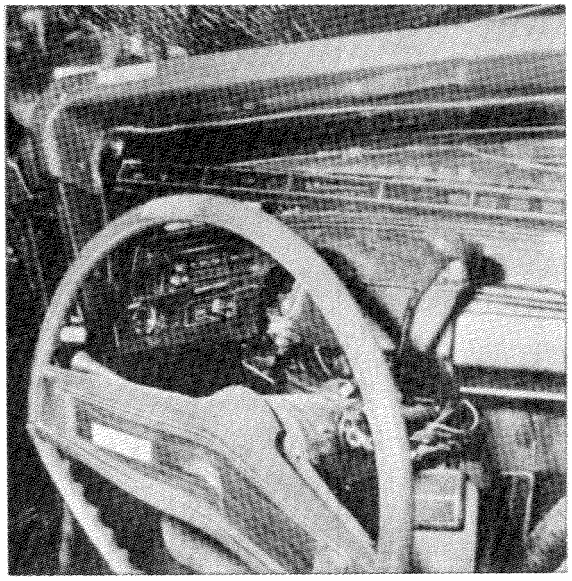


Photo 12

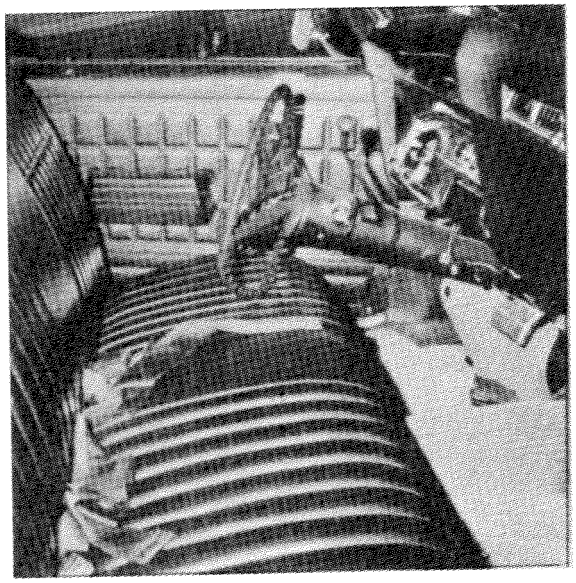


Photo 13

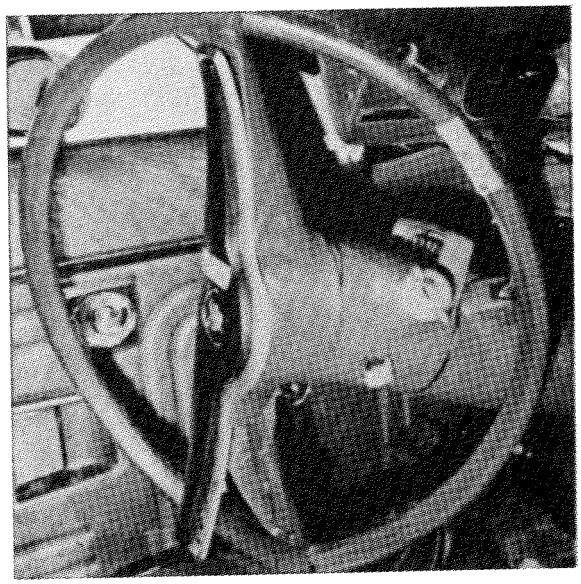
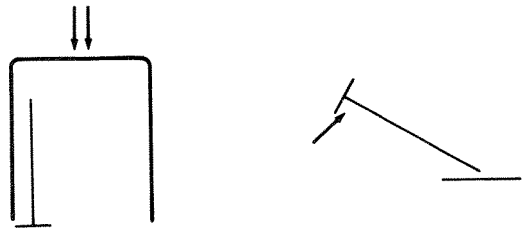


Photo 14

not yield as readily when contacted on the edge in this fashion).

This case demonstrates a limitation in the current measurement technique for column rotation. Although measured properly, the final column angle obtained by lifting the column back into position does not indicate the correct angle of the column during maximum engagement.

72-185A. The driver may accentuate impact induced vertical rotation as in Case 71-314A or may be the major cause of such rotation as in Case 72-185A. In the latter case, the driver who was slumped under the wheel at impact, moved forward and upward contacting the bottom rim of the wheel (photographs 15 and 16) beneath her ribs. (A



Case No.	VDI	Maximum Crush	EAD Compression	Shear Capsule	Column Rotation
72-185A	12–FCEN	21″	0.0″	1.3″	+2.1°

Photo 15

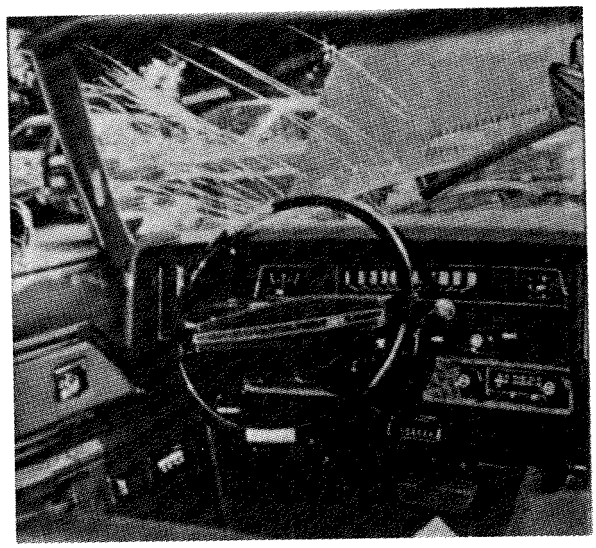


Photo 16

Photo 17

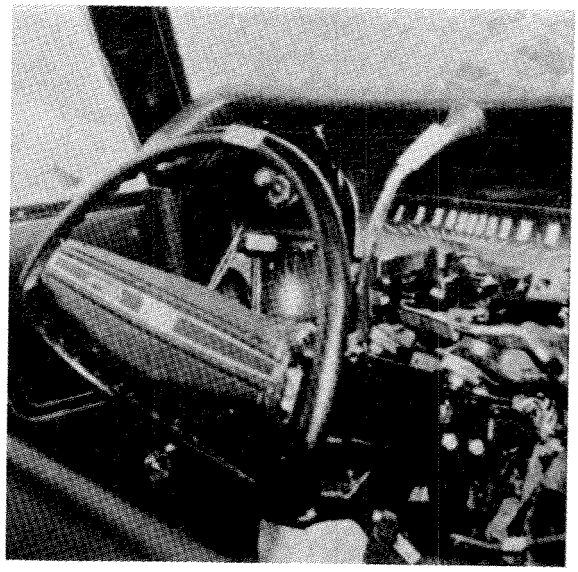
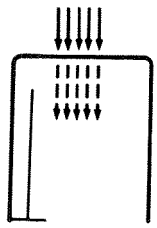
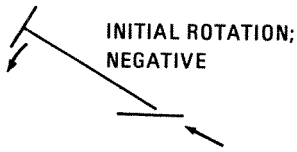
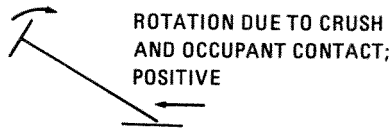


Photo 18







Case No.	VDI	Maximum Crush	EAD Compression	Shear Capsule	Column Rotation
72-58A	12–FCEW	41"	6.3"	1.9"	+5.7°

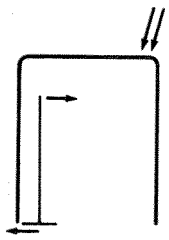
combination of alcohol and tranquilizers apparently resulted in the driver's loss of consciouness prior to impact.)

72-58A. In case 72-58A, center front contact with a large tree limited lateral movement of the column to the right (photographs 17 and 18). Initial crush compressed the EAD and forced the front area upward inducing a negative rotation at the upper end of the column. At maximum engagement, the extent of crush reversed column rotation, inducing a positive rotation at the upper end of the column. The lap belted occupant apparently bottomed in the seat and then contacted the lower rim while the column was rotated upward, increasing positive column rotation.

72-307A. The right front impact shown in photographs 19 and 20 deformed the left front structure to the right displacing the forward end of the column in that direction, and the upper end to the left. This type of impact induces "binding" of the EAD system. The driver contacts the system at an angle which is less than the optimum condition.

Vehicle Rotation (Figure 5)

Vehicle rotation is minimal in low speed front impacts but increases as speed increases for certain front area impacts. Because the driver's body attempts to continue in the direction it was moving



Case No.	VDI	Maximum Crush	EAD Compression	Shear Capsule	Column Rotation
72-307A	01–FREW	40"	0.0"	0.6"	+3.0°

Photo 19

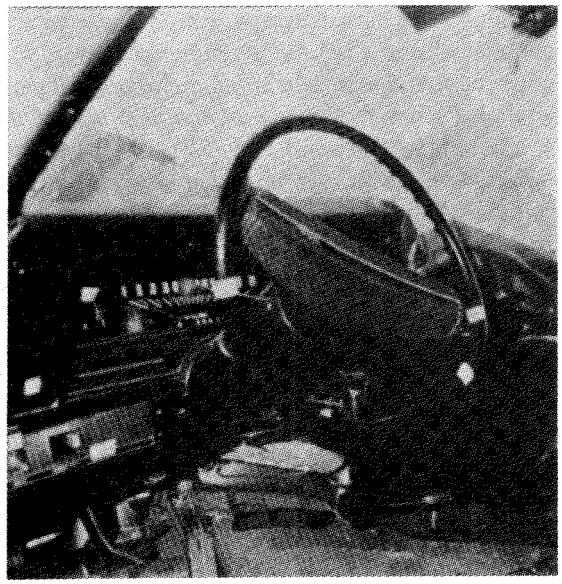


Photo 20

prior to impact, vehicle rotation helps determine where the driver contacts the vehicle interior.

Impacts to the extreme left or right of the vehicle generally produced limited or no rotation and had minimal influence on the area contacted by the driver (5A and 5E). When maximum rotation occurred, the impact area generally involved more than the corner area but less than one-half of the front area. When the

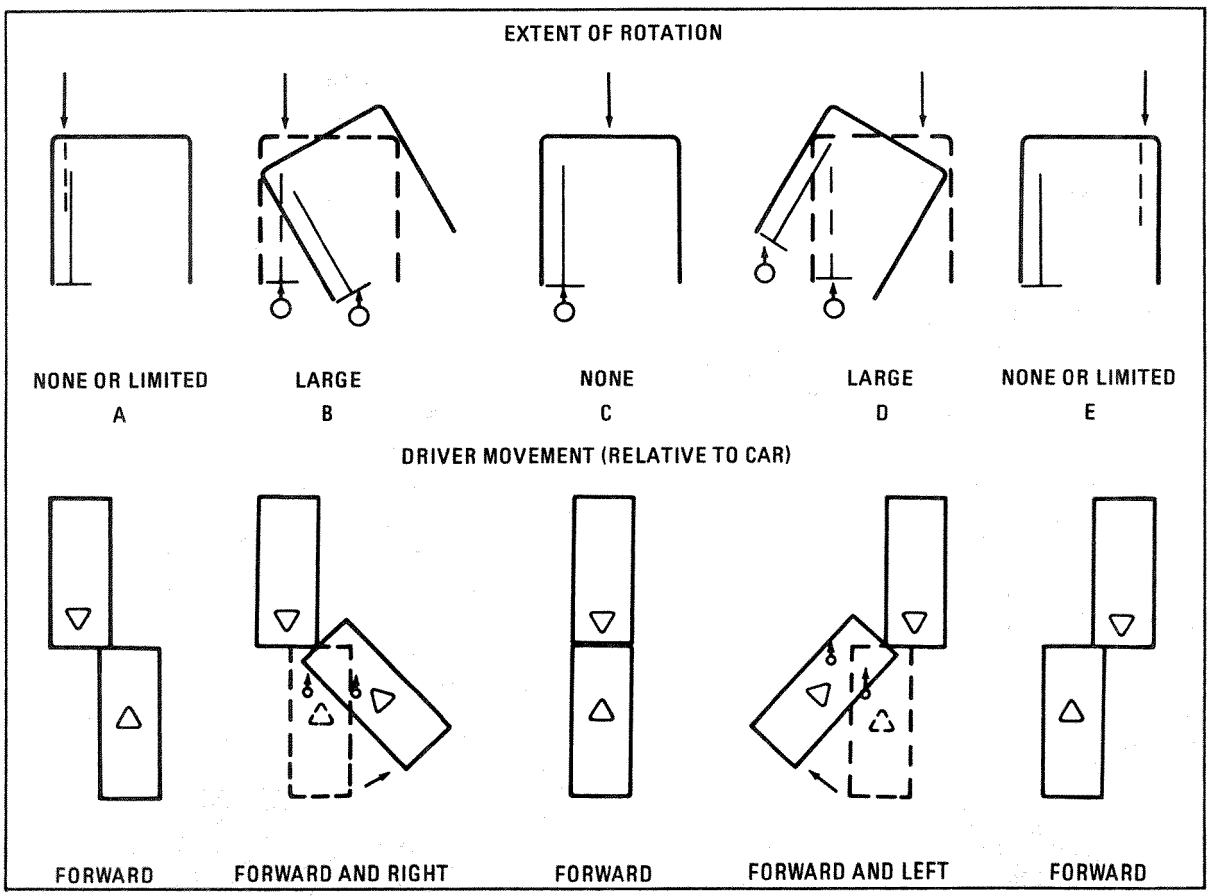
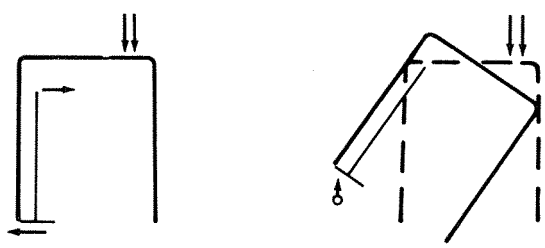


Figure 5. Vehicle Rotation

left front area was engaged, the vehicle rotated counterclockwise (5B). As a result, the driver, who continued to move straight ahead on his original path, in relative terms moved toward the right side of the car as it rotated beneath him (5B). Thus, he often contacted the steering wheel while moving toward the right side of the car rather than moving straight ahead into the wheel (the optimum condition). In extreme cases of vehicle rotation, the driver often missed the wheel entirely. The converse of this situation occurred when the right front area of the car was impacted (5D) (Case Histories 71-445A, 71-447A). In center front, or entire front impacts, the driver moved directly forward into the wheel (5C).

Three case histories appear on the following pages. Two cases validate the predicted behavior. The third case illustrates an exception to the described behavior.



Case Histories

71-445A, 71-447A. Cases 71-445A and 71-447A both involve clockwise vehicle rotation and driver movement to the left (relative to the vehicle) following a right front impact. In both cases the extent of crush was sufficient to move the forward end of the column to the right and upper end to the left. In case 71-447A (photographs 21 and 23) the driver probably assisted impact forces in rotating the upper column to the left.

71-413A. Photographs 24 to 26 illustrate Case 71-413A in which vehicle rotation was less pronounced than in the preceding cases. Here, the influence of impact forces was greater than the effect of vehicle rotation. Consequently, the driver moved toward the area of impact. In photograph 26, the tape on the instrument panel and windshield incicates driver contact points and indirectly indicates the trajectory of the driver (forward and to the right). Note that this case is an exception to the rotation pattern discussed previously.

Direction of Force (Figure 6)

One of the most consistent effects of direction of force is to vary the trajectory of the driver. In

Case No.	VDI	Maximum Crush	EAD Compression	Shear Capsule	Column Rotation
71-445A	12–FREW	40"	3.8"	3.8"	-1.0°*
71-447A	12–FREW	60"	2.6"	4.0"	-3.0°**

*Actual rotation was positive.
**Possible positive rotation.

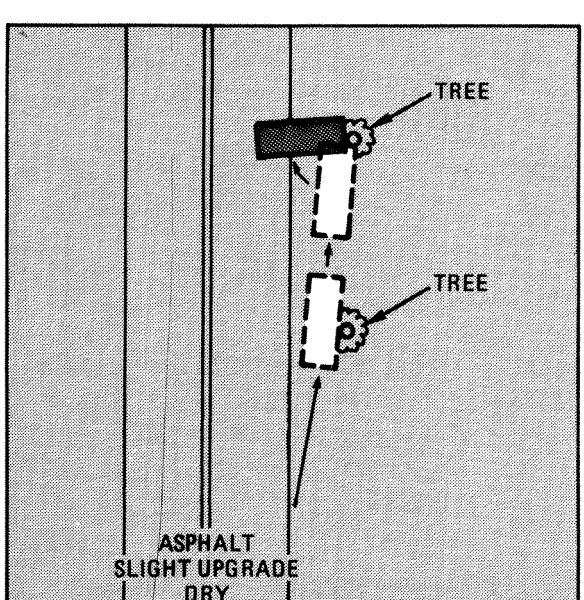


Photo 21

Photo 22

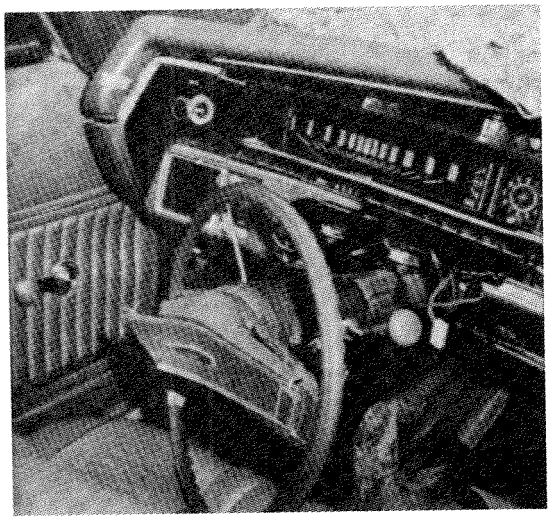


Photo 23

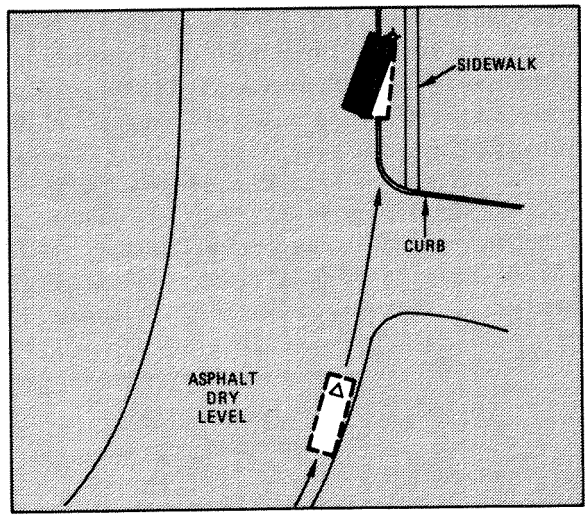
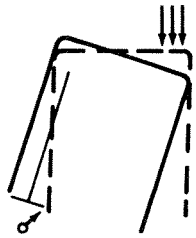


Photo 24

impacts where the resultant direction of force is 12 o'clock, the driver generally approaches the steering assembly on a path parallel to the longitudinal axis of the steering column (6G). When the resultant direction of force is nearer to 11 or 1 o'clock, the driver follows a trajectory which results in an angular approach to the longitudinal axis of the steering column. (This trajectory is almost directly opposite to the resultant direction of force on the vehicle – Case



Case No.	VDI	Maximum Crush	EAD Compression	Shear Capsule	Column Rotation
71-413A	12–FZEN	21"	0.9"	0.4"	-5.8°*

*Actual rotation was positive.

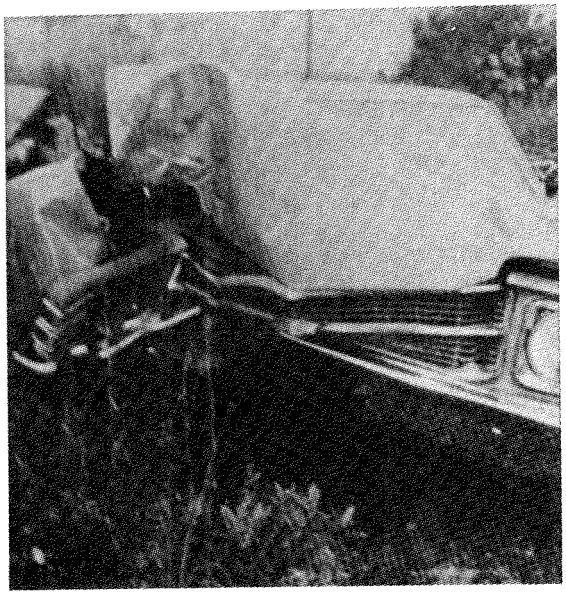


Photo 25

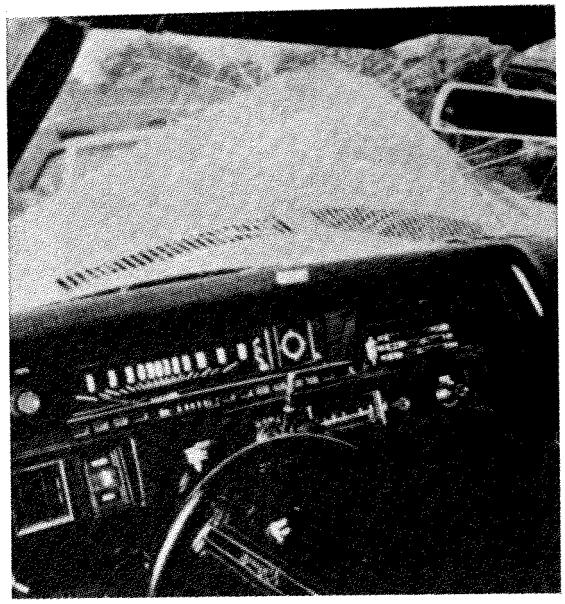


Photo 26

Histories 71-469A, 73-7, 71-77A, 71-334A.) If the impact is severe, the driver will "cock" the steering wheel and/or the steering column and obtain minimal EAD compression as a result of "binding" of the telescoping elements.

The direction of force in combination with the area of impact generally tends to reduce or accentuate vehicle rotation. For example, a 12 o'clock impact to the left front produces vehicle rotation as discussed earlier (6A). A similar 11 o'clock impact tends to reduce the amount of rotation, while a 1 o'clock impact increases rotation (6B and Case History 72-31A). The converse is true for a right front impact.

A second effect of direction of force is to move the forward end of the steering column laterally in the direction that the front area of the car is crushed and deformed. In impacts to the left front, deformation from a 1 o'clock impact often moves the forward end of the column to the left and, as it pivots

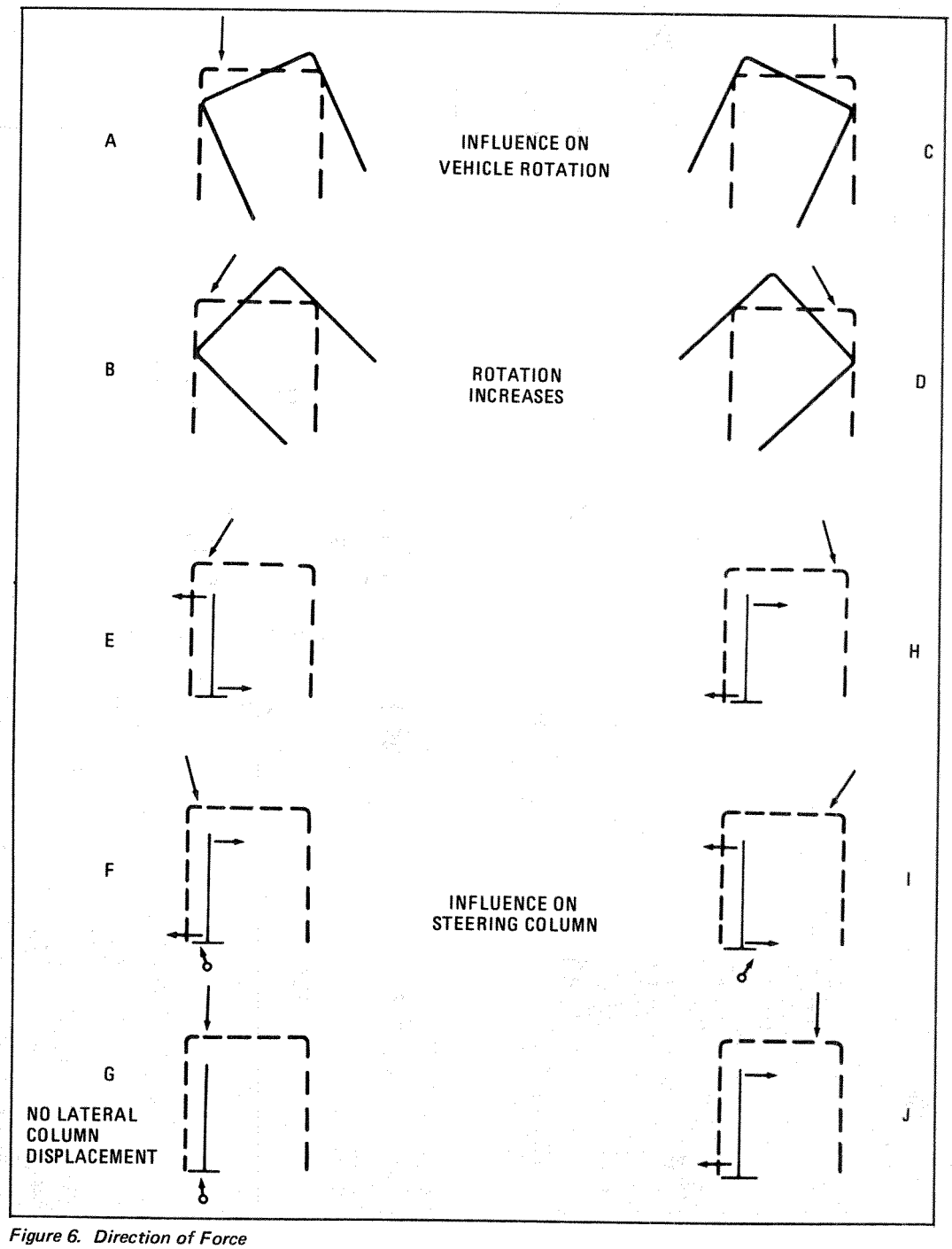


Figure 6. Direction of Force

at the instrument panel, the upper end moves to the right (6E). An 11 o'clock impact to the same area deforms the front to the right and the forward end of the column moves in that direction while the upper end moves left (6F and Case History 71-335A). No lateral displacement occurs in 12 o'clock impacts to this area (6G).

An 11 and 12 o'clock impact to the right front area generally deforms the car body to the right and the forward end of the column moves to the right, the upper end to the left (6H and 6J). At 1 o'clock impact to the right front often deforms the body to the left so that the forward end of the column moves left while the upper end moves right (6I).

A series of case histories confirming the predicted patterns is presented in the following pages. The combined influence of a number of factors is illustrated in Case History 71-386A.

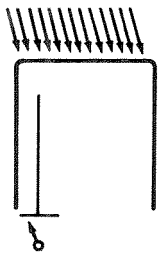
Case Histories

71-469A, 73-7, 71-77A, 71-334A. Cases 71-469A (photographs 27 and 28) and 73-7 involve frontal impacts with an 11 o'clock resultant direction of force. Both drivers moved forward and to the left sustaining severe thorax injuries from contact with the steering assembly. EAD compression was minimal.

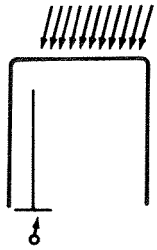
Cases 71-77A and 71-334A (photographs 29 and 30) are similar to the above but involve a 1 o'clock resultant direction of force. Both drivers moved forward and to the right, sustaining severe thorax

Case No.	VDI	Maximum Crush	EAD Collapse	Shear Capsule	Column Rotation
71-469A	11–FDEW	19"	0.0"	0.8"	0.0°
73-7	11–FDEW	23.5"	0.2"	0.5"	-0.4°
71-77A	01–FZEW	19"	0.4"	1.0"	+2.5°
72-334A	01–FDEW	48"	1.2"	1.1"	+4.5°

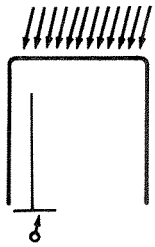
71–469, 73–7



71–77A



71–334A



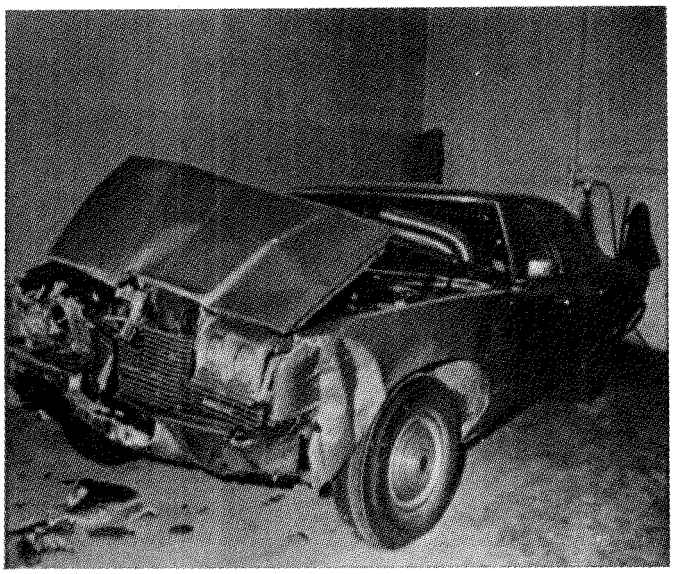


Photo 27

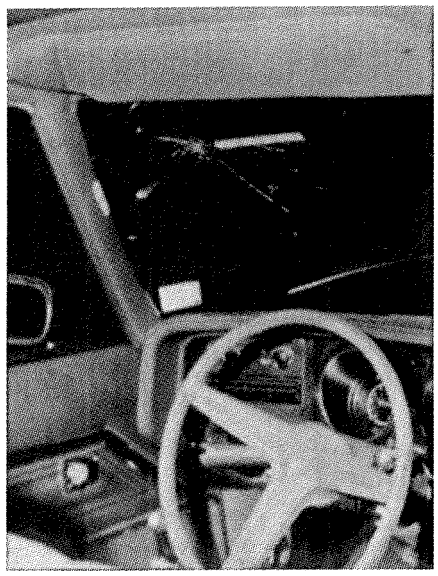


Photo 28

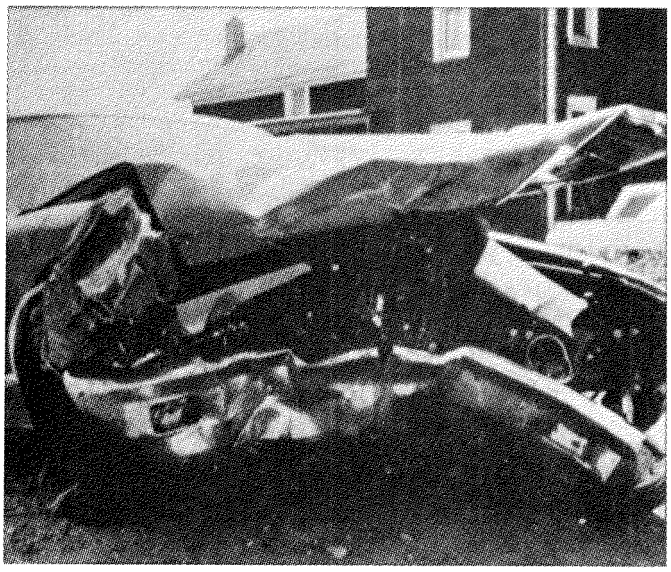


Photo 29

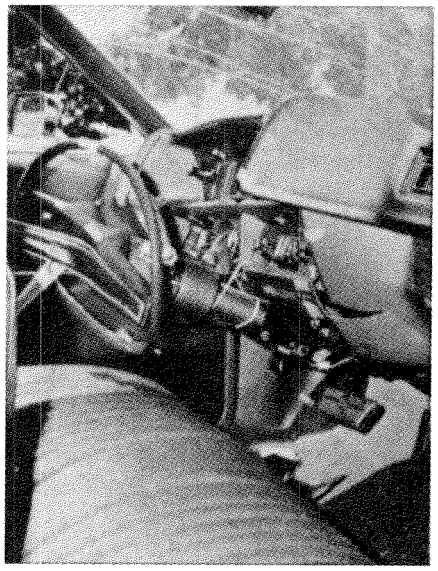
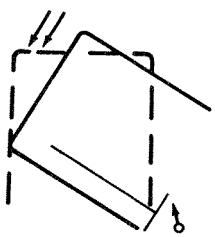


Photo 30

injuries from steering assembly contact. Again, EAD compression was minimal.



72-31A. A 1 o'clock left front impact in this case increased counterclockwise vehicle rotation (photographs 31 to 33). Note in photograph 33 that the driver moved to the right contacting the right windshield and instrument panel areas (taped). Driver contacts in these areas are generally indicative of minor or no involvement with the steering assembly.

71-335A. One effect of direction of force (in combination with area of impact and accident severity) is to accentuate lateral movement of the forward end of the steering column. Case No.

Case No.	VDI	Maximum Crush	EAD Compression	Shear Capsule	Column Rotation
72-31A	01–FLEW	32″	0.4″	1.0″	+12.5°

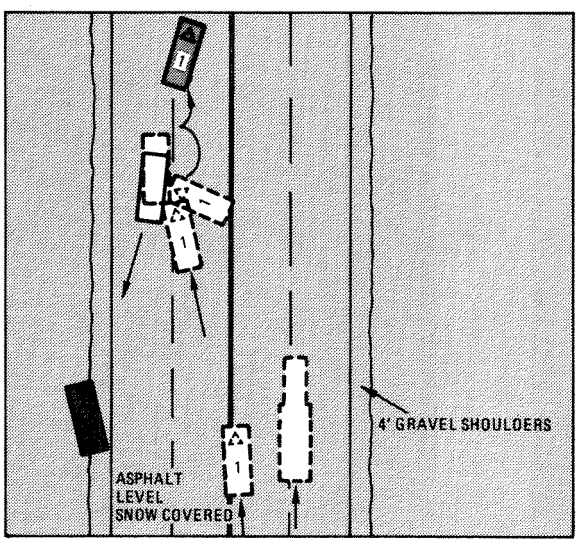


Photo 31

Photo 32

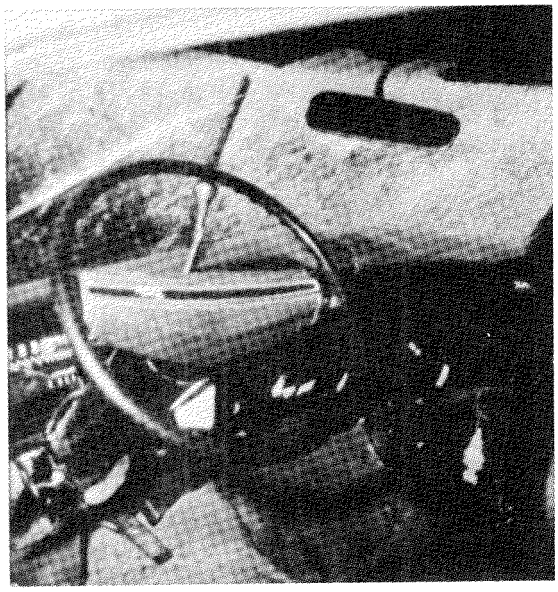
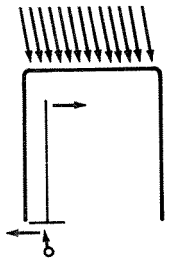


Photo 33



71-335A demonstrates the combined effect of occupant and column movement, as a result of an 11 o'clock direction of force applied during the accident sequence. Contact was distributed across the frontal area. However, the 11 o'clock force direction distorted the entire frontal structure to the right and rotated the upper end of the column to the left. The driver also moved to the left, sustaining a severe thorax injury as a result of contacting the steering assembly. EAD compression was minimized by the driver's trajectory to the left and by upper column rotation to the left.

71-386A. All of the factors discussed to this point have a distinct effect on EAD performance. The influence of a combination of these factors appears in Case No. 71-386A. This vehicle contacted a bridge railing and utility pole with the right frontal area, moving the forward end of the column laterally to the right and the upper end to the left (photographs 36 to 38). The column also rotated upward (positive) as a result of the extent of crush. Following maximum engagement, the vehicle rotated in a clockwise direction and the driver moved to the left relative to the steering column.

Height of Impact

The height of an impact and the direction of force application, particularly to the left front of the car, generally increases or limits EAD compression. A high

Case No.	VDI	Maximum Crush	EAD Compression	Shear Capsule	Column Rotation
71-335A	11–FDEW	43"	1.0"	0.0"	+2.5°

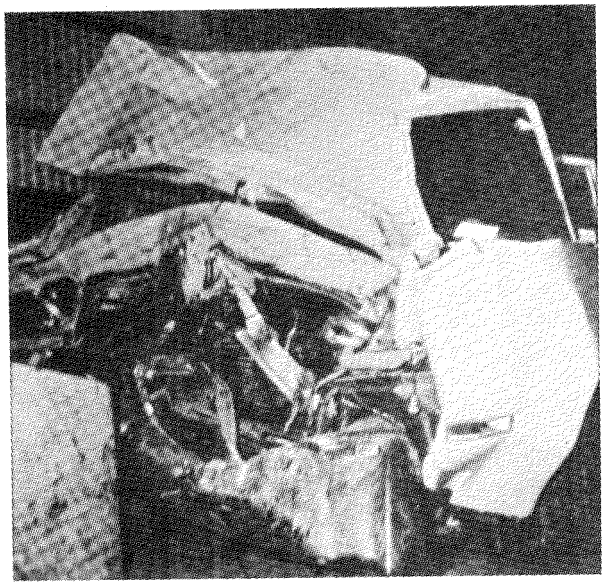


Photo 34

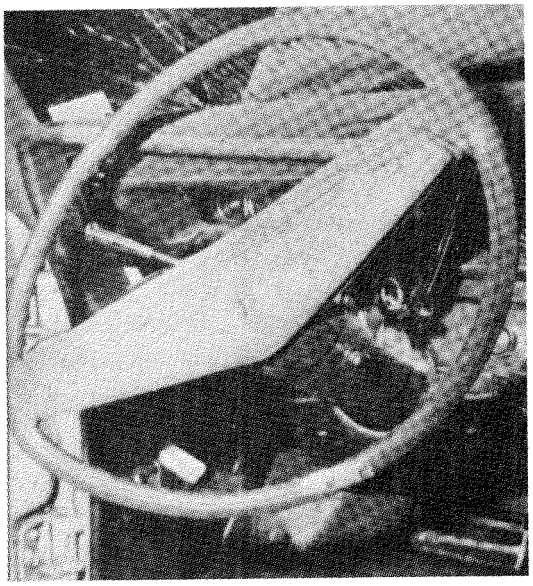
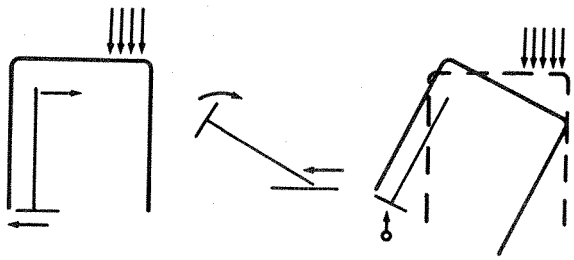


Photo 35

Case No.	VDI	Maximum Crush	EAD Collapse	Shear Capsule	Column Rotation
71-386A	2–FZEW	77"	0.5"	0.8"	+26.0°



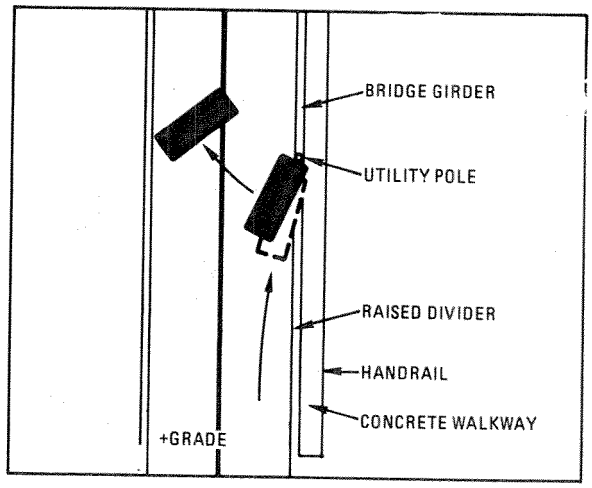


Photo 36

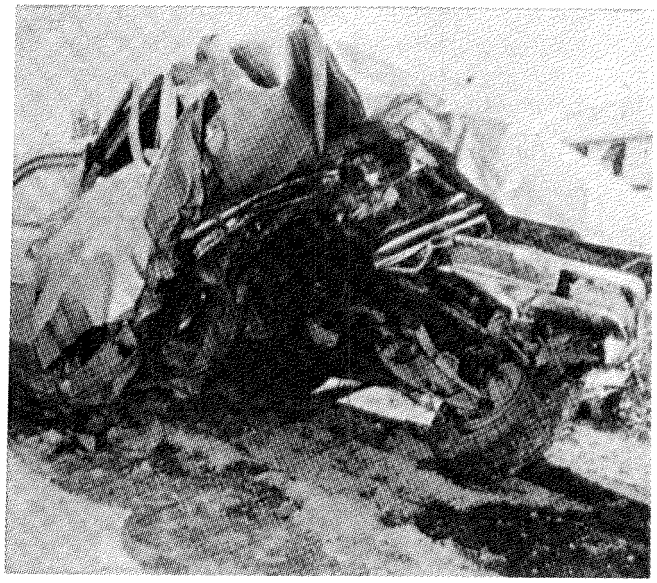


Photo 37

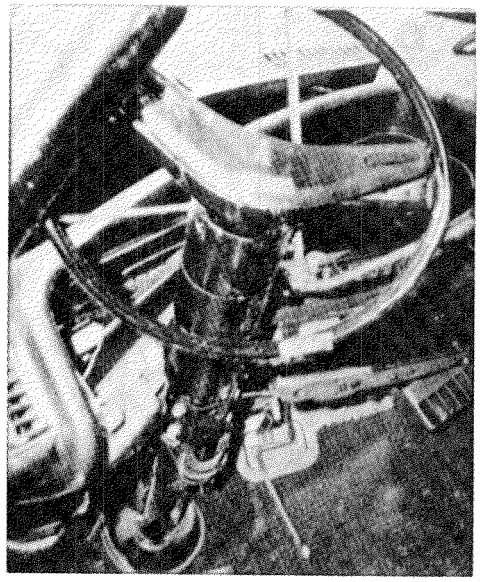


Photo 38

impact (above the bumper level) forces the column to rotate about the forward end causing the upper end to move down and producing a more favorable condition for driver contact (Figure 7A and Case Histories 73-87, 71E-60A). A low impact generally causes the column to rotate about the panel mounting so that the upper end of the column moves upward, a less favorable condition for the driver (Figure 7B). A low impact with a marked vertical component of force (an impact with a culvert would be an example) often produces a favorable condition by forcing the front of the column upward (Figure 7C).

Case histories illustrating the predicted effects of impact height appear on the following pages. Where other factors modify predicted behavior their influence also is shown (see Cases 72-71A, 71E-38A, and 72-75A, accident severity).

Case Histories

73-87, 71E-60A. Cases 73-87 and 71E-60A are examples of a high impact, i.e., an underride, which forced the upper end of the column downward, producing a more favorable condition for driver contact. The table below and photographs 39 and 40 demonstrate the negative column rotation and

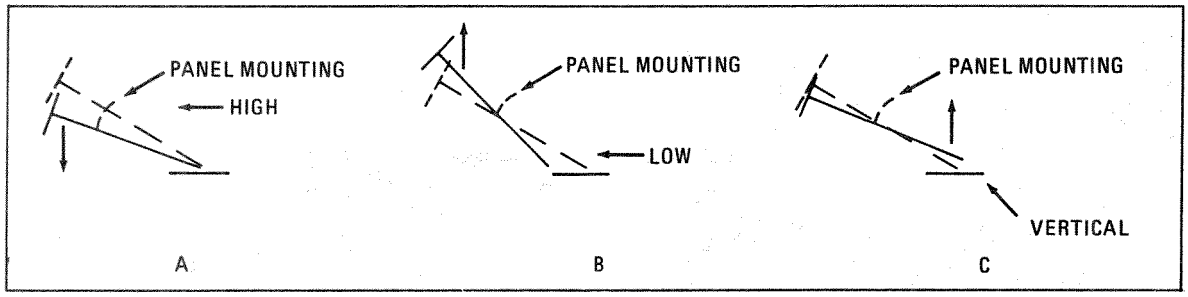
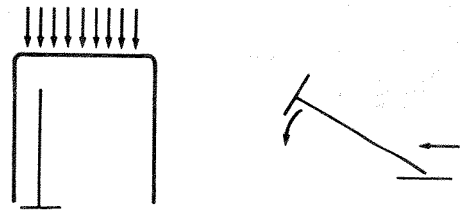


Figure 7. Height of Impact

marked EAD compression resulting from this impact configuration.

If a high impact is extremely severe, deformation of the steering column may occur, producing positive (upward) rotation of the upper end of the column. This is illustrated in photographs 41 and 42 below (Case 72-71A).



Case No.	VDI	Maximum Crush	EAD Collapse	Shear Capsule	Column Rotation
73-87	*12–FDEW	27.5"	6.0"	3.0"	-4.0°
71E-60A	12–FDMA	24.0"	5.5"	5.2"	-2.9°

*Case 73-87 is a borderline underride impact. The entire vertical height of the frontal area is involved. However, the amount of crush is more extensive above chassis frame height.

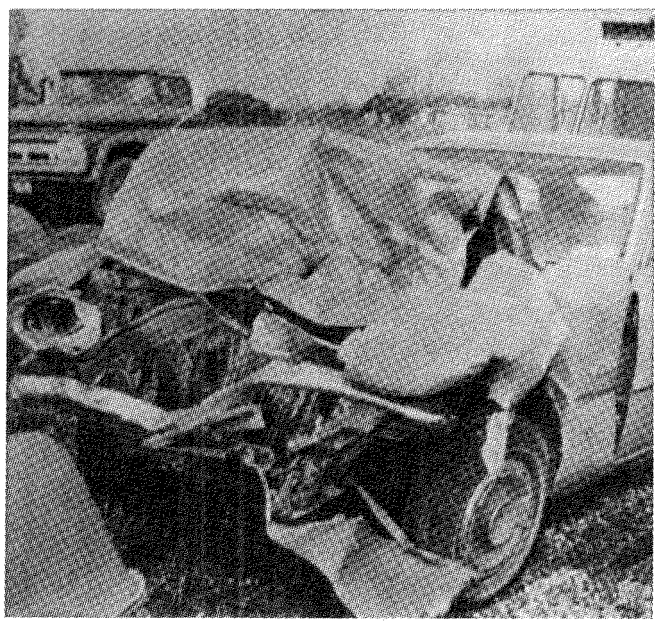


Photo 39

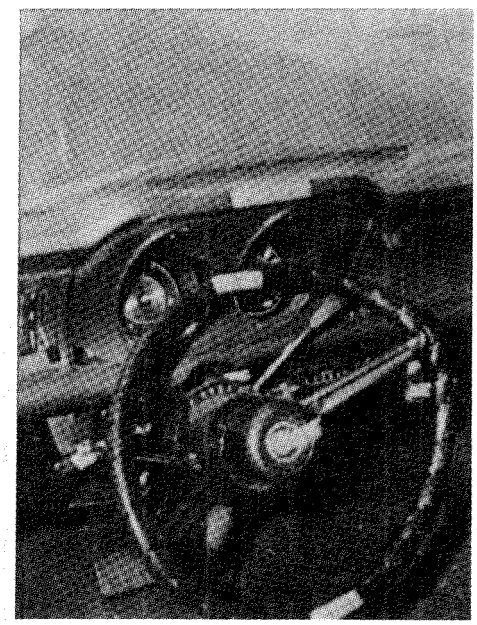


Photo 40

71-405A. A frontal impact which forced the frontal structure downward and caused the upper end of the column to rotate upward (positive) is illustrated in photographs 43 and 44. Rotation was increased by the extent of the crush in this case (71-405A).

72-54A. An example of a frontal impact with a marked vertical component of force is illustrated in

Photo 41

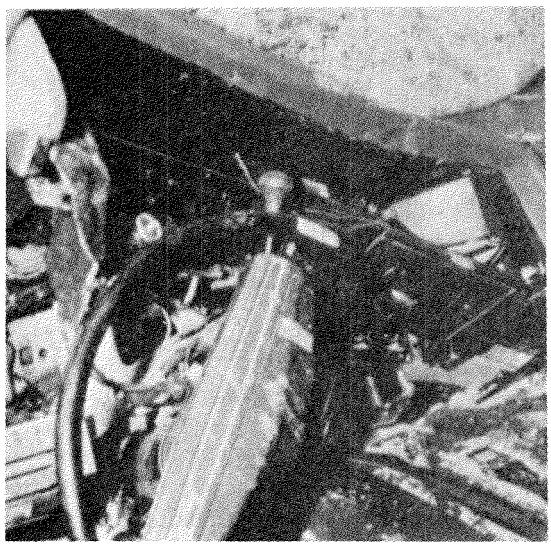
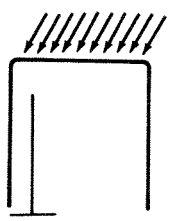


Photo 42

Case No.	VDI	Maximum Crush	EAD Collapse	Shear Capsule	Column Rotation
72-71A	01–FDHW	78″	3.1″	4.0″	+15.6°



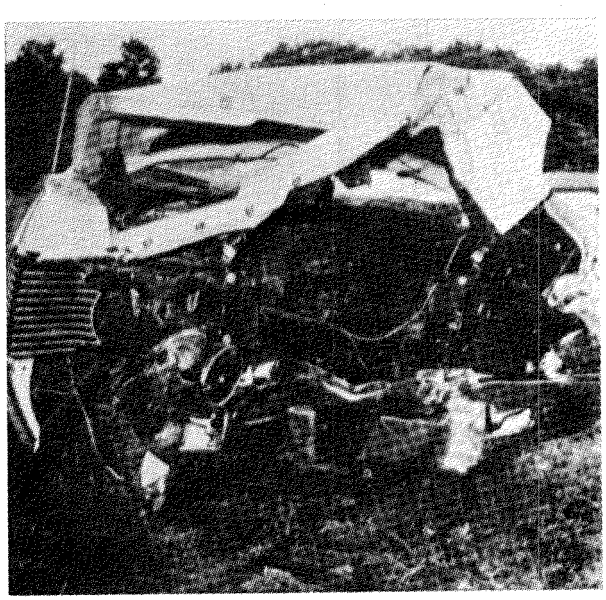


Photo 43

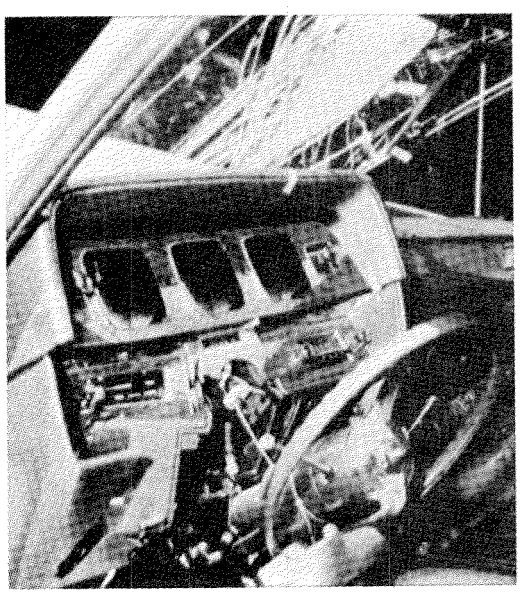


Photo 44

Case No.	VDI	Maximum Crush	EAD Collapse	Shear Capsule	Column Rotation
71-405A	01–FDEW	41″	4.0″	0.6″	+11.8°

photographs 45 and 46 (Case 72-54A). This vehicle struck a snowbank and then began rising as it plowed through the bank. Photograph 45 illustrates the extent of the upward deformation of the front of the vehicle. This movement forced the forward end of the column upward, inducing a negative (downward) rotation of the upper end of the column. Note the extent of EAD compression relative to crush.

71E-38A, 72-75A. As noted earlier for high frontal impacts, severe impacts also may produce positive rather than negative column rotation in this situation. Photographs 47 (71E-18A) and 48 (72-75A) illustrate frontal damage to two cars in which the severity of the accident resulted in positive (upward) rotation although vertical forces were applied to the vehicle. In case 72-75A, positive rotation was accentuated by the driver who bottomed in the seat, then struck the lower wheel rim bending it under and forcing the column upward.

STATISTICAL SUMMARY OF CASE DATA

In this section, all 549 frontal impact cases are analyzed in order to determine the effect of the accident variables discussed previously on EAD

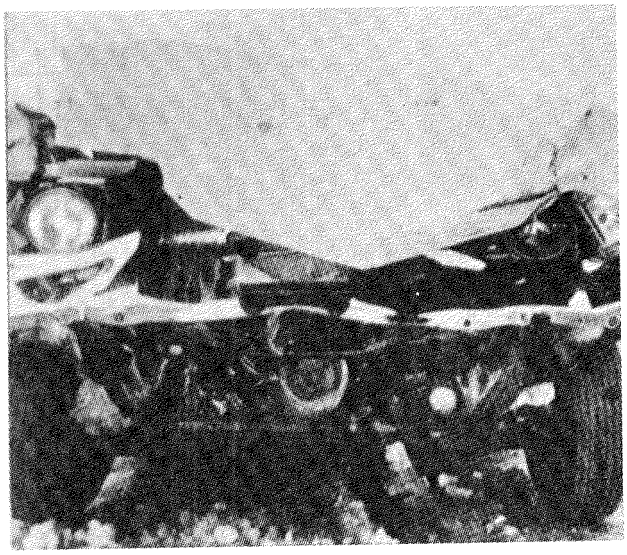


Photo 45

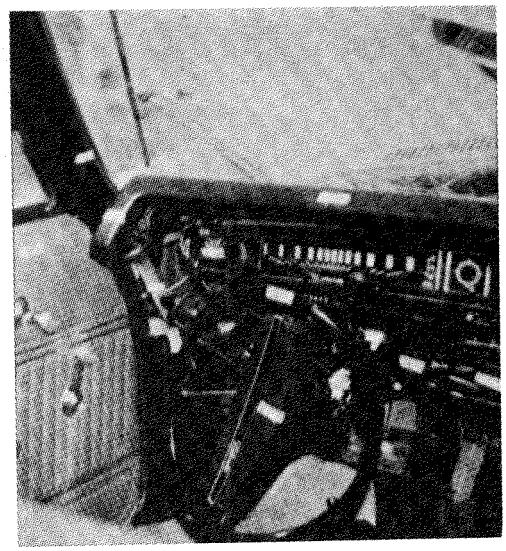
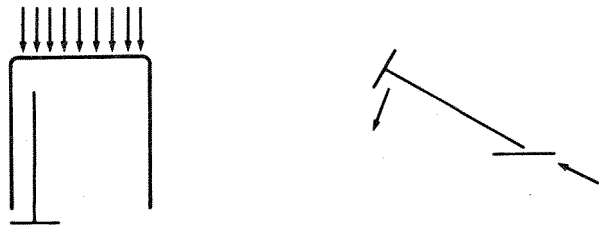


Photo 46

Case No.	VDI	Maximum Crush	EAD Collapse	Shear Capsule	Column Rotation
72-54A	12–FDEW	29"	6.8"	7.3"	-8.8°



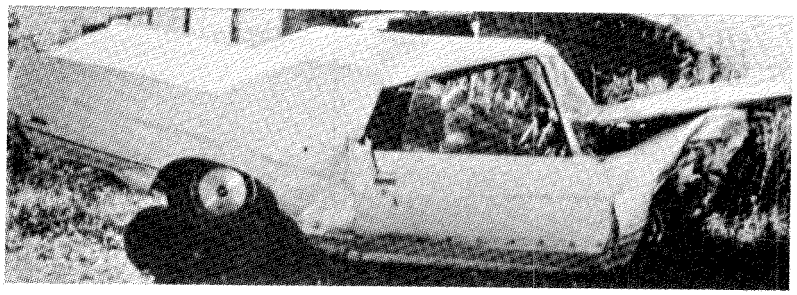
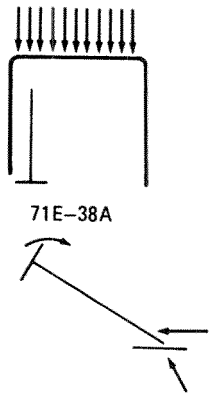


Photo 47



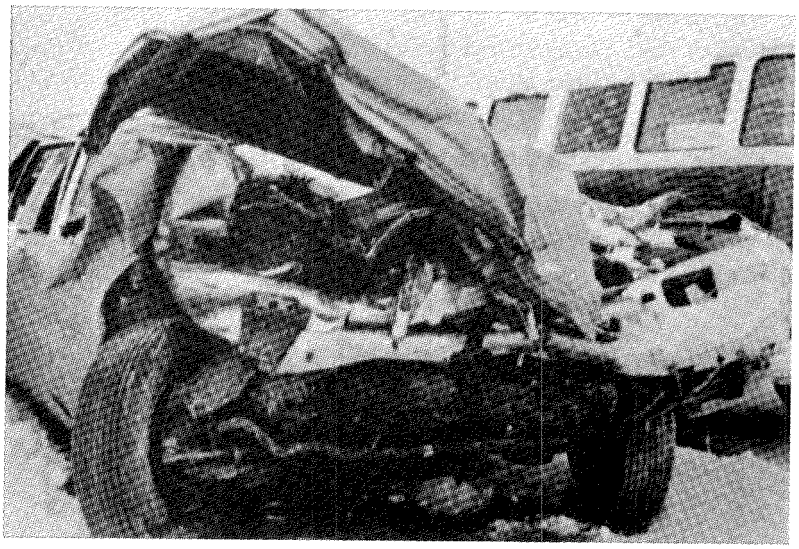
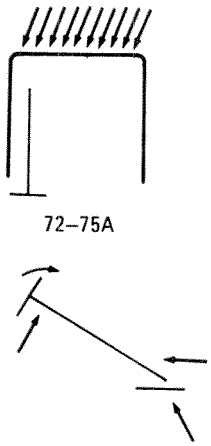


Photo 48



Case No.	VDI	Maximum Crush	EAD Collapse	Shear Capsule	Column Rotation
71-E-38A	12–FDEW	55″	4.3″	3.0″	+1.3°
72-75A	01–FDEW	51″	1.7″	1.1″	+3.6°

compression and driver injury. Relevant factors such as vehicle crush, car make and restraint use also are discussed. Only impacts from 11 to 1 o'clock are included.

Maximum Vehicle Crush and EAD Compression

Figure 8 shows the extent of EAD compression when various amounts of vehicle crush occurred. Vehicle crush is shown in 10" increments and EAD compression in 1" increments.

In each category, the largest proportion of cars appears in the 0″ compression category and the next largest proportion is the .1″ to 1.0″ category. Overall, over half of the cars had no compression of the EAD and, even in the severest category – 41″ or more of crush – 50.7″ percent of the cars had one inch or less of compression.

The proportion of cars with no EAD compression decreased steadily from a high of 82.1 percent to a low of 34.7 percent as vehicle crush increased. Overall, 76.4 percent of the cars studied had 1″ or less of EAD compression regardless of the amount of vehicle crush.

Vehicle Crush, EAD Compression and Car Make

The amount of EAD compression within various vehicle crush ranges is shown for U.S. car makes in Table 1. In each crush category, the incidence of EAD compression was greater among General Motors cars than among Ford cars. As vehicle crush increased, the difference between the two makes with respect to the incidence of EAD compression also increased. The proportion of General Motors cars that had 1.1″ or more of compression also was greater

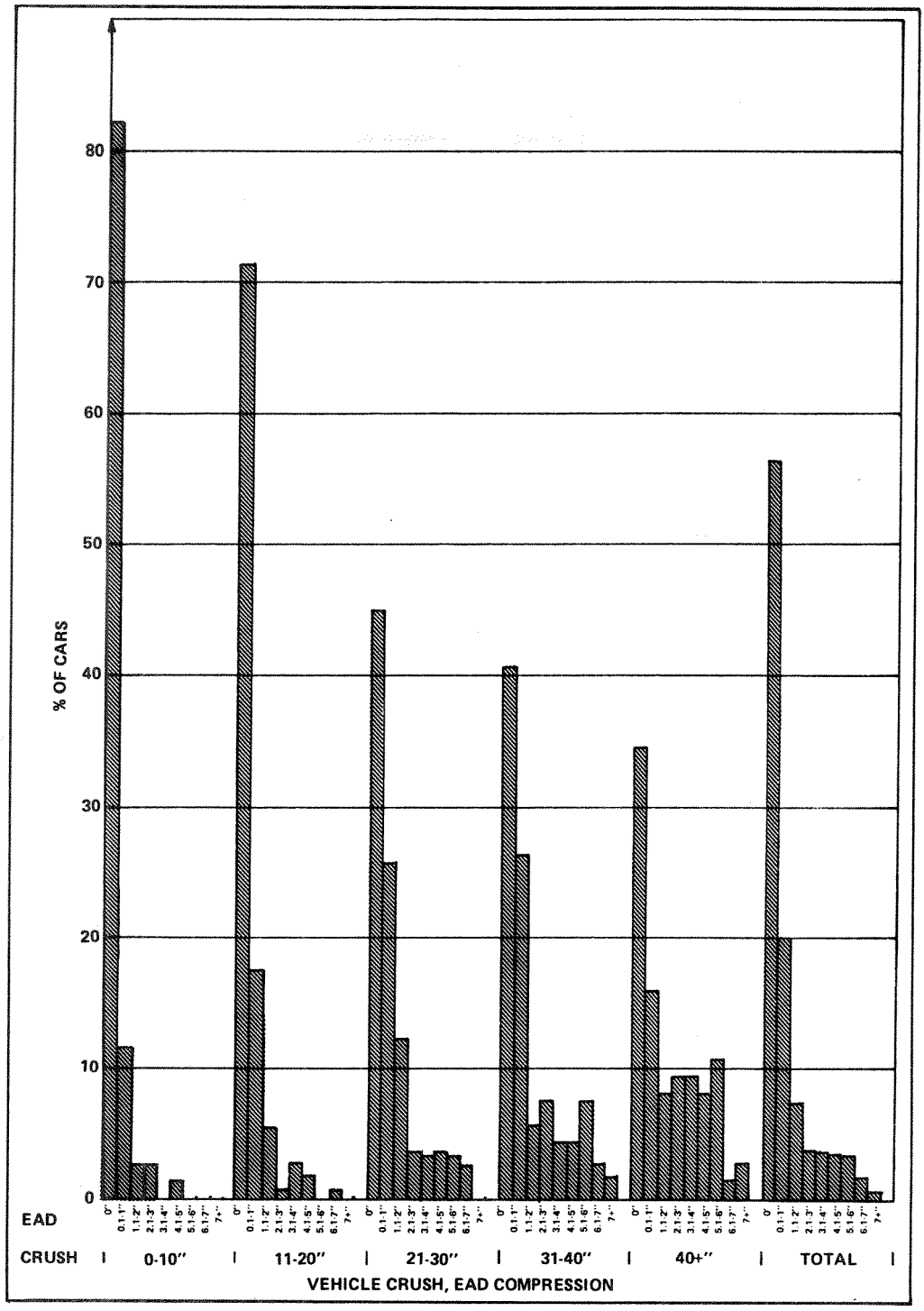


Figure 8. Vehicle Crush, EAD Compression

than that for Ford, except in the 0 to 10″ crush category. Thus, it appears that the Ford EAD was stiffer or less yielding than that of General Motors.

The Chrysler EAD data tended to resemble that of General Motors more than that of Ford. However, the volume of data available for both Chrysler and American Motors was limited.

Vehicle Crush, EAD Compression and Restraint Use

Data in Table 2 show that in each vehicle crush range, there is less EAD compression when lap belts are worn than when they are not. It is believed that this reflects the fact that accident severity, direction of force and other accident variables discussed in the

Table 1 Vehicle Crush, EAD Compression and Car Make					
Vehicle Crush	EAD Compression	AMC	Chrysler	Ford	GM
0-10"	0	100.0	85.0	81.5	77.8
	.1-1.0"	0	15.0	7.4	14.8
	1.1+	0	0	11.1	7.4
11-20"	0	63.6	48.5	85.4	73.4
	.1-1.0"	9.1	36.4	12.5	13.9
	1.1+	27.3	15.1	2.1	12.7
21-30"	0	9.1	38.7	59.6	43.5
	1.-1.0"	45.5	16.1	25.0	27.4
	1.1+	45.4	45.2	15.4	29.1
31-40"	0	0	25.0	56.5	36.1
	.1-1.0"	0	37.5	21.7	27.8
	1.1+	100.0	37.5	21.8	36.1
41"+	0	66.7	35.7	50.0	18.7
	.1-1.0"	33.3	0	15.4	21.9
	1.1+	0	64.3	34.6	59.4

previous section is greater than driver influence on EAD compression. Although it is possible that the lap belt limits the amount of force applied to the steering assembly, the injury data in Table 3 do not substantiate this.

Vehicle Crush, EAD Compression and Driver Injury

For vehicle crash of 20" or less, AIS 2 or worse thorax injury occurred when there was little or no EAD compression. When 1.1" or more of compression occurred (see Figure 9), this compression appeared to be beneficial to the driver and no AIS 2 injuries occurred.

When vehicle crush was 21" to 30", the probability of AIS 2 or worse thorax injury was nearly equal in both EAD compression categories. At vehicle crush levels above 31" the probability of an AIS 2 or worse thorax injury was greater when EAD compression exceeded 1" than when it was 1" or less. At these higher crush levels, the apparently more favorable injury experience of drivers with limited EAD compression occurs because many of them did not contact the wheel, or contacted it lightly as a result of vehicle rotation and direction of force.

The probability of an AIS 2 or worse head injury was essentially the same throughout the entire vehicle crush range when EAD compression was 1" or less (Figure 10). When EAD compression exceeded 1" the occurrence of head injury was markedly lower up to a crush level of 30 inches. Above 30" of crush, the proportion of head injury was essentially the same as for EAD compression of less than 1 inch. Thus, there is a decrease in head injury associated with EAD compression in low to middle crush ranges.

Table 2 Vehicle Crush, EAD Compression and Restraint Use										
Vehicle Crush	0-10		11-20		21-30		31-40		41+	
EAD	0-1	1+	0-1	1+	0-1	1+	0-1	1+	0-1	1+
No Restraint	92.9	7.1	86.9	13.1	67.8	32.2	66.0	34.0	50.0	50.0
Lap Belt	94.4	5.6	96.8	3.2	81.5	18.5	71.4	28.6	50.0	50.0
Lap and Shoulder	100.0	0	100.0	0	66.7	33.3	50.0	50.0	66.7	33.3

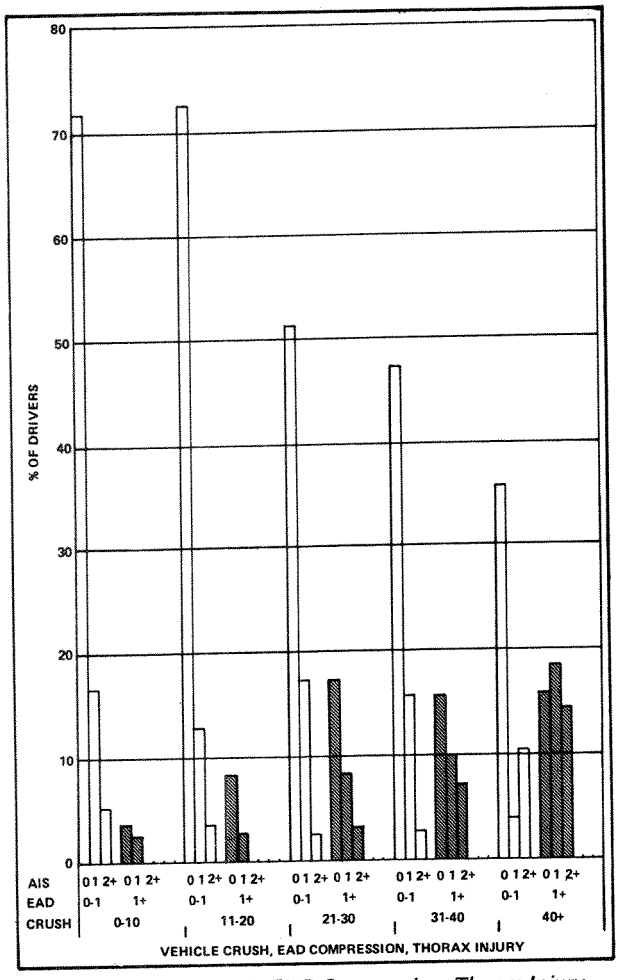


Figure 9. Vehicle Crush, EAD Compression, Thorax Injury

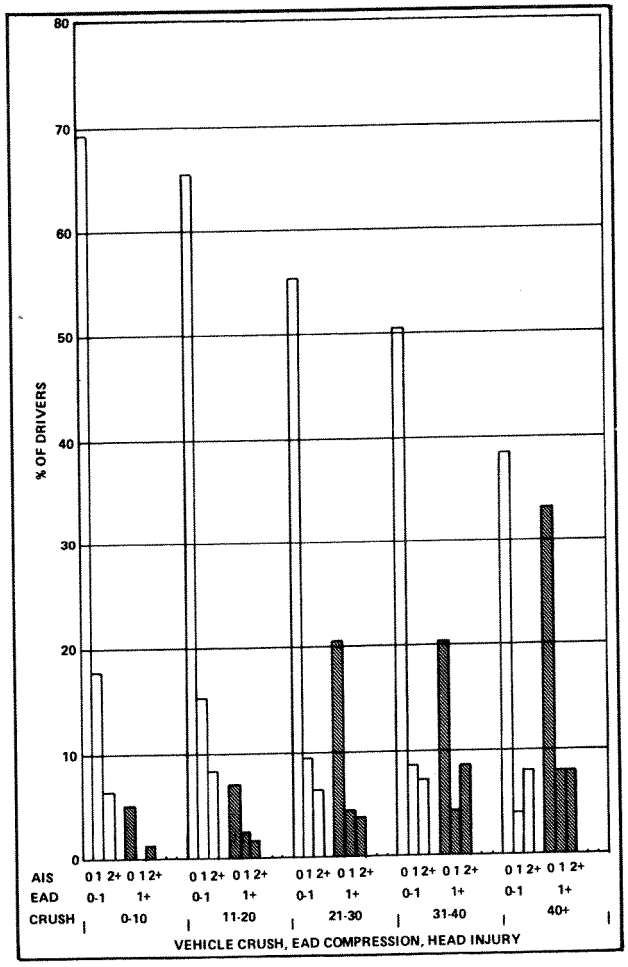


Figure 10. Vehicle Crush, EAD Compression, Head Injury

Restraint Use and Injury

Data in Table 3 indicate that the use of the lap belt alone did not appear to reduce wheel-caused Thorax injury. Use of both lap and shoulder restraints reduced these injuries markedly. However, the data volume was small (15 cases).

The proportion of head injuries was somewhat higher when the occupant was unrestrained than when the lap belt was worn. Lap and shoulder restraint use appeared to reduce head injury occurrence. As noted, available data volume for the lap and shoulder straint condition was small.

CONCLUSION

Maximum compression of the EAD occurs when forces are applied in a plane perpendicular to the steering column. The angle of force application is influenced by many factors, including the frontal area of the car impacted, the direction of force, vehicle rotation, accident severity and driver approach angle. Generally, as the angle of force application shifts from the perpendicular, the compression of the EAD decreases.

Table 3
Restraint Use and Injury

	Thorax Injury		
AIS	No Restraint	Lap Belt	Lap and Shoulder
0	70.2	68.5	93.3
1	21.3	23.4	6.7
2 or more	8.5	8.1	0
n	423	111	15

	Head Injury		
AIS	No Restraint	Lap Belt	Lap and Shoulder
0	74.7	67.6	80.0
1	14.7	18.0	13.3
2 or more	10.6	14.4	6.7
n	423	111	15

Oscillation of the steering assembly upon impact also appears to influence the compression of the column. The role of oscillation is difficult to assess because it is impossible to predict the position of the wheel when the driver contacts it. It seems clear that in certain impacts, wheel oscillation operates to either increase or limit EAD performance.

Any vertical component of force in a frontal impact was found to be particularly useful in predicting column behavior. It is suggested that this information be added to the Collision Deformation Classification because of its implications in terms of occupant injury.

The present method of measuring the steering column angle after impact was found to be inadequate in a number of cases. Negative column rotation often was measured by the investigator because damage prevented the column from fitting back into normal position at the panel. However, examination of case photographs and vehicles indicated that this could be detected and properly recorded. In some cases an accurate measurement was not possible, but the proper direction of rotation could be identified from panel and column damage.

A review of compliance test procedures should be considered in view of the fact that, even in frontal impacts from 11 to 1 o'clock where at least one occupant required hospital treatment, approximately 76 percent of the columns studied compressed one inch or less.

Results of this study should not be interpreted as arguments against the EAD concept. These analyses do indicate the need for improved implementation.

REFERENCES

1. "Automotive Crash Injury Research Primer of Facts and Figures," Cornell University, August 1961.
2. Huelke, D.F., and Gikas, P.W., "Causes of Deaths in Automobile Accidents – Final Report," Office of Research Administration, Report No. 06749-1-F, The University of Michigan, April 1966.
3. Anderson, T.E., "Analysis of Vehicle Injury Sources," Calspan Corporation, Report No. ZM-5010-V-2R, September 1972.
4. Garrett, J.W., "A First Look – Differences in Injuries in 1968 and Pre-1968 Automobiles," Society of Automotive Engineers, Report No. 680542, August 1968.
5. Nelson, W.D. and Wilson, R.A., "Field Accident Research," Proceedings General Motors Corporation Automotive Safety Seminar, July 1968.
6. Huelke, D.F. and Chewning, W.A., "The Energy Absorbing Steering Column," Highway Safety Research Institute, Report Bio-7, 1968.
7. Skiels, P.C., "Field Experience with the General Motors Energy Absorbing Steering Column," 14th Annual Canadian Highway Safety Council, 1968.
8. Patrick, L.M., "Analysis of Automobile Accident and Experimental Data – Steering Assembly Impacts," Biomechanics Research Centre, Wayne State University, 1968.
9. Gadd, C.W. and Patrick, L.M., "System Versus Laboratory Tests for Estimating Injury Hazard," Society of Automotive Engineers, Report No. 680053, 1968.
10. Nahum, A.M., Siegel, A.W., and Brooks, S., "The Reduction of Collision Injuries: Past, Present, and Future," Society of Automotive Engineers, Report No. 700895, 1970.
11. Levine, D.N. and Campbell, B.J., "Effectiveness of Lap Seat Belts and the Energy Absorbing Steering System in the Reduction of Injuries," Highway Safety Research Center, November 1971.
12. O'Day, J. and Creswell, J.S., "Can the Effect of Changes in Vehicle Design be Seen in Mass Accident Data," HIT LAB Reports, University of Michigan, February 1971.
13. DuWaldt, F.A., "Basic Research in Crashworthiness II – Steering Column Investigation," Calspan Corporation, Report No. YB-2987-V-20, May 1973.
14. Searle, J.A., "Analysis of Achievements to Date in the Safety Design of Steering Assemblies," Motor Industry Research Association, Report No. C195173, April 1973.
15. "VSDSS Research Studies," New York State Department of Motor Vehicles DOT-HS-800-780, PB 200 984, January 1973.
16. Griffin, L.I., "Analysis of the Benefits Derived from Certain Presently Existing Motor Vehicle Safety Devices: A Review of the Literature," Highway Safety Research Center, December 1973.
17. McLean, A.J., "Collection and Analysis of Collision Data for Determining the Effectiveness of Some Vehicle Systems," Motor Vehicle Manufacturers Association (not yet published).

NHTSA'S EVALUATION OF AIR CUSHION RESTRAINT SYSTEM EFFECTIVENESS (ACRS)

DONALD F. MELA
Office of Statistics and Analysis
National Highway Traffic Safety Administration

INTRODUCTION

The General Motors Air Cushion Restraint System (ACRS), popularly known as the "air bag," is being installed in several thousand 1974 U.S. General Motors passenger car production models.

There is intense interest in the performance of this passive restraint system because the occupant passive restraint safety standard* is presently scheduled to take effect in two years (September 1976). Therefore, the evaluation of the ACRS effectiveness is a high-priority program for NHTSA. Let me describe how NHTSA is carrying out the evaluation.

The material in the paper developed by a task group of members of NHTSA's Office of Statistics and Analysis (formerly Office of Accident Investigation and Data Analysis). William E. Scott, C. J. Kahane, John Keryeski and Scott Lee were the principal contributors, and should be given credit for the work that is described here.

General Motors Corporation in 1973 announced its aim to produce and sell 50,000 full-size ACRS-equipped cars in the 1974 model year, and 100,000 1975 model cars. The introduction of these cars began about January 1, 1974. Subsequently, there has been a drastic reduction in the expected rate of introduction of ACRS vehicles. It now appears (May 1974) that the number of ACRS cars on the road in the United States by the end of 1974 will not exceed 8,000 cars. So the original NHTSA estimates of crash data availability have also had to be reduced.

OBJECTIVES

The basic objectives of the evaluation program are:

1. Assess the injury-reducing effect of ACRS.
2. Determine operational characteristics of the ACRS.
3. Evaluate public/owner acceptance.

This discussion will be concerned mainly with the first objective, but objectives two and three will also be discussed briefly. As noted above, the drastic reduction that has occurred in the rate of introduction of ACRS vehicles has made it necessary to replan. However, much of the general approach and many of the specifics will either be preserved or will be utilized at a later date, when more ACRS vehicles are on the road. Therefore, it is felt that the information should be of interest to participants in this Conference, particularly since it affords the opportunity to comment and to influence the later conduct of the evaluation.

GENERAL CONSIDERATIONS

Based upon the expected sales rate, it had been estimated that there would be about 12,000 ACRS vehicles in crashes by the end of August 1975. This includes all accident types from minor to total demolishment. It also includes crashes in which the ACRS is not deployed (ACRS is designed only to deploy in impacts with a significant frontal component). Of these 12,000 crashes, about 4,000 would damage the vehicle severely enough to require towaway and 1,300 crashes would result in some injury. Table 1 provides a further classification, according to severity level of the injury.

The estimates in Table 3 were developed from previous accident experience for vehicles similar to the ones expected on the road. The injury frequencies presented would be expected if all occupants in the control group were unrestrained. The observed

Table 1
INJURY SUMMARY

ACCIDENT SEVERITY	PARTS OF THE BODY AFFECTED					
	HEAD	NECK	THORAX	ARMS	UPPER LEGS	LOWER LEGS
VDI = 1						
VDI = 2						
VDI = 3						
VDI = 4						
VDI = 5						

Table 2
PREDICTED ACRS CRASH EXPERIENCE
(Predicted in December 1973)

VEHICLES/EVENTS	LEVEL OF INVOLVEMENT
12,000	ALL POLICE REPORTED ACCIDENTS
4,000	TOWAWAYS
1,300	TOWAWAYS WITH MINOR OR GREATER INJURY
360	TOWAWAYS WITH MODERATE OR GREATER INJURY
120	TOWAWAYS WITH SEVERE TO FATAL INJURY
20	TOWAWAYS WITH FATAL INJURY

*Standard 208.

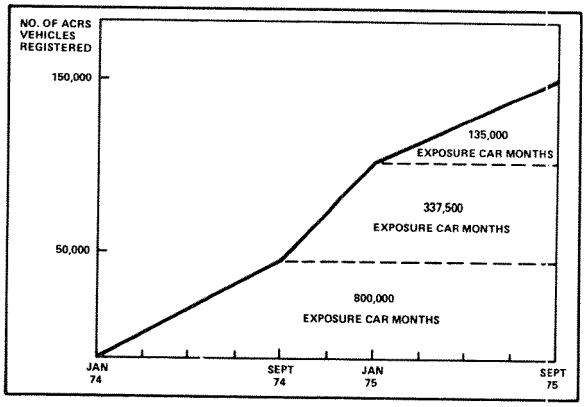


Figure 1. Original Schedule for Planning Accident Investigations (Basis for Table 1).

frequencies of injuries in ACRS crashes should be much lower.

To compare the ACRS with active restraint systems such as safety belt systems, we ask two general questions:

1. Is the overall safety performance of the ACRS better than that of cars equipped with devices such as the ignition interlock or buzzer supplements to safety belt systems?
2. How do vehicle occupant injury rates compare for persons involved in crashes who are protected by
   a. ACRS,
   b. Three-point belt,
   c. Lap belt, or
   d. With no restraints?

This identifies several groups for comparisons with the ACRS fleet.

1. Interlock Fleet – 1974 and 1975 model years.
2. Buzzer Fleet – 1972 and 1973 model year cars.
3. Car occupants wearing lap and shoulder belts, 1968 and later model years.
4. Car occupants wearing lap belts.
5. Unrestrained occupants.

Since the ACRS will be in full-size 1974-75 Cadillacs, Buicks and Oldsmobiles, these control groups should be restricted to cars of the same size and body construction, viz., full-size 1973-75 Cadillacs, Buicks, Oldsmobiles, Pontiacs and Chevrolets.

Rationale for Objective #1

Ideally, it would be best to compare injury performance of the ACRS and control group for the "same" exposure or accidents. There are two well-established approaches to the ideal of "sameness" that is unattainable in real-life:

1. Assume that, overall, ACRS and control-group cars have similar exposure or accident experience and that any difference in gross injury rates is attributable to the restraint system alone.
2. Use some criteria to stratify the exposure units or accidents into sub-groups. The stratification is of a sort that corresponding sub-groups may be compared using approach (1). A statistical method is employed to aggregate the comparison of the sub-groups into a net comparison of ACRS and control group.

Approach (1) is rejected because there is no evidence to support its underlying assumption. ACRS-car owners may well be highly unrepresentative of the automobile population with regard to both exposure and accident experience. In fact, some have conjectured that ACRS buyers would be extremely safety-conscious, while others felt they would be reckless drivers eager to evade the interlock system.

Therefore, Approach (2) was selected. However, the use of "exposure units" had to be rejected. At this time, there is no wide recognized method for stratifying the exposure units – e.g., sub-dividing the miles driven into "dangerous" and "non-dangerous" categories – making it impossible to use fleet exposure as a basis for stratifying until such theory is developed.

On the other hand, there are four well-established means for stratifying accidents into severity classes: accident descriptors used by the police, such as "pre-impact speed"; economic descriptors such as "dollar damage"; damage descriptors that can be measured by looking at post-crash photographs, such as "inches of crush"; and highly sophisticated engineering descriptors such as "velocity-vector change during impact."

Accident and economic descriptors are highly inadequate because past experience show them to be poorly correlated with real accident severity, subject to large errors of measurement, and subject to different methods of measurement in different States – an important consideration since the sparse ACRS data will require collection in more than one State. Dollar damage has year-to-year inconsistencies due to secular economic trends.

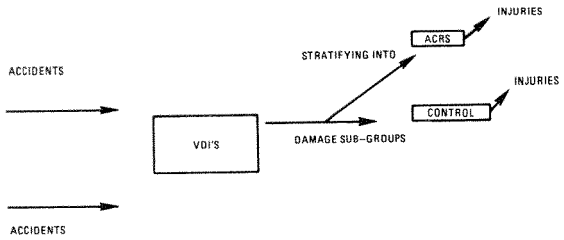
The engineering descriptors were considered the best measures of accident severity, but were rejected for three reasons: (1) some of the descriptors, such as "vehicle aggressiveness" have not yet been adequately defined; (2) the methods for measuring them, either crash recorders or the Calspan computerized accident reconstruction program, would not be available in time; and (3) these methods are exceedingly expensive.

There is a damage descriptor presently being used on an international basis – the Vehicle Deformation Index (VDI). It is described in the SAE recommended practice J224a, Reference 3. *With vehicles that are of*

essentially the same body style and with similar crashworthiness characteristics, the VDI is considered to give a good comparative measure of the magnitude of impact forces sustained by a vehicle. The VDIs may be readily and inexpensively calculated by looking at photographs of the damaged vehicle.

It was decided to collect photographs and construct VDIs while, left open, was the option to extend data collection at a later time in obtaining engineering descriptors. This procedure was followed just in case the VDIs failed to come up to expectations, or in case there were unforseen advances in the theory and measurement of engineering descriptors.

An examination of the inputs determined the following analytic approach:



Finally, the expected data on injury production were examined. Usually "injury severity" is coded for each *occupant* of the vehicle, but it can also be construed as a vehicle characteristic by using some composite measure such as "worst injury in vehicle." The latter approach was rejected because, for example, if the control group consists of occupants wearing "lap belts," then "worst injury in vehicle" cannot be fairly defined for a vehicle in which some occupants used belts and others did not.

Thus, injuries per occupant were considered, but here again there were several possibilities: occupant injuries could be stratified according to seated position, or seated position could be ignored. For this study, a middle course was taken. Since the ACRS would have no influence whatever on back-seat occupants, only front-seat occupants would be studied. Moreover, further stratification would dilute the statistical significance of the sparse ACRS data. Hence, a restriction of output "injury production" to "injuries to front-seat occupants" was effected.

The ACRS system is not expected to totally eliminate all injuries, but rather to lower their *severity*. It is, therefore, imperative that the measure of injury production be not a "yes-no" tally, but a severity scale.

The doctor-reported Abbreviated Injury Scale (AIS) defines severity levels in precise medical terms that are fairly consistently interpreted nationwide and thus, would properly distinguish intermediate levels of injury. It was decided to obtain a medical report on each injured front-seat occupant, and to compare ACRS and control-group injury production at each severity level from minor through fatal, but with special interest in the intermediate levels.

Following the decision to obtain injury rates at each level of severity, analytic methods for comparing the respective rates of ACRS and Control Groups were considered. In addressing the primary question, we queried whether statistical comparison should be a qualitative or quantitative statement. A *qualitative statement* would take the form: "Front-seat occupants of ACRS cars had a significantly lower severe-injury rate than unrestrained occupants"; a quantitative statement would be: "Front-seat occupants of ACRS cars had a 25 ± 5 percent lower severe-injury rate than unrestrained occupants."

The qualitative statements are second-best for a cogent reason: when comparisons are made between the ACRS and belts, it is expected that the ACRS may perform *better* with regard to severe injuries and worse on less severe injuries.* This is expected because the bag, if it deploys, should be superior to the belt; however there will be many less severe injuries in crashes where the bag (through design) fails to deploy.

Therefore, unless cost-benefit analysis is used to determine *how much better* and *how much worse,* it will be impossible to decide which restraint system is superior. (See Tables 2 and 3 for examples of quantitative results.)

Three acceptable statistical methods for comparing injury rates for a population stratified by damage severity were considered:

1. Stratify the ACRS and control-group front-seat occupants into four or five classes, according to the VDI extent and impact direction of the crash involved vehicle. Compare injury rates for corresponding strata. Take an appropriate weighted average over the strata to estimate net injury reduction.
2. For each front-seat occupant of an ACRS car, search the file of control-group cases and find the one which best matches the ACRS car with regard to numerous vehicle damage descriptors and other characteristics. Obtain two groups of vehicles of equal size who are so "similar" that any significant difference in gross injury rate of occupants is entirely attributable to the different restraint systems.
3. For each front-seat occupant of an ACRS car, search the file of control-group cases and find

*Unbelted occupants of ACRS cars may suffer minor injuries in crashes below the deployment speed.

about 20 that closely match the ACRS car. Compare the ACRS occupant's injuries to the median injury or to other percentiles of the corresponding 20 or, perhaps, do a Ridit analysis, Reference 4. Some of the ACRS occupants will fare better than the median, some the same, and some worse. An appropriate non-parametric test can be applied to the list of comparisons to determine which system, if any, is significantly better.

Only Methods (1) and (2) give adequate quantitative comparisons. Method (1) is preferred because it requires fewer and less detailed data than for (2) to attain a given level of precision.

In the contingency that the ACRS fleet is much smaller than expected (which now appears to be the case) Method (1) will not yield statistically significant quantitative comparisons. This is precisely where Method (3) is best, that is, when the test group is small and the control group large.

Hence, if things were still to go as originally planned Method (1) would be used to obtain quantitative results, but due to the apparent reduced ACRS fleet contingency, Method (3) will probably be used for qualitative results.

Most of the conceptual model was constructed simply by carefully examining the objectives. Figure 2 displays the model constructed. The details of the operational plans were not described in the rationale presented in the preceding pages. These include:

1. Details of notification, sampling, data collection, and analysis chosen to respond to objectives #2 and #3 (e.g., the collecting of more detailed data on injuries in deployment accidents).
2. Sampling techniques used to optimize statistical efficiency – i.e., maximize statistical precision per data-gathering dollar (e.g., use of stratified rather than simple random sampling).

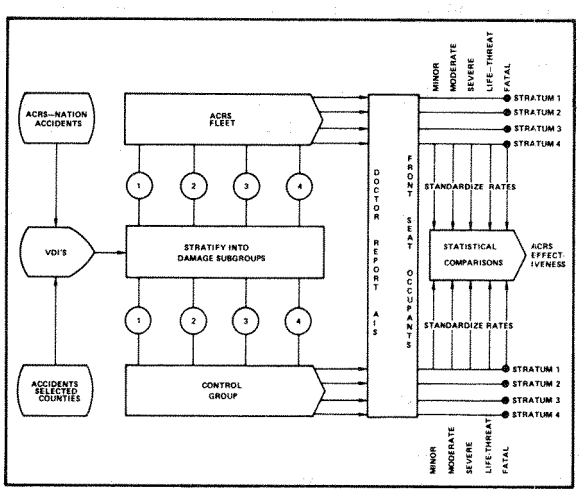


Figure 2. ACRS evaluation program model.

3. Real-world constraints that force the collection of extraneous data or prevent the collecting of data.

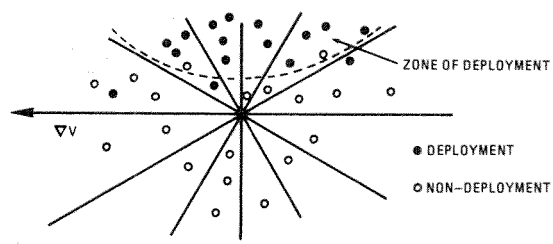
Rationale for Objective #2

Evaluate the performance of the ACRS in those accident events for which protection was designed.

The first point which must be clarified is the definition of "Operational Characteristics." This is taken to include the following:

1. The types of crashes which deploy the ACRS.
2. The types of injuries that result from ACRS deployment.
3. The tendency of the ACRS to deploy inadvertently.

The Types of Crashes Which Deploy the ACRS: What one would ideally want would be a polar plot as shown where



the principal direction of force (θ) and the change in velocity of the vehicle during impact (ΔV) would be plotted for accidents in which the bag both did and did not deploy. The "zone of deployment" would thus characterize those accidents which result in ACRS deployment.

To construct such a plot one would need detailed information concerning a significant number of accidents: "detailed" means either crash recorders on the vehicles or a computerized accident reconstruction program used together with an investigating team to estimate ΔVs and directions of force from physical evidence at the scene. This degree of data-taking cannot be done on every accident and hence no true engineering description of the "zone of deployment" can be obtained.

An inferior but perhaps useful description of the type of accidents that result in ACRS deployment, is a graph of mean or median VDIs and principal direction of force, such as shown in Figure 3.

One would not expect the correlation of deployment with the VDI to be as strong as with ΔV. Both of these data reductions require data on crashes where the bag did not deploy. So it is necessary for NHTSA to investigate a representative sample of ACRS non-deployments.

A third option is merely to summarize those VDIs and θs where ACRS deployment occurred. This could

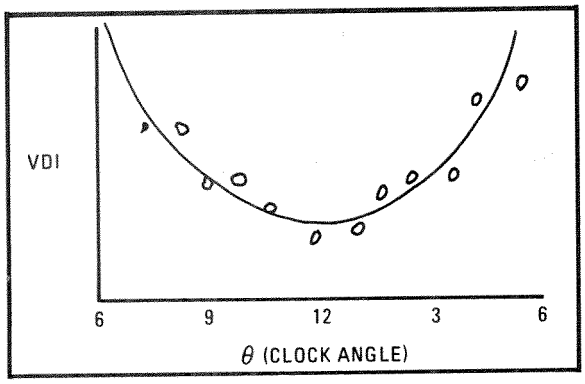
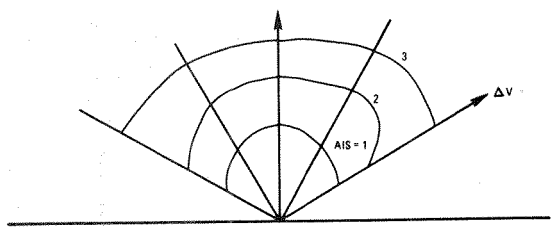


Figure 3. ACRS deployment by VDI and principal direction of force (θ).

be misleading since we could not say whether accidents outside this range would have caused deployment. However, the data might be useful to point out some unexpected deployment modes (e.g., side impacts). This could be done, of course, using the data from the hundreds of deployments that will be investigated.

The Types of Injuries That Result from Crashes Where the ACRS Deploys: Again the ideal is a polar plot showing injury levels (AIS) as a function of ΔV and θ using only deployment data, for example, as shown below:



However ΔVs will not be obtained, so we must depend upon the next best indicator of crash severity, VDI, and consider plots like the ones in Figures 4 and 5.

Again, one would not expect the correlation of injury level with VDI to be as strong as with ΔV.

It is expected that among 460 random deployments there would be at most 240 injuries (.52 x 460), of which only about 80 (.175 x 460) will be moderate or worse, 32 severe or worse, 17 life-threatening or fatal, and 8 fatals. Obviously then, constructing the low-injury level contours should be possible (with up to 100 data points). The moderate injury contours (with up to 20-25 data points) might be possible. Clearly, insufficient data will preclude constructing the higher injury severity contours. This justifies additional investigations of accidents with severe and worse injuries.

So far only the severities of the injuries where deployment occurs has been discussed. Certainly we

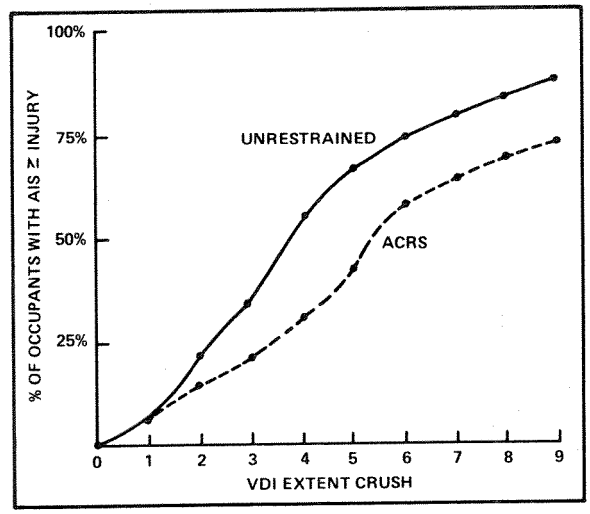


Figure 4. Injury rates and accident severity.

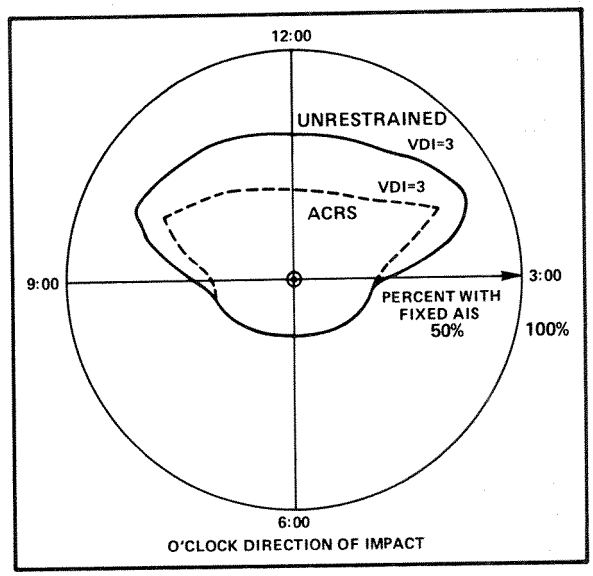


Figure 5. Injury rates and accident severity for a fixed extent of crush.

also would like to know about the nature and mechanism of the injuries.

Lower limb injuries at all levels of accident severity and perhaps some neck injuries in otherwise fatal level crashes can be expected. Fatal injuries will probably be due to ejection, or severe underride/override frontals. Table 2 exemplifies a tabulation summarizing these injuries.

The Tendency of the ACRS to Deploy Inadvertently: What is desired is the inadvertent firings per vehicle mile. It is probable that NHTSA will know of inadvertent firings. It will require more effort to determine the total vehicle mileage accumulated at any point in time. Estimates will probably have to be made based on owner surveys and odometer readings.

Rationale for Objective #3

To evaluate owner/public acceptance we must determine how the general population of new car buyers will react when they purchase a new car equipped with an ACRS.

The method to achieve this would require owner surveys to determine the reaction to the General Motors ACRS. There are problems with this approach in that those people who will purchase the GM luxury class vehicles with the relatively expensive ACRS option are likely not representative of the new car buyer population. For the purpose of this survey, this and other similar potential biases must be regarded as setting upper bounds for favorable reaction. It is simply anticipated that the biases are not significant and that this group of purchasers will be similar to that of the general population of new car buyers. There would be less concern if the ACRS option was available to all potential new car buyers.

The owner survey function is planned to be carried out under contract. The following indicates the type of information that can be obtained.

DATA TO BE OBTAINED FROM OWNER

1. Demographic Information
   - Age
   - Sex
   - Occupation
   - Education
2. Vehicle Data
   - Model
   - Air bag
   - Air bag with lap belt
   - Date of purchase
3. Use of Vehicle
   - Work
   - Pleasure
   - Other
   - Annual mileage (driven by respondant)
   - Mileage at time questionnaire is answered
4. Restraint System (Previously Owned/Driven Car)
   - Year and model of previous car
   - Use of lap and shoulder belts in this car
   - If 1972-73 car, was warning system defeated or circumvented
5. Restraint System (Air Bag Car)
   - How learned of air bag option
   - Why chose air bag car
   - Understanding of system operation
   - Why or why not chose additional lap belt
   - Use of lap belt
   - Reason for use or nonuse of lap belt
   - Modification or tampering with system(s)
   - Satisfaction/security derived from system and basis for such
   - Reaction to system by friends, relatives, etc., when they are driven in car
6. Restraint System Defects (Air Bag Car)
   - Nature of defect
   - How it was discovered
   - Was it repaired, by whom, cost
7. Accident Involvement
   - Has ACRS vehicle been involved in an accident
   - Did air bags deploy
   - Was anyone in ACRS vehicle injured
   - No treatment required, treated and released, hospitalized
   - Major damage was sustained to the front, rear, or sides
   - Was vehicle towed from the scene
   - Repair cost

ANALYSIS OF THE DATA

Objective #1. Injury Reduction

From general considerations, it was concluded that the best course of action was to collect ACRS crash data; to obtain or collect control-group crash data; to stratify each data set into four classes according to vehicle damage; to estimate, for various AIS injury levels, the injury rates for front-seat occupants in test and control groups, for each stratum; to estimate the "true" sizes of each stratum for the "total" vehicle population; to obtain the net injury rates by taking the averages weighted by the "true" sizes of the strata; and to give some confidence range on the percentage injury reduction (or increase) due to ACRS with respect to the other restraint system.

For each crash, the Vehicle Damage Index (VDI), including "damage extent" (measured on a 0-9 scale) and "direction of impact" (measured by "clock direction") will need to be obtained. Also required for each injured front-seat occupant will be a medical report, including the overall injury severity as measured using the Abbreviated Injury Scale (AIS).

Data of these kinds must be collected by technicians, who, in turn, must be notified by the police that an accident has occurred. A realistic notification threshold is that either the car had to be towed from the scene or that an occupant had a disabling injury. Non-injury, non-towaway accidents are more elusive.

Table 3 gives the nationwide totals for the 20-month time-frame from January 1, 1974, to August 31, 1975, for towaway-involved, front-seat occupants of ACRS cars, and for front-seat occupants of the 1973-1975 full-sized GM cars equipped with

TABLE 3

EXPECTED NUMBER OF INJURIES AMONG TOWAWAY-INVOLVED FRONT SEAT OCCUPANTS (20-MONTH PERIOD – FULL SIZED GM CARS – NATIONWIDE)

Cummulative Injury Severity	ACRS Group	Control Groups		
	ACRS 1974-75	Unrestrained 1973-75	Lap/Shoulder 1973-75	Lap 1973
All Towaway-involved front-seat occupants	5,300	220,000	130,000	50,000
All AIS ≥ 1	1,660	68,700	40,150	15,600
All AIS ≥ 2	430	17,600	10,700	4,000
All AIS ≥ 3	140	5,760	3,500	1,300
All AIS ≥ 4	46	1,860	1,100	420
All Fatalities	22	900	500	200

other restraint systems. It also gives the number of injuries more severe than or equal to a given AIS level, that would be expected if none of the restraint systems were effective.

The reader who has some experience in statistical hypothesis-testing will quickly note that nearly the entire national ACRS occupant population would be needed to detect significant injury reduction at the life-threatening or fatal levels. Each of the control groups, on the other hand, is much larger and need only be sampled.

The four damage strata into which the towaway crashes are likely to be subdivided are defined as follows:

1. Major Frontal Impacts: VDI ≥ 3 and 11:00-1:00; or VDI ≥ 4 and 9:00-3:00; or VDI ≥ 4 and rollover.
2. Nonmajor Frontal Impacts: VDI ≤ 2 and 11:00-1:00.
3. Nonmajor Side Impacts and Rollovers: VDI ≤ 3 and 9:00-3:00; or VDI ≤ 3 and rollover.
4. Rear-end Impacts: All towaways with 4:00-8:00 o'clock angles of impact.

It is anticipated that within each stratum, the ACRS and control groups will be closely matched with regard to accident severity.

The strata were defined this way because it is expected that nearly all of the (non-inadvertent) deployments will be concentrated in one stratum. Essentially, the ACRS occupants are "unrestrained" outside Stratum 1 (or, at worst, Strata 1 and 2). Hence, outside these strata, we may replace or supplement the sparse ACRS-Group data with more easily obtained "unrestrained" data. Further, the other strata would, as a result, contribute no variance when comparing with unrestrained. As a result, the precision of our estimates would greatly increase.

Furthermore, this method is, in a sense, self-checking. VDIs were justified in the foregoing rationale because they were believed to be a good surrogate for engineering descriptors of crash severity for vehicles of similar makes and models. Since the sensor threshold is defined in terms of an engineering descriptor (velocity vector change during impact), most deployments will be in Stratum 1 if and only if this rationale is correct.

The following discussions give the formulas to be used to give net injury rates. A definitive measure of the precision of these rates for various sample sizes is under development.

Let t be the total number of front seat occupants of ACRS towaways in the data collection. Let $t_1, \ldots, t_4$ be the number of occupants in each stratum.

Let $t', t'_1, \ldots, t'_4$ be the corresponding numbers of *unrestrained* occupants in the various control groups.

Let $t'', t''_1, \ldots, t''_4$ be the lap-shoulder belted and $t''', t'''_1, \ldots, t'''_4$ be the lap-belted.

Let $T = t + t' + t'' + t'''$ be the total number of front-seat occupants in all of the data.

Let $T_i = t_i + t'_i + t''_i + t'''_i$ be the totals for the ith stratum. T_i/T will be used to give an estimate of the "true" size of the ith stratum.

Let x_i be the number of occupants of ACRS cars who are in the ith stratum and who sustained injuries of at least some specified AIS (e.g., the number with severe or worse injuries). Let x'_i, x''_i, x'''_i be corresponding numbers for control groups.

Then x_i/t is the sample estimate of the injury rate for the ith stratum of ACRS occupants and x'_i/t'_i is the estimated rate for unrestrained occupants.

The net rate for injuries at some AIS level to ACRS occupants in towaways is estimated by $R = (x_1/t_1)(T_1/T) + \ldots (x_4/t_4)(T_4/T)$.

The estimated net rate for injuries to unrestrained front seat occupants is given by $R' = (x'_1/t'_1)(T_i/T) + \ldots (x'_4/t'_4)(T_4/T)$.

If the assumption is correct that most of the deployments fall in Stratum 1, R' may be replaced by $\tilde{R}$. Then:

$$\tilde{R} = (x_1/t_1)(T_i/T) + (x'_2/t'_2)(T_2/T) + (x'_3/t'_3)(T_3/T) + (x'_4/t'_4)(T_4/T)$$

Now, $\tilde{R}$ and R' have a large covariance, and this will improve the precision of $(R' - \tilde{R})/R'$, i.e., the estimate for the net injury due to ACRS.

Improved Sampling Plan

There appears to be a method for greatly improving statistical efficiency – i.e., getting the same degree of precision with a much smaller number of investigations. In the first approach collection of many thousands of noninjury towaways is only for the purpose of seeing which stratum they were in – i.e., finding $t_1, \ldots, t_4; t'_1, \ldots, t'_4$; etc. This is wasteful of data. In fact, after looking at a random sample of only 25-50 percent of the noninjury towaways in ACRS and each control group one may estimate with great precision the distribution of the remaining

noninjury towaways among the 4 injury causing strata. Hence, one need only keep a tally of all noninjury towaways and extrapolate their distribution among the strata from the small sample. The disabling injuries, of which there are few, already have to be investigated to find the AIS. It is necessary also to collect VDIs for all of them; there are not enough of them so that they can be precisely distributed among the strata on the basis of a less than 100% sample.

The more efficient sampling plan, then, consists of:

1. Keeping a tally of all towaway crash involved front-seat occupants in ACRS and control groups, and collecting police accident reports.
2. Get the AIS and the vehicle VDI for all occupants for whom the police report stated that they were taken to a hospital – this will include most AIS > 2 injured.
3. Get the VDIs for a random sample of crashed vehicles for which no occupants required treatment. The appropriate sample sizes are now being determined by NHTSA.

As in the first approach, the net injury rates R and R' are calculated and $(R' - \tilde{R})/R'$ is estimated. The only difference is that t_i, t_i', and T_i are algebraically calculated from a smaller sample, and are subject to more variance than before. Sample sizes will be chosen to optimize the balance of the variance of the t's against the variance of the x's.

The control groups can be obtained by using existing data that classifies VDI and AIS, or by collecting new data. Each control group should be somewhat larger than the ACRS group. In order to obtain the precision desired, we need an ignition interlock crash involvement that will produce 7,500 towaways in the 20-month study period and a 1973 vehicle crash involvement which will have a like number. Fleets of that size will also give us slightly more lap/shoulder belted front seat occupants than ACRS occupants, and twice as many unrestrained occupants as in ACRS vehicles. As indicated in the rationale for Objective #1, these cars should be of the same makes and models as the ACRS cars, i.e., they should be full-sized GM cars, preferably Buicks, Oldsmobiles, and Cadillacs.

Neither Calspan intermediate level data nor the Multi-disciplinary Accident Investigation files which are available have anywhere near the sufficient number of crashes. Motors Insurance Corporation file, which is at this time proprietary to GM, is unusable because of missing data. The VDIs are not coded for noninjury accidents, so it is impossible to determine which of the damage strata they belong to.

Therefore, it will be necessary for NHTSA to collect the needed control group data.

The control group must obviously be collected in some pre-defined areas of the country where either NHTSA or MVMA* accident investigation teams are presently located and where all moderate to fatal injury producing crashes in those regions could be investigated. A systematic sample of the "minor" and "no injury" towaways would also be collected. As for the ACRS group, every moderate to fatal ACRS injury in the country needs to be collected because of the smallness of the population. With regard to the "minor" to "no injury" ACRS towaway crashes, there are two approaches to collecting the sample towaways not involving occupants brought to a medical facility:

1. Collect every ACRS towaway in areas where control group data is collected,** and
2. Collect the towaways across the nation by a systematic sampling procedure.

Approach #1 appears to involve less travel and thus to be less costly, but it is fraught with difficulties:

- It presumes that teams must cover areas large enough to contain the required sample of ACRS towaways.
- The distribution of accidents among the damage strata in the areas may be unrepresentative of the nation. The only clue as to whether they are representative is to compare the percentage of the area towaways resulting in deployment to the national percentage. This clue is worthless if the assumption were incorrect that the deployment population closely resembles Stratum 1 (major frontals). Therefore, Approach (2) is highly recommended.

DATA COLLECTION

Two distinct data collection systems are employed to cover both ACRS equipped vehicles involved in crashes and control group crashes. ACRS data are collected nationally by Multidisciplinary Accident Investigation Teams (MDAI) operating in five regions around the country. Control group crashes are also investigated by the five MDAI teams but in selected counties in each of the team regions.

Notification of the occurrence of a crash involving an ACRS equipped vehicle and the initiation of an investigation is based on a 24-hour-per-day, 7-days-a-week operation of a National Response Center (NRC) located at DOT Headquarters in

*Motor Vehicle Manufacturers Association

**Currently, and until more ACRS cars are on the roads, all possible ACRS towaways are being investigated.

Washington, D.C. The control group sampling system depends heavily upon the liaison that has already been established between the MDAI teams and the police agencies in their counties for notification of crashes involving acceptable control group vehicles.

Through the help of the NHTSA regional administrators and the respective governors' highway safety representatives in their regions, it is expected the ACRS evaluation program will receive wide publicity and cooperation. In addition, an explanatory letter containing the NRC toll free telephone number 800/424-8802 was mailed out to all police jurisdictions having a population of 15,000 or greater requesting that the NRC be notified of all traffic crashes in their area involving an ACRS equipped vehicle. General Motors has also requested owners and dealers to provide information. Once the NRC has been contacted, a member of the NHTSA's Accident Investigation staff makes the decision as to what depth the accident will be investigated. The level of investigation is determined by accident severity, whether it involved towing the vehicle from scene or not as well as injury severity based on the Abbreviated Injury Scale (AIS) developed under the auspices of the American Medical Association.

In the case of crashes involving a 1973-1975 GM full size car (control group) the procedure is basically the same except the NRC is not notified. Notification and response are accomplished at the regional level. Local and State police operating in the selected counties will notify the team whenever a crash involving a vehicle meeting control group criteria occurs. If preliminary information provided to the MDAI team indicates that an occupant was taken to a treatment facility, a technician is dispatched to carry out an intermediate level type of investigation. Intermediate level investigations provide police and medical reports plus photographs of the vehicle to provide for developing a VDI.

When the crash did not require an occupant to be transported to a treatment facility, the MDAI staff will apply the systematic sampling procedure specified by NHTSA. If the procedure selects the crash, an intermediate level investigation will be made. The sampling fractions will be determined by the optimization procedure mentioned in "Analysis of the Data." There will probably be different fractions for buzzer and for ignition interlock cars.

REFERENCES

1. J.D. States, H.A. Fenner, Jr., E.E. Flamboe, et al., "Field Application and Research Development of the Abbreviated Injury Scale," SAE Print 710783, New York. Society of Automotive Engineers, 1971.
2. Committee on Medical Aspects of Automotive Safety: "Rating the Severity of Tissue Damage I. The Abbreviated Injury Scale," Journal of American Medical Association, 215:277-280, 1971.
3. Irwin D.J. Bross, "How to Use Ridit Analysis," Biometrics, March, 1958.
4. Collision Deformation Classification – SAE J224a – SAE Recommended Practice, New York: Society of Automotive Engineers, 1972.

SYNTHESIS OF STATISTICAL DATA ON TRAFFIC ACCIDENTS IN FRANCE, WEST GERMANY, ITALY AND UNITED KINGDOM

F. HARTEMANN and C. TARRIERE*
Association Peugeot-Renault
* Associated with Biomecanic-Accidentology and Crashworthiness Groups (C.C.M.C.) in carrying out this report.

A statistical balance-sheet of traffic accidents should correspond to a classification of accidents by categories established with carefully selected parameters.

The choice of these parameters only depends on the nature of questions propounded, to which statistics are wished to answer.

In the present communication, accidents were described and gathered so as to give the first answers available in France, Germany, Italy and United Kingdom to the following synthetic questions:

- How many killed and injured people would be saved, should such safety measures be adopted, capable of such performances under such impact conditions?
- What characteristics should the "equivalent" tests have so as to be very representative and to avoid either too little or too much severity, which in either case might make false the choice of countermeasures, delay their adoption or alter the benefit/cost ratio?

We refered to official documents published by National Departments of Transportation, Insurance Companies publications (particularly H.U.K., Germany) and data collected by bi-disciplinary accident investigation teams (University of Birmingham, PEUGEOT-RENAULT Association, FIAT, the teams of which analysed 3.000 cases, VOLKSWAGEN and DAIMLER-BENZ teams).

Because of numerous divergences between the different sources regarding criteria used in impacts classification, we have retained in the comparative classifications only the most homogeneous data, at the risk of not showing interesting but isolated informations: that is to say pertaining to one source or another, but statistical reach of which looked insufficient.

1. DESCRIPTION OF ACCIDENTS BY USERS CATEGORIES AND TYPES OF OBSTACLES

To complete the informations given by the figures I to IV, we can notice that the rate of killed occupants in cars compared with injured occupants are:

1 for 38 in GERMANY,
1 for 23 in FRANCE,
1 for 28 in ITALY,
1 for 55 in U.K.

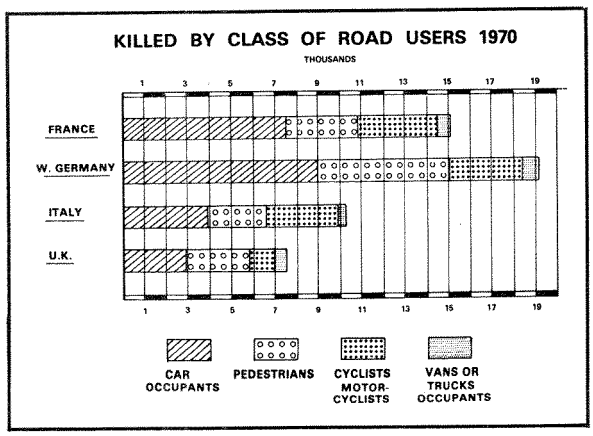


Figure 1

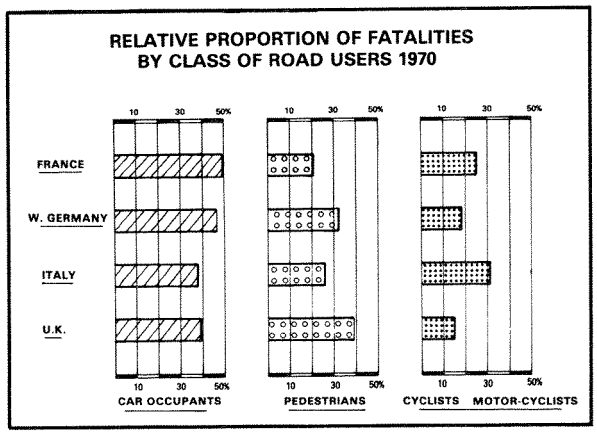
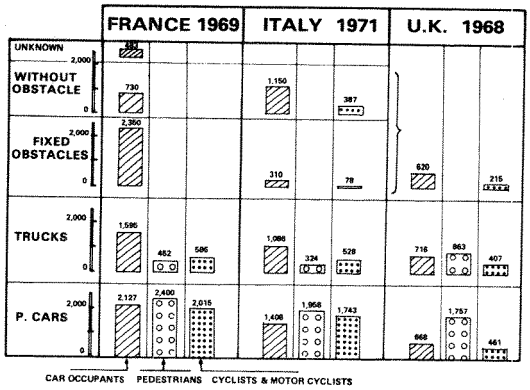
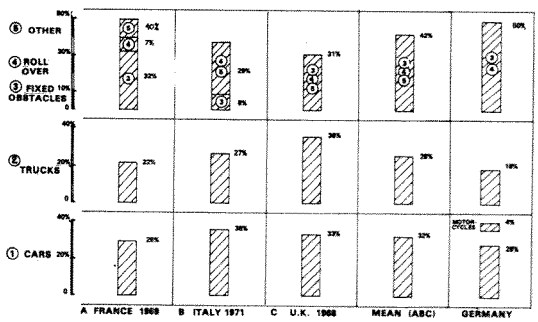


Figure 2



DISTRIBUTION OF CAR OCCUPANTS PEDESTRIANS, CYCLISTS, MOTOR-CYCLISTS KILLED BY TYPE OF COLLISION

Figure 3



RELATIVE PROPORTION OF CAR OCCUPANTS FATALITIES BY TYPE OF COLLISION

Figure 4

On the other hand, cyclists, motorcyclists and pedestrians represent more than 50 % of road users killed in these countries.

In France, Italy and U.K. (1), in 1970, 32 % of the car occupants were killed in a collision against another car, 26 % were killed in a collision against a truck and 42 % died in an accident against fixed obstacle or without any obstacle.

The severity of the collisions for car occupants differs greatly depending the type of obstacle. Table I shows that, in case of collision against truck, the risk of being killed is 5 times higher than against another car, and, in the case of a collision against fixed obstacle, this risk is nearly 7 times higher.

TABLE I	OBSTACLES		
	Fixed obstacles	Truck	Cars
Killed	7.4	5.4	1.1
Severe injured	29.0	21.8	10.0
All injured	100.0	100.0	100.0

Car agressivity against other road users (other cars occupants, pedestrians, cyclists and motorcyclists) can be characterized as follows:

TABLE II	Car occupants	Cyclists & Motor.	Pedest.	Total
Killed by car impact	4,203 29 %	4,219 29 %	6,115 42 %	14,537 100 %

It is evident that cyclists, motorcyclists and pedestrians must be protected in priority from car agressivity.

2. DESCRIPTION OF CARS COLLISIONS

The description of frontal, lateral, rear impacts and rollovers is rather similar from one country to another (Figure V). But we have to take care not to consider the rate of each type of impact as absolutely accurate. In each country, this rate varies according to the mean severity of the analysed sample and to the type of collisions. So the percentages mentioned below, calculated from other samples, are quite different of those on Figure V.

	Frontal collisions
• U.K. (Birmingham sample) 59 killed occupants	49.9 %
• GERMANY (H.U.K.) all accidents car to car	52.5 %
• FRANCE 257 killed (front seat occupants)	51.9 %

The analysis of a sample representative of European accidents, as currently conducted by the C.C.M.C. should allow us to work out more accurate percentages by degrees of seriousness for lesions associated with each type of impact.

The relative importance of rear impacts in Italy is probably due to a more important highway traffic than in France and U.K.; and the proportion of rear impacts in Germany is not, in this respect, as high we could expect.

(1) The statistics established by H.U.K. in Germany are not always comparable with those of other countries because the investigation consists of cases of motor vehicles third party liability insurance, in such a way that only collisions between two vehicles are studied.

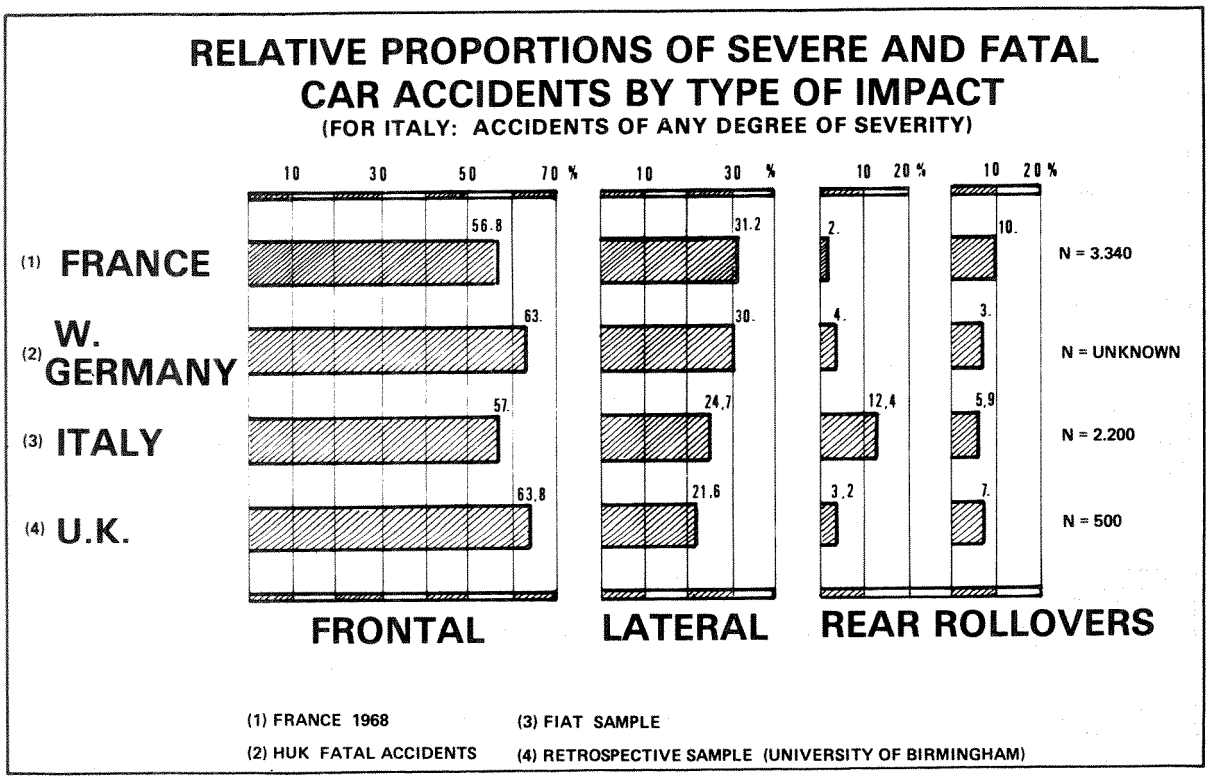


Figure 5

2.1. Frontal impacts

a) Location of deformation and impact direction

The axial impact (Fig. VI) amounts to 60 to 70 % of the frontal impacts. In most cases (24 to 41 % of the frontal impacts), it is offset. Axial impact with distributed deformation occurs in only 19 % of the frontal impacts.

The accidents of any degree of severity involving a front-right or front-left two-third deformation are very frequent (Fig. VII a, VII b, VII c, VII d). This type of impact represents 53 % of the frontal impact involving severe or fatal injuries in Italian and French samples (Fig. VIII).

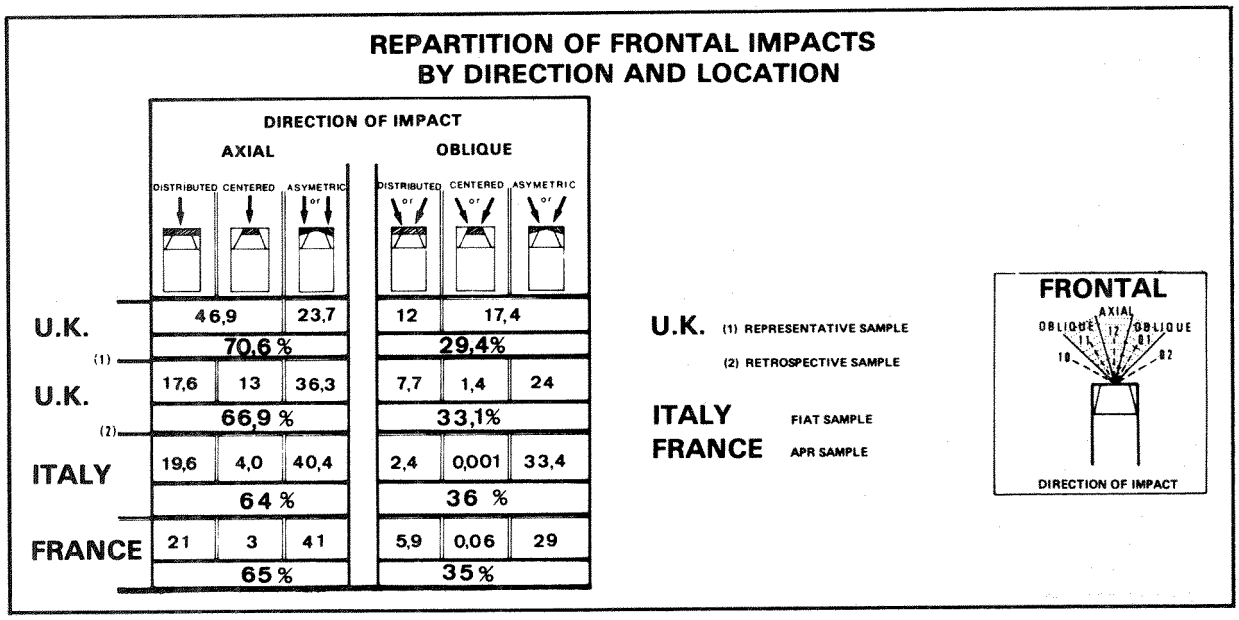


Figure 6

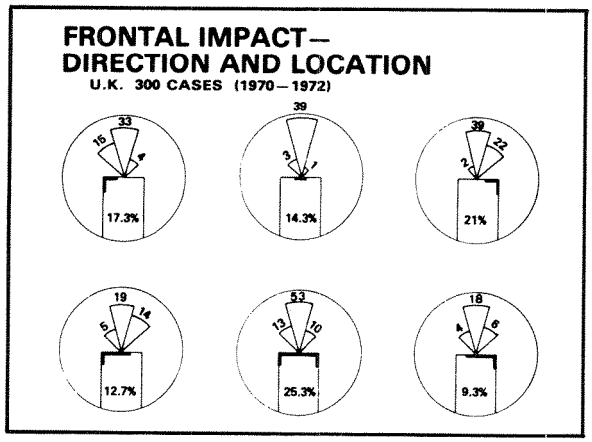


Figure 7(a)

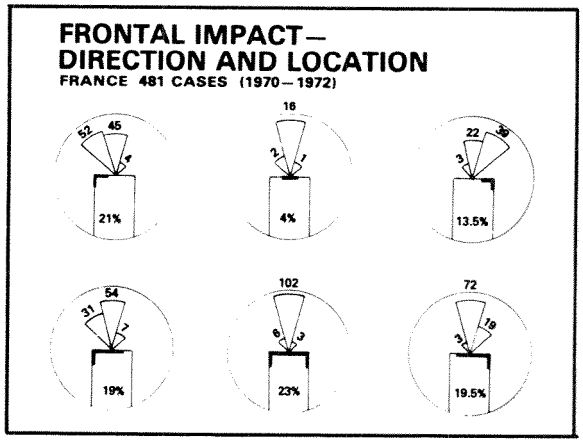


Figure 7(b)

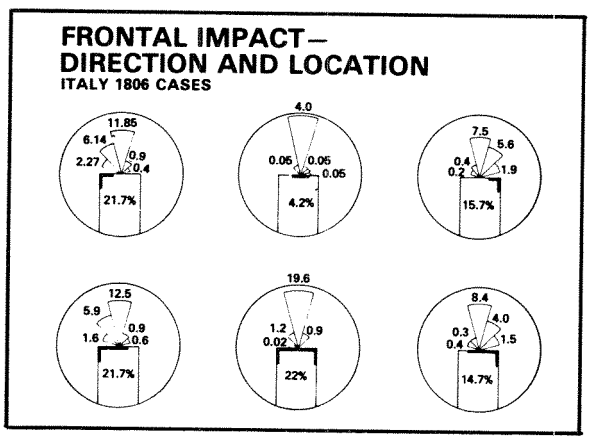


Figure 7(c)

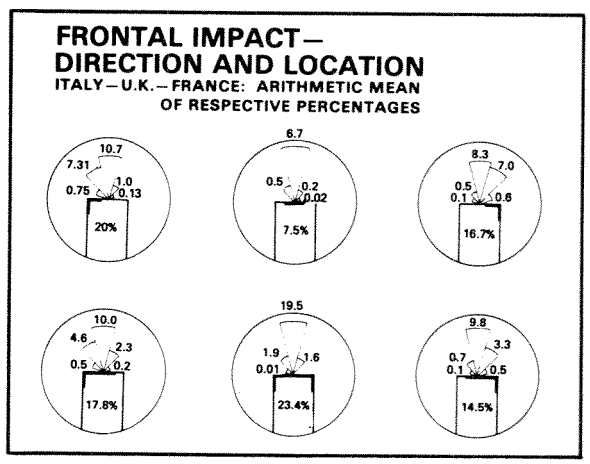


Figure 7(d)

b) Violence of frontal impacts

ΔV - Meaning

ΔV means the speed variation sustained by any undistorted portion of the vehicle during force application at impact interface.

Its calculation mode, which takes into account, among others, the ratio of confronted masses, allows a more accurate estimate of the true speed variation than the method used heretofore, which only definies the speed by comparaison with the wall impact speed required in order to achieve the same distortion.

ΔV and impact violence — unrestrained occupants.

Whenever occupants are not restrained, they impact vehicle inner structure at a speed substantially equivalent to that shown by ΔV for them, ΔV is a pertinent parameter of impact violence. Under such circumstances, the curve shown on Fig. IX, which describes the statistical distribution ΔV's observed in real accident also describes impact violence distribution for unbelted occupants.

ΔV and standardized impact tests : restrained occupants

The generalization of devices coupling the occupant to the vehicle creates fully different impact conditions: impact violence no longer depends on ΔV but also on the distance within which speed variation takes place (or the duration of such speed variation).

It would therefore be necessary, in order to define standardized tests, also to define impact conditions such that they reproduce a deceleration law near to the one observed in real accidents involving restrained occupants.

The speed, or ΔV, whose choice is a function of the reduction percentage of the number of victims aimed at (the curve in Fig. IX provides an indication thereof) makes up one parameter there should be associated therewith an extra parameter (e.g. impact duration) which is absolutely necessary in order to secure impact conditions conveniently simulating conditions under which bodily lesions occur in real accidents. Fig. IX b shows that the statistically very frequent impact (C) is over-estimated in a simulation by perpendicular impact against rigid barrier: the

reproduction of ΔV (50 km/h) does not however simulate the real violence conditions. A 30° barrier impact (B) makes up a better simulation.

Frontal collision violence

At least until we get news elements given by the next exploitation of 4,000 cases analysed by C.C.M.C., we may consider that the distribution of speed variation suffered by occupants during impact can be in the following order (Fig. IX) (*).

For any degrees of injuries severity, Speed variation (ΔV) Cumulative %	20-22 km/h 50 %	30-34 km/h 75 %	43-48 km/h 90 %	
For severe and fatal injuries, Speed Variation (ΔV) Cumulative %	44 km/h 50 %	50 km/h 65 %	60 km/h 82 %	65 km/h 90 %

FRONTAL IMPACTS

SEVERE AND FATAL CASES — LOCATION AND DIRECTION

		ITALY (480 cases)		FRANCE (80 cases)		MEAN (of percentage)	
LEFT OR RIGHT	(10)	1.8		—		0.9	
	11	4.4	22.7	8.7	11.4	6.6	17.6
	12	13.7		1.2		7.5	
	01	4.6		2.5		3.5	
	(02)	1.2		—		0.6	
LEFT OR RIGHT	(10)	2.5		—		1.2	
	11	10.6	43.4	26.1	57.1	18.3	50.2
	12	26.6		22.3		24.5	
	01	6.2		8.7		7.4	
	(02)	2.9		—		1.5	

		ITALY (480 cases)		FRANCE (80 cases)		MEAN (of percentage)	
	(10)	0.8		—		0.4	
	11	1.0	20.0	—	24.9	0.5	22.5
	12	18.0		24.9		21.5	
	01	1.0		—		0.5	
	(02)	—		—		—	
	(10)	—		—		—	
	11	—		—		—	
	12	3.3	3.5	—	4.9	1.6	3
	01	0.2		4.9		2.5	
	(02)	0.2		—		0.1	
		100. %		100. %		100. %	

Figure 8

FRONTAL IMPACTS

SPEED VARIATION DISTRIBUTION (ΔV) - UNBELTED OCCUPANTS

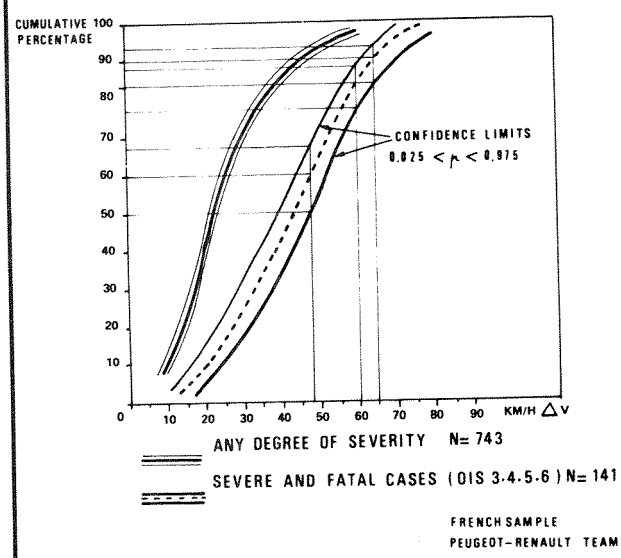
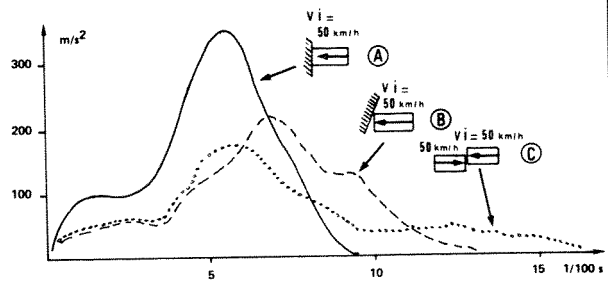


Figure 9(a)

ILLUSTRATION OF OVER-ESTIMATING THE VIOLENCE OF THE MOST FREQUENT COLLISIONS (TYPE B C) THROUGH THE SO CALLED "EQUIVALENT BARRIER TEST A".



IN CASES B AND C ΔV IS CLOSE TO 50 km/h, AS IS IMPACT SPEED (Vi). THE CURVES CLEARLY SHOW HOW MUCH THE VIOLENCE OF B - C IMPACTS IS OVER-ESTIMATED WHEN SIMULATED BY AN A TYPE IMPACT, i.e. 50 km/h TEST AGAINST BARRIER.

Figure 9(b)

(*) Notice the signification of confidence limits on the Fig. IX: 0.0250 is the probability of making error by asserting that the real ΔV (according to a given percentage) is superior to the left line; 0.975 is the probability of being right by asserting that the real ΔV is inferior to the right line.

c) Violence, direction of impact and deformed areas

On Figure X, the distribution of cases is represented according to the speed variation (ΔV) for each type of severe and fatal impacts.

FRONTAL IMPACTS

DISTRIBUTION (in %) OF SEVERE AND FATAL CASES BY LOCATION, DIRECTION AND ΔV

		ΔV (km/h)	<45	46 55	56 65	66 75	75 85>
LEFT		11	2.5	1.2	5.0		
		12		1.2			
		01					
RIGHT		11					
		12					
		01	2.5				
LEFT		11	5.0	2.5	12.5	3.7	
		12	6.2	5.0	1.2		
		01					
RIGHT		11	1.2	1.2			
		12	2.5	3.7	2.5	1.2	
		01	3.7	5.0			

	ΔV (km/h)	<45	46 55	56 65	66 75	75 85>
	11					
	12	8.7	7.5	5.0		3.7
	01					
	11					
	12	1.2	1.2	2.5		
	01					
FRENCH SAMPLE (80 cases)	Σ =	33.8	28.8	28.8	4.9	3.7

Figure 10

This is the way available statistical data must be synthetized in order to objectively characterize the conditions wherunder the equivalent tests must be conducted. The only missing parameter is deceleration distance. It is calculated on γ mean value, which we recently checked to be relatively easy to determine in real accidents provided the following factors are known: masses equivalent speeds and closing distance between gravity centers.

2.2. Lateral Impacts

There is a small number of lateral impacts against pole or tree (10 %). The most frequent are car to car. The French, Italian and U.K. samples are comparable concerning the perpendicular and angled impacts proportions. But we observed important differences as to the relative frequency of deformations at the level of the passengers compartment. These differences are undoubtedly, due to differences of injuries average severity registered in each sample (Fig. XI a, b).

The higher the average severity, the more important the proportion of lateral impacts deforming the passengers compartment.

REPARTITION OF LATERAL IMPACTS BY DIRECTION

	DIRECTION OF FORCES: PERPENDICULAR			OBLIQUE		
	PASSENGER COMPARTMENT	FRONT OR BACK	TOTAL	PASSENGER COMPARTMENT	FRONT OR BACK	TOTAL
U.K (189 cases)	28 %	6 %	34 %	39 %	27 %	27 %
FRANCE (72 cases)	22,4 %	5 %	27,4 %	59 %	13,6 %	72,6 %
ITALY (708 cases)			24 %			76 %

Figure 11(a)

REPARTITION OF LATERAL IMPACTS BY LOCATION OF DEFORMATIONS

	LOCATION OF DEFORMATIONS: PASSENGER COMPARTMENT	FRONT OR BACK
U.K (1)	67 %	33 %
U.K (2)	47 %	53 %
ITALY	38,8 %	61,2 %
FRANCE	81 %	19 %

U.K (1)	RETROSPECTIVE SAMPLE	189 cases
U.K (2)	REPRESENTATIVE SAMPLE	127 cases
FRANCE	A.P.R SAMPLE	72 cases
ITALY	FIAT SAMPLE	708 cases

Figure 11(b)

If the lateral impact occurs at high speed, the relative movements of cars are such that even when the car is impacted at the front wheel level, its passengers compartment is then liable of deformation.

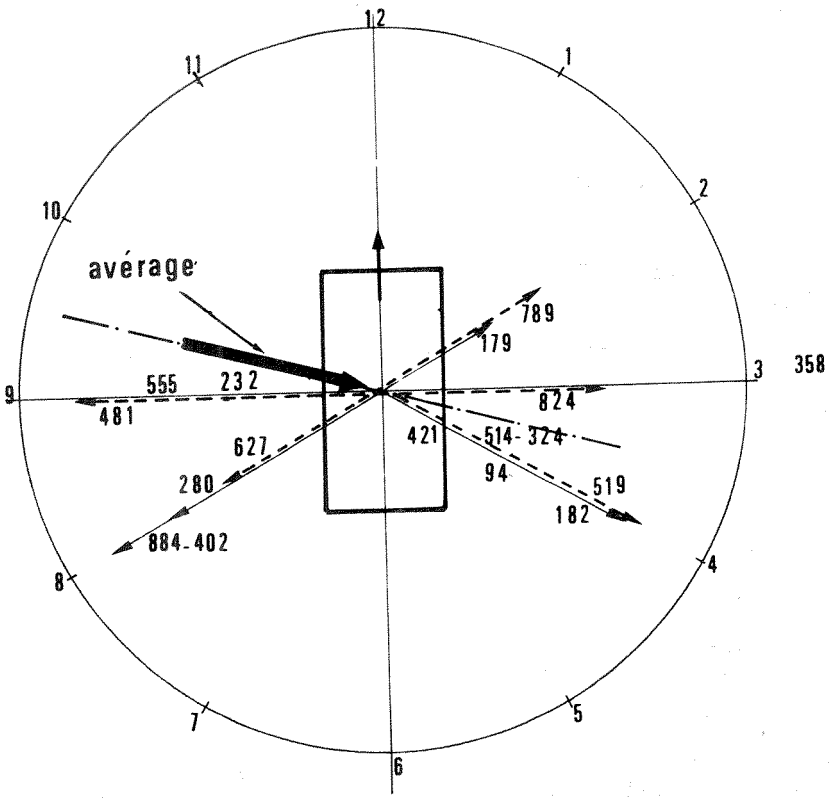
Occupants trajectory

A synthesis of lateral collision severe cases is represented in Figure XII which outlines the average direction of deceleration forces which are sustained by occupants (supposed to seat on the left side for simplication).

Distribution of speed variations

The distribution illustrated exclusively concerns vehicles impacted in vehicle-vehicle collisions. There should be noted the relationship between ΔV and impact speed as a function of masses confronted with each other (right hand side of figure) and it should also be noted that it is necessary to treat lateral fixed barrier collisions separately, since the test procedures to be defined for simulation are quite different from those which reproduce impacts of two vehicles (Fig. XIII).

LATERALLY IMPACTED CARS WITH KILLED AND MAJOR INJURIED OCCUPANTS



ESTIMATED

Case Vehicle	ΔV	ΔV x	ΔV y	
884	50	44	-25	KILLED
232	30	30	/	
179	20	17,5	+10	
421	20	17,5	-10	
402	50	44	-25	
358	75	75	/	
280	40	35	-20	
182	45	39	-22,5	
94	30	26	-15	
555	40	40	/	
Averages		36,8	-10,75	
824	35	35	/	INJURED
514	25	22	-12	
324	25	22	-12	
481	50	50	/	
519	45	39	-22	
627	30	26,5	-14,5	
789	30	26	+16	
Averages		27,58	-6,81	
Averages		32,2	-8,78	

$\longrightarrow$ means $\Delta V . \frac{\vec{Vr}}{|\vec{Vr}|}$

ΔV : Variation of speed for impacted vehicle

$\vec{Vr}$: Relative speed of the two vehicles at impact

Figure 12

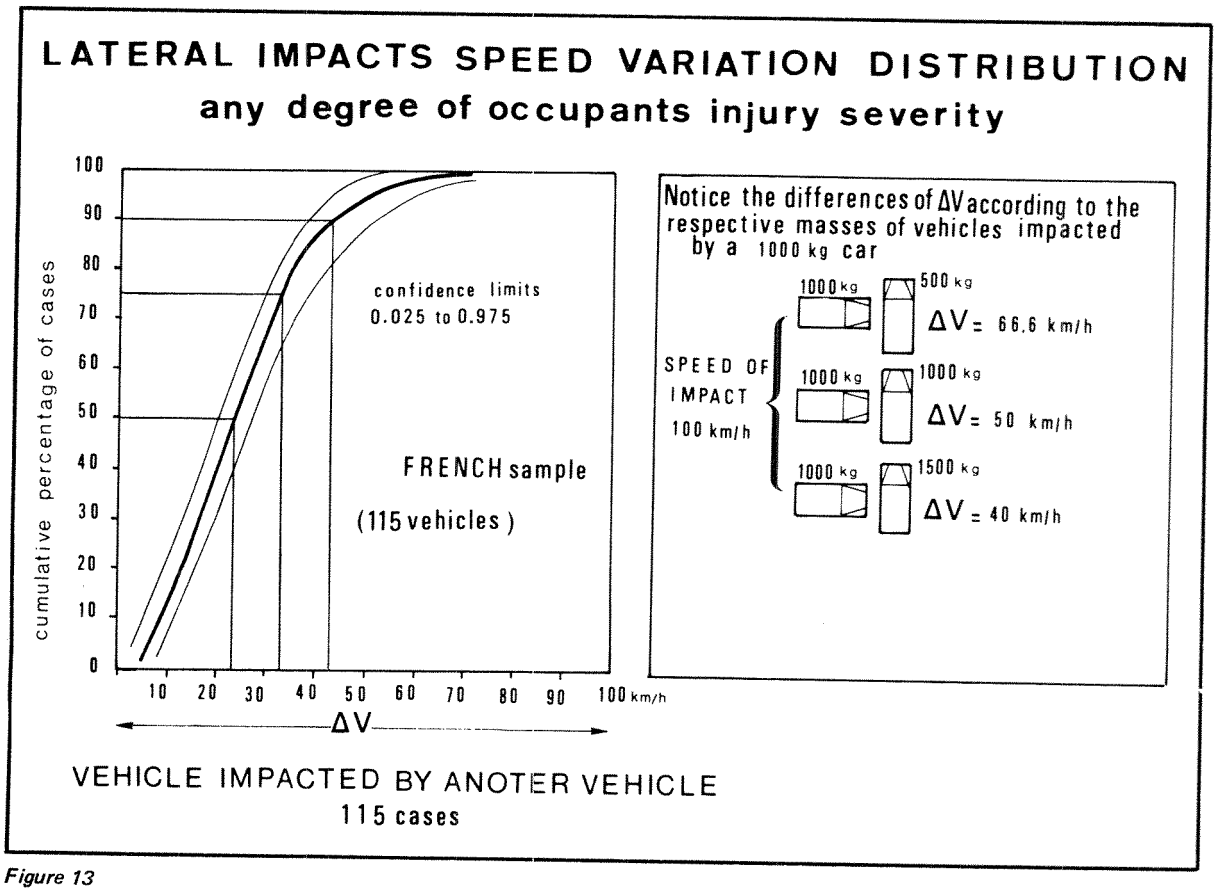


Figure 13

2.3. Rear impacts and rollovers

Rear impacts

The very low severity of rear impacts justify to describe them only briefly.

Rear impact, in U.K. (Birmingham sample) represents 10 % of the representative sample accidents and 3,5 % of the retrospective severe accidents sample.

In France, it causes death of 3 % of motorists.

Rear centered impact, simulated by impact against pole occurs in 6 % of the rear impacts. Damages are distributed on the whole rear of the vehicle in 30 or 40 % of cases.

At least 50 % of impacts of any degree of severity for the occupants occur at a speed inferior to 20 km/h and 75 % inferior to 48 km/h.

Rollovers

It is particularly difficult to appreciate exactly rate and severity of the rollovers because most of the statistics do not show whether the resulting injuries are caused by the rollover or possible impact (very frequent) that occurs before.

In France, we consider that rollover was directly responsible in 1969 and 1970 for 7 or 8 % car occupants fatalities.

We must add that we attribute the rollover severity to the eventual ejection.

It is a rare case when the occupant projection violence against pillars, roof, etc. (inside the vehicle) is great to the extent that it leads to very serious lesions; and if there is, sometimes, association of serious lesions with a substantial sinking of the roof, it comes from the fact that the relative speed of occupant relatively to the roof is high, and that the latter is rigidified on account of its hearing on a fixed obstacle (shaft, post or other).

2.4. Efficiency of safety belts

As before pointed out by C.C.M.C. on the fourth conference E.S.V. in Kyoto, the probability of being seriously injured or killed in frontal impacts is six time lower for belted occupants than for unbelted ones in case of accident with ΔV lower than 55 km/h, that is to say for 94 % of the Whole frontal impacts.

For the 116 occupants involved in accidents with $\Delta V > 45$ km/h (the 10 % most severe impacts of the French sample, Peugeot-Renault), the proportion of killed is as following:

	$\Delta V > 45$ km/h (Frontal Impacts)	
(1) Involved Car Occupants	(2) Killed	(2) 1 × 100
Belted : 37	7	18.9
Unbelted : 79	32	40.5

As the above table shows, the probability of being killed for belted occupants is twice smaller than for unbelted people, although the violence is very high.

For a lot of cases, fatal or severe injuries observed on belted occupants do not indicate the safety belt efficiency limit for a given violence, but rather reflect certain technological or practical inadaptation to the safety belt, such as deceleration space reduction due to intrusion, belted occupant on front seat hit by a rear passenger, slack worn belt and so on... (reference X).

By avoiding the extraneous factors quoted above, allowing safety belt to work with its optimum efficiency, we must undoubtedly increase the belted saved occupant percentage.

CONCLUSIONS

1. Proportion of pedestrians, cyclists and motorcyclists victims of fatal accidents is currently over 50 % of the killed total. We can expect that measures reducing accident severity for the occupants, such as speed limits, fixed obstacles protection, passive safety measures, etc., will considerably increase the relative number of people killed outside cars. The cost/benefit studies should take that into account as far as possible.
2. Car agressivity must be reduced to protect pedestrians by priority.
3. At present time, we have to improve lateral impact studies. It is likely that we could release some ideas about counter measures benefits of which would be greater than those we can expect from improved collision performance above 60 km/h; in frontal impact for example.
4. Number and severity of collisions against heavy trucks suggests studying efficient solutions reducing the damages resulting from underriding the truck overhanging structure. These solutions may be less expensive if they consist to reduce heavy truck agressivity rather than to improve cars resistance.
5. It would be useful that all the national organisms (Transportation Departments, Insurance Companies, etc.) who establish accident statistics, adopt a common method of injuries and accidents description and classification in every European country. In this way, a worldwide cooperation (O.C.D.E., C.C.M.S., E.E.V.C...) must be achieved.

When describing accidents, it is essential not to neglect any of the data which can be collected rather easily such as rollovers, ejections, designation of impacted obstacle, for example, and which are absolutely necessary to make a choice among safety measures as well as to control the efficiency of the latter on running vehicles.

These improvements are no replacement for detailed bidisciplinary analysis, but they would allow us to better evaluate the relative importance of the various problems that these analysis allow to raise.

6. The definition of the so called "equivalent tests" preceeded the accurate, physical and statistical description of impact conditions which the motorists undergo in accidents. This advance was justified: safety measures had to be taken without delay. Now one must take care that the test conditions are well fitted to a reality which we know how to describe better and better. If one forgets that the tests are to simulate this reality while including some parameters liable to arise in the the future, on account of vehicle technological evolution and traffic condition changes, one runs the risk to give a bad orientation to the safety measures, to work toward solutions which could prove disappointing, or to brush aside some of them wrongly: this is the risk of a scientific and economic squandering.

BIBLIOGRAPHY

1. S.J. ASHTON, J.L.G. HARDY and G.M. MACKAY : " The use of the vehicle deformation index and collision speed assessments ". Accident Investigation Workshop. N.A.T.O.-C.C.M.S. Brussells, June 1973.
2. S.J. ASHTON & G.M. MACKAY : Characteristics of car accidents. A draft. July 1973.
3. B. BAYLATRY & C. BERLIOZ : Distribution et gravité des collisions en fonction de la partie heurtée du véhicule et de l'obstacle heurté. Second International Technical conference on ESV. Stuttgart, octobre 1971.
4. M. DANNER & K. LANGWIEDER : Verletzungen von Personenwagen Insassen bei Unfällen und deren Bezug auf äquivalente Testgeschwindigkeiten. Automobiltechnische Zeitschrift, June 1973.
5. FIAT : Analysis of road accidents in Italy. Third International Technical Conference on ESV. Washington D.C., June 1972.
6. J.L.G. HARDY & G.M. MACKAY : Field accident damage as a basis for crash tests. International Conference on the Biokinetics of Impact. Amsterdam, June 1973.
7. K. LANGWIEDER : Passenger injuries in collisions and their relation to General Speed Seals. Proceedings of Seventeenth STAPP Car crash Conference. 1973.
8. G.M. MACKAY : Field studies of traffic accidents in Europe. Forth International Technical Conference on ESV. Kyoto, March 1973.
9. C. PROST-DAME : Rating accident severities For Occupants. Kyoto. March 1973.
10. C. TARRIÈRE : Efficacité des ceintures 3 points en accidents réels. (C.C.M.C.) Kyoto, mars 1973.
11. C. TARRIÈRE et F. HARTEMANN : Comment meurt-on dans un accident de voiture? International Conference on the Biokinetics of Impacts. Amsterdam, June 1973.
12. P. VENTRE : Proposal for test evaluation of compatibility between very different passenger cars. Kyoto, March 1973.

THE ROAD SAFETY CHECKLIST—A PERIODICAL INQUIRY REGARDING TRAFFIC

J. C. BLUET
Organisme National de la Securite Routiere
ONSER

SUMMARY

The purpose of the inquiry regarding traffic is to gather qualitative and quantitative information as to the behavior of the drivers, so as to relate them to the characteristics of those involved in deaths and bodily injuries and thus to appraise the criteria of exposure to the risk.

We illustrate this method of comparison by assessment of the effectiveness of safety seat belts.

It is found that in 1972, in the absence of a regulation requiring use of the belt and of any general speed limit, the risk of being involved in a death or bodily injury was greater if the person used his seat belt than if he did not; but that since use of the belt has been made mandatory, the risk has been reduced by nearly two-thirds. This relative risk is now approximately 0.6.

This is attributable to the fact that use of a seat belt is reflected in an initial phase (optional use), from the behavioral standpoint, in an assumption of risk which may exceed the norm, and is reflected in a second phase (mandatory use) in a conversion of involvement in death or bodily injury into involvement in property damage; as these two factors have contrary effects, it is normal to observe such a variation in the rate of risk over two years.

Taking this rate of risk into account in the calculation of effectiveness, there results, with reference to country roads, an estimation of the combined effectiveness of the speed limit and of the use of seat belts of 63 percent as regards deaths; 36 percent as regards injuries; and 38 percent overall.

PURPOSE

Road Safety Checklist means an aggregate of data relative to accidents on the one hand and to traffic on the other, regularly gathered and processed expeditiously. Comparison of these two types of data should meet the following goals:

- Provide check of decisions on the subject of road safety, by observation of the behavior of the users and the consequences thereof, and thus testing the effectiveness of the steps taken.
- To detect and assess abnormal risks assignable to any infrastructural, driver or vehicle categories, as well as the trend thereof in time.

The primary official data relative to safety, systematically gathered and used in France annually, have to date been:

- The circustances of accidents involving death or bodily injury and the location thereof as found by the police in connection with reports.
- The average daily traffic on national roads and motorways recorded by standing counts (some 900 points) and by spot counts.

The data are analyzed by the Roads and Motorways Technical Studies Service (SETRA) of the Roads and Road Traffic Department.

The processing of accidents is handled on the basis of forms transmitted through the Departmental Administrations; it accordingly proves to be quite burdensome and involves substantial distribution time lags (in the range of ten months). Only a summary analysis of accidents and traffic is published monthly.

But analysis of the safety trend requires comparison of homogeneous data with an adequate degree of acuity; the trend in the number of accidents of a certain kind is inadequate to assess the trend in safety; it is necessary to be enabled to relate it to that in the number of users exposed to the risk of accident of that kind. This is why ONSER has made surveys of roads and drivers, in order to assess the exposure to risk relative to various categories of drivers. In this way it has been possible to assess the relative risks of accident in relation to the driver's alcoholemia and to vehicle age, brand and model; for example, in two surveys of 7,500 drivers made at random in 1969 and 1970, it was found that 3 percent of them exhibited alcoholemia exceeding 0.8 grams (of pure alcohol per liter of blood) as compared with 9 percent of intoxicated drivers involved in a death or bodily injury, which reflects an increase in exposure to the risk due to alcoholemia equal to 3. But such surveys are made only one year (sometimes two), for purposes of special studies, not systematically in order to follow the trend of the phenomenon. It is accordingly necessary to have a medium of intermediate analysis of accidents and traffic which reconciles the gathering of homogeneous information at the lowest cost and with greatest expeditiousness; this is the purpose of random surveys relating both to accidents and to traffic.

The ONSER commenced observing traffic on the various types of French roads in February, 1972 on the initiative of SETRA and then on that of the Road Safety Commission. In addition, another system of gathering by random (1/15) sampling of accident reports has been in effect since that year. Comparison

of these two types of collective data with reference to the same geographical areas should furnish us with quite extensive information within the next year.

We present here only the road survey undertaken for methodological purposes and to be taken over thereafter by the administrative authorities involved; we will illustrate with reference to a specific problem – the effectiveness of the seat belt – the advantages resulting from application thereof.

NATURE OF THE SURVEYS UNDERTAKEN

There are two types of surveys:

- Periodic "infrastructure-traffic" survey gathering information "on the wing", i.e., without stopping the driver.
- "Vehicle-driver" survey made at gas stations, requiring the active participation of the user and seeking to characterize the vehicle/driver pair and the trips made on the various types of roads of the system.

The French road system is classified into primary categories on the basis of the traffic which it carries:

- Corridor motorways: 850 km.
- Link motorways: 2,000 km.
- National roads in the "primary system": 25,000 km.
- Heavy-traffic national roads outside the primary system 35,000 km.
- Other national roads: 20,000 km.
- Heavy-traffic departmental roads: 6,000 km.
- Other departmental roads: 274,000 km.
- Rural and local roads: 700,000 km.

By way of illustration, the average daily traffic estimated by the SETRA in the 1970 census was in the range of 3,000 vehicles/day on the entirety of the national roads.

The purpose of the periodic "infrastructure-traffic" survey is to assess: intensity and make-up of the traffic; distances covered; rate of seat-belt-equipped vehicles; rate of use of seat belts; rate of use of helmets by cycle drivers; rate of vehicle occupancy; and speeds by brands and models and the speed-limit violations.

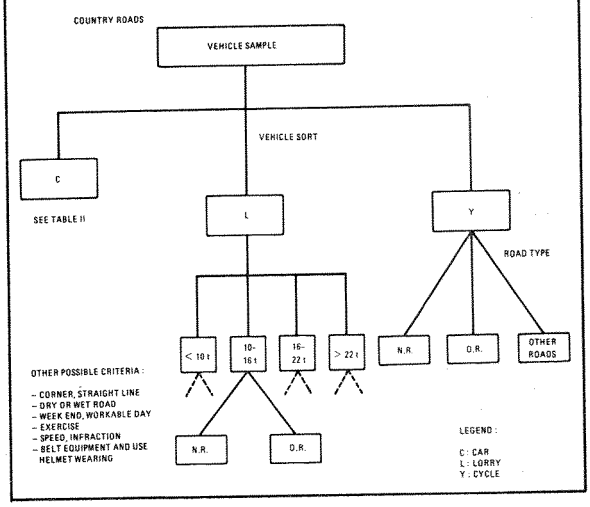
Gathering of the information has improved progressively since February 1972. Currently, the observation relates to: motorways; heavy-traffic national roads (including centers of less than 5,000 inhabitants); departmental roads (including centers of less than 5,000 inhabitants); passages of primary-system roads through centers of less than 20,000 inhabitants; and centers of more than 20,000 inhabitants (to come).

The survey on the heavy traffic national roads (HTNR) is distinguished from the others by its scope: 20 administrative departments spread throughout France have been chosen so as to obtain adequate representativeness of the traffic. In each department there are set up, at random in proportion to the length of the HTNR system in kilometers, 80 daytime observation points per month and 80 nighttime points per quarter, every hour of the 24 being accounted for. Each point provides 30 minutes of observation during the day and 1 hour at night. This represents 960 daytime observation points, the anticipated observed daytime traffic being: 75,000 passenger cars; 10,000 trucks; and 1,500 cycles.

The precision of the average of the speeds measured (interval of confidence 95 percent) is in the range of 6 km/hr monthly, and 2 km/hr annually, and the rate of non-recognition of brands and models averages 10 percent at night.

As compared with the SETRA automatic count, the "Checklist" survey has the advantage of greater geographical scope, and, above all, of obtaining qualitative information (vehicle brands and models, belts, etc.); on the other hand the accuracy of the automatic information (traffic flow) is less on the roads involved. By way of illustration we give hereunder the tables of sorting criteria and the analytical forms relative to speed measurements and to the use of belts.

TABLE 1
SORTING CRITERIA



APPLICATION TO AFTER-THE-FACT ASSESSMENT OF SEAT BELT EFFECTIVENESS

Trend of Use Rates from August 1973 to April 1974

The French regulations relating to equipment of vehicles with seat belts are set forth in the decree of

ONSER.

TABLEAU DE BORD SECURITE - PORT DE LA CEINTURE

SETEC-ECONOMIE

Département : _____ Route : _____ Jour : _____ 1 Sens ☐ Etat de la chaussée _____ N° Point _____

Commune : _____ P.K. : _____ Heure : _____ 2 Sens ☐ Intempéries _____ N° ½ h au Pt _____

V.L. IMMATRICULATION	2 ROUES	VEHICULE EQUIPE DE CEINTURE — PLACE DU CONDUCTEUR	PLACE DU PASSAGER AVANT — EQUIPEE	PLACE DU PASSAGER AVANT — NON EQUIPEE	VEHIC. NON EQUIPE	Nbr. DE PERSON. A BORD
6 loc 1 lim 2 out 3 étr 4	5 v 6 m	7 b 1 nb 2 né 3	8 b 1 nb 2 vid 3	occ 4 vid 5	9 1	10
11 loc 1 lim 2 out 3 étr 4	5 v 6 m	12 b 1 nb 2 né 3	13 b 1 nb 2 vid 3	occ 4 vid 5	14 1	15
16 loc 1 lim 2 out 3 étr 4	5 v 6 m	17 b 1 nb 2 né 3	18 b 1 nb 2 vid 3	occ 4 vid 5	19 1	20
21 loc 1 lim 2 out 3 étr 4	5 v 6 m	22 b 1 nb 2 né 3	23 b 1 nb 2 vid 3	occ 4 vid 5	24 1	25
26 loc 1 lim 2 out 3 étr 4	5 v 6 m	27 b 1 nb 2 né 3	28 b 1 nb 2 vid 3	occ 4 vid 5	29 1	30
31 loc 1 lim 2 out 3 étr 4	5 v 6 m	32 b 1 nb 2 né 3	33 b 1 nb 2 vid 3	occ 4 vid 5	34 1	35
36 loc 1 lim 2 out 3 étr 4	5 v 6 m	37 b 1 nb 2 né 3	38 b 1 nb 2 vid 3	occ 4 vid 5	39 1	40
41 loc 1 lim 2 out 3 étr 4	5 v 6 m	42 b 1 nb 2 né 3	43 b 1 nb 2 vid 3	occ 4 vid 5	44 1	45
46 loc 1 lim 2 out 3 étr 4	5 v 6 m	47 b 1 nb 2 né 3	48 b 1 nb 2 vid 3	occ 4 vid 5	49 1	50
51 loc 1 lim 2 out 3 étr 4	5 v 6 m	52 b 1 nb 2 né 3	53 b 1 nb 2 vid 3	occ 4 vid 5	54 1	55
56 loc 1 lim 2 out 3 étr 4	5 v 6 m	57 b 1 nb 2 né 3	58 b 1 nb 2 vid 3	occ 4 vid 5	59 1	60
61 loc 1 lim 2 out 3 étr 4	5 v 6 m	62 b 1 nb 2 né 3	63 b 1 nb 2 vid 3	occ 4 vid 5	64 1	65
66 loc 1 lim 2 out 3 étr 4	5 v 6 m	67 b 1 nb 2 né 3	68 b 1 nb 2 vid 3	occ 4 vid 5	69 1	70
71 loc 1 lim 2 out 3 étr 4	5 v 6 m	72 b 1 nb 2 né 3	73 b 1 nb 2 vid 3	occ 4 vid 5	74 1	75
76 loc 1 lim 2 out 3 étr 4	5 v 6 m	77 b 1 nb 2 né 3	78 b 1 nb 2 vic 3	occ 4 vid 5	79 1	80

V.L. IMMATRICULATION	2 ROUES	VEHICULE EQUIPE DE CEINTURE — PLACE DU CONDUCTEUR	PLACE DU PASSAGER AVANT — EQUIPEE	PLACE DU PASSAGER AVANT — NON EQUIPEE	VEHIC. NON EQUIPE	Nbr. DE PERSON. A BORD
6 loc 1 lim 2 out 3 étr 4	5 v 6 m	7 b 1 nb 2 né 3	8 b 1 nb 2 vid 3	occ 4 vid 5	9 1	10
11 loc 1 lim 2 out 3 étr 4	5 v 6 m	12 b 1 nb 2 né 3	13 b 1 nb 2 vid 3	occ 4 vid 5	14 1	15
16 loc 1 lim 2 out 3 étr 4	5 v 6 m	17 b 1 nb 2 né 3	18 b 1 nb 2 vid 3	occ 4 vid 5	19 1	20
21 loc 1 lim 2 out 3 étr 4	5 v 6 m	22 b 1 nb 2 né 3	23 b 1 nb 2 vid 3	occ 4 vid 5	24 1	25
26 loc 1 lim 2 out 3 étr 4	5 v 6 m	27 b 1 nb 2 né 3	28 b 1 nb 2 vid 3	occ 4 vid 5	29 1	30
31 loc 1 lim 2 out 3 étr 4	5 v 6 m	32 b 1 nb 2 né 3	33 b 1 nb 2 vid 3	occ 4 vid 5	34 1	35
36 loc 1 lim 2 out 3 étr 4	5 v 6 m	37 b 1 nb 2 né 3	38 b 1 nb 2 vid 3	occ 4 vid 5	39 1	40
41 loc 1 lim 2 out 3 étr 4	5 v 6 m	42 b 1 nb 2 né 3	43 b 1 nb 2 vid 3	occ 4 vid 5	44 1	45
46 loc 1 lim 2 out 3 étr 4	5 v 6 m	47 b 1 nb 2 né 3	48 b 1 nb 2 vid 3	occ 4 vid 5	49 1	50
51 loc 1 lim 2 out 3 étr 4	5 v 6 m	52 b 1 nb 2 né 3	53 b 1 nb 2 vid 3	occ 4 vid 5	54 1	55
56 loc 1 lim 2 out 3 étr 4	5 v 6 m	57 b 1 nb 2 né 3	58 b 1 nb 2 vid 3	occ 4 vid 5	59 1	60
61 loc 1 lim 2 out 3 étr 4	5 v 6 m	62 b 1 nb 2 né 3	63 b 1 nb 2 vid 3	occ 4 vid 5	64 1	65
66 loc 1 lim 2 out 3 étr 4	5 v 6 m	67 b 1 nb 2 né 3	68 b 1 nb 2 vid 3	occ 4 vid 5	69 1	70
71 loc 1 lim 2 out 3 étr 4	5 v 6 m	72 b 1 nb 2 né 3	73 b 1 nb 2 vid 3	occ 4 vid 5	74 1	75
76 loc 1 lim 2 out 3 étr 4	5 v 6 m	77 b 1 nb 2 né 3	78 b 1 nb 2 vid 3	occ 4 vid 5	79 1	80

TABLEAU DE BORD SECURITE - MESURES DE VITESSES

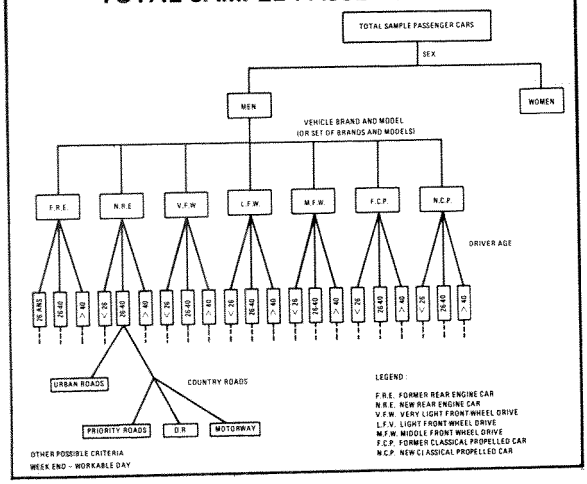
SETEC-ECONOMIE

N° départ	Commune	Route	P.K.	Voie	Heure	Etat de la surface	Intempéries	Nb. sens	Ligne droite ☐ Virage ☐ Epingle ☐ Courbe ☐	Vites. limit.

N° Point _____ N° ½ h _____

VITESSES	Véh. lim. 90	RENAULT R.4	R.5	R.6	R8 R.10	R12 R.15	R16 R17	4 CV Dauph.	CITROEN 2 CV. Dyane	Ami 6, 8	G.S.	I.D. D.S.	PEUGEOT 104	204	304	404	504	203 403	SIMCA 1000	1100	1300 1500	Aut. V.L. Fran.	FIAT	FORD	Aut. V.L. Etr.	Four-gonn.	V.L. à Cara.	CARS Sans Disque	Avec Disque	CAMIONS FRANÇAIS Sans Disque	1 Disque 60 65	75	85	2 Disques Sans	60	80	90	CAR ETRAN	CAR ETRON	2 Roues Un	Les Cas
N° COL.		6	7	8	9 10	12	14	16	18	20	22	24	26	28	30	32	34	36	38	40	42	44	46	48	50	52	54	56	58	60	62	64	66	68	70	72	73	74	76	77	78	80
≤ 20																																										A
21 à 40																																										B
41 à 50																																										C
51 à 60																																										D
61 à 70																																										E
71 à 80																																										F
91 à 90																																										G
	L 90	R4	R5	R6	R8.10	R12.15	R16.17	4CV.D	2CV.D	Ami 6.8	GS	I.D.DS.	104	204	304	404	504	2.403	1000	1100	1300 1500	A.V.L.F.	FIAT	FORD	A.V.L.E.	Fourg.	VL+C	C. S.D	C.A.D.	S.D.	60.65	75	85	80	80	90	S.I.	S.M.	2 R.		
N° COL.		6	7	8	9 10	12	14	16	18	20	22	24	26	28	30	32	34	36	38	40	42	44	46	48	50	52	54	56	58	60	62	64	66	68	70	72	73	74	76	77	78	80
91 à 100																																										H
101 à 110																																										I
11 à 120																																										J
121 à 130																																										K
131 à 140																																										L
141 à 150																																										M
> 150																																										N
N° COL.		6	7	8	9 10	12	14	16	18	20	22	24	26	28	30	32	34	36	38	40	42	44	46	48	50	52	54	56	58	60	62	64	66	68	70	72	73	74	76	77	78	80

TABLE 2
TOTAL SAMPLE PASSENGER CARS



February 5, 1969, in particular making mandatory the installation during manufacture of front seat belts in cars manufactured from April 1, 1970. Since July 1, 1973, contemporaneously with general application of speed limits, the use of belts has been made mandatory in the country in vehicles equipped therewith.

The "Checklist" relative to the heavy traffic national roads permits monthly check of the observance of this regulation in relation to the equipment of the vehicles. Four categories of vehicles are distinguished according to their registration: "local" (i.e., registered in a contiguous department), "other French," and "foreign"; a fifth category embraces all light vehicles registered in France. Locals account for about 60 percent of this category.

It is found that the aggregate of the vehicles observed have a rate of equipment in the range of 60 percent at present, the equipment rate of the "local" category being slightly lower (on the order of 55 percent), the average age of the motor-vehicle population being higher in the rural areas. This equipment rate follows a regularly rising curve.

The same thing is not true of the rate of use; this rate relates the number of belted front-seat drivers and passengers to the corresponding users having but not using a belt. It is found that this rate has dropped during autumn in the "local" and "adjoining" categories in the absence of punishment for failure to use the belt. This state of affairs has resulted in specific steps being taken, the police now issuing tickets for failure to wear a belt when a car in violation is stopped; since November a rise in the use rate back to the initial level in the range of 64 percent has been noted. The rates of users is obtained by multiplying the two foregoing rates; it is observed that more than a third of the users actually use their seat belts on the heavy traffic national roads (HTNR).

The curves hereunder, Figures 1, 2, and 3, reflect the monthly trends in the 2nd quarter of 1973 and in the 1st quarter of 1974 in the rate of use; and, in the rate of equipment and of belted users in the "local" and "all-France" categories, with reference to the rate of use before the regulations of July 1, 1973. This rate was on the order of 30 percent for this road system versus about 15 percent on the lighter traffic national and departmental roads. The series giving the rates of belt use by categories of users appear in Table III; the accuracy of the monthly estimates relative to the aggregate of the users is in the range of 4.5 percent, the interval of confidence being 90 percent.

Assessment of Safety Belt Effectiveness

We should be able to measure the effectiveness by comparing the rate of belt use observed on the roads and the rate of accidents for belted users on belt-equipped vehicles. We are thus limited to a homogeneous aggregate of relatively new vehicles while eliminating other risk factors. The difficulty of

TABLE 3
BELT USE RATE (HTNR CHECKLIST)
IN PERCENT

	1973					1974			
Categories	Aug.	Sept.	Oct.	Nov.	Dec.	Jan.	Feb.	March	April
Local	56	51	44	49	53	56	57	54	59
Adjoining	68	68	58	63	74	71	73	72	75
Other French	66	66	65	69	71	70	72	80	75
Foreign	63	63	66	74	45	49	45	51	43
Overall	63	58	52	58	62	62	64	64	67

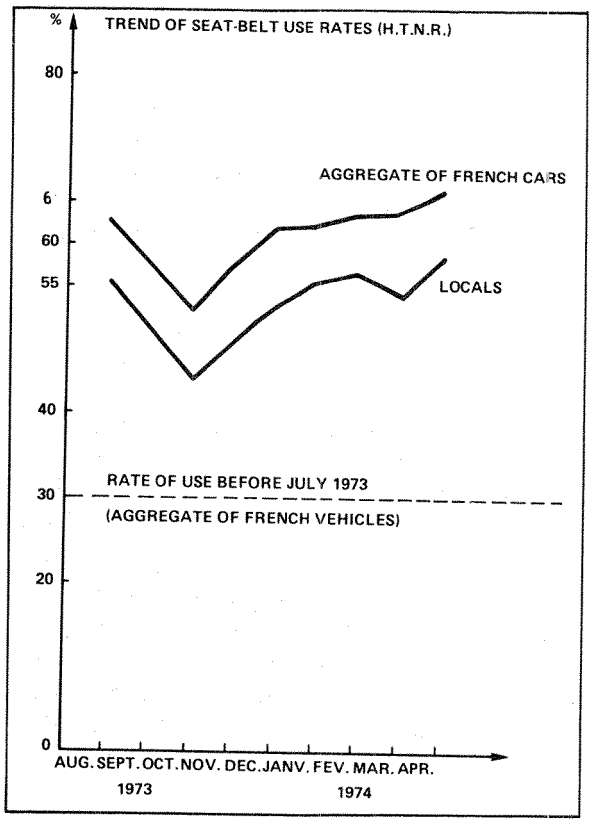


Figure 1

such measurement lies in the comparison of homogeneous statistics.

When we consider *traffic*, we have monthly information as to the daytime behavior of drivers and front-seat passengers since August, 1973, on the heavy traffic road system even while passing through centers of less than 5,000 inhabitants. Annual information by road categories has been included.

In an appraisal of *accidents*, the Constabulary has since October, 1973, provided statistical statements of deaths and bodily injuries for belt-equipped drivers and front seat passengers according to: use or nonuse of belts; in centers of less than 5,000 inhabitants; and on motorways and all other roads. Considering the situation prior to 1973, we can refine these statistics, for example, by isolating according to the same categories, the daytime accidents on the HTNR system; on the other hand, the availability of detailed index for 1973 will be waiting on the fall of 1974.

Heretofore, belt-use effectiveness was calculated by comparing ratios, for a given sample of fatal or injurious accidents, of deaths or injuries to passengers according to belt use or nonuse. In fact, such reference is ambiguous because the use of a belt induces, from the behavioral standpoint, an assumption of risk which may be greater than average; and, from the *accident* view, a conversion of

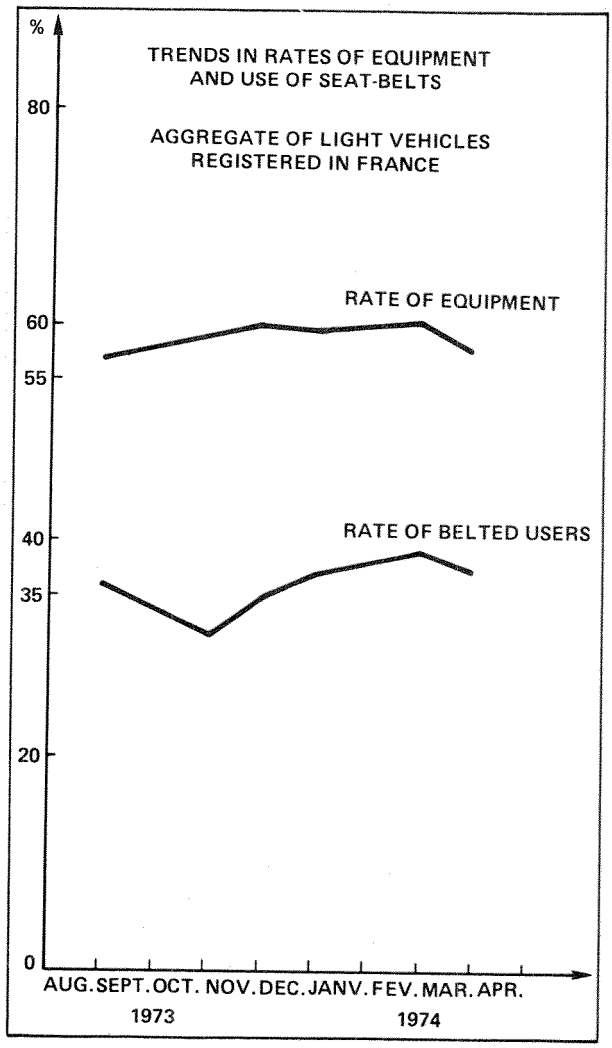


Figure 2

the harm from bodily injury into property damage. These two factors work against each other.

In appraising all events, it would appear that the concept of risk for fatal or injurious accident according to belt use or nonuse, must be evaluated in calculating effectiveness of belt use.

To illustrate the methods of calculation, we will use the following notations:

	Beltusers	Non-Beltusers
Road users (drivers and front-seat passengers)	C	$\bar{C}$
Those involved in deaths and injuries	A	$\bar{A}$
	V	$\bar{V}$
Victims (killed and injured)	(K + I)	$(\bar{K} + \bar{I})$

The rates of gravity reflect the ratios of victims to those involved in fatal or injurious accidents, according to whether or not they were using their belts: $v = V/A$; $\bar{v} = \bar{V}/\bar{A}$.

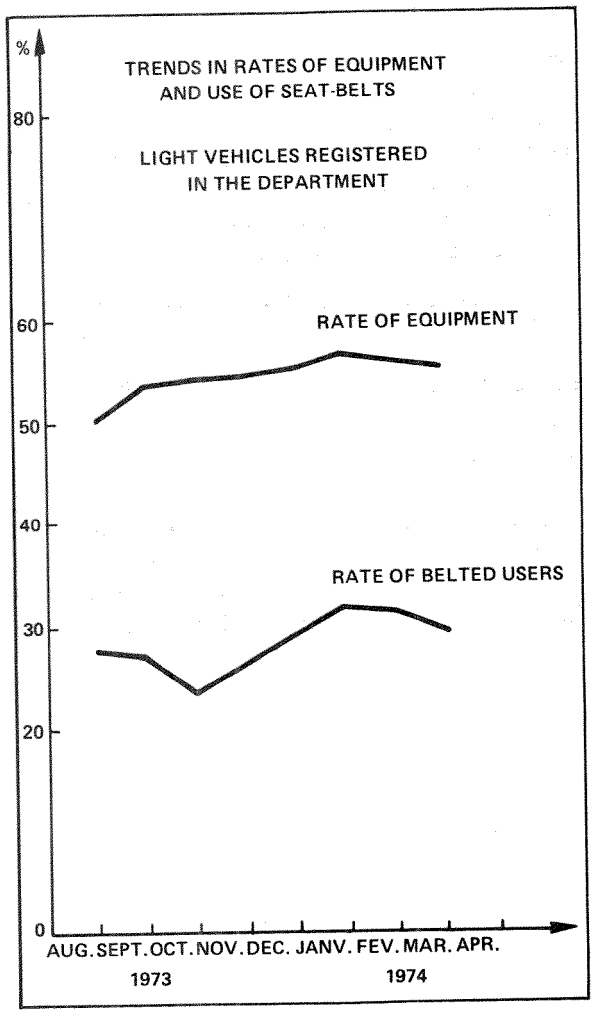


Figure 3

The effectiveness measured in reduced gravity is expressed by the relative gain in frequency: $\lambda = (\bar{v} - v)/\bar{v} = 1 - (V/\bar{V})/(A/\bar{A})$, and can be calculated separately with reference to those killed, the seriously injured, and the injured and victims in entirety. By applying this coefficient to the number of victims not using their belts, we obtain the saving in victims with belt use.

The risk of being involved in a fatal or injurious accident, according to belt use or nonuse is: $r = A/C$; or $\bar{r} = \bar{A}/\bar{C}$, and, consequently, the relative risk resulting from nonuse is: $\tau = (\bar{A}/A)/(\bar{C}/C)$.

The effectiveness measured in reduction of the risk for fatal or injurious accidents is: $\lambda' = 1 - (V/\bar{V})/(C/\bar{C})$.

We apply these calculations to the two stated period which correspond to very different behaviors, while considering a certain generalization on use of belts combined with generalization on speed limits for the two periods:

1. The year 1972.
2. The period October, 1973-April, 1974, covered by monthly statistics.

Effectiveness for 1972 on the Heavy Traffic National Roads (HTNR). The sources are daytime accidents recorded on the HTNR involving belt-equipped vehicles, and daytime belt-use rate on the HTNR. This rate is 26 percent, or $C/(C + \bar{C}) = 0.26$; when $\bar{C}/C = 2.85$. On the other hand: $\bar{A}/A = 1.95$.

The relative risk for nonbelt use is in the range of 0.7. Paradoxically, there was more risk of an injurious accident if the belt was used (in 1972). This situation is based on these factors: rate of belt use (about 10 percent) is very low even on this category of roads; speed is limited to 110 km/hr only on 13,000 kilometers of national roads (in the country); and the factor of driver behavior must accordingly preponderate in the interpretation, which is to say, a greater assumption of risk is laid on belt use (belted drivers).

On the other hand, effectiveness, measured in terms of reducing the aggregate of killed and seriously injured, is significant – on the order of 36 percent (from 50 percent of past fatalities alone). However, effectiveness was only marginally significant for the slightly injured – about 10 percent. And if we refer *not* to the casulty figures but to the belted drivers, such effectivenss becomes negative.

Effectiveness for October, 1973-April, 1974. The sources are monthly accident statistics relating to belt-equipped vehicles involving belted or unbelted drivers and front-seat passengers in centers of less than 5,000 inhabitants, on motorways, and all other roads; and monthly rate of daytime belt use on the HTNR system with centers included. This rate over the entire period is about 60 percent; the proportion of belt-equipped vehicle belted users is accordingly: $C/(\bar{C} + C) = 0.60$, when $\bar{C}/C = 0.67$.

The accident statistics disclose little variability in the frequencies relating users and nonusers of belts, according to the categories of roads.

	$K/\bar{K}$	$I/\bar{I}$	$V/\bar{V}$	$A/\bar{A}$
Entirety of the system (Constabulary)	.42	.74	.71	.88
Country and nonmotorway portion	.41	.77	.74	.88
On motorways	.34	.72	.63	.82

However, a lower relative proportion of belted drivers killed on motorways is noted.

On the HTNR system the intermediate figure: $\bar{A}/A = 1.18$ can be assumed, hence the mean risk of being involved in an injurious accident if the belt is not used is: $(\bar{A}/A)/(\bar{C}/C) = 1.75$. This risk has thus more

than doubled by 2.6 between 1972 and the end of 1973. *This is due to the combined effect of mandatory belt use and generalized speed limits.*

When measuring the effectiveness of the belt in reduction of severity alone, according to the condition of the victims and the categories of the roads, we see:

Effectiveness in	Killed	Injured	Victims
Entirety of the system	53%	17%	20%
In the country and off motorways	54%	13%	16%
On motorways	58%	12%	22%

This "apparent" effectiveness is in the same range of magnitude as for 1972 (46% killed and 19% injured).

These appear to be low numbers from the standpoint of the injured, in view of the conversion of bodily injury into property damage involvement – a situation distorting the basis of reference. In order to eliminate this phenomenon, reference must be made to belted and unbelted users. Thus, over the period in question, we have certain assessments only for the HTNR system. It can be estimated that for the rest of the system, the behavior of the locals observed on HTNR can serve as a reference, for a use rate of 53 percent. On motorways, assessments are in process and a rate of 80 percent can be forecast.

On these assumptions the effectiveness of belts, combined with that of the speed limit would be:

Effectiveness = λ'	Killed	Injured	Victims
In the country	63%	36%	38%
On motorways	91%	82%	85%

Thus, in the calculation of *effectiveness* this application brings out the advantage of taking account of the risk of involvement in an injurious or fatal accident according to whether or not the seat belt is used.

THE FREQUENCY OF CORRESPONDING VEHICLE DAMAGE IN CRASH TESTS AND ACTUAL ACCIDENTS

DR. M. DANNER and
K. LANGWIEDER
German Association of Liability, Accident and Motor Traffic Insurers – (HUK-Verband)

INTRODUCTION

For the purpose of uniformly determining the crash behavior of motor vehicles, internationally laid-down crash tests are today conducted (Reference 1). These tests are designed to accurately reproduce the most frequent and serious types of damage which occur in actual traffic accidents today; on the other hand, these should be authentically reproducible with the least possible difficulty from the test and measuring aspect.

A study of actual traffic accidents shows, however, that present day crash tests reproduce actual accident circumstances only to an inadequate degree. The most frequent types of damage in actual accidents are not embraced, and various crash tests furnish only a small proportion of actual accident circumstances.

On the basis of 813 actual accidents involving 1,626 damaged vehicles, a frequency categorization of types of damage is therefore set-out in this report, and the question examined as to what proportion of all car accidents involving injuries to occupants is reflected by present-day crash tests.

INVESTIGATION METHOD

This study is based on material from the HUK-Verband accident investigation on "Interior Safety of Automobiles," which now incorporates some 30,000 car accidents involving injuries to occupants. The evaluation modus was elucidated in detail at the 3rd ESV Conference in Washington (Reference 2).

The 813 accidents involving 1,626 vehicles, which form the basis of this report, pertain to cases in which detailed pictures of the damaged cars existed. As opposed to the overall material embracing a total of 30,000 accidents, these accidents show no divergencies as regards the nature of the accident, the injuries and the degree of damage at significance level p = 1.0%. The investigation on "Interior Safety of Automobiles" relates to car accidents within the field of motor vehicle third party liability. While these accidents thus reflect in representative form the accident circumstances in car/motor vehicle accidents, they cannot demonstrate the true proportion of car accidents from other causes.

For this reason, the frequency of various types of damage was initially compiled in respect of car/motor vehicle accidents only.

The proportion of car accidents from other causes with injuries to occupants can, however, be gathered from Reference 3 and Figure 1. The nature of the collision and the type of damage in respect of these accidents is known from investigations conducted by the HUK-Verband on more than 1,000 car accidents from other causes (References 4, 5, 6) and from Reference 7. Based on car/motor vehicle accidents, the frequency allocation of types of damage in all car accidents with injuries to occupants was thus established.

Damage sustained to the vehicles themselves was reproduced in the form of 22 Damage Categories (Figure 2). In determining the type of damage, only the primary damage was evaluated, but not incidental, minor damage to other surfaces of the vehicle. The types of damage in the case of angled collisions, e.g. Damage Category C, were compiled on the basis of equal degree of damage and not

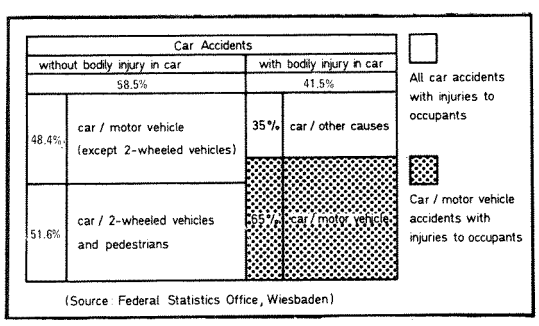


Figure 1. Categorization of Car Accidents in the Federal Republic of Germany

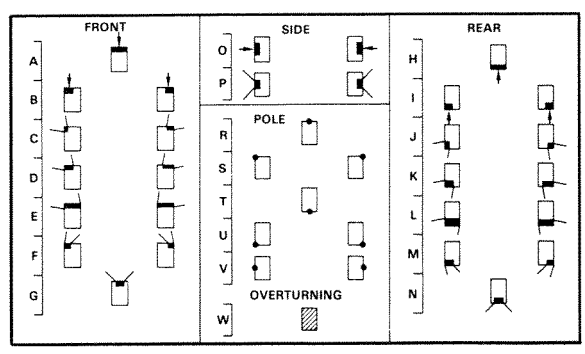


Figure 2. Types of Damage to Vehicles in Actual Accidents

categorized according to the angle of impact, as this can often not be subsequently determined with the required accuracy in actual accidents. Accidents in which the vehicle overturned were allotted to Damage Category W, which thus embraces all types of overturning.

FREQUENCY ALLOCATION OF TYPES OF DAMAGE IN CAR/MOTOR VEHICLE ACCIDENTS

A review of the damage sustained by 1,626 vehicles is compiled in Figure 3. In 52.4% of all car/motor vehicle accidents, the left hand side (driver's side) sustained damage. Where side-on collisions occurred, damage to the driver's side was twice as frequent as on the front passenger seat side. Center damage to the rear-end of the vehicle, however, prevailed.

	CENTRE+	LEFT+	RIGHT+	TOTAL
FRONT	14.4	36.7	18.5	69.6
SIDE	–	8.8	4.0	12.8
REAR	7.4	6.9	3.3	17.6
TOTAL	21.8	52.4	25.8	100.0

+RELATIVE TO LONGITUDINAL DIRECTION OF CAR

Figure 3. Damaged Surfaces of 1,626 Cars in Car/Motor Vehicle Collisions (%)

Damage to the front-end surface predominated with 69.6%. Particularly in the case of head-on/side-on collisions of vehicles with minor overlapping, a problem arose in allocating the damage to the front-end and side surfaces. This differing allocation explains the many divergencies in well-known literature (References 8, 9), and it would seem desirable here to agree on a uniform system.

An exact allocation according to the individual types of damage in car/motor vehicle accidents with injuries to occupants is shown in Figures 4(a) – 4(c). The percentages refer in each case to the overall proportion of the respective type of damage. By far the most frequent were head-on collisions with side impact and partial overlapping. Damage Category C alone accounted for 23% of all damage in car/motor vehicle collisions. In second place, was Damage Category D with 17.3%. This type of damage, by virtue of an overlapping of 2/3 of the front-end surface, frequently points to a high accident severity, whereby injuries to the driver by possible impact against the A stanchion and to the front seat passenger by side-on impact against the steering column and steering wheel are increased. In third and fourth place, with more or less the same frequency, were head-on collisions, without side impact, with complete overlapping – Damage Category A – and with one-sided overlapping – Damage Category B.

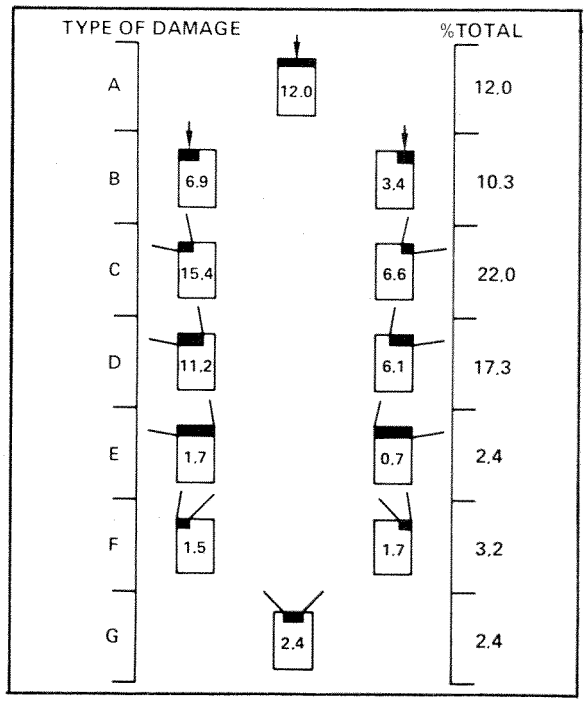


Figure 4(a). Types of Damage (front) in Car/Motor Vehicle Accidents with Passenger Injuries

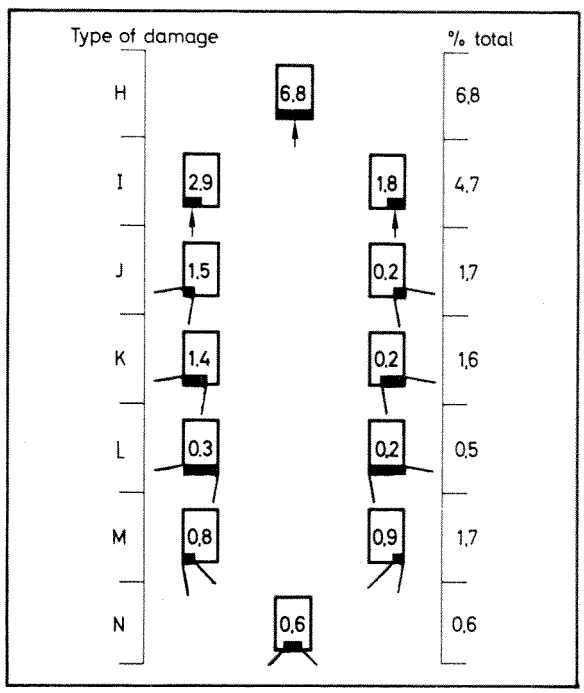


Figure 4(b). Types of Damage (rear) in Car/Motor Vehicle Accidents with Passenger Injuries

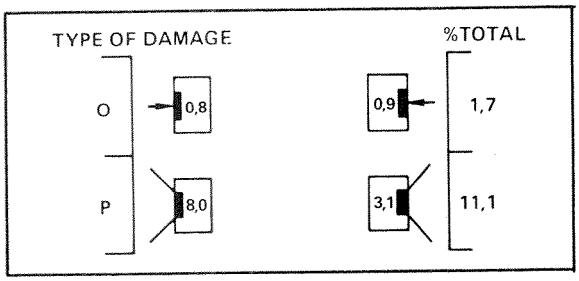


Figure 4(c). Types of Damage (side) in Car/Motor Vehicle Accidents with Passenger Injuries

In the case of rear-end collisions, Damage Category H with 6.8% clearly predominated. Only Damage Category I with 4.7% showed any other notable percentage.

In the case of side-on collisions, Damage Category P with 11.1% occurred exceptionally frequently. Injuries were in general relatively slight, however, as grazing damage to the vehicles was also included in this category. A quite different picture, on the other hand, is revealed by Damage Category O, which by virtue of the almost right-angled collision of the vehicles led to a high degree of deformation and often to serious injuries to the occupants in the area of collision. The injuries in head-on collisions, side impact, collision in the area of the doors and rear-end collisions are demonstrated (Reference 10) on the basis of several thousand accidents.

FREQUENCY OF TYPES OF DAMAGE IN ALL CAR ACCIDENTS WITH INJURIES TO OCCUPANTS

Based on these results, the order of types of damage in all car accidents with injuries to occupants was then determined by additionally taking into account car accidents from other causes.

The various types of damage in the latter category of accidents are known (References 4, 5, 6, 7), from which ensued an overturning percentage of 8% and a collision frequency with trees or poles of 11.5% of all accidents with injuries to occupants. Other types of collisions were allocated according to the respective type of damage.

Figure 5, shows a general survey of the dominating proportion of damage to the front-end surface and the left-hand side (driver's side). Analogous to Figures 4(a) and 4(c), Figures 6(a)-6(d) show the respective types of damage incorporating car accidents from other causes. No change has occurred in the order of the types of damage – here again Damage Category C dominates with 16.2%, followed by Damage Category B (12.9%) and Damage Category A (12.5%). In car collisions with trees, center impact – Damage Category R – predominates with 5.6% (Figure 7). The various types of damage in accidents in which the car turns over (Figure 8) are at present being analyzed at the HUK-Verband in a special study.

	Centre*	Left*	Right*	Total
Front	20,6	28,4	17,0	66,0
Side	–	7,6	4,7	12,3
Rear	5,5	5,0	3,2	13,7
Overturning	–	–	–	8,0
Total	26,1	41,0	24,9	100,0

*relative to longitudinal direction of car

Figure 5. Damaged Surfaces of Cars in All Car Accidents (%)

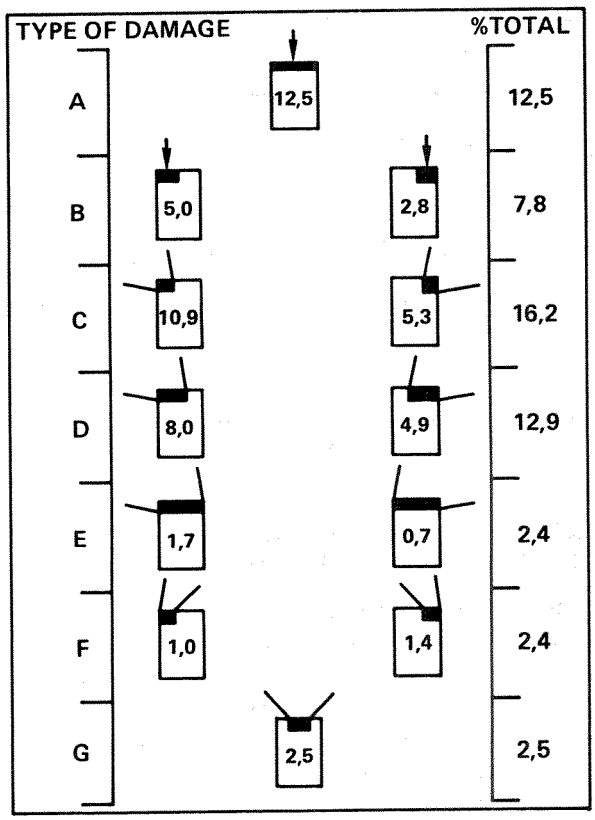


Figure 6(a). Types of Damage (front) in All Car Accidents with Passenger Injuries

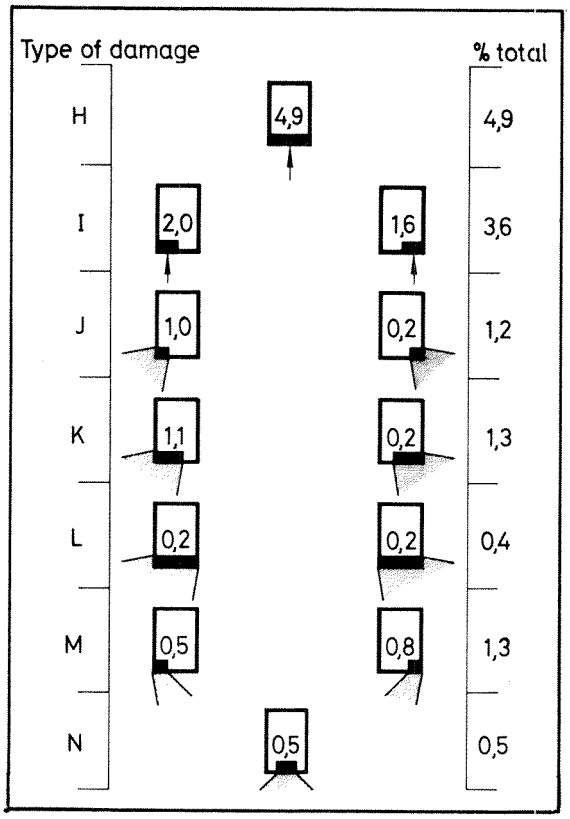


Figure 6(b). Types of Damage (rear) in All Car Accidents

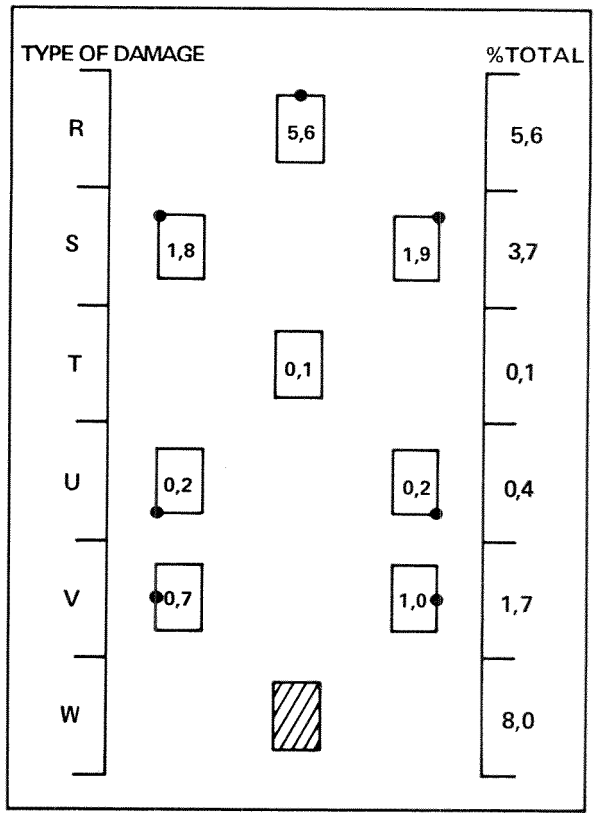


Figure 6(d). Types of Damage (tree, overturning)

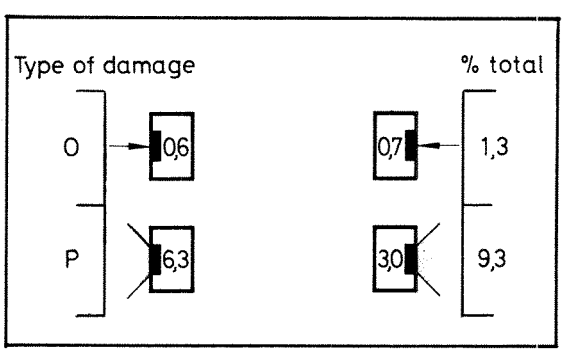


Figure 6(c). Type of Damage (side) in All Car Accidents

RELATIVE PROPORTION OF ESV CRASH TESTS TO ACTUAL TYPES OF DAMAGE

In comparing what proportion of actual vehicle damage can be reproduced by internationally laid-down ESV Crash Tests (Figure 7) according to Reference 1, various reservations must be made:

- On the basis of available accident material on car accidents with injuries to occupants, an assessment of Tests 1 and 6 cannot be made, as here in the majority of cases only material damage is involved.
- The ESV Tests 3/4 and 8/9 had to be combined, as these tests are difficult to assess due to the few results available and especially as pictorial material is only seldom published.
- In actual accidents, collisions mostly occur between motor vehicles and therefore inevitably reveal a different picture to that which the same force would exert in the case of a barrier collision. By applying the types of damage to the approximate corresponding ESV tests, maximum values ensue, which must be appropriately corrected. These correction values between the maximum and true relative proportion of ESV tests thus represent approximate values, which ensued from assessment of the respective vehicle damage and can be adjusted.

A comparison with the types of damage (Figure 6) shows how often ESV tests correspond to actual accident circumstances. The true significance of the head-on, distributed collision with a fixed barrier and the head-on pole test is clearly evident. Five further test procedures show percentages of between 1.7% and 2.5% (Figure 8). Of these, the rear-end impact with a moving barrier is of particular importance. Rear-end impact with the pole, on the other hand, has no significance as regards actual accidents and is, at the most, of applicable value in the case of very low speeds. The right-angled collision between vehicle

or pole and the door surface of the test car lies, in this list, below 2% of all actual accidents, but nevertheless represents approximately the maximum deformation of the vehicles in Damage Category P with a total of 10%.

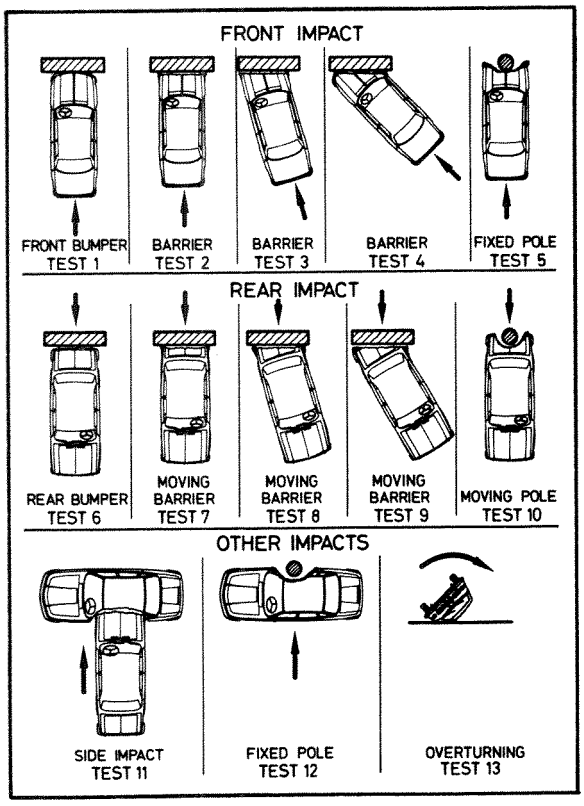


Figure 7. Type of Damage R

Test	1	2	3/4	5	6	7	8/9	10	11	12	13
Max. %	–	12.5	2.4	5.6	–	4.9	2.9	0.1	1.3	1.7	8.0
Correction %	–	50	80	100	–	50	80	100	100	100	25
Actual %	–	6.2	1.9	5.6	–	2.5	2.3	0.1	1.3	1.7	2.0
Position	–	1	6	2	–	3	4	9	8	7	5

Figure 8. Type of Damage W

SYNOPSIS

By means of present-day ESV Crash Tests, some 20%-25% of actual accidents with injuries to occupants can be reproduced. This percentage ensues above all from the frequency of head-on tests against a fixed barrier and pole, and from rear-end impact against a moving barrier. The Damage Categories C and D, which occur most frequently in actual accidents – i.e. angled head-on collisions with overlapping – are, on the other hand, not taken into account in the ESV Crash Test Program.

One-sided overlapping in non-angled head-on collisions (Damage Category B) with a percentage proportion of 7.8% of actual accidents also cannot be demonstrated by present-day ESV tests.

A test commensurate with Damage Categories B or C/D should at least be incorporated into the ESV program, even though difficulties could occur as regards reproductivity of the actual accident. This would apply, in particular, if the fixed barrier were to be replaced by another vehicle or by a moving barrier, on which a standardized structure is fixed. Such a test would, however, hold out prospects of achieving particularly realistic results.

One-sided and angled crash tests should be conducted in such manner that the most serious damage occurs on the driver's side, where damage in actual accidents dominates by far. In actual accidents, the driver is more often than not in the direct area of collision, which also has a significant bearing on the many accidents in which the driver is alone in the vehicle.

Earlier studies conducted by the HUK-Verband (Reference 10) have already shown that present-day ESV crash speeds embrace 90% of actual collision speeds. This question will, however, again be analyzed in great detail in the new investigations on car accidents with injuries to occupants, which the HUK-Verband is now carrying out during the course of 1974 and in which approximately 120,000 car accidents will be evaluated. In this connection, it would be of great advantage if automobile manufacturers were to at least make available pictorial material from the more important crash tests with serial production or ESV vehicles, as the coherence between types of damage and actual accident speeds, on the one hand, and crash tests and equivalent speeds, on the other, could thereby be analyzed. Only by constantly improved adaptation of present-day crash tests to actual accident circumstances is it possible to apply current research developments to the full advantage and benefit of vehicle safety.

REFERENCES

1. Verband der Automobilindustrie (VDA), Technische Anforderungen für Experimentier-Sicherheits-Personenkraftwagen, 21. Dez. 1970.

2. Danner, M.; "Interior Safety of Automobiles – A Study of the German Motor Insurers on Road

Accidents involving Passenger Injuries," Washington (1972), Report on the Third International Technical Conference on Experimental Safety Vehicles.

3. Deutsches Statistisches Bundesamt, Wiesbaden, Strassenverkehrsunfälle, Reihe 6.
4. HUK-Verband, Hamburg (1972), Bericht Nr. 1 "Innere Sicherheit im Auto," eine Untersuchung des HUK-Verbandes über 10,271 Pkw-Unfälle mit Insassenverletzung.
5. HUK-Verband, Hamburg (1974), Bericht Nr. 3 "Innere Sicherheit im Auto," eine Untersuchung des HUK-Verbandes über 28,936 Pkw-Unfälle mit Insassenverletzung.
6. HUK-Verband, Hamburg (1973), "Ursachen und Begleitumstände der Verkehrsunfälle mit schwerem Personenschaden in der Bundesrepublik Deutschland," Samelband M.
7. Berlioz, M. C., "Distribution and Gravity of Collisions as a Function of the Damaged Parts of the Vehicle and the Obstacle Hit," Report on the Second International Technical Conference on Experimental Safety Vehicles, Sindelfingen (1971).
8. Mackay, G. M., "Some Collision Aspects of British Road Accidents," Automobile Engineer (Oct. 1969) Vol. 59, No. 3.
9. Hartmann, F.; Patel, A.; Tarrière, C.; Thomas, C., Analyses médicales et techniques des collisions sur autoroute bilan de deux années d' investigation, XIV. FISITA (1972).
10. Langwieder, K. "Passenger Injuries in Collisions and their Relation to General Speed Scale," XVII. STAPP-Car-Crash-Conference 1973, Oklahoma.

ACCIDENT INVESTIGATION AS AN AID TO PLANNING THE FUTURE OF CAR SAFETY

BARBARA E. SABEY and
I. D. NEILSON
Transport and Road Research Laboratory

INTRODUCTION

In planning the future of car safety, investigations of accidents, their attendant circumstances, the vehicles involved, and casualties incurred, provide an invaluable guide to the priorities to be applied both in accident prevention and reduction of injury. On a national scale, statistical analyses of data collected as a routine process can be used to indicate the broad perspective of types of conflict between different road users and classes of vehicles and the most pressing needs for remedial action. At a much more local level, investigations in which vehicle features, injuries, and the detailed circumstances of accidents can be examined in depth make possible a basic understanding of factors leading to the accidents and the injuries incurred. This paper gives an outline of some of the accident investigation techniques used in Great Britain for this purpose, together with illustrative examples of the studies which have led to estimation of the effects of possible safety measures and establishment of priorities in car safety.

STATISTICAL ANALYSES

On a national basis, data for all known accidents involving death or injury are reported by the Police in standardized form (see Fig 1). These data, which include information about casualties, vehicles involved, and the attendant circumstances, are collected by the Department of the Environment and stored on computer at the Transport and Road Research Laboratory to provide the basic national road accident data bank. In recent years injury accidents in Great Britain have totalled over a quarter of a million annually, and the casualties incurred have numbered about 360,000, including 7,500 deaths.

The most important facets of car involvements are indicated by an analysis of the 80,000 fatal and serious injury accidents reported in 1971. Of these, 60,000 accidents involved cars (including taxis), the types of conflicts and casualties incurred being summarized in Table 1. The main sources of accidents and casualties lie in four main categories of conflict.

1. Single vehicle car accidents, 10,795, in which no other vehicle or pedestrian is involved. These are indicative of loss of control, which is frequently associated with tire or brake deficiencies including locking of wheels through braking.
2. Car striking a pedestrian, 17,551 accidents, which emphasizes the vulnerability of pedestrians. These accidents mostly occur in urban areas where car speeds are relatively low.
3. Car and two wheeler collisions, 10,758 accidents. In these accidents most of the injuries occur to the drivers or passengers of the two wheeled vehicles, indicating also the vulnerability of the two wheelers.
4. Two car accidents, 10,350, in which car occupants are the main sufferers. On average, there are 1.2 casualties per car in these accidents.

Broadly speaking, therefore, the main areas for remedy indicated by these statistics are the prevention of loss of car control and the protection of pedestrians, two wheeler riders, and car occupants.

Knowledge of the distribution of damage to cars is a useful indication of the relative frequency of various types of accident and the need for occupant protective features. The national statistics can help in this respect in that they provide information on the sides of the car damaged and whether it overturned. Table 2 gives three damage distributions as examples: these relate to numbers of cars involved in accidents in which some person was either fatally or seriously injured. The sides of the car, front, back, left and right are denoted by F, B, L and R respectively.

Most single vehicle accidents result from driving off the road or from loss of control. Single car accidents in which the car overturns occur about 40 percent as frequently as those not involving overturning and the first column shows that the

TABLE 1
TYPES OF CONFLICTS WITH CARS IN FATAL AND SERIOUS INJURY ACCIDENTS IN GREAT BRITAIN IN 1971

Car in conflict with	Number of accidents	Car casualties Driver	Car casualties Passenger	Other vehicle casualties Driver	Other vehicle casualties Passenger	Pedestrian casualties
No other vehicle or pedestrian	10795	4091	10684	–	–	–
Pedestrian	17551	429	299	–	–	18420
Pedal cycle	3266	134	91	3252	–	26
Moped	693	27	20	675	–	2
Other 2-wheeler	6799	409	333	6544	1427	91
Cars (incl. taxis)	10350	12231	13302	←		530
PSV	719	495	489	73	483	122
Light goods	2294	1380	1224	1197	1043	199
Medium goods	449	355	319	82	48	35
Heavy goods	2340	2089	1637	221	51	74
Other vehicle	658	460	466	151	108	68
More than one other vehicle (not specified)	4134	3781	3723	1484	720	235
TOTAL	60048	30881	32587	13679	3873	19802
		100822				

MINISTRY OF TRANSPORT
SCOTTISH DEVELOPMENT DEPARTMENT
WELSH OFFICE

Form Stats. 19 (Rev. 1968)

REPORT OF ROAD ACCIDENT RESULTING IN PERSONAL INJURY

POLICE REFERENCE No. .....

CASUALTIES

CASUALTY CLASS (1st 2nd 3rd 4th 5th 6th) — 1

- Driver or rider 12
- Passenger 11
- Pedestrian 0

Of:—
- Pedal cycle 1
- Moped solo 2
- Motor scooter solo 3
- Motor cycle solo 4
- Combination (m/p, m/s, or m/c) 5
- Car or taxi 6
- Public service vehicle 7
- Goods vehicle 8
- Other vehicle 9

SEVERITY OF INJURY — 2

- Killed † 1
- Seriously injured 2
- Slightly injured 3

SEX OF CASUALTY — 3-4

- Male 12
- Female 11

AGE in years †

CAR PASSENGER *
- In front seat 12
- In rear seat 11

PEDESTRIAN ACTION — 5
- Crossing road at pedestrian crossing 12
- Crossing road within 50 yds. of ped. crossing 11
- Crossing road elsewhere 0
- In road, not crossing 1
- Masked by stationary vehicle 2
- On footpath or verge 3
- On refuge or centre strip 4

PSV PASSENGER
- Boarding or alighting 5

CAR OR VAN OCCUPANT *
- Safety belt in use 1
- Safety belt fitted but not in use 2
- Safety belt not fitted 3

VEHICLE OCCUPIED — 6
- First vehicle 1
- Second vehicle 2
- Third vehicle 3
- Fourth vehicle 4
- Fifth or subsequent vehicle 5

VEHICLES INVOLVED

REGISTRATION NUMBERS
1st ..... 2nd .....
3rd ..... 4th .....

1st 2nd 3rd 4th
7 8 9 10
(7) (7) (7)

TYPE — 11-12

MAKE — 13

MODEL OR SIZE — 14

SKIDDING
- Skidded 12
- Jacknifed 11

TOWING & ARTICULATION
- Articulated vehicle 1
- Caravan 2
- Other tow 3

VEHICLE DEFECTS — 15, 16, 17
- Load 12
- Tyres—report for process 11
- Brakes 0
- Lights, front 1
- Lights, rear 2
- Brake lights 3
- Trafficators 4
- Vision—glass wipers, washers 5
- V.E. report requested 6

12—0 / 1—6

AGE (entered by Ministry) — 1st 2nd 3rd 4th 20

MANOEUVRES — 21
- Entering or leaving layby 12
- Entering or leaving hard shoulder 11
- Reversing 0
- Parked 1
- Stopping 2
- Starting 3
- Turning round 4
- Turning left or waiting to 5
- Turning right or waiting to 6
- Going ahead but held up 7
- Going ahead overtaking moving or held up vehicle 8
- Going ahead, other 9

OVERTURNING — 22
- Overturned 11

PART DAMAGED — 23, 24
- Front 1
- Back 2
- Left 3
- Right 4
- All four sides 5
- None 6

SEATS OCCUPIED (CARS ONLY) * 25
- Front Passenger 11
- Rear: One passenger 1
- Rear: Two passengers 2
- Rear: 3 or more passengers 3

DRIVERS AND RIDERS

Vehicle: 1st 2nd 3rd 4th

SEX — 18
- Male 12
- Female 11

BREATH TEST
- Positive 1
- Negative 2
- Not required 3
- Failed to provide 4

LEARNER-DRIVER — 19
- "L" driver 11

DRIVER'S ACTIONS
- Disobeyed junction control 1
- Disobeyed double centre or offset line 2
- Failed to give precedence at pedestrian crossing 3

DRIVER'S OR PASSENGER'S ACTION
- Opened door negligently 4

Vehicle: 1st 2nd 3rd 4th

AGE GROUP (estimated) — 26
- Under 17 years 12
- 17 years 11
- 18 years 0
- 19 years 1
- 20 years 2
- 21 years 3
- 22—24 4
- 25—28 5
- 29—34 6
- 35—54 7
- 55—64 8
- 65 or over 9

*Report these items until 31.12.70, at which date a review becomes operative.

ATTENDANT CIRCUMSTANCES

POLICE FORCE — 27 28

LOCAL AUTHORITY — 29 30 31

DAY OF WEEK .....

DATE — 32 33 34 (Day Mth)

TIME (24-hour clock) — 35 36 37 (Hour 5Min.)

ROAD CLASS — 38
- M 1
- A, motorway standard 2
- A 3
- B 4
- C or unclassified 5

ROAD NUMBER — 39 40 41 42

SPEED LIMIT — 43
- 30 mph or less 3
- 40 mph 4
- 50 mph 5
- 60 mph 6
- 70 mph 7

ROAD TYPE — 44
- Clearway in force 11
- One-way street 1
- Dual carriageway 2
- Other road 3

LANE MARKINGS — 45
- No lane markings 1
- Two lanes 2
- Three lanes with offset double lines 3
- Normal three lanes 4
- Four lanes 5
- Lane markings unknown 6

LIGHT — 46
- Daylight 1
- Dark, with street lights over 20 feet high, lit 2
- Dark, with street lights under 20 feet high, lit 3
- Dark, street lights unlit 4
- Dark, no street lighting 5

WEATHER — 47
- Raining 1
- Snowing 2
- Fog 3
- Other 4

ROAD SURFACE CONDITION — 48
- Dry 1
- Wet 2
- Snow or ice 3

JUNCTION DETAIL — 49
- Roundabout 1
- 'T' or staggered junction 2
- 'Y' junction 3
- Crossroads 4
- Multiple junction 5
- Other junction 6
- Using private drive or entrance 7
- Not at or within 20 yards of junction 8

JUNCTION CONTROL — 50
- Authorized person 1
- Automatic traffic signal 2
- Stop sign 3
- 'Give Way' sign or marking 4
- Uncontrolled 5

PEDESTRIAN CROSSING (If on or within 50 yds.) — 51
- Manually controlled 1
- Light-controlled at junction 2
- Light-controlled not at junction 3
- Uncontrolled 4

MOVEMENTS BEFORE ACCIDENT — 52
- Stationary vehicle involved 11
- One moving vehicle 1
- Two moving vehicles, same direction 2
- Two moving vehicles, opposite directions 3
- Two moving vehicles, different roads 4
- More than two moving, all same direction 5
- More than two moving, same road, not all same direction 6
- More than two moving, not all same road 7

CARD CLASS (entered by Ministry) — 53

SPECIAL CONDITIONS AT SITE — 54 55 56
- Dog in carriageway 12
- Other animal in carriageway 11
- Object in carriageway 0
- Parked vehicle(s) (contributory) 1
- Automatic level crossing involved 2
- Lamp post hit 3
- Telegraph pole hit 4

CLASS OF ACCIDENT — 57
- Fatal 1
- Serious 2
- Slight 3

LOCATION — 58 59 60 61 62 63

† If a person under 15 is killed, attach or write below a brief account of the accident.

53-5967 8/68 TSB Co. Ltd.

Figure 1. National Road Accident Report Form For Great Britain

TABLE 2
SIDES OF CAR DAMAGED IN FATAL AND SERIOUS INJURY ACCIDENTS IN GREAT BRITAIN IN 1969

Sides damaged	Single vehicle accidents not involving pedestrians: Overturning	Single vehicle accidents not involving pedestrians: Not overturning	Two vehicle accidents: Not overturning
Total number of cars	2881	7625	35388
Percentage with damage:			
F	3	33 ←	33 ←
B	0	0	5
L	3	3	6
R	3	2	7
FB	1	1 }	1 }
FL	4	21 }	11 }
FR	4	9 }	17 }
BL	0	1	1
BR	0	0	3
LR	2	1	0
FBL	1	2	1
FBR	2	1	1
FLR	10	8 ←	4 ←
BLR	1	0	0
FBLR	65 ←	12 ←	3 ←
No damage	0	5	5
Not specified	1	1	2
TOTAL	100	100	100

damage usually extends all round the car. When the car does not overturn the damage is usually due to striking some roadside object. The second column shows that the front of the car is most often damaged, though some side impacts are severe enough to distort all the sides and so the FBLR impacts are not necessarily exclusively frontal directions of impact.

Impacts between a car and another vehicle are much more frequent. Some of the damage is sustained in impacts with roadside objects following the vehicle impact. Both cars are included separately in the third column if the impact is car to car. Frontal impacts, with the damage sometimes extending around the car, are by far the most frequent. Side impacts are by no means negligible and are more prevalent than in the single vehicle accidents.

As another example of the usefulness of the national data, a simple comparison of severity of injury incurred by car occupants wearing seat belts or not (Table 3) indicates the substantially lower percentages of fatal and serious injuries to casualties wearing seat belts, especially in urban areas. These reductions tend to underestimate the effectiveness of seat belts in reducing injury because they neglect accidents in which seat belts eliminate injury.

INVESTIGATIONS IN DEPTH

Two main types of depth investigation are carried out at the Transport and Road Research Laboratory

TABLE 3
SEVERITY OF INJURY OF DRIVERS AND FRONT SEAT PASSENGERS IN CARS, WHEN WEARING OR NOT WEARING SEAT BELTS (1971)

	Fatal and serious injuries as a percentage of all injuries			
	Drivers		Front Seat Passengers	
	Urban	Rural	Urban	Rural
Seat belt: worn	14	31	14	28
not worn	20	38	20	36
not fitted	20	37	21	35

with the object of investigating both primary and secondary safety. For the former, a multidisciplinary team attends the scene of accidents ("on the spot" investigations) to obtain at first hand details of how and why accidents occur and to get a better understanding of the contributory role played by the road and environment, the vehicle and road user. For the latter, a medical team examining injuries of road casualties undergoing hospital treatment links with an engineering team examining vehicles concerned in order to establish how and why injuries occur.

On-the Spot Investigations and Accident Prevention

In the most recent investigation, which took place between March 1970 and February 1974, details of 2130 accidents were obtained by the team, who were on call 24 hours a day. Arrangements with the local police enabled the team to be called immediately to the scene of accidents, where they took photographs, made extensive notes of each vehicle including its damage and position, features of the road scene, signs of debris, tire marks and any other environmental factors. A scale plan of the site was drawn up. Then, with the agreement of the persons concerned, drivers and pedestrians involved were interviewed. Finally, the team discussed each accident in detail to assess the contributory factors. All the information was recorded on a specially designed form and transferred to a computer bank.

The four year study was divided into two phases: for the first two years' data, 1164 accidents, results are available. In the second two years, special attention was given to contributory vehicle factors which included a thorough examination of the braking systems of vehicles involved.

The most important outcome of the analysis is the assessment of contributory factors. These are considered in three main groups: those concerning the road design and environment; those concerning the vehicle, its design and condition; and, thirdly, human factors – driver or pedestrian error and driver impairment. The relative contributions of these three groups and their interactions are summarized in Table 4. Thus, vehicle factors were contributory in 23-1/2 percent of accidents – usually linked with some human factor.

A more detailed breakdown of the vehicle design and maintenance features contributing to accident occurrence shows that defects play a much larger part than do obviously poor design characteristics. Table 5 indicates the type of defect for all types of vehicles and cars. Tire and brake defects are the most prevalent. Incorrect tire pressures indicate differentials of at least 55 kN/m^2 (8 lb/in^2) and were associated with loss of control of the vehicle; they were more associated with wet roads, while deflations were more associated with dry roads.

TABLE 4
CONTRIBUTORY FACTORS IN ROAD ACCIDENTS

	Contributory factor		
	Road and Environment	Road User	Vehicle
Number of accidents in which factor is contributory	366	1,066	278
Percentage	32	91½	23½
Interactions (percentage contributions)			
Single factor	1½	50	3
Double factors	25		15
		½	
Treble factors		5	

Traffic Medicine and Injury Reduction

In the course of studies which began in 1965, over 1550 vehicle occupants and 200 pedestrians have been interviewed and examined in hospital by medically qualified investigators, and vehicles concerned examined in detail. A further 1150 injured occupants or pedestrians have been interviewed and examined in hospital but the vehicles involved not investigated. Over 150 postmortems on road accident victims have been attended. These detailed investigations have enabled firm conclusions to be drawn as to mechanism by which certain injuries are incurred – leading to recommendations for changes in vehicle design to alleviate the problem. The injury data also provide a basis for experimental work on limits of human tolerance.

Results and applications of this work are reported in three other papers being presented to this Conference: Some patterns and causes of injuries in car occupants (E. O. C. Grattan): Towards pedestrian safety (V. J. Jehu): and Human injury tolerance level determination from accident data using the OPAT dummy (J. G. Wall).

ESTIMATION OF THE EFFECTS OF POSSIBLE SAFETY MEASURES

Accident studies inevitably relate to the existing situation involving vehicles built within the preceding ten years, some being designed before then. What is wanted is an estimate of the improvements in safety to be expected of measures under discussion. It is now customary to produce cost benefit ratio estimates, but these are not the only outcome of accident studies; more important is the indication of what the problems really are and what may be effective means of dealing with them. The procedures can best be explained by two examples, one in the field of car occupant protection, and one of car primary safety.

TABLE 5
VEHICLE DEFECTS IDENTIFIED AS CONTRIBUTORY TO THE ACCIDENT OCCURRENCE
(FROM ON-THE-SPOT INVESTIGATION OF 1164 ACCIDENTS)

Defect	Contributory factor			
	All vehicles		Cars	
Tire deflation before impact	18	96	13	79
Illegal tread depth (less than 1mm)	19		16	
Incorrect pressure	59		50	
Brake defect	64		42	
Steering defect	9		6	
Light defect	9		5	
Mechanical failure	25		10	
Electrical failure	3			
Loading fault	11		1	
Overall poor	11			
TOTAL	228			

Protecting Occupants of Cars Struck in the Rear

Though Table 2 shows that about 10 percent of cars involved in fatal and serious accidents receive damage to their rears, a similar breakdown for casualties shows lower percentages. Of all 1971 car occupant fatalities in Great Britain, only 5 percent were in cars damaged in the rear – the corresponding percentages for seriously and slightly injured being 7 percent and 13 percent respectively. These percentages exclude cars damaged on all four sides although these may include a few major rear impacts. Further study of the national accident data for 1971 shows several other features of these accidents. Over a half of these cases involved damage to the fronts as well as to the rears, presumably in shunts and double impact accidents. In many of these, the frontal impacts were probably more serious than those to the rear. In all rear impact accidents almost 60 percent of the casualties were drivers and only 20 percent of the fatalities and 14 percent of the seriously injured were in the rear seats. Asymmetric impacts to the rear occurred slightly more often than symmetric ones. These accidents occurred equally often on rural and urban roads.

Detailed studies, described above, show how these injuries were sustained. The rear seat occupants were usually injured by collapse of the back of the car around them when cars or larger vehicles struck them heavily in the rear, this accounting for very close to 1 percent of all car occupant casualties for both fatal and serious injury casualties.

Drivers and front passengers made up 3 percent of all fatal and 4.5 percent of all serious injury casualties to car occupants by being in accidents in which the major impacts were at the rear. The detailed studies are not yet sufficiently numerous to say how many were injured by each of the three most important causes: (a) even more complete structural collapse than those injuring rear occupants; (b) collapse of front seat backs or seats, projecting their occupants into the rear of cars which had just been damaged in the rear impacts; and (c) by heads or other parts of the body striking the B posts or other structures usually in asymmetric rear impacts.

Apart from these, there were two other kinds of injuries: (a) a very few resulting from fires following rear impacts; and (b) in slight injury cases or in accidents wrongly classified as being damage only, whiplash neck injuries occur in number amounting to at least 5 percent of all noted car occupant slight injuries.

This information is sufficient to plan safety countermeasures. The strengthening of front seats against collapse rearwards in severe impacts would be worthwhile on cost benefit grounds. The inclusion of head restraints on high seat backs would be desirable; in spite of increased costs, this would reduce the number of painful and, in some cases, long-lasting whiplash injuries. Redesign of the inside face of B posts and adjacent items may be justified by benefits in frontal and side impacts as well as those from the rear. Good design practice may eliminate some of the few fires leading to injury without increasing costs appreciably. Clearly major improvements would come only if rear compartment structural strengths were greatly increased, such action coming after incorporating the minor measures already discussed. Developments in the vehicle safety program should show how costly this might be. This present study indicates that the additional benefits of major structural strengthening might reduce fatal and serious injury car occupant casualties by about 2½ percent.

Accidents Resulting from Tires at Reduced Pressures

A study of the TRRL "on-the spot" accident data in conjunction with experimental work by Holmes and others has helped to explain the accident situation resulting from use of tires at reduced pressure. Tires lose pressure over weeks, hours, minutes, or a few seconds: in the last case, they may be said to have burst. With slow leaks or small punctures the driver gradually alters his driving to match the changing handling of his car without in many cases being aware of the changes. With fast leaks the driver may be caught unaware by an unexpected response at the first bend or lane change after a length of straight road. The situation arises almost immediately with a 'burst' and, in a violent form, if the tire comes off the rim. Three accident situations may be described:

1. Car control has been lost wherein a tire at low pressure remains on its rim; alternatively, the pressure distribution between the four tires may be quite different from that recommended. A skilled driver could almost certainly drive such a car satisfactorily. In these accidents there are usually road and driver factors contributing as well and it is possible that the tire contribution may not have been important. Data from Table 5 and further details in TRRL Leaflet LF 374 suggest that this occurs in 5 to 10 percent of all accidents involving cars.
2. Control has been lost of a car – later found to have one or more tires deflated and off the rim. However, detailed investigations do not prove that the tire came off the rim before the driver lost control, but point to it having done so while the

car was swerving or striking a curb, or some other obstacle. In such cases the tire must have been below 60 kN/m^2 (9 lb/in^2) at the instant of loss of control; this situation is really similar to *1* except that the change in handling and the probability of the low tire pressure contributing to the accident would have been greater.

3. Control has been lost after a tire has deflated and, in some cases, off its rim.

Situations 2 and 3 together contribute to just over 1 percent of all accidents involving cars studied on the on-the-spot survey. Detailed evidence was not sufficient to state clearly the reason for the accident. On motorways, this situation is relatively much more common, possible occurring in 16 percent of all accidents (Godley, TRRL Report LR 498). The higher speeds and smooth and straight driving conditions on motorways are thought to account for the difference.

Two remedial measures are possible. Tires and rims which cannot separate at low tire pressure would be valuable in situation 3, particularly for cars used on motorways. Low-pressure warning devices would be useful in all three situations, assuming that reliable devices can be produced cheaply enough to give an acceptable cost benefit ratio and that drivers heed their warnings.

Summing-up

These two examples illustrate briefly the way in which consideration of aspects of car design and performance in relation to accident occurrence may indicate the appropriate need for improvement and remedy. Similar considerations of other aspects have led to an assessment of priorities for car safety action.

ACKNOWLEDGEMENTS

The work described in this paper forms part of the program of research for the Transport and Road Research Laboratory. The paper is contributed by permission of the Director.

SOME PATTERNS AND CAUSES OF INJURY IN CAR OCCUPANTS

E. GRATTAN and
J. A. HOBBS
Transport and Road Research Laboratory
Department of the Environment

ABSTRACT

An analysis has been made of the patterns of injury among 768 unbelted and 111 belted occupants of cars involved in accidents which resulted in at least one person being detained in a hospital.

The causes of severe and of fatal injury received by the occupants are described for the different directions of vehicle impact, for the different seating positions, and for the different regions of the body injured.

The chances of sustaining injury are compared for the unbelted and belted front seat occupants in the sample both for the body as a whole and for the different parts of the body.

INTRODUCTION

The challenge implicit in the design of the experimental safety vehicle is the reduction of injury in car occupants. It may be helpful therefore to examine some of the injury patterns which occur in this group of road users, in vehicles currently in use, in order to demonstrate the type, incidence and severities of injury for the different seating positions and vehicle impact directions, and also to indicate some of the leading causes of the more severe injuries sustained, and the relationship between seat belts and injury. There were 768 unbelted occupants and 111 belted occupants at risk in the sample of car accidents studied by the Transport & Road Research Laboratory, in each of which at least one occupant was sufficiently seriously injured to be detained in a hospital. 73 percent of the cases involved frontal, 18 percent side and 2 percent rear impacts, while 7 percent were overturning accidents. Injuries have been classified as minor, moderate, severe or fatal, TRRL Leaflet No. 130, 1974.

OVERALL PATTERN OF INJURY

The overall pattern of injury among unbelted fatally injured car occupants was investigated some years ago by Gissane and Bull (1964) and the conclusion was reached that rear seat occupants were less likely to be killed in car accidents than front seat occupants. Among the 768 unbelted occupants at risk in our sample the incidence of the different severities of injury for different categories of occupant is shown in Table 1.

Table 1
PATTERN OF INJURY AMONG 768 UNBELTED CAR OCCUPANTS AT RISK
Percentage of Occupants Sustaining Injury by Category of Occupant and by Severity of Injury

Category of Occupant \ Severity of Injury	No Injury %	Minor Injury %	Moderate Injury %	Severe Injury %	Fatal Injury %
Drivers (370)	10	18	26	33	13
Front Seat Passengers (213)	12	21	26	31	10
Rear Seat Passengers (185)	23	31	27	13	6

It will be seen that rear seat occupants were injured much less frequently than front seat occupants and that this reduction in injury occurred in the fatal and severe categories, there being no reduction in moderate injury and small increase in the number of slightly injured. These results generally confirm earlier results that rear seats are safer than front seats but the findings are more comprehensive since they cover the whole range of injury severities.

ANALYSIS OF INJURY PATTERNS

If only the severe injuries in the sample of unbelted occupants are considered and the figures are broken down to show the different regions of the body sustaining such injury for different categories of occupant, and for different impact directions, it is apparent that the leading injuries received vary according not only to seating position but to the type of collision. These results are shown in Table 2.

It will be noted in frontal impacts that among drivers the commonest severe injury sustained was to the thigh (fracture of the femur), that among front seat passengers the region most commonly receiving severe injury was the head, and that among rear seat passengers it was the thigh. It will also be noted that the risk of severe injury to the chest was greater in drivers than in front seat passengers and that the risk of severe injury to the head was higher in front seat passengers than in drivers.

Table 2
NUMBERS OF SEVERE INJURIES SUSTAINED AMONG 768 UNBELTED CAR OCCUPANTS AT RISK BY REGION OF BODY, BY DIRECTION OF IMPACT AND BY SEATING POSITION

Direction of Impact and Category of Occupant / Region of Body	Head	Neck	Back (Spine)	Upper Limb	Chest	Abdomen	Pelvis	Hip-Joint	Thigh	Knee	Lower Leg	Ankle and Foot
FRONTAL												
Drivers (268)	30	1	2	14	27	11	5	19	36	20	21	12
Front Seat Passengers (154)	25	1	1	4	9	8	3	10	13	2	5	3
Rear Seat Passengers (129)	6	1	2	–	5	2	2	1	9	1	2	1
SIDE												
Drivers (66)	7	2	1	3	8	2	3	2	11	1	3	–
Front Seat Passengers (34)	3	1	–	2	3	3	4	–	7	1	1	–
Rear Seat Passengers (38)	1	–	–	–	1	1	2	–	2	1	1	–
OVERTURN												
Drivers (29)	–	1	1	–	–	–	–	–	–	–	1	–
Front Seat Passengers (21)	2	1	3	–	2	–	–	–	–	–	–	–
Rear Seat Passengers (17)	1	–	–	–	1	–	–	–	–	–	–	–
REAR												
Drivers (7)	No Severe Injuries Sustained											
Front Seat Passengers (4)												
Rear Seat Passengers (1)												

For side impacts the pattern was somewhat different; although the numbers are rather small the thigh (fracture of the femur) was the leading region of the body sustaining severe injury among both drivers and front seat passengers, head injury was about equally common in drivers and front seat passengers, but chest injury, as in frontal impacts, was more common in drivers; it will also be noticed that the risk of sustaining a fracture of the pelvis was higher in side impacts than in frontal impacts, for all categories of occupant, whereas the risk of receiving a dislocation of the hip was greater in frontal impacts, as was the risk of receiving a fracture of the lower leg.

For overturning accidents the numbers are small but the risk of receiving spinal injury appeared to have been much greater than in other forms of collision. In the rear impact accidents in this sample there were no severe injuries.

If a similar analysis is undertaken for the fatal injuries sustained it will be noted, Table 3, that

Table 3
NUMBERS OF FATAL INJURIES SUSTAINED AMONG 768 UNBELTED CAR OCCUPANTS AT RISK BY REGION OF THE BODY, BY DIRECTION OF IMPACT AND BY SEATING POSITION

Direction of Impact and Category of Occupant / Region of Body	Head	Neck	Back (Spine)	Upper Limb	Chest	Abdomen	Pelvis	Hip-Joint	Thigh	Knee	Lower Leg	Ankle and Foot
FRONTAL												
Drivers (268)	9	2	–	–	17	10	–	–	–	–	–	–
Front Seat Passengers (154)	7	1	–	–	7	2	–	–	–	–	–	–
Rear Seat Passengers (129)	4	–	–	–	3	3	–	–	–	–	–	–
SIDE												
Drivers (66)	3	1	–	–	3	2	–	–	–	–	–	–
Front Seat Passengers (34)	5	1	–	–	4	1	–	–	–	–	–	–
Rear Seat Passengers (38)	–	–	–	–	–	–	–	–	–	–	–	–
OVERTURN												
Drivers (29)	2	–	–	–	2	1	–	–	–	–	–	–
Front Seat Passengers (21)	–	–	–	–	–	–	–	–	–	–	–	–
Rear Seat Passengers (17)	2	–	–	–	–	–	–	–	–	–	–	–
REAR												
Drivers (7)	No Fatal Injuries Sustained											
Front Seat Passengers (4)												
Rear Seat Passengers (1)												

among drivers involved in frontal impact accidents the region of the body most commonly receiving fatal injury was the chest, whereas among front and rear seat passengers the risk of the chest and the head sustaining fatal injury was about the same.

For side impacts the head and the chest received fatal injury in about equal proportions in both drivers and front seat passengers. In overturning accidents, although the numbers are small, if all seating positions are taken together the head appears to have been more frequently injured than the chest. There were no fatalities in the rear impact accidents.

CAUSES OF INJURY

A knowledge of injury patterns is clearly of importance in the design of vehicles incorporating safety features, but it is necessary to know not only the parts of the body most commonly receiving injury, and its severity, but how the injuries were caused. For the regions of the body injured which have been discussed the leading causes of severe or of fatal injury are shown in Tables 4-10.

It will be seen, Table 4, that among drivers involved in frontal impact collisions the components most commonly causing severe injury to the head were the windshield pillar or header rail or the steering wheel and column assembly; to the chest or abdomen the steering wheel and column assembly; to the thigh, the knee and hip joint the facia panel; and to the lower leg the parcel tray or bulkhead. Table 5 shows that among front seat passengers the component most frequently causing severe injury to the head was the windshield, while to the chest, the abdomen, the hip joint and the thigh it was the facia panel. Among rear seat passengers the numbers were too small for conclusions to be drawn, and in a high proportion of cases no injury correlation could be ascertained.

In side impacts, Tables 6 and 7, the most common cause of severe injury to drivers for most parts of the body, although not for the head, was intrusion of the side structure of the vehicle. Among front seat passengers intrusion of the side structure of the vehicle was also the commonest cause of injury. There were too few rear seat passengers for conclusions to be drawn although as for front seat occupants intrusion appeared to have been the most common cause of severe injury. When injury was related to intrusion of the side structure of the vehicle the severely injured occupant was usually seated on the side of the vehicle sustaining intrusion.

In overturning accidents, Table 8 and 9, the numbers are small but it will be seen that severe injury to the back (fracture of the spine) was related to the roof in both of the front seat occupants sustaining such injury. In a larger sample of seriously injured car occupants investigated at an earlier date the incidence of fracture of the spine was found to be five times greater in overturning accidents than in the frontal impact collisions, the leading cause of spinal fracture being intrusion of the roof.

No causes for severe injury in rear impact collisions could be determined since no severe injuries were seen.

Table 4
INTERIOR OR EXTERIOR STRUCTURES CAUSING SEVERE INJURY TO DIFFERENT REGIONS OF THE BODY IN FRONTAL IMPACTS AMONG 268 UNBELTED DRIVERS AT RISK

Structure Causing Injury \ Region of Body	Head	Neck	Back (Spine)	Upper Limb	Chest	Abdomen	Pelvis	Hip-Joint	Thigh	Knee	Lower Leg	Ankle and Foot
Roof	3	1	–	–	–	–	–	–	–	–	–	–
Windshield Pillar or Header Rail	9	–	2	1	–	–	–	–	–	–	–	–
Windshield	4	–	–	–	–	–	–	–	–	–	–	–
Steering Wheel and Column Assembly	10	–	–	1	21	10	4	–	3	3	–	–
Dash	–	–	–	1	–	–	–	8	9	8	–	–
Parcel Tray	–	–	–	–	–	–	–	5	3	4	9	–
Gear Lever	–	–	–	–	–	–	–	–	–	–	–	–
Heater	–	–	–	–	–	–	–	–	–	–	–	–
Bulkhead or Toeboard Intrusion	–	–	–	–	–	–	–	3	–	–	8	9
Seat Back	–	–	–	–	–	–	–	–	–	–	–	–
A Post Intrusion	–	–	–	–	–	–	–	–	5	–	–	–
External Object	3	–	–	–	–	–	–	–	–	–	–	–

Table 5
INTERIOR OR EXTERIOR STRUCTURES CAUSING SEVERE INJURY TO DIFFERENT REGIONS OF THE BODY IN FRONTAL IMPACTS AMONG 154 UNBELTED FRONT SEAT PASSENGERS AT RISK

Structure Causing Injury \ Region of Body	Head	Neck	Back (Spine)	Upper Limb	Chest	Abdomen	Pelvis	Hip-Joint	Thigh	Knee	Lower Leg	Ankle and Foot
Roof	–	–	–	–	–	–	–	–	–	–	–	–
Windshield Pillar or Header Rail	–	–	–	–	–	–	–	–	–	–	–	–
Windshield (or Bonnet)	8(4)	1	–	–	–	–	–	–	–	–	–	–
Steering Wheel and Column Assembly	–	–	–	–	–	–	–	–	–	–	–	–
Dash	–	–	–	–	9	6	1	4	7	–	–	–
Parcel Tray	–	–	–	–	–	–	–	3	1	1	–	–
Gear Lever	–	–	–	–	–	–	–	–	–	–	–	–
Heater	–	–	–	–	–	–	–	3	–	–	–	–
Bulkhead or Toeboard Intrusion	–	–	–	–	–	–	–	–	1	1	3	1
Seat Back	–	–	–	–	–	–	–	–	–	–	–	–
A Post Intrusion	–	–	–	–	–	–	1	–	4	–	–	–
External Object	–	–	1	–	–	–	–	–	–	–	–	–

Table 6
INTERIOR OR EXTERIOR STRUCTURES CAUSING SEVERE INJURY TO DIFFERENT REGIONS OF THE BODY ON SIDE IMPACTS AMONG 66 UNBELTED DRIVERS AT RISK

Structure Causing Injury \ Region of Body	Head	Neck	Back (Spine)	Upper Limb	Chest	Abdomen	Pelvis	Hip-Joint	Thigh	Knee	Lower Leg	Ankle and Foot
Roof	2	–	1	–	–	–	–	–	–	–	–	–
Windshield Pillar or Header Rail	2	–	–	–	–	–	–	–	–	–	–	–
Windshield	–	–	–	1	–	–	–	–	–	–	–	–
Steering Wheel and Column Assembly	–	–	–	–	–	–	–	–	1	–	–	–
Dash	1	–	–	–	1	1	–	–	1	1	–	–
Parcel Tray	–	–	–	–	–	–	–	–	–	–	–	–
Gear Lever	–	–	–	–	–	–	–	–	1	–	–	–
Heater	–	–	–	–	–	–	–	–	–	–	1	–
Bulkhead or Toeboard Intrusion	–	–	–	–	–	–	1	1	1	–	–	–
Seat Back	–	–	–	–	1	–	–	–	–	–	–	–
Side Structure Intrusion	–	2	–	–	6	1	2	1	4	–	1	–
External Object	1	–	–	–	–	–	–	–	–	–	–	–

Of 768 unbelted occupants 66 were fatally injured. In frontal impact accidents there were 48 fatalities. The causes of death are shown in Table 10. The component most commonly causing fatal injury among drivers was the steering wheel and column assembly, the resulting injury being to the chest or to the abdomen, and among front seat passengers the facia panel, the injury being to the chest. Among 7 rear seat passengers dying there was no leading cause of fatal injury. In side impacts there were 12 fatalities. Intrusion of the side structure of the vehicle was the most common cause of fatal injury among drivers, the resulting injury being to the chest or abdomen, as it was also among front seat passengers. There were no fatalities amongst the rear seat passengers in side impact accidents. In overturning accidents there were six fatalities, in only two of which, both drivers, was it possible to determine the

Table 7
INTERIOR OR EXTERIOR STRUCTURES CAUSING SEVERE INJURY TO DIFFERENT REGIONS OF THE BODY IN SIDE IMPACTS AMONG 34 UNBELTED FRONT SEAT PASSENGERS AT RISK

Structure Causing Injury \ Region of Body	Head	Neck	Back (Spine)	Upper Limb	Chest	Abdomen	Pelvis	Hip-Joint	Thigh	Knee	Lower Leg	Ankle and Foot
Roof	–	–	–	–	–	–	–	–	–	–	–	–
Windshield Pillar or Header Rail	2	–	–	–	–	–	–	–	–	–	–	–
Windshield	–	–	–	–	–	–	–	–	–	–	–	–
Steering Wheel and Column Assembly	–	–	–	–	–	–	–	–	1	–	–	–
Dash	–	–	–	–	–	–	–	–	–	–	–	–
Parcel Tray	–	–	–	–	–	–	–	–	–	–	–	–
Gear Lever	–	–	–	–	–	–	–	–	–	–	–	–
Heater	–	–	–	–	–	–	–	–	–	–	–	–
Bulkhead of Toeboard Intrusion	–	–	–	–	–	–	–	–	–	–	–	–
Seat Back	–	–	–	–	–	–	1	–	1	–	–	–
Side Structure Intrusion	–	–	–	1	3	2	3	–	1	1	1	–
External Object	1	1	–	–	–	–	–	–	–	–	–	–

Table 8
INTERIOR OR EXTERIOR STRUCTURES CAUSING SEVERE INJURY TO DIFFERENT REGIONS OF THE BODY IN OVERTURNING ACCIDENTS AMONG 29 UNBELTED DRIVERS AT RISK

Structure Causing Injury \ Region of Body	Head	Neck	Back (Spine)	Upper Limb	Chest	Abdomen	Pelvis	Hip-Joint	Thigh	Knee	Lower Leg	Ankle and Foot
Roof	1	–	–	–	–	–	–	–	–	–	–	–
Windshield Pillar or Header Rail	–	–	–	–	–	–	–	–	–	–	–	–
Windshield	–	–	–	–	–	–	–	–	–	–	–	–
Steering Wheel and Column Assembly	–	–	–	–	–	–	–	–	–	–	–	–
Dash	–	–	–	–	–	–	–	–	–	–	–	–
Parcel Tray	–	–	–	–	–	–	–	–	–	–	–	–
Gear Lever	–	–	–	–	–	–	–	–	–	–	–	–
Heater	–	–	–	–	–	–	–	–	–	–	–	–
Bulkhead or Toeboard Intrusion	–	–	–	–	–	–	–	–	–	–	–	–
Seat Back	–	–	–	–	–	–	–	–	–	–	–	–
Side Structure	–	1	–	–	–	–	–	–	–	–	–	–
External Object	–	–	–	–	–	–	–	–	–	–	–	–

Table 9
INTERIOR OR EXTERIOR STRUCTURES CAUSING SEVERE INJURY TO DIFFERENT REGIONS OF THE BODY IN OVERTURNING ACCIDENTS AMONG 21 UNBELTED FRONT SEAT PASSENGERS AT RISK

Structure Causing Injury / Region of Body	Head	Neck	Back (Spine)	Upper Limb	Chest	Abdomen	Pelvis	Hip-Joint	Thigh	Knee	Lower Leg	Ankle and Foot
Roof	–	1	2	–	–	–	–	–	–	–	–	–
Windshield Pillar or Header Rail	–	–	–	–	–	–	–	–	–	–	–	–
Windshield	–	–	–	–	–	–	–	–	–	–	–	–
Steering Wheel and Column Assembly	–	–	–	–	–	–	–	–	–	–	–	–
Dash	–	–	–	–	–	–	–	–	–	–	–	–
Parcel Tray	–	–	–	–	–	–	–	–	–	–	–	–
Gear Lever	–	–	–	–	–	–	–	–	–	–	–	–
Heater	–	–	–	–	–	–	–	–	–	–	–	–
Bulkhead or Toeboard Intrusion	–	–	–	–	–	–	–	–	–	–	–	–
Seat Back	–	–	–	–	–	–	–	–	–	–	–	–
Side Structure	–	–	–	–	1	–	–	–	–	–	–	–
External Object	–	–	–	–	–	–	–	–	–	–	–	–

Table 10
INTERIOR OR EXTERIOR STRUCTURES CAUSING FATAL INJURY TO DIFFERENT REGIONS OF THE BODY AMONG 56 FATALLY INJURED UNBELTED FRONT SEAT OCCUPANTS, FOR DIFFERENT IMPACT DIRECTIONS

Direction of Impact and Structure Causing Injury / Category of Occupant and Region of Body	DRIVERS				FRONT SEAT PASSENGERS			
	Head	Neck	Chest	Abdomen	Head	Neck	Chest	Abdomen
FRONTAL								
Header Rail	–	–	–	–	2	–	–	–
Hood	–	1	–	–	1	–	–	–
Steering Wheel and Column Assembly	–	1	15	7	–	–	–	–
Dash	–	–	–	–	–	–	6	1
A Post Intrusion	–	–	2	2	–	–	–	–
External Object	2	–	–	–	2	–	–	–
Other Structures	–	–	–	–	1	1	–	–
SIDE								
Roof	2	1	–	–	1	1	–	–
Side Structure Intrusion	–	–	3	2	–	–	2	1
Steering Wheel and Column Assembly	–	–	–	–	–	–	1	1
External Object	1	–	–	–	2	–	1	–
OVERTURN								
Steering Wheel and Column Assembly	–	–	–	1	–	–	–	–
Other Structures	1	–	–	–	–	–	–	–
REAR	No Fatal Injuries							

cause of fatal injury. As already noted in the rear impact collisions there were no fatalities.

SAFETY BELTS AND INJURY

It is of interest to compare the incidence of different severities of injury among the unbelted and belted front seat occupants. There were 422 unbelted and 96 belted front seat occupants who were involved in frontal impact collisions; the injury data are given in Table 11. It will be seen that the greatest reduction in injury was in the fatal category; the smaller reduction in other injury categories was presumably because most of those occupants who would otherwise have been killed survived with lesser injuries and because the same process of down-grading of severity of injury received occurred throughout the whole range of injury severities. The proportion of those occupants with no injury was about three times greater among the belted than among the unbelted occupants in this sample of frontal impact accidents.

If the frontal impact accidents are broken down to show the incidence of severe or fatal injury for the different regions of the body, Table 12, it will be noticed as would be expected that among the 518 front seat occupants the chances of receiving severe or fatal injury for most regions of the body were lower among the belted occupants than among the unbelted. However the lower leg and ankle or foot appear to have been more frequently injured in the belted occupants. It will also be noticed among the belted drivers that there was no reduction in head injury, but that there was a considerable reduction in head injury among belted front seat passengers. About half the head injuries in the belted drivers were due to the head hitting the steering wheel, which suggests that the steering wheel is a hazard to the head even when the driver is restrained.

Table 11
PERCENTAGE OF UNBELTED AND BELTED FRONT SEAT OCCUPANTS RECEIVING INJURY, BY SEVERITY OF INJURY. FRONTAL IMPACTS ONLY.

Severity of Injury \ Category of Occupant	Unbelted Front Seat Occupants (422) %	Belted Front Seat Occupants (96) %
No Injury	7	23
Minor Injury	22	16
Moderate Injury	26	22
Severe Injury	35	33
Fatal Injury	10	6

CONCLUSIONS

A number of conclusions may be drawn from the injury data presented in this paper, all of which are related to safety design requirements for passenger cars. Some of the more important findings are summarized briefly.

Table 12
PERCENTAGE OF BELTED AND UNBELTED OCCUPANTS WITH SEVERE OR FATAL INJURIES BY REGIONS OF THE BODY AMONG 518 FRONT SEAT OCCUPANTS AT RISK INVOLVED IN FRONTAL IMPACT ACCIDENTS

Category of Occupant \ Region of Body	Head %	Neck %	Back (Spine) %	Upper Limb %	Chest %	Abdomen %	Pelvis %	Hip %	Thigh %	Knee %	Lower Leg %	Ankle and Foot %
BELTED												
Drivers (65)	14	2	–	5	12	3	–	3	8	3	12	6
Front Seat Passengers (31)	6	–	–	–	13	–	–	–	6	3	6	6
All Front Seat Occupants (96)	11	1	–	3	13	3	–	2	7	3	10	6
UNBELTED												
Drivers (268)	15	1	1	4	16	8	2	7	11	7	6	3
Front Seat Passengers (154)	23	2	–	3	12	5	2	6	5	1	2	2
All Front Seat Occupants (422)	18	2	1	4	15	7	2	7	9	5	5	3

It has been shown that unbelted rear seat occupants were much less frequently injured than unbelted front seat occupants; also in frontal impact accidents the leading severe injury to unbelted drivers was fracture of the femur, whereas in unbelted front seat passengers the region most commonly sustaining severe injury was the head, but the risk of chest injury was greatest in drivers. In side impacts the incidence of fracture of the pelvis was higher than in frontal impacts and in overturning accidents the risk of receiving a fracture of the spine was greater than in other forms of collision. In rear impact accidents no severe or fatal injuries were seen.

Among the unbelted fatalities the region of the body most commonly receiving fatal injury in drivers in frontal impact accidents was the chest; in other seating positions and for other types of accidents the chest and the head received fatal injury in about equal proportions.

The various structures of the vehicle causing severe injury to the different regions of the body are tabulated in detail in Tables 4-9. It will be noted that in frontal impacts severe injury to the head in unbelted drivers was most commonly caused by the steering wheel or the windshield pillar or header rail, and that severe injury to the chest or abdomen was most commonly caused by the steering wheel. Lower limb fractures or dislocations were most frequently caused by knee impact on the facia for the knee or for the hip joint, but by angulation of the thigh over the lower edge of the facia for the femur, by impact of the lower leg on the parcel tray for the tibia and fibula and by bulkhead or toeboard intrusion for the lower leg or the ankle and foot. Among unbelted front seat passengers the facia was the leading cause of severe injury for the chest, the abdomen, the hip joint and the thigh, and the windshield was the leading cause for the head.

In side impacts whether for drivers or for passengers, the main cause of severe injury to most parts of the body was intrusion of the side structures of the vehicle. In overturning accidents the leading cause of injury to the spine was the roof, usually with roof intrusion. There were no severe injuries in the rear impact accidents in the sample.

The structure most commonly causing fatal injury to unbelted drivers in frontal impacts was the steering wheel and column assembly and to unbelted front seat passengers the facia panel. In side impacts intrusion of the side structures of the vehicle was the principal cause of death for all occupant positions.

Comparison of the injuries sustained by the unbelted and belted front seat occupants in the sample shows that there was a considerable reduction in injury among the belted occupants chiefly in the fatal category, and that when the injuries sustained were broken down by regions of the body it is to be noted that the incidence of severe or fatal head injury in drivers was not reduced, as was the case in front seat passengers, probably because in half the belted drivers the head made contact with the steering wheel.

The patterns and causes of injury which have been discussed briefly in this paper are of value not only as an indication of where protection is most required but they form an essential part of the experimental studies on human tolerance to impact, some aspects of which are considered in a separate paper.

REFERENCES

1. W. Gissane and J.P. Bull, "Fourth Annual Progress Report," Birmingham Accident Hospital: Road Injuries Research Group, 1964.
2. Transport and Road Research Laboratory, "The Classification of Injuries Sustained in Road Accidents." Leaflet No. 130 Issue 3. Crowthorne: Transport and Road Research Laboratory, 1974.

ACKNOWLEDGEMENTS

The work described in this paper forms part of the program of the Transport and Road Research Laboratory, and the paper is published by permission of the Director. We would like to record our thanks to numerous colleagues at the Transport and Road Research Laboratory and in particular to Mrs. M. Keigan for preparing the data on which this paper is based.

ADDENDUM

It should be noted that Tables 4-9 show only the most common cause or causes of severe (or in Table 10 of fatal) injury for the different regions of the body, and that injuries due to ejection are not included in them.

SECTION 3

TECHNICAL SEMINARS

PART TWO – HUMAN FACTORS

Chairman: B. ALDMAN
Department of Traffic Safety
Chalmers University of Technology
Goteborg, Sweden

Themes: Human Tolerance Levels to Injury;
Use of Dummies and Similar Test Devices;
Impact Testing and Accident Injuries

Introduction by the Chairman:

I am privileged to serve as Chairman of the Technical Session on "Human Tolerance to Injury, the Use of Dummies and Similar Test Devices, Impact Testing, and Accident Injuries." It has been mentioned in several presentations that there is a lack of data on human tolerance and biomechanics of the human body during accident situations. We have, I think, a rather complex problem. First, there is the problem of really describing the accident situations in enough detail to describe the input to the human body. Then, the human body is not a standardized mechanical system. It appears in different sizes with different component combinations with different material characteristics, and so on, adding up to what results in biological variations. So far, we have been guided by some simple figures on whole body impact tolerances and tolerance indices which, at this early stage of development of restraints, have served quite well for the basic design of systems where a transfer of kinetic energy from the human body to the vehicle structure begins early in the accident sequence and continues throughout the entire accident.

The application of this simple principle has influenced the injury rate to such an extent that we are now asking for refinement where the scientist can calculate, with a certain probability of injury to the population, the risk, be it car occupants, or other road user categories, in case of different accident situations. There is no single simple way of obtaining such data.

We will hear today about in depth accident investigation tests with different types of human substitutes, an effort to extrapolate from various sources; information that will eventually result in the knowledge necessary to define the requirements for a future production vehicle.

NHTSA PROGRAMS IN BIOMECHANICS

ARTHUR E. HIRSCH
Acting Head, Biomechanics
Office of Vehicle Safety Research
NHTSA

The biomechanics program at NHTSA is focused on problems involving protection in the crash environment. Our approach to the solution to these problems is through the generation of "Crash Protection Standards," and "Protection System Concepts." Biomechanical input to the development of standards compliance test procedures is through discovery of human crash injury tolerance levels and development of anthropomorphic dummies and instrumentation. Protection system concepts are tested for feasibility through experimentation and computer model exercise.

Our tolerance program involves a variety of projects, both experimental and analytical in nature, including, at present, studies of head, neck, chest, and leg injury.

Head impact and whiplash studies conducted by one of our sponsored laboratories have shown that head rotation and skull deformation are significantly related to head injury.[1]* A second study[2] has examined the relationship between the occurrence of skull fracture, and the velocity and curvature of the impactor. A serious deficiency in our studies of the relationship between brain concussion and crash intensity has been the lack of precise techniques for detecting the onset and measuring the severity of the concussion. Our researchers[3] are investigating the application of the "Somato-Sensory Evoked Potential" technique to this problem. A sensory stimulus is generated in some predetermined sequence, such as a continuous series of finger pinches, and the electrical response pattern of the brain is monitored. This input pattern is continued during the test and post-test. Changes in the response pattern and time of return to the baseline situation may be used as indicators of the quality and quantity of neurological deficit.

Other current work on head injury consists of four associated studies – two experimental and two analytical. The experimental studies involve heavily instrumented primates which undergo head impact in a variety of modes[4] and nonimpact whiplash type head motion.[5] The data from these studies are being used for validation of a finite element model of the head and brain[6] currently under construction, and as input to a nonlinear multivariate analysis computer program[7] which is capable of digesting large masses of test data and synthesizing an empirical injury predictive model. This model has the potential of continuous refinement and validation as additional data is fed in.

Experimental studies of chest injury are being initiated[8] as an extension of the cadaver impact work conducted by Kroell et al.,[9] where fresh cadavers were struck in the chest and the response of the thoracic cage recorded. The data output will be subject to the same scrutiny under the nonlinear multivariate analysis as the head data, in hopes of developing one single quantitative predictor of impact injury.

Experiments using fresh cadavers are underway to determine the relationship between knee impact loading and knee, femoral, and hip injury. This work is being conducted in direct support of Federal Motor Vehicle Standard 208.

In regard to analytical studies bearing on tolerance, in addition to the finite element model of the head and the multivariate analysis we are supporting a finite element and lumped parameter model of the thoracic region.[11] In conjunction with this modeling effort, there is an experimental program[11] which is aimed at model validation, and another one which is attempting to determine the constitutive properties of the chest tissue.[12] The model validation experiments involve a comparison of actual chest responses to a blow with those predicted by model exercise. The constitutive properties of the

*Note: Superscript numbers refer to reports, ongoing work under DOT contract or work which will be initiated under DOT contract by June 30, 1974.

chest tissue are required for insertion of real-life properties into the model.

As has been stated, a second major element in our program in support of "Crash Protection Standards" preparation is the development of human simulation for "Standards Compliance Testing." If the testing for compliance of a protection system must be dependent upon the performance of a human surrogate during a specified crash with the system, it is necessary not only to specify injury level in some measurable terms, but also to have a human simulator whose responses will reliably and validly provide these measures. Our efforts in this area, as in tolerance, include experimental and analytical studies. The experimental efforts are aimed at developing techniques for obtaining, in a nondestructive manner, and on a large sampling of the population, those measurements of the family of man which are required for simulation. These measures are not available in the anthropometric literature since we require a description of man as a dynamic-mechanical system rather than as a space occupier or something to clothe. Our analytical studies consist largely of development and exercise of a large three spatial dimension computer model of a man and portions of his immediate crash environment.[13] In its present stage of development the model is almost ready for parametric sensitivity exercises in the vehicle occupant, pedestrian, and motorcyclist modes. Validation studies are currently underway in these three areas.

As an adjunct to compliance test procedures, but also applicable to tolerance and other biomechanical pursuits, we are currently conducting studies which are leading to the development of several useful measurement tools. The first of these is an interface pressure gauge[16] which will enable us to measure the pressure as a function of time, between two impacting surfaces, for a variety of shapes and compliance characteristics.

We also have under development an accelerometer which will measure the six rigid body motions and will be small enough to be imbedded in a mouthpiece. In addition, we are planning a companion microminiaturized transmitter to be placed in the mouth, which will permit us to telemeter the data output. It is expected that, if these plans are realized, we will be able to measure head motions in boxers and obtain head injury data on living subjects hitherto unobtainable.

The third major element in our program is "Protection System Concepts" feasibility studies. Under this element we are conducting crash tests with a variety of restraint systems, passive and active, in realistic types of crash situations. Our test subjects are, in the main, living and cadaver humans. In one study we have subjected volunteers to frontal barrier crashes of 30 miles per hour, using air bags[18] and passive energy absorbing belt systems.[19] We are also subjecting fresh cadavers to barrier impacts of up to 40 mph in standard lap-shoulder belt systems. The report on the results of these tests will be available shortly.[20]

A detailed, heavily instrumented study is underway to measure the contact loading on a cadaver pedestrian associated with an effort to provide data for a pedestrian model validation.[21]

We have also conducted[22] and are continuing[23] to conduct studies which lead us to believe that an air bag protective system adapted for a motorcycle will be of considerable value in effecting a reduction in death and injury in that mode.

Finally we are developing computer models which, when validated, will permit us to do feasibility studies by computer exercise. There are models of the crash victim in the vehicle,[13] the pedestrian-vehicle impact,[14] and the motorcycle-rider crash[15,22] under preparation or in validation stage.

In summary, the NHTSA Biomechanical Studies Group has under contract or in negotiation more than thirty-two separate projects directly bearing on the area of human tolerance, human simulation, and protection systems. We expect that we will, within a reasonable period, be able to provide valid injury criteria for all the significant body areas, specifications for a valid, reliable family of anthropomorphic dummies, computer tools for assistance in the design and evaluation of advanced protection concepts, and ultimately perhaps, computer programs which will be employed for standards compliance testing.

References and Footnotes

1. "Tolerances for Cerebral Concussion from Head Impact and Whiplash in Primates," Journal of Biomechanics, Vol. 4, pp. 13-21, Pergamon Press, 1971, by Ommaya, A.D. and Hirsch, A.E.

2. "Breaking Strength of the Human Skull vs. Impact Surface Curvature," Report DOT-HS-800-583 on Contract DOT-FH-11-7609, June 1971, Wayne State University.

3. "Head Injury Mechanisms," Report DOT-HS-800-959 on Contract DOT-HS-081-1-106 1A. October 1973, National Institutes of Health.

4. "Validation Studies for Head Impact Injury Model," Contract DOT-HS-031-3-749, Highway Safety Research Institute. (Report has not been published yet.)

5. "Validation Study for Whiplash Injury," Contract DOT-HS-168-3-587, National Institutes of Health. (Report has not been published yet.)

6. "Finite Element Model of Head and Neck," Contract DOT-HS-289-3-550 IA, Naval Civil Engineering Laboratory. (Report has not been published yet.)

7. "Nonlinear Multivariate Modeling of Head Injury," Contract DOT-HS-4-00882, Adaptronics, Inc. (Report has not been published yet.)

8. "Quantification of Thoracic Response and Injury," Contractor to be selected.

9. Kroell, Charles K., et al., "Impact Tolerance and Response of the Human Thorax," paper 710851, Fifteenth Stapp Car Crash Conference, November 1971, Society of Auto. Engineers, Inc., Two Pennsylvania Plaza, New York, N.Y. 10001.

10. "Knee and Hip Study," Contract DOT-HS-081-1-106 Mod 2, West Virginia University. (Report has not been published yet.)

11. "Thoracic Impact Injury Mechanism," Contract DOT-HS-243-2-424, Franklin Institute. (Report has not been published yet.)

12. "Thoracic Model Improvement: Experimental Tissue Properties," Contract DOT-HS-031-3-763, Highway Safety Research Institute of Michigan. (Report has not been published yet.)

13. "Improvements in Crash Victim Gross Motion Simulation," Contract DOT-HS-053-2-485, Cornell Aeronautical Laboratory (Calspan). (Report has not been published yet.)

14. "Pedestrian Model Parametric Studies," Contract DOT-HS-356-3-719, Boeing. (Report has not been published yet.)

15. "Dynamics of Motorcycle Impact," Contract DOT-HS-126-3-643, Denver Research Institute. (Report has not been published yet.)

16. "Piezoelectric Contact Pressure Gauge" Contract DOT-HS-4-00835 IA, National Bureau of Standards. (Report has not been published yet.)

17. "Biomechanical Instrumentation and Model Validation," DOT-PPA-HS-410, Transportation Systems Center. (Report has not been published yet.)

18. "Impact Dynamics; Air Bags," Contract DOT-HS-017-1-017 IA, Wright-Patterson Air Force Base. (Report has not been published yet.)

19. "Impact Biodynamics Passive Belts," Contract DOT-HS-063-1-081 IA, Naval Air Development Center. (Report has not been published yet.)

20. "Kinematics and Impact Injury of Belted Occupants," Contract DOT-HS-146-3-753, Wayne State University. (Report has not been published yet.)

21. "Contact Loads Duration; Experimental Study," Contract DOT-HS-146-3-711, Wayne State University. (Report has not been published yet.)

22. "Dynamics of Motorcycle Impact," Final 3 vol. report DOT-HS-800-586,587,588, on Contract FY-11-7307, July 1971, Denver Research Institute.

23. "Dynamics of Motorcycle Impact 1971-1973," final 3 vol. report, DOT-HS-800-588-906,907,908, on Contract DOT-HS-126-1-186, September 1973.

GM-ATD 502 ANTHROPOMORPHIC DUMMY – DEVELOPMENT AND EVALUATION

J. A. TENNANT, R. H. JENSEN,
R. A. POTTER
Advance Product Engineering
General Motors Engineering Staff

INTRODUCTION

There is a critical need in the safety community for an anthropomorphic test dummy that will provide meaningful human injury assessments in simulated automobile crashes. A collateral need is for the development of repeatable testing procedures and for dummy features that enhance the accuracy of the initial test setup.

In response to this need a new anthropomorphic test dummy system (GM-ATD 502) was developed by General Motors in a joint program with the National Highway Traffic Safety Administration. Two of the resulting test dummies (Figure 1) have been delivered to NHTSA along with detailed manufacturing drawings and a final report. This paper summarizes the design, development, and performance of the test device. For additional details, the reader is referred to the final report (1)* delivered to NHTSA covering this program.

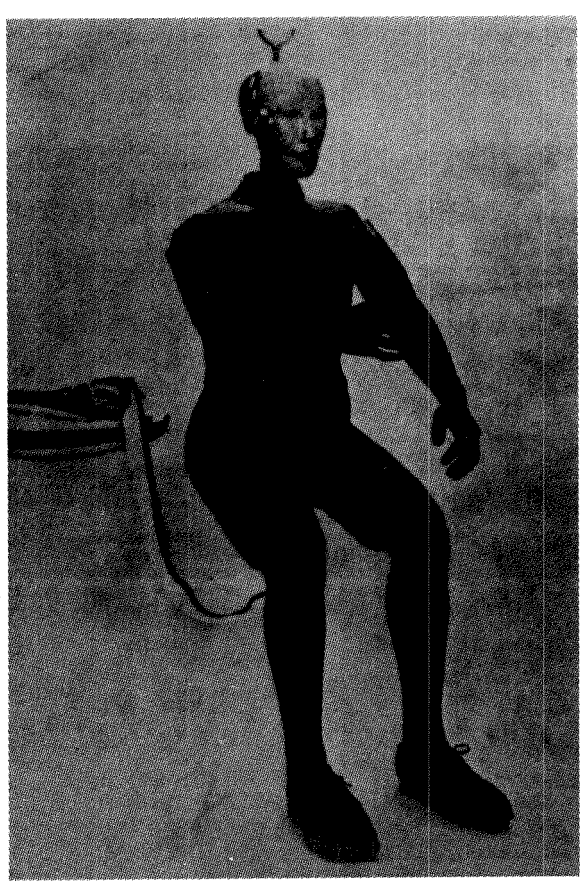


Figure 1

The goals of the program in order of priority are summarized as follows:

- Repeatability and reproducibility of the assembled dummy performance
- Development of repeatable testing procedures.
- Ease and accuracy of dummy setup to initial test position.
- Anthropometric and biomechanical fidelity.
- Durability and maintainability
- Quality of instrumentation.
- Cost of manufacturing.

This order of priority was used to resolve conflicts arising in meeting the respective goals. In particular, conflicts arising between repeatability and biomechanical fidelity were resolved in favor of the former. This position was taken because biomechanical fidelity cannot be documented unless satisfactory repeatability is achieved. In regard to our main objective of performance repeatability and reproducibility for the assembled dummy, it should be observed that hardseat test results on early prototype dummies showed coefficients of variation** as high as ten percent for head and chest S.I. This level of repeatability was also typical of commercially available test dummies. It was concluded that a reasonable goal for the GM-ATD 502 would be a coefficient of variation of less than five percent for each response measured.

In the past, dummies have been designed to the anthropometric "erect" seated position of SAE J963. However, dummies are used in the "automotive seated position" rather than this erect position. The GM-ATD 502 has been designed for the automotive seated position but can be made to assume the erect position for anthropometric measurement purposes by means of the adjustable neck bracket and by straightening the curved lumbar (Figure 2). This dummy departs from previous practice in that it can be disassembled for measurement of segment weight and center-of-gravity locations rather than sectioned or sawed at non-separable planes as implied by SAE J963. The disassembly procedure for this dummy corresponds closely with an anatomically based segmenting scheme, and the inertial properties specified for the dummy subassemblies are based on

*Numbers in parentheses designate References at end of paper.
**See footnote under "Test Results.."

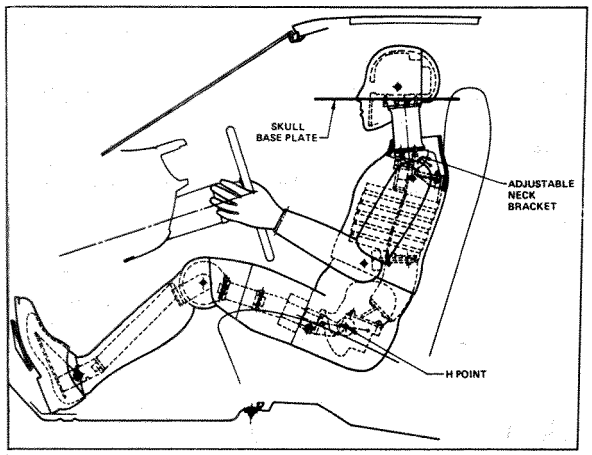


Figure 2

available anthropometric data interpreted with reference to this segmenting scheme. This approach allows subassembly weight and center of gravity location to be controlled more accurately, and it allows the user to check these properties whenever it is desired. Also, location for both the head and chest accelerometer packages differ from those specified by SAE J963. In the head, the accelerometers are located at the center of gravity of the head alone (without the 2/3 of the neck as called for in J963). Accelerometers in the chest are located at the center of gravity of the "essentially rigid thorax" rather than at the center of gravity of the shoulders and 1/3 neck, as formerly used.

Analytical models of the dummy in a crash environment were employed to study the sensitivity of the dummy response to changes in joint resistances, link lengths, and inertial properties. This information was then used to aid the selection of design tolerances for the dummy components.

A further design goal was durability and ease of maintenance. Up to twenty tests on a single prototype were conducted without serious malfunction. Commonization of parts (e.g., elimination of right- and left-hand components) was maximized to assist maintainability. Parts are well documented with drawings and material specifications which provide for efficient manufacture of this test device. All parts except the vinyl head skin and molded polyacrylate neck and lumbar spine were manufactured by outside vendors.

FEATURES AND CONSTRUCTION

The main subsystems comprising the dummy are described in this section.

Head

The skull of the GM-ATD 502 (Figure 3) features more human like geometry (2) than previous state-of-the-art dummies and is a precision aluminum casting which is easy to manufacture and has uniform wall thickness and symmetry.

The thickness of the vinyl skin has been specified and closely controlled to assure biomechanical fidelity and repeatability head response in hard surface impact. The skin fits well to the skull contour, and retainers are used to prevent the skin from moving relative to the skull.

The head assembly is ballasted to attain the design weight, center-of-gravity location and mass moment of inertia about a lateral axis through the center of gravity. An adaptor plate simplifies neck-to-head attachment.

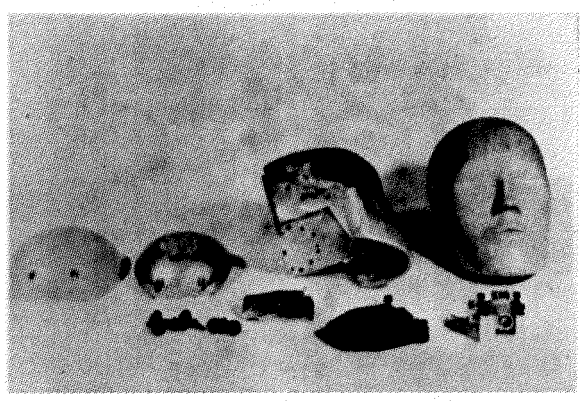


Figure 3

Neck

The neck proper (Figure 4) is composed of a polyacrylate elastomer. This material provides repeatable response and is durable. Integrally molded end plates provide the attachment to the head and to the neck bracket.

The neck bracket, which provides the neck-to-spine attachment, can be adjusted to maintain the base plane of the head horizontal for current vehicle seat back angles. It is made of cast aluminum for weight control and ease of manufacture and it has a positive locking adjustment.

Shoulders

The shoulder assembly (Figure 5) forms an articulated linkage between the spine and arms. The elevation/depression clavicular link motion is controlled by a rubber block in conjunction with a Delrin pivot. Anterior-posterior excursions are

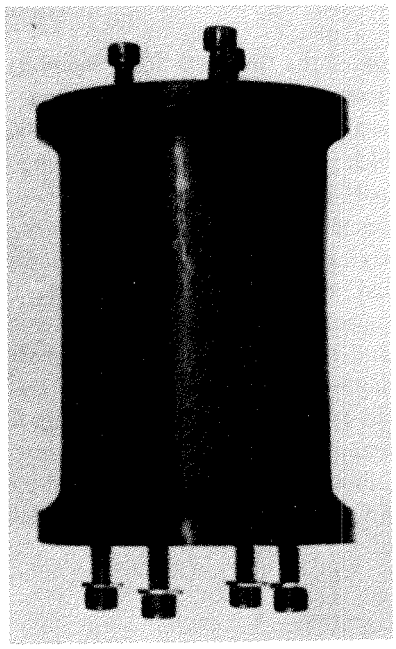


Figure 4

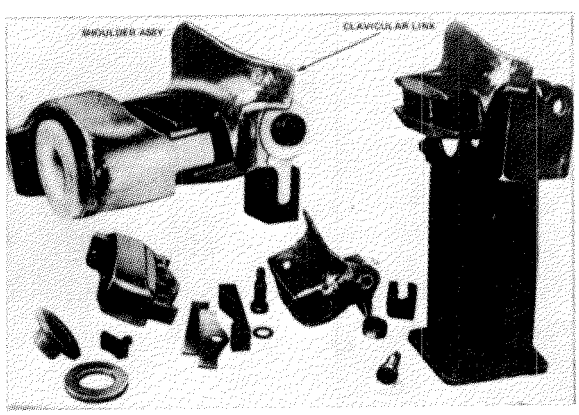


Figure 5

controlled by a cast urethane member for self-centering. Upper arm flexion involves joint bushings, and cushioned rubber stops limit travel and prevent metal-to-metal contact. The cast aluminum clavicular link and clavicle are designed for weight control, durability and the maximum expected loads.

The whole assembly is smoothly contoured to provide a good belt-to-shoulder interface. A neck flange assures a repeatable shoulder belt location and minimizes incidental belt damage and torso jacket tearing. This was a problem area in previous dummy designs.

Thorax

The thorax assembly (Figures 6 and 7) consists of a spine and rib cage covered by a removable chest jacket. The whole assembly is ballasted for accurate weight and center of gravity location control.

The thoracic spine, of welded steel construction, has the following structural functions. It provides a neck bracket attachment, the clavicular link pivot, and the rib attachment support and location. A lumbar spine attachment is also provided. Finally, it houses the triaxial accelerometer instrumentation block for the chest.

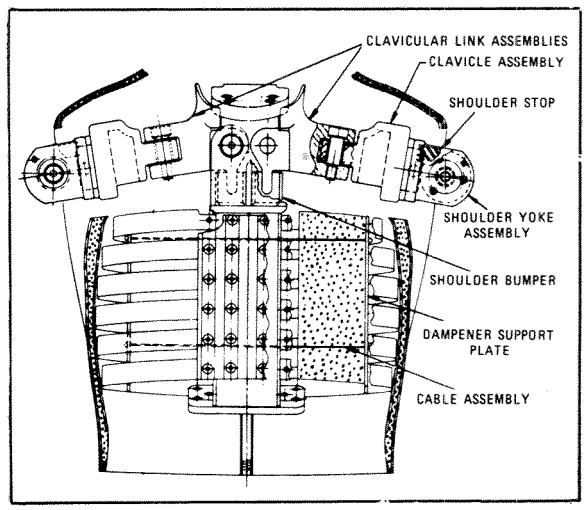


Figure 6

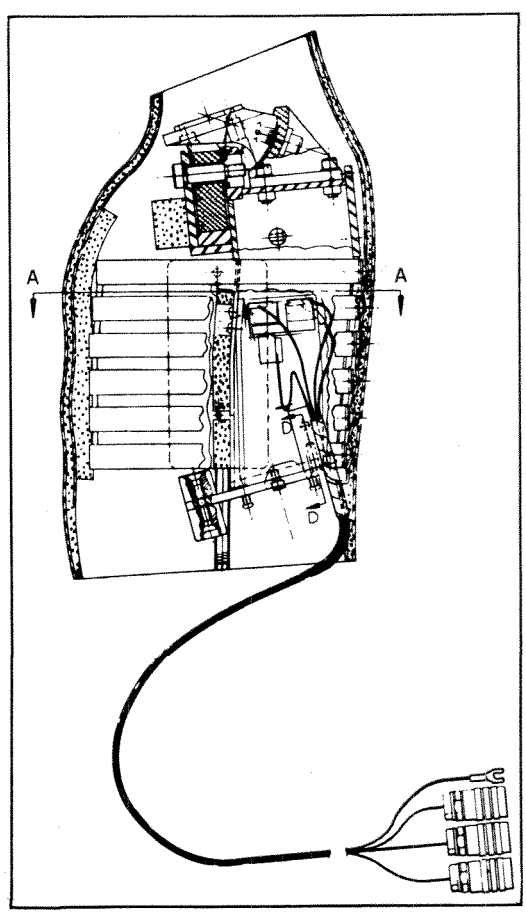


Figure 7

The rib-cage assembly is designed for maximum practicable dynamic deflection. It consists of steel ribs attached to the rear of the spine, with helper leaves for stress control. The ribs are contoured to simulate the human form and are backed up by polyethylene foam damper assemblies. These dampers eliminate spurious vibrations and provide vertical rigidity and buckling control. An aluminum sternum completes the rib-cage assembly, and provides rigidity and durability.

Lumbar Spine

The lumbar spine is a molded, curved polyacrylate rubber member similar to the neck. The curved contour provides the more humanlike, in-car seating position. Molded-in end plates provide attachment means to the thoracic spine and pelvis. Dual cable stabilizers assure lateral seating control, yet still permit fore and aft flexibility.

Joints

The knee joint is typical of the new joints designed for the GM-ATD 502 (Figure 8). It consists of a cast aluminum kneecap with a steel clevis. Adjustable from 1 g to 3 g preloading, it maintains its setting over several tests, and constant torque is held at varying joint angle and velocity. The joint operates on a clutch principle. The clutch force is provided by a urethane spring held by a floating nut against Delrin pressure plates. The pivot bolt in the joint is isolated from rotational forces. The number of operating parts of this joint have been reduced from 49 to 7 compared to a current, commercially available dummy.

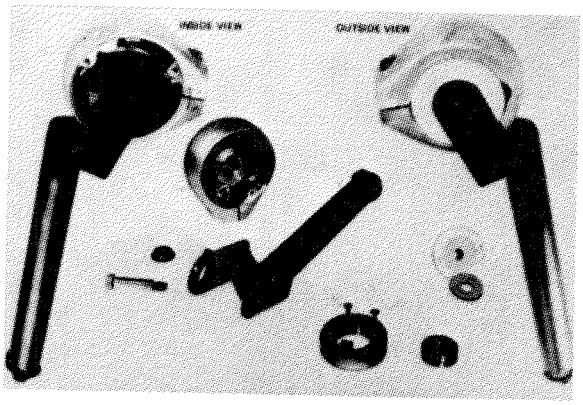


Figure 8

Other joints incorporating this new design are the elbow and arm/shoulder joints. Flesh contours have been modified to eliminate interference with joint motion.

ANTHROPOMETRY AND BIOMECHANICS

Background

If the test dummy is to be an indicator of the injury-producing potential of an automotive environment, it must be build with an appropriate degree of humanlike characteristics. These characteristics fall into the two categories of biomechanics (structure and performance) and anthropometrics (geometry, weights, and measurements). Human injury assessments are usually based on analyses of dummy head acceleration, chest acceleration, and femur loads. Maximizing the equivalence of human and dummy response for injury assessment requires the following:

- Knowledge of human response and the factors affecting that response.
- Characterization of human response into dummy specifications and performance requirements.
- Design and development of dummy systems to comply with these requirements.

Anthropometry

Anthropometry is the basis for humanlike crash-dummy design geometry. In addition to biomechanical fidelity, realistic results from a crash test depend on the dummy having humanlike shapes, dimensions and weights (anthropometry). The seating position, shape, location of joint centers, centers of gravity, and the weights of the various dummy segments are among the primary factors which determine the kinematics of the dummy in a crash test.

In the past, an "erect" seated position as shown in the SAE Recommended Practice J963 has been the standard seated position for anthropometric measurement of humans. A large amount of data on human measurements exists for this position, and has been previously used to specify and design crash test dummies. However, humans do not sit in a vehicle in this erect seated position; they sit in what will be referred to as the "automotive seated" position, that is with a slump. The placement of test dummies in automotive-type seats for crash simulations determines their initial position for that test and is an important factor which affects the responses measured in the dummy during the crash test. If these responses are to be meaningful, then the initial seating position of the dummy should be based on the configuration which humans assume in a similar seat (Figure 2).

However, the human is also capable of seating himself in an upright anthropometric position on hard surfaces, Yet, the test dummy can have but one natural position and must be forced to take the other.

Past dummies have been built to the upright anthropometric position and could not be forced into the automotive seated position. The GM-ATD 502 is built to be seated naturally in the automotive seated position and can be forced to take the upright anthropometric position. This change is consistent with the goals of improved ease of test setup, repeatability, and fidelity.

The human changes from a slouched seated position to an upright position by straightening the lumbar and cervical sections of his spine, and by rotating his pelvis. These changes appear to increase the overall height. This dummy has this slouch as its natural position. To go to the upright position with this dummy requires changes similar to those that occurred in the human. Specifically, the curved lumbar spine must be straightened, the pelvis is rotated forward, and the neck bracket is adjusted to bring the head up. These changes cause the dummy to "grow" to the erect seated height. It is important to note that the GM-ATD 502 has not been built shorter to be more compatible with automotive seated position of humans. Instead, it has been designed to obtain a natural humanlike automotive seating position.

To describe the typical position of a 50th percentile male person in an automotive environment is a difficult problem because very little data are available. It appears that the problem may best be solved by defining the position of the dummy with respect to the seat.

The position of the dummy with respect to the seat is probably best defined by relating the dummy H-point to the location of a human H-point in the same seat (3). The SAE J826a 3-D seating manikin ("H-Point Machine") is a tool which was based on human H-point location studies and which was designed as a tool to measure this point for automotive seat designers. Therefore, if the dummy sits with its H-point coincident with that indicated by the J826a 3-D manikin, the position of the lower body is humanlike as possible with current knowledge. The GM-ATD 502 dummies tend to sit in this manner.

It is also felt that the SAE J941b "eyellipse" is too broad to define exactly the location of a 50th percentile adult male because it includes both men and women drivers of a range of percentiles and includes seat travel.

The SAE J941b data has been reevaluated to locate the 50th percentile adult male eyepoints for various seat back angles. The GM-ATD 502 has been designed to sit with its "eye" location as close as possible to this point. However, a more definitive and rigorous analysis of this SAE J941b data is required in order to assure correct positioning of the 50th percentile eye point.

The external body dimensions specified in SAE Recommended Practice J963 are the ones used in this program (Table 1 and Figures 9 and 10). They are representative of the 50th percentile American adult male. To more completely and realistically define the

TABLE 1
EXTERIOR BODY DIMENSIONS

Dimensional Symbol	Description	Design Value* (Inches)	SAE J963 (Inches)
I	Shoulder – Elbow Length	14.4	14.1±0.3
J	Elbow Rest Height (Erect)	9.5	9.5±0.5
L	Popliteal Height	17.3	17.3±0.2
M	Knee Height (Sitting)	21.4	21.4±0.3
N	Buttock Popliteal Length	18.45	19.5±0.3
O	Chest Depth	8.8	9.0±0.4
P	Buttock Knee Length	23.2	23.3±0.3
Q	Thigh Clearance	5.9	5.7±0.3
R	Elbow – Fingertip Length	18.3	18.7±0.5
S	Foot Length	10.4	10.5±0.2
T	Head Length	7.7	7.7±0.2
U	Sitting Height (Erect)	35.7	35.7±0.5
V	Shoulder Breadth	18.3	17.9±0.4
W	Foot Breadth	4.0	3.8±0.3
X	Head Circumference	22.7	22.5±0.5
Y	Chest Circumference	38.0	37.7±1.0
Z	Waist Circumference (Sitting)	34.0	33.0±1.0
AA	Head Breadth	6.0	6.1±0.2
AE	Occiput to Z-Axis	1.3	–

*Referenced to the Erect Seated Position

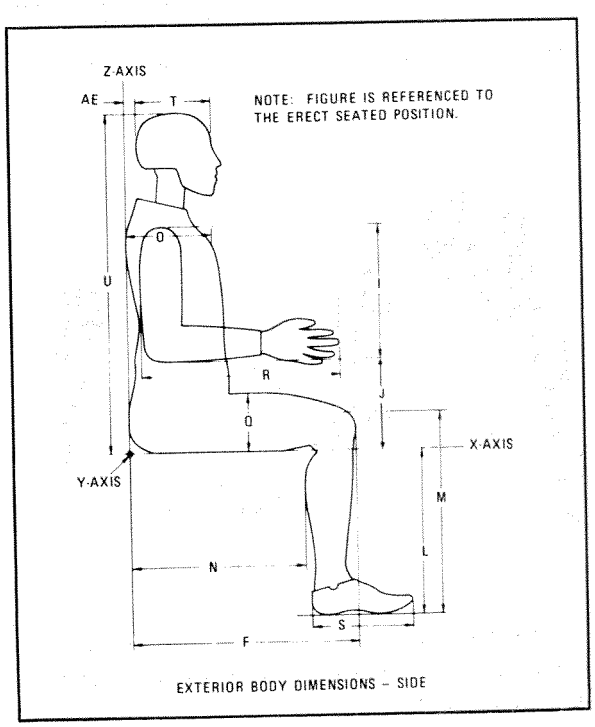


Figure 9

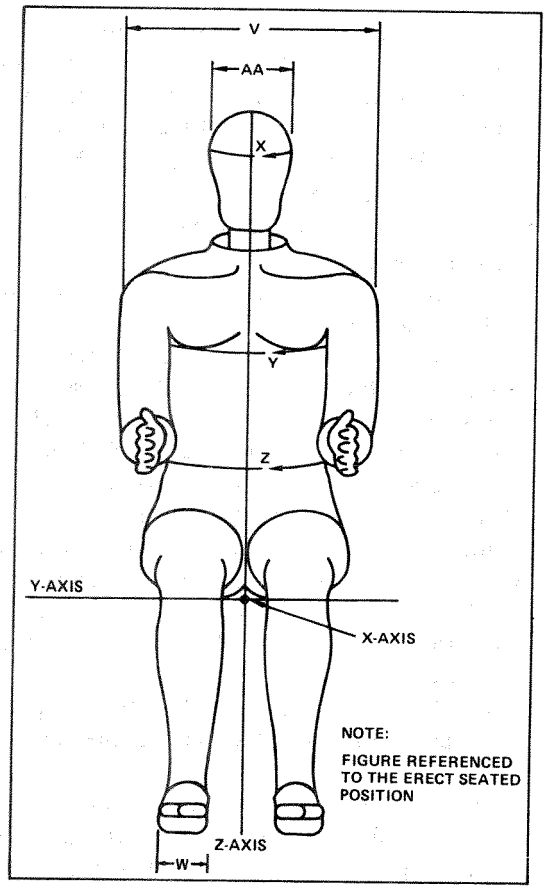


Figure 10

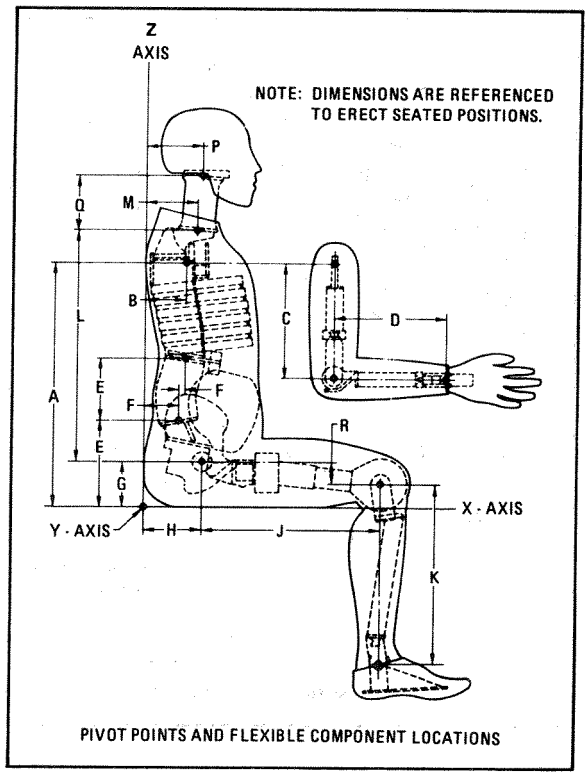


Figure 11

geometric and inertial properties of the dummy, the placement of important joint pivots have also been defined (Table 2 and Figure 11). Section planes between the dummy head, neck, and upper and lower torso have been specified; and requirements for the weight and center-of-gravity locations for these segments have been presented (Table 3 and Figure 12). Additional specifications for the head include anatomically based coordinate directions and the mass moment of inertia about a lateral axis through the head center of gravity.

TABLE 2
BODY PIVOT AND FLEXIBLE COMPONENT LOCATIONS

Dimensional Symbol	Description	Nominal Design Value* (Inches)
A	Shoulder Pivot to X-Axis	21.9
B	Shoulder Pivot to Z-Axis	3.5
C	Shoulder Pivot to Elbow Pivot	10.3
D	Elbow Pivot to Wrist Pivot	9.8
E	Lumbar Lower Centerline Point to X-Axis	7.7
E[1]	Lumbar Vertical Height	5.6
F	Lumbar Lower Centerline Point to Z-Axis	2.9
F[1]	Lumbar Horizontal Offset	0.5
G	H-Point to X-Axis	4.0
H	H-Point to Z-Axis	5.0
J	H-Point to Knee Pivot	15.7
K	Knee Pivot to Ankle Pivot (Horiz.)	16.3
L	H-Point to Neck Lower Centerline Point	20.8
M	Neck Lower Centerline Point to Z-Axis	4.5
P	Upper Neck Centerline Point to Z-Axis	4.9
Q	Neck Vertical Height	4.9
R	H-Point to Knee Pivot (Vertical)	1.9

*Referenced to the Erect Seated Position

TABLE 3
WEIGHTS OF SUBASSEMBLIES (SECTIONED AS PER FIGURE 12)

Segment	Anthropometric Goal[1] (Lbs)	X Dummy[2] Design Value (Lbs)	Delivered #5 (Lbs)	Dummies[3] #6 (Lbs)
Head	10.0	10.00 ± 0.05	10.03	10.04
Neck	3.1	2.70 ± 0.05	2.68	2.69
Upper Body	38.2	36.1 ± 0.4	36.0	36.1
Lower Body	50.8	40.3 ± 0.4	40.3	40.1
Upper Arms (Both)	8.5	8.5 ± 0.2	8.3	8.5
Lower Arms and Hands (Both)	7.5	9.9 ± 0.2	9.9	9.9
Upper Legs (Both)	26.8	36.0 ± 0.4	35.6	35.9
Lower Legs and Feet (Both)	19.0	20.0 ± 0.4	19.9	19.6
TOTAL	163.9	163.5[4]	162.7	162.8

1 Refer to Report No. DOT-HS-299-3-369 (1).
2 Differs from column 1 because all anthropometric goals could not be met without redesign/remanufacturing.
3 Tolerance on measurement is ±1.0%.
4 Sum of component tolerances is ±2.1 Lb.

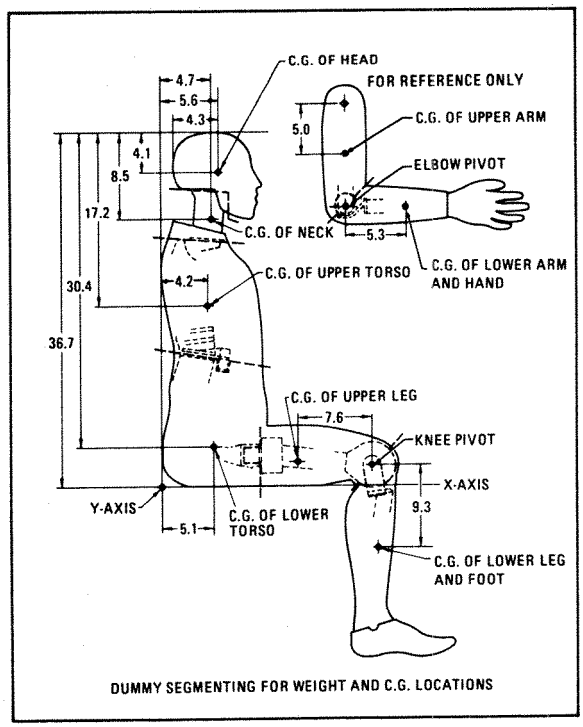


Figure 12

Biomechanics

To assure humanlike results, a test dummy should be designed to biomechanical performance goals obtained from both system and component tests. Little or no biomechanical data exist at the present time which might serve as a basis for the specification of humanlike response requirements for an anthropomorphic dummy on a total system basis.

There exists, however, some biomechanical data which can be used to specify subsystem (component) performance. These data were reviewed at the onset of the program and the major biomechanical performance goals were defined. These included:

Head

Test Objectives and Methodology. A drop test procedure is described, and requirements are presented for dummy head response to the drop test. These requirements are an effort to evaluate dummy and human head impact responses so that the accelerations measured in dummy heads could have some meaning as a basis for human head injury as it is presently assessed. The correlation between dummy and human head response is based on generation of the same peak resultant accelerations upon impact of human and dummy heads with a rigid, flat surface. The relevancy of this correlation however to human tolerance is not well established. Performance was characterized by a drop test response of 120-150 g's peak resultant acceleration for a 6-inch drop and 200 to 250 g's for a 12-inch drop onto a steel plate. During the course of the program another review of the biomechanical data suggested that a performance of 225 to 275 g's for a 14.8-inch drop height would be more appropriate. The GM-ATD 502 head complies with all of these requirements.

Test Procedure

- Drop testing of dummy heads onto a rigid, flat surface was chosen as a basis for study and specification of dummy head mechanical performance for two reasons: (1) the response of the head to this test depends almost entirely on the characteristics of the head so that differences in response are primarily due to differences in head characteristics, and (2) the response of dummy heads to this test can be compared with the response of cadaver heads under similar test conditions in an attempt to establish the correlation of dummy and human head impact responses.
- A detailed description of the drop test equipment used and procedure followed is presented in Reference 1.
- In the past, mechanical resonance of dummy heads has obscured the acceleration pulses measured during impact events and has led to head injury assessments which were difficult to interpret. To preclude this undesirable resonance, the resultant acceleration versus time which occurs during drop test from heights of 6-inches, 14.8-inches (a more appropriate performance requirement), and 35-inches should be unimodal to the extent that oscillations occurring after the main acceleration pulse are less than five percent zero-to-peak of the main pulse.

Test Results and Evaluation.

- The dummy head response requirements were established in response to this NHTSA contract and have been the performance goals during the development of the GM-ATD 502 head. Table 4 shows the results of the forehead drop tests and Figure 13 shows a typical acceleration vs. time curve for the 14.8-inch drop height.
- The foregoing results indicate compliance with the biomechanical goals for performance as previously stated. Also, the 35-inch drops show the heads to be quite free of obvious mechanical resonancess.

Neck

Test Objectives and Methodology. This is a systems test of head-neck-chest complex on a complete dummy in a belted sled test environment. Biomechanical data have been collected in this configuration, and necessary requirements for fidelity

TABLE 4 FOREHEAD DROP TEST RESULTS		
	Head	
	D5	D6
6-inch Drop Height		
Peak Resultant Acceleration (Mean of 3 Tests)	130.8g	127.9g
Coefficient of Variation (%)	0.65	0.27
14.8-inch Drop Height		
Peak Resultant Acceleration (Mean of 3 Tests)	258.7g	250.3g
Coefficient of Variation (%)	2.66	2.84

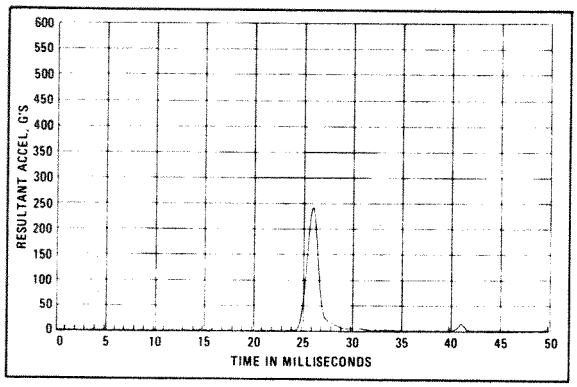


Figure 13

have been formulated. This test was required by the Contract. The Contract test description is slightly modified to specify the input velocity more narrowly and to clarify the intent of the original working. Also, the occipital condyles have been more accurately defined since the original contract was written. Neck biofidelity has been characterized by the flexion and extension corridors of Mertz, et al. (4). The GM-ATD 502 neck performance lies within the flexion corridor but does not meet the peak torque or area ratio requirements as specified by the referenced criteria.

Test Procedure. The neck is mounted between the head and chest structure of the dummy. The chest is reclined 15 degrees rearward from the erect position and the anterior-posterior axis of the head is horizontal. The initial head position is maintained until the onset of the prescribed loading. A description of the test equipment used and procedures followed is detailed in Reference 1.

Test Results and Evaluation. Typical results for the molded polyacrylate neck when subjected to the flexion and extension dynamic loadings are illustrated in Figure 14. The torque levels and area ratios (R) for flexion and extension do not reach the required values.

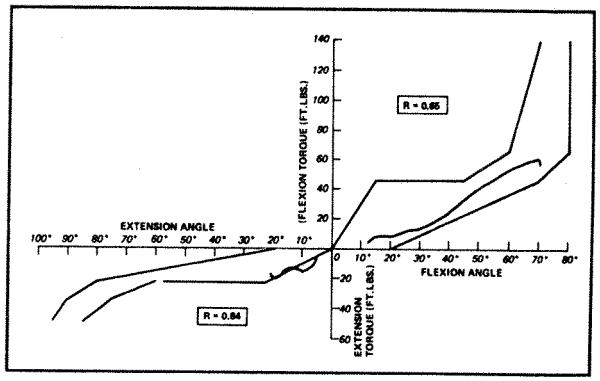


Figure 14

Chest

Test Objectives and Methodology. This is a test of the whole dummy system for which biofidelity has been specified. While the characteristics of the entire dummy are involved, the results depend primarily on the dynamic response of the torso structure. The thoracic impact corridors by Lobdell et al., as cited in Reference 5, were the biomechanical goals during chest development. The chest response approaches, but does not comply with the corridors.

Test Procedures. The complete dummy is seated on a hard, horizontal surface with its head, neck, and torso in an upright posture, unsupported and unrestrained and the limbs extended horizontally. The upper torso is impacted in the anterior-posterior direction at a point in the midsagittal plane at a specified distance from the top of the head. A detailed description of the impact test equipment used and procedure followed is presented in Reference 1.

The Dynamic Impact Test of the Thorax suffers from many of the same type of problems applicable to the Biomechanical Systems Test of the Neck. This is also a system test because the cadaver testing was done with whole cadavers. It means that the performance of structures such as the neck, lumbar, and arms are also a part of the results of this test. Therefore, the rib-cage performance indicated by this test changes if any of the other components perform differently, and the development of the rib cage depends on these other components being in the final design stage and being repeatable.

Also, since the dummy is seated on a surface and upon impact to the thorax is allowed to move, the friction between the dummy pelvis and legs and the seating surface could affect the results. Again, the test

is done in this way because the cadaver work was done in a similar manner.

Test Results and Evaluation. Three dynamic impact upper torso tests were run at each of the two specified velocities (11.0 mph and 16.0 mph) on the two complete test dummies. Typical results of the dynamic load deflection characteristics of the assembly are shown in Figures 15 and 16.

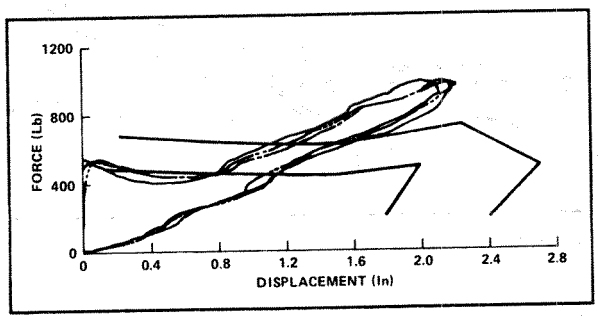


Figure 15

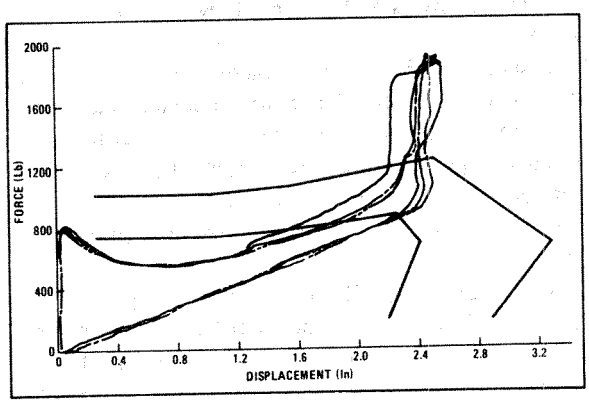


Figure 16

The biomechanical performance corridors were not met. Additional development effort would be necessary to obtain a chest design that would meet the Contract specification and perform satisfactorily on a system test basis.

Knees

Biomechanical criteria for static knee penetration into a deformable material were used to evaluate the knee structure. Responses approached but did not comply in all instances.

The requirements and criteria used for the development of the GM-ATD 502 were based on currently available knowledge of human impact response and the factors affecting that response. These requirements are a set of conditions *necessary* for similarity of dummy and human response, but they are *not sufficient* to assure humanlike response to general impact conditions even at the subsystem level. The inability to definitively specify a set of requirements sufficient to assure that the GM-ATD 502 is humanlike is due primarily to a lack of appropriate human impact response data.

For further details regarding the above mentioned tests, the documentation of the procedures involved and of the data collected, the reader is referred to Volume II of Reference 1.

SYSTEM TESTS

Experimental Design

Background. The end result of a dummy design is testing in simulated automotive crash environments with various restraint system designs. These, of course, involve a total dummy and are therefore referred to as system tests. However the question that the designer must face is which of the many combinations of restraint systems, automotive interior environments, dummy setups, test procedures, and acceleration pulses should be used as the primary development system test for the dummy?

One approach would be to pick several representative tests (e.g., a three-point belt test, an air bag test, etc.) and assess the dummy's performance for each system and several environments. However, this was not considered a practical solution because extensive development would be needed on each type of test to eliminate the effects of test to test hardware variation from indicated dummy performance.

Other considerations that could highly influence the design of the experiments are the test environment and conditions under which the biomechanical data used for formulating the performance goals were collected. Unfortunately biomechanical data representative of total system performance are extremely limited, and the restraint system and test equipment used for the conduction of these tests cannot be satisfactorily extrapolated to a realistic automotive crash environment.

Consequently, the restraint condition chosen was the three-point, lap-shoulder belt system. This system was selected for several reasons:

- It is representative of three-point belt systems typical of present-day automotive restraint systems.
- Field accident data are available on this type of restraint system, providing some knowledge of its effectiveness.
- External sources of variability are less difficult to control with this system.
- Major components (neck, chest, pelvis, and limbs) of the dummy are exercised when this system is used.

In implementing the above testing configuration, special efforts were devoted to minimize all extraneous variables that could affect test results. Pilot tests were conducted to study potentially important variables (other than the dummy) affecting the results of 3-point restraint sled tests. As a result of this study, the following dominant factors were identified:

- Initial seated position of the dummy
- Seat characteristics
- Belt geometry and webbing characteristics
- Sled pulse
- Instrumentation set
- Locking retractors
- Belt load transducers
- Time between tests on dummy
- Temperature and humidity

Test Fixture Used

To ascertain the performance of the dummy with as little influence as possible from these external variables, a "hard seat" fixture was designed (Figure 17). Its uniform, rigid, seating surfaces allowed the variability associated with typical automotive seats to be eliminated and aided in precise control of the dummy's initial seated position. It provided a common static seating environment (for dummy setup and belt restraint placement) and a common dynamic environment for repeatable accelerations (forcing function on the test dummies), and dummy interaction with the seat surface and belt restraints. By minimizing test variables due to seat structure and belt restraints, resulting variability in dummy kinematics and acceleration profiles would then to a great extent be the result of the test dummy hardware and/or manufacturing.

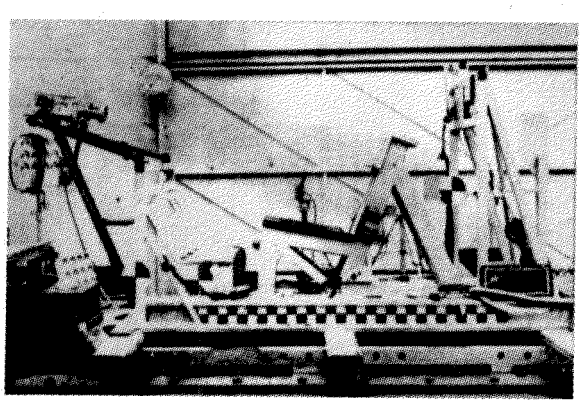


Figure 17

Seating geometry of a compact vehicle (front passenger location) was chosen for the sled test fixture. The H-point to heel hard (intersection of the toe and floor pans) vertical and horizontal dimensions were used with the seat located in midposition as the coupling distance for the test fixture.

Analytical and experimental test hardware parameter studies were conducted to investigate seat pan angle and lap and shoulder belt angles. A seat back angle of 21 degrees (measured from the vertical) was selected as simulating the test dummy sitting attitude in the compact vehicle bucket seat. This 21-degree angle is the seating manikin's (SAE J826a) torso angle used in the design of the compact vehicle seating package and is the angle to which the fixed hard seat structure was built.

The criterion used in selecting the seat pan angle was to attempt to match the resultant pelvic acceleration of the test dummy with that recorded in a production bucket seat. Matching the resultant acceleration tends to exercise the test dummy in a manner similar to the production vehicle environment. Seat pan angles ranging from -5 degrees (negative slope) to +15 degrees were investigated via hardware tests and analytical modeling. The +10 degree seat pan ramp angle was chosen because earlier tests indicated this angle produced pelvic acceleration components most similar to those recorded in production seats.

The revised Calspan occupant dynamics simulator MODROS (6, 7) was exercised to further optimize seat pan and lap belt angles. The simulations verified the test investigation and showed a flattening of the response curves at a seat pan angle of 10 degrees and lap belt angles of 60 degrees (measured from the horizontal).

These choices thus met two important criteria: First, they would produce dummy responses comparable with production seat tests, and, second, effective seat belt angle changes due to test setup variability would produce minimum changes in the dummy response.

A platform for mounting a triaxial accelerometer at the same vertical elevation as the test dummy H-point was incorporated so that the sled acceleration environment at the dummy/sled test fixture seat interface could be monitored.

The seat pan steel surface was covered with a 0.50-inch sheet of 50 durometer natural rubber. This was deemed necessary to semi-isolate the test dummy's pelvis from the rigid seat structure and to provide some surface friction between the test dummy and the surface of the seat.

The seat back was padded with 2.00 inches of Uniroyal Ensolite, Type AH, in an attempt to attenuate the rebound acceleration of the test dummy. Data trends from the tests indicate that the

rebound was repeatable but at a higher level than anticipated.

Positioning Templates

Because development testing indicated that test dummy setup location was important for repetitive results, positioning templates were designed and fabricated to accurately locate the test dummy's head, chest, and H-point relative to the reference grid and the test fixture. The locating probes on the templates were set using a carefully positioned and restrained test dummy on the hard seat fixture.

It is believed that this positioning template promotes more accurate and repeatable test dummy setups than the devices in current use that only locate one side of the test dummy. Figure 18 illustrates the positioning template used.

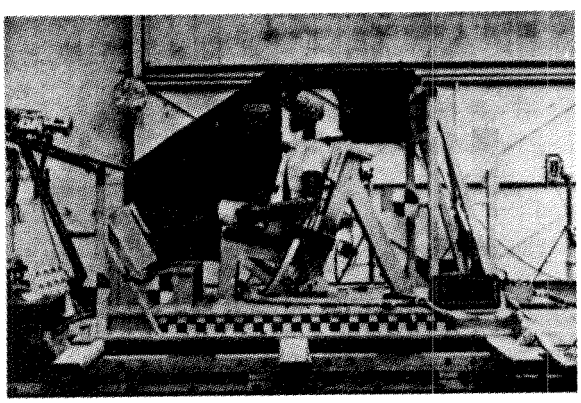


Figure 18

Belt Selection

A three-point belt system was selected for restraining the test dummy and providing dummy/fixture interaction during acceleration environment exposure. The belts interfaced with the test fixture using adjustable anchors that provided 2.00 inches of linear adjustment.

Lap belt anchors fore and aft locations were determined by using FMVSS 210 procedure for measuring belt angles. FMVSS 210 identifies measuring procedures for lap belt angles as using a line tangent to the belt centerline passing through the design H-point or a point 2.50 inches forward and 0.375 inch above the design H-point. The latter point was chosen for defining belt angles.

Impact Sled Parameters

In selecting the acceleration time function for HYGE sled tests using the hard seat test fixture, repeatability and reproducibility considerations were given top priority. A haversine wave was chosen because of its basic geometric shape and because it could be repeatably generated by a number of other testing laboratories with various types of equipment. The haversine also has the characteristics of not having large discontinuities, thus improving the predictability and the ability to control the input (sled forcing function) from test to test.

The impact sled parameters (simulation of floor pan average acceleration during barrier impacts for a production compact size vehicle) were chosen for the development of haversine simulation. The peak accelerations, pulse duration, and rise and decay rates of the haversine pulse were modified so that the velocity time curve best approximated the conventional simulation. Pulse amplitude and frequency were 33 g and 6.85 Hz, respectively. A comparison of a compact vehicle acceleration pulse simulation and the developed haversine pulse-acceleration and velocity curves are shown in Figures 19 and 20. The only time interval at which the two velocity simulations deviate from each other is early in the impact event. At this time, the displacements are relatively small and are judged to have a minimum effect on overall simulation and on the resultant dynamic performance of the test dummy.

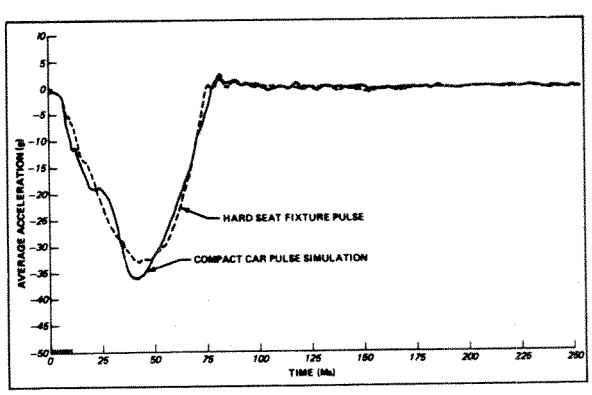


Figure 19

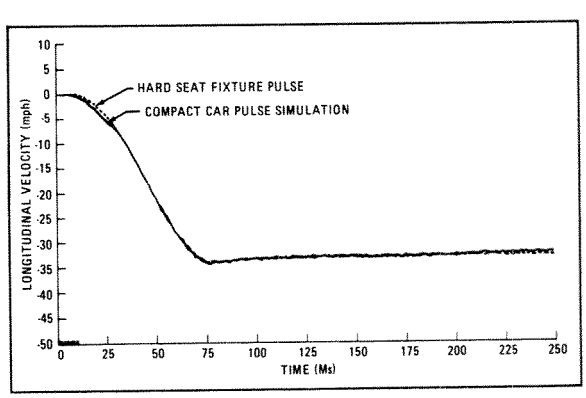


Figure 20

PERFORMANCE CRITERIA

Repeatability as used in this program means the similarity of test results when a single dummy is given several repeat tests. Reproducibility is the similarity of test responses when more than one dummy is subjected to the same test conditions. Repeatability depends not only on each dummy component performing the same in test after test, but also depends on the control of the test setup, data recording, and many other variables which are external to the dummy. Reproducibility depends upon repeatability and also upon the degree to which the dummies in question are manufactured alike. Reproducibility of a dummy design cannot be truly evaluated on prototype hardware. Furthermore, the small sample size limits the statistical significance of the results.

During the course of the project several variables were explored as repeatability measures of dummy performance. These included the following:

- Root-Integrated-Mean-Square-Difference (RIMSD)*
- Severity Index(8)
- Three milliseconds cumulative acceleration(8)
- Mean of resultant acceleration over the entire event
- Head Injury Criteria-1(8)

The Head Injury Criteria (HIC-1) and the Severity Index (S.I.) appeared to be the most sensitive indicator of repeatability. The severity index is an integrated quantity using data from the whole response event; it also magnifies local differences in the acceleration traces due to the 2.5 power weighting of the calculation. To express input parameters on a common basis with the dummy response, the S.I. was also calculated for the sled and fixture acceleration, and it was used as a repeatability measure.

TEST METHODOLOGY

First Test Series

Three test series were conducted. The first test series involved "Design No. 1" and consisted of a total of 18 tests on three dummy systems (six each). It had for primary objective the development of a common data base to properly assess the repeatability and reproducibility (R & R) of the prototype hardware. The tests were structured such that R & R values could be identified not only for the dummy hardware, but also for the instrumentation used, the sled and the fixture. The dummies were tested with the arms initially straight (hands placed outside of thighs) to minimize upper limb joint effects. To determine the contribution of the instrumentation used, a plan was developed for this test series to interchange, in an organized manner, accelerometer assemblies among the test dummies.

Three sets of instrumentation were identified. Each set consisted of three uniaxial accelerometers mounted in a head accelerometer mount block, three accelerometers mounted in the chest accelerometer mount block, and two femur load cells. When rotating the instrumentation from dummy to dummy, head and chest units were never interchanged.

To determine the repeatability and reproducibility of the prototype hardware (test dummy) and the instrumentation sets, the following experiment was designed, so that an analysis of variance could be conducted:

		Dummy Number		
		1	2	3
Instrumentation	1	X11	X12	X13
	2	X21	X22	X23
Set Number	3	X31	X32	X33

This is a full 3^2 factorial experiment (9, 10) (two factors at three levels, i.e., instrumentation and dummies) which results in nine tests. To increase the statistical confidence in the results this entire experiment was conducted twice. The basic statistical model for this test plan is:

$$F_{ijk} \text{ or } X_{ijk} = \mu + A_i + B_i + AB_{ij} + \Sigma_{k(ij)}$$

Where i = j = 1, 2, 3 levels of factors A and B

And K = 1, 2 replication of the experiment.

This will produce the mean response, μ, the effect of Factor A (dummy number), the effect of Factor B (instrumentation set number), and an assessment of interaction and error (AB_{ij} and $\Sigma_{K(ij)}$ respectively). Analyzing each column of data yields a measure of test-to-test repeatability. Analyzing each row of data gives a measure of reproducibility as well as the analysis of variance of the total experiment.

Second Test Series

After the first test series, a complete redesign of the *joints* and of the shoulder complex was executed. This new design, referred to as "Design No. 2," represented the prototype hardware for the final design developed in this program. It was evaluated in

*This is a statistical technique for comparing the similarity of two curves and can be utilized in a "goodness of fit" test. It can be computed for any recorded variable and is calculated over the entire time history.

a second test series comprised of a total of 16 tests involving *two* dummies (eight tests each).

The testing procedures in this series were the same as those of the first series with the exception of the initial arm position, which was in accordance with the FMVSS 208 test procedure (that is, arms were bent at the elbow, upper arm against the seat back, hands along thighs).

Because the effects of the instrumentation on the test results had already been investigated in the initial test series, the experiment for the second series of tests was simplified.

Two sets of instrumentation were chosen from the three sets used in the initial test series. This was done, again, without interchanging the positions of head and chest accelerometer packs.

Test dummy D2 was evaluated in a series of four tests with one set of instrumentation. The instrumentation was then switched and D2 was again evaluated in a second series of four tests. This constitutes a 2^1 experiment with four replicates (instrumentation at two levels and dummies at one). The analysis of variance can then be conducted as described earlier.

Finally, dummy D3 was subjected to eight repeat tests without instrumentation exchange because instrumentation effects had been identified.

Third Test Series

The third and final test series was conducted to validate and document the dynamic performance of the two NHTSA Contract deliverable test dummies (D5 & D6) and an identical unit (D4) built under the General Motors in-house program. It was conducted with instrumentation sets identified for each test dummy. The instrumentation sets selected were from the previous test series and were not rotated because this series was strictly a validation test (i.e. one test per dummy).

TEST RESULTS

Results for the first (Design No. 1) and second (Design No. 2) test series are summarized in Tables 4 and 5 respectively. The mean response value (X), standard deviation (S), and the coefficient of variation (C.V.)*, are tabulated for the parameters defined in the first column. A separate listing of the results for each dummy system (D_i)** is given as well as the combined results for Designs No. 1 and No. 2 (last columns of Tables 4 and 5).

*C. V. = Repeatability and Reproducibility measure = standard deviation expressed as % of mean. i.e. C. V. = $(S/\bar{X})100$.

**i = 2, 3, 4 for Design No. 1
i = 2, 3 for Design No. 2

As defined, the coefficient of variation does provide a measure of "repeatability" when computed from data resulting from several tests of the *same* dummy. Similarly, the coefficient of variation computed from data resulting from *several* dummies provides a combined measure for both the repeatability and reproducibility of the test design.

Comparing coefficients of variation between Tables 5 and 6, we observe an improvement in R & R from 6.7% and 6.8% to 3.7% and 4.5% for the HIC-1 and head SI respectively. The improvement in the chest R & R is even more pronounced, the chest SI having gone from 8.0% down to 4.1%. These results are also illustrated in Figures 21 and 22 where the mean head and chest SI's for both Designs 1 and 2, along with their standard deviations are plotted. The overall mean anc combined standard deviations (R & R) are also shown for each design. As mentioned earlier the mean level difference between the two designs results from arm positioning. When testing Design 1 the arms were straight with hands on knees. This was done to improve the repeatability of the tests by eliminating the erratic behavior of the elbow joint. With the improved joints used in Design 2, the arms were bent at the elbow with hands along the thighs (MVSS 208 position).

TABLE 5
SLED TEST RESULTS
(DESIGN NO. 1)

Dummy No.	D2			D3			D4			D2 + D3 + D4		
No. of Tests	6			6			6			18		
Parameters	$\bar{X}$	S	C.V.	$\bar{X}$	S	C.V.	$\bar{X}$	S	C.V.	$\bar{X}$	S	C.V.
HIC-I	702.0	57.0	8.2	690.0	48.0	7.0	737.0	27.0	3.7	709.0	48.0	6.7
Head SI	1068.0	79.0	7.4	1055.0	78.0	7.4	1136.0	40.0	3.5	1086.0	73.0	6.8
Chest SI	571.0	25.0	4.3	505.0	38.0	7.5	517.0	34.0	6.5	531.0	43.0	8.0
Left Femur (Lbs)	207.7	8.8	4.2	206.7	14.6	7.1	241.5	13.7	5.7	218.6	20.4	9.4
Right Femur (Lbs)	182.9	10.7	5.9	204.5	16.5	8.1	201.6	23.8	11.8	196.4	19.4	9.9
Lap Belt (Lbs)	1465.5	67.6	4.6	1479.3	37.8	2.6	1467.0	84.7	5.8	1470.6	62.6	4.3
Shoulder Belt (Lbs)	2869.3	89.3	3.1	2826.2	88.0	3.1	2937.5	76.0	2.6	2877.7	97.4	3.2
Tongue Belt (Lbs)	3483.0	63.2	1.8	3429.7	63.5	1.9	3523.2	85.2	2.4	3478.6	77.8	2.2
Sled Velocity (mph)	34.5	.1	.4	34.5	.2	.5	34.5	.1	.3	34.5	.4	.4
Fixture SI	232.2	2.5	1.1	231.4	2.1	.9	230.5	2.1	.9	231.4	2.2	1.0

TABLE 6
SLED TEST RESULTS
(DESIGN NO. 2)

Dummy No.	D2			D3			D2 and D3		
No. of Tests	8			8			16		
Parameters	$\bar{X}$	S	C.V.	$\bar{X}$	S	C.V.	$\bar{X}$	S	C.V.
HIC-1	619.8	20.1	3.3	588.4	9.7	1.6	604.1	22.3	3.7
Head SI	933.5	44.2	4.7	889.1	24.0	2.7	911.3	41.3	4.5
Chest SI	435.9	19.7	4.5	440.9	17.3	3.9	438.4	18.1	4.1
Left Femur (Lbs)	247.2	9.0	3.6	270.1	15.7	5.8	258.6	17.1	6.6
Right Femur (Lbs)	241.8	24.9	10.3	253.1	21.4	8.5	247.4	23.2	9.4
Lap Belt (Lbs)	1716.3	42.6	2.5	1751.8	52.8	3.0	1734.0	49.9	2.9
Shoulder Belt (Lbs)	2557.9	41.6	1.6	2554.0	37.8	1.5	2555.9	38.5	1.5
Tongue Belt (Lbs)	3358.6	44.4	1.3	3426.5	137.2	4.0	3392.6	104.5	3.1
Sled Velocity (mph)	34.3	.2	.6	34.3	.1	.3	34.3	.2	.5
Fixture SI	231.6	2.9	1.3	230.3	1.5	.7	230.9	2.4	1.0

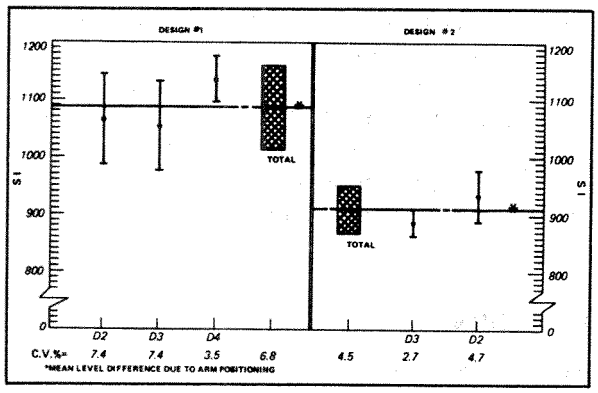


Figure 21

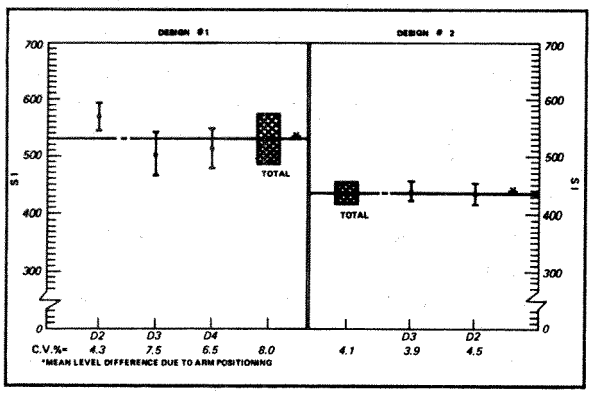


Figure 22

The results for the final design (Design No. 2) are also illustrated in nonstatistical terms by Figures 23, 24, and 25. Plotted are time histories for six consecutive tests for the head accelerations, head SI's, and for the chest SI's. These plots are generated with "Macran" processing.* The data included here are based on only six of the eight tests conducted for the dummy because the processing program is limited to

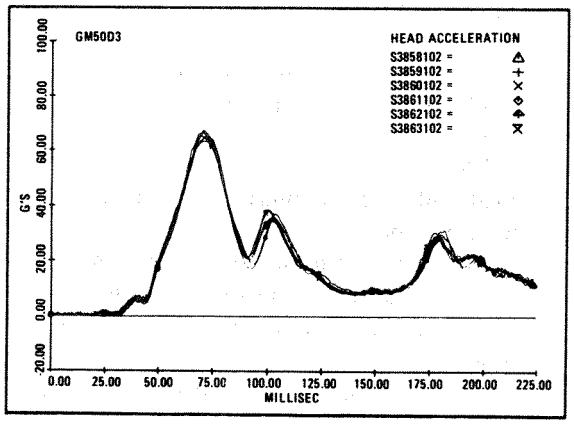


Figure 23

*University computing corporation proprietary software package.

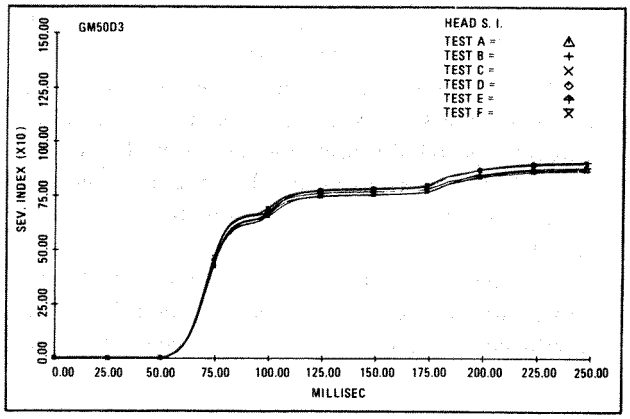


Figure 24

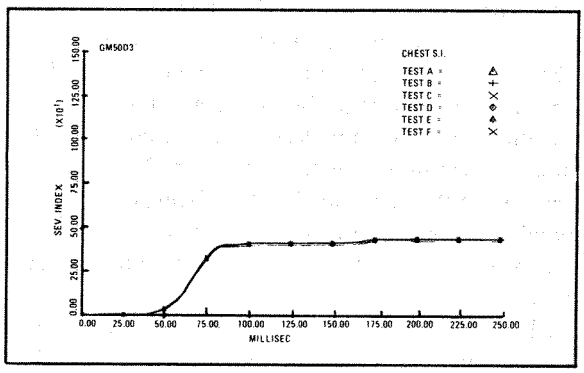


Figure 25

a maximum of six tests in its present form. Also illustrated is a multiplot summary for the mean accelerations of the sled, head, chest, pelvic and test fixture accelerations (Figure 26). This allows the reader to better correlate the relative responses of the above subsystems.

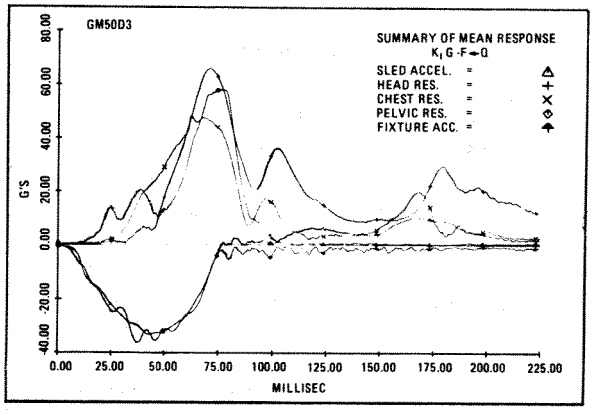


Figure 26

The high repeatability of the *sled* pulse and of the *fixture* response can also be observed from Tables 5 and 6. The coefficient of variation for the sled pulse

is less than 1%. The fixture response was measured at the H-point elevation level. The trial accelerations resultant converted to a severity index was used as a sensitivity measure. Note that the fixture has a one percent repeatability using the data from the 16 tests on both dummies. It was slightly higher (1.3 percent) on the D2 series and slightly lower (0.7 percent) on the D3 series.

Belt loads measured in this series of tests also prove to be repeatable and reproducible. The coefficient of variation was four percent or less on the lap, shoulder, and tongue data for either a single dummy or the data from both dummies. It is concluded that this excellent repeatability is due to a very consistent sled pulse, a rigorous setup procedure, and a repeatable and reproducible dummy.

An exception to the above, as can be noted in Table 6, is the repeatability of the femur loads. This is probably due more to the low mean values in a belt test than any inherent lack of consistency in the dummy construction or the test setup. The standard deviation of the femur loads is low, being less than 25 pounds. A more appropriate evaluation of femur load repeatability would be a test where substantial loads are generated.

Also included are the validation tests results for the two anthropomorphic dummies (D5 and D6) delivered to NHTSA. These results are illustrated in Table 7.

TABLE 7
PERFORMANCE VALIDATION RESULTS – NHTSA CONTRACT DUMMIES

	Dummy D5	Dummy D6
Head and Chest Responses		
HIC	581.0	530.0
Head SI	880.5	776.3
Chest SI	436.5	433.8
Femur and Belt Loads (Lb)		
Left Femur	283.4	311.0
Right Femur	301.4	340.1
Lap	1725.0	1550.0
Shoulder	2475.0	2500.0
Tongue	3725.0	3525.0
Sled and Fixture Parameters		
Sled Velocity (mph)	34.54	34.49
Fixture SI	232.20	231.00

In summary it can be concluded that the GM-ATD 502 with its biomechanically improved head and further improved neck, chest, and joints can be expected to give repeatable responses under five percent in a hard seat, three-point belt restraint system test. This conclusion is postulated on the assumption of well controlled setup and test procedures.

DISCUSSION

General Assessment

A summary of the important characteristics of the GM-ATD 502 presented in this paper is given below. The part 572 test device defined by the U. S. Department of Transportation MVSS 208 for compliance testing is used as the state-of-the-art reference for comparison purposes (11). Based on this frame of reference, the following assessments are made

- *Repeatability* due to
  - Shoulder design with better shoulder belt interface.
  - Joint design which gives more uniform resisting torque.
  - Lumbar spine design which allows a natural automotive seated position without preloading of flexible parts for improved setup.
- *Reproducibility* due to
  - Data package that represents a milestone in depth of design documentation for manufacturing of identical dummies.
  - Dummy segmentation techniques which allow better control of inertial properties.
  - Subassembly weights that are controlled (rather than total weight) for more accurate weight distribution.
- *Ease of use and maintenance* because
  - Joints no longer require setting between each test.
  - Separate left- and right-hand parts have been eliminated in many places for smaller parts inventory.
- *Biofidelity* because
  - Skull and head contours are based on a study of human shapes.
  - Head impact performance is based on a drop test which is correlated with cadaver testing.
  - More humanlike automotive seated positioning is used.
  - Neck and chest were designed towards biomechanical corridors.

This program has also identified improved system and component test procedures and instrumentation

for dummy development and evaluation. In particular, it is felt that the "hard-seat" fixture and positioning template developed greatly enhanced the test environment by minimizing external variables.

However, it should be pointed out that because of the absence of quantitative human-injury criteria correlated with anthropomorphic dummy and product performance, there is no assurance that the measurements made with the GM-ATD 502 can be well correlated to human injury tolerance levels. Furthermore, there is no assurance that changes in the responses of the dummy due to variations in test environments can be correlated to human response sensitivity. Both of these requirements must be met before the degree of protection afforded by a restraint can be truly evaluated.

Finally, it should be pointed out that since the performance goals of this development program differed from the requirements set forth in Regulation 572, this test dummy does not meet the 572 requirements.

RECOMMENDATIONS

It is recommended that this testing device not be specified for use in the determination of injury criteria in any occupant protection standard that would apply to all types of restraint systems. It is our concern that since complete biomechanical fidelity has not been achieved with this dummy, its use would generate misleading test results.

It is recommended that when the performance of various test dummy designs are compared, they be evaluated under the same set of controlled test conditions. Furthermore, it is recommended that the type, location, and calibration procedure of all instrumentation be specified.

Continuing effort in the simulation of human impact response should be concentrated in the following areas of biomechanics, anthropometrics, and test dummy hardware:

Biomechanics

- Human response and cadaver data should be obtained for various types of system tests (e.g., three-point belt systems, air cushions, and padded interiors). This would provide a basis for validating dummy responses in controlled, simulated collision environments.
- Sensitivity studies should be conducted to determine how changes in impact environments affect human response. These data would serve as a basis for comparing dummy and human response sensitivities.
- To aid in dummy component development, biomechanical tests should be conducted with the objective of isolating the response characteristics of a given body segment.
- Biomechanical performance data on human joint resistance should be gathered and incorporated into performance specifications for dummy joints.
- The dynamic deformation and friction characteristics of human soft tissue should be documented and incorporated into dummy specifications.

Anthropometrics

- Accurate determination of the human body segment masses, moments of inertia, center of gravity locations, pivot locations, and segment mobility is required.
- The external contours of the test dummy need to be defined as rigorously as was done for the head of the GM-ATD 502 dummy.
- A formalized procedure for crash dummy positioning in automotive seats based on human anthropometric data should be developed.

Test Dummy Hardware

- Repeatability and reproducibility tests for dummy components should be developed which have a meaningful relationship with that component's response in systems tests.
- The sensitivity of dummy system and component responses to such factors as temperature, time between tests, and number of tests should be investigated.
- Further development work should be pursued on the various dummy subassembly designs as the above information becomes available.

REFERENCES

1. "Anthropomorphic Test Dummy," Final Report No. DOT-HS-299-3-569 3 volumes prepared for DOT, NHTSA Wash., D. C. 20590. Dec. 1973. Available through National Technical Information Service, Springfield, Virginia 22151.
2. R. P. Hubbard and D. G. McLeod, "A Basis for Crash Dummy Skull and Head Geometry," presented and published in the Human Impact Response Symposium held at the General Motors Research Laboratories, Warren, Michigan, October 1972, Plenum Press, New York, 1973.
3. R. W. Roe, "A Description of Driver Workspace Anthropometry from 50th-Percentile Male Data," Safety and Human Performance

Laboratory, GM Design Staff, Report No. 72-1011, November 1972.

4. H. J. Mertz, R. E. Heathery, and C. C. Culver, "Performance Requirements and Characteristics of Mechanical Necks," presented and published in the Human Impact Response Symposium held at the General Motors Research Laboratories, Warren, Michigan, October 1972, Plenum Press, New York, 1973.
5. T. E. Lobdell, C. K. Kroell, D. C. Schneider, W. E. Hering and A. M. Nahum, "Impact Response of the Human Thorax," presented and published in the Human Impact Response Symposium held at the General Motors Research Laboratories, Warren, Michigan, October 1972, Plenum Press, New York, 1973.
6. J. P. Danforth and C. D. Randall, "Modified ROS Occupant Dynamics Simulation Users Manual," General Motors Research Publication GMR-1254, prepared for use of Motor Vehicle Manufacturers Association, Inc.
7. D. J. Segal, "Revised Computer Simulation of the Automobile Crash Victim," CAL No. V-5-2759-V-2, Cornell Aeronautical Laboratory, Inc., Buffalo, N.Y.
8. "Occupant Crash Protection Head Injury Criterion." Department of Transportation National Highway Traffic Safety Admin. (49 CFR Part 571), Docket No. 69-7; Notice 17.
9. C. R. Hicks, "Fundamental Concepts in the Design of Experiments," New York: Holt, Rinehart and Winston, 1964.
10. Kan-Chen Peng, "The Design and Analysis of Scientific Experiments: An Introduction with some Emphasis on Computation," Reading, Mass.: Addison-Wesley Press, 1967.
11. Part 572, "Anthropomorphic Test Dummy," Occupant Crash Protection, Federal Motor Vehicle Safety Standards, Federal Register, Vol. 38, No. 147, August 1, 1973.

RELATING AIR CUSHION PERFORMANCE TO HUMAN FACTORS AND TOLERANCE LEVELS

PART I– PROGRAM DEVELOPMENT

LOUIS C. LUDSTROM
Environmental Activities Staff
General Motors Corporation

INTRODUCTION

The development of the crash deployed air cushion restraint system for use in passenger cars required many special programs over a number of years to be reasonably certain the cushions would perform as designed in the protection of both children and adults in a wide range of environmental exposures and accident situations. This paper will identify the major programs and summarize their findings. Details are not repeated in this paper, but will be found in referenced material. When significant, the results of continuing projects will be updated.

Technical presentations at the First International Conference on Passive Restraints (1969), cosponsored by the U.S. Department of Transportation (DOT) and the U.S. auto industry left many, but not all participants, with the belief that air cushion restraint systems were feasible from a technical standpoint, but that much remained to be done before they could be considered practicable. In 1969, the National Highway Traffic Safety Administration (NHTSA) issued their first notice of proposed rulemaking on passive restraints. The intent was to provide restraints that would save more lives and injuries than were being realized with belt restraint systems. A major shortcoming of belt restraints was their lack of use by most occupants. Less than 3% were using both the lap and shoulder belt, and roughly 15% were using the lap belt alone.

INITIAL GENERAL MOTORS CORPORATION AIR CUSHION DEVELOPMENT PROGRAM

The initial General Motors (GM) programs to develop the completely passive air cushion restraint system for front seat passengers were related to such basic considerations as:

- Sensor requirements for selected vehicle crash decelerations
- Air cushion size, shape, fold and deployment
- Source of inflation
- Inflation rate and energy release
- Cushion venting
- Placement of components
- Reliability
- Noise control
- Multiple collision protection

Status reports on these programs were presented at the Second International Conference on Passive Restraints in May of 1972. These included: "Air Cushion Restraint Systems Development and Vehicle-Application" and "Special Problems and Considerations in the Development of Air Cushion Restraint Systems."

These reports indicated much had been accomplished in resolving the early unknowns, primarily technical, related to this new and untried restraint system. Although compromises were necessary, no insurmountable mechanical problems were identified. Of continued concern, however, was the unknown effect of air cushion deployment on humans and the unknowns relative to passive restraint performance in a wide range of real world highway accidents. Anthropomorphic dummies were useful in measuring the dynamics of the crash deployed system, but could not indicate possible trauma from the inflation or deployment of the cushion itself. This could only be accomplished through some type of human volunteer program.

INITIAL HUMAN VOLUNTEER TEST PROGRAM

One of the very important programs in the development of air cushion restraint systems involved testing with human volunteers at the Holloman Air Force Base. This work was sponsored by the NHTSA in conjunction with the Aerospace Medical Research Laboratory of Wright-Patterson Air Force Base. General Motors cooperated by supplying the front seat passenger's air cushion hardware. This program included a series of 41 static deployments to determine the effect of cushion deployment on occupants out of normal seating positions such as with crossed legs, leaning forward, or turned sideways. Because of the severity of cushion deployment, one modification was made following the initial static tests: the orifice size at the bag inlet was reduced from 1.0 inch diameter to 0.6 inch to diminish the rapidity of inflation. Subsequent testing included 35 sled runs with human volunteers starting at 8.6 G (15.1 mph) and culminating at 21.7 G (31.5 mph).

The results of this work were reported at the Second International Conference on Passive Restraints in May of 1972, in a paper entitled, "Human Volunteer Testing of GM Air Cushions." The tests of both the static deployments and simulated crashes on the impact sled were successfully accomplished and pointed up some design areas requiring additional attention. The deployment of the air cushion continued to be overly severe and the rebound characteristics were not considered satisfactory for the interior of a passenger car. As a result, a redesign program was initiated.

INITIAL PROGRAM TO EVALUATE EFFECT OF AIR CUSHION DEPLOYMENT ON CHILDREN

One of the most difficult items to evaluate is the effect of crash deployed passive restraints on children. Small anthropomorphic dummies have not been developed with sufficient biomechanical fidelity to give the development engineer confidence in the system performance as measured with these devices. The Wayne State University in Detroit was therefore requested to evaluate the air cushion inflation severity and occupant rebound characteristics as they might affect children. Their work utilizing live baboons in laboratory experiments indicated a potential inflation hazard to small children who might be out of the normally seated position. The result of this program also stimulated the redesign program for the passenger restraint system.

AIR CUSHION RESTRAINT SYSTEM PRODUCTION DESIGN PROGRAM

The second generation air cushion restraint for front seat passengers was developed and incorporated in a production design program. The possible inflation hazard experienced with the first generation design was reduced by providing dual sensing of impact severity and control of cushion inflation. During impacts of low severity, a low inflation of the cushion would be used. For high severity impacts, a faster deployment of the air cushion was provided. The actual inflator is an augmented type consisting of a steel vessel with stored gas and two internal generators which is capable of producing and releasing two levels of inflation energy. In addition, the passenger cushion material was changed to permit the entire fabric to partially vent the gas during occupant restraint. This provided improved occupant dynamics and rebound control. Protection is provided for a variety of occupant weights and severity of collisions by the air cushion being able to exhaust pressure in relation to the occupant force against it.

The gaseous products of the air cushion inflation are non-toxic. Extensive tests and evaluations by General Motors and independent laboratories have verified that all gaseous products of an air cushion inflation are well below toxic levels.

The driver air cushion restraint was concurrently designed with its own gas generator triggered from the common crash detector. This development program permitted storing the folded driver cushion within the hub of the steering wheel. A newly designed energy absorbing steering column was also provided to match the dynamic characteristics of the driver and air cushion system.

The lower instrument panel knee restraint was constructed of a sheet metal substrate covered with 1/2 inch of foam and a vinyl cover. This knee restraint performed two very important functions in the driver system: It controlled the driver motion by limiting the lower torso movement to retain the driver on the seat in an upright position and also absorbed the energy of the lower torso while keeping the femur loads within reasonable limits.

Results of the production design program were presented in the following technical papers: "Front Passenger System – Design and Production" and "Driver System – Design and Production" (June, 1973, Automotive Safety Engineering Seminar) and the "General Motors Driver Air Cushion Restraint System" (Society of Automotive Engineers, May, 1973).

SECOND HUMAN VOLUNTEER TEST PROGRAM (PASSENGER CAR SYSTEM) – AND PROTECTION OF CHILDREN

The second generation GM air cushion restraint system for passengers was evaluated with human volunteers in August, 1972, at the Holloman Air Force Base. The results were considered satisfactory and indicated a significant reduction in inflation hazards and rebound characteristics. An additional series of tests at Wayne State University indicated the second generation air cushion restraint would reasonably control the inflation hazard to small children.

HUMAN VOLUNTEER TEST PROGRAM – DRIVER AIR CUSHION RESTRAINT SYSTEM

Three important programs were conducted for General Motors by Southwest Research Institute in evaluating the driver air cushion restraint system. The first of these involved a series of static deployments to determine what inflation hazards, if any, might exist. The driver cushion as initially submitted did

not produce severe inflation hazards; however, it was found possible to modify the fold pattern thereby reducing the likelihood of abrasions to areas of the body contacted during deployment of the air cushion. Refinements of this type to crash deployed systems cannot be accomplished through available anthropomorphic dummy testing. Properly programed human volunteer tests must continue to be required in these development programs.

The second series of human volunteer tests of the driver air cushion restraint involved a "startle" program. In this case, drivers with no prior experience with air cushions were subjected to unanticipated deployments while driving on a private road system. In all cases, the drivers recovered from the initial element of surprise and retained control of the car, bringing it to a safe stop on the roadway. Although it was impossible to pretest volunteers representing every segment of the population, this test series provided confidence that most drivers have the capability of recovering from unanticipated events of this type. The complete system program, of course, was proceeding with the development of designs for the highest possible reliability. This would keep the number of inadvertent firings to an absolute minimum and avoid exposing drivers to inadvertent firing of the air cushion on the highway. Nevertheless the volunteer test program indicated that if through some unforeseen event, inadvertent firing did occur, the driver would likely have the ability to bring the vehicle to a safe stop.

The third human volunteer test program conducted by Southwest Research Institude for General Motors involved the sled testing of human volunteers with the driver air cushion restraint system. Details of this work are included as Part III of this paper. In summary, the results of these controlled direct frontal impact simulations were successful.

PROGRAM TO PREDICT EFFECTIVENESS OF RESTRAINT SYSTEMS

Although data were avilable from the impact sled and full scale controlled barrier crash tests of proposed air cushion restraint systems, an intensive program was necessary to utilize these laboratory test results in predicting the effectiveness of restraint systems in the field. This work was initially published at the GM Automotive Safety Engineering Seminar, June, 1973: "Restraint System Effectiveness – A Study of Fatal Accidents." In summary, the projected effectiveness was estimated as follows:

- Air Cushion – 18%
- Air Cushion and Lap Belt (when used) – 29%
- Lap Belt (when used) – 17%
- Lap and Shoulder Belt (when used) – 31%

For example, these data mean that for an air cushion alone, an individual has an 18% chance of survival over a spectrum of reported accidents where occupants have been killed. Until sufficient field reports covering accidents with air cushion restraints are available, predictions of this type must be used. They must also be updated as frequently as possible. The effectiveness of these systems will change as air cushion or belt systems are improved. This, then, is a continuing program.

AIR CUSHION RESTRAINT SYSTEM FIELD TRIAL PROGRAM

Recognizing the importance of obtaining actual field data on the new air cushion restraint system, General Motors built driver and front seat passenger air cushion restraint systems into a fleet of 1,000 1973 model Chevrolet Impalas. These were constructed between October, 1972, and February, 1973. About one half of these cars have been used in GM fleets and one half have been leased to various companies and police fleets. The initial results of the program were described in a paper, "The 1,000 Air Cushion Field Trial Program" at the GM Automotive Safety Seminar, June, 1973. This program is continuing into its second year of operation. As of May 17, 1974, this fleet had accumulated 35.1 million miles of operation. During that time, 358 accidents have been recorded, most of them in the low severity range. There have been 12 accidents of sufficient severity to deploy the air cushion restraint system. There has been one inadvertent firing of the passenger cushion only. In this case, the driver brought the car to a safe stop. The woman passenger suffered a minor injury to one hand.

Of considerable significance is the fact that in the 35 million miles of exposure, the air cushion has deployed in the type of accidents and at the approximate severity levels anticipated in the engineering design. As additional information is obtained, refinements in the accident sensing and deployment systems may be desirable; however, with the limited amount of information presently available, immediate changes do not seem to be necessary. Data from these accidents continue to be analyzed in depth. Most of them have been covered by the NHTSA MDAI field teams. Their reports are being made public. Injuries to the occupants of cars in accidents with air cushion restraints have been minor with one exception – a driver leg fracture which was considered a moderate injury. In that same accident, an infant lying unrestrained on the seat was fatally injured. The cause of death cannot be determined with certainty.

Conclusions should not be drawn from such a limited sample of accidents which, for the most part, were in the low severity range. None have exceeded 30 mph equivalent barrier speed, and most have been less than 20 mph. Because the data do not lend themselves to a statistical analyses of the effectiveness of air cushions, a comparison has been made between air cushion deployment accidents and similar production car accidents (Chart A). These data indicate there would not be a significant difference in injury severity between the unrestrained and air cushion restrained drivers involved in most of these accidents.

PROGRAM OF CRASH TESTING GM'S PRODUCTION AIR CUSHION

Because it was known that the 1,000 cars in the Field Trial Program would not experience a sufficient number and variety of severe accidents to quickly evaluate the air cushion effectiveness, a program was developed to crash test a number of the production vehicles. To date, 16 full scale tests have been completed to supplement the pre-production evaluation programs, the compliance tests, and the field accident data. This crash test program has been planned to study the impact configuration, the mileage and environmental effects as well as occupant size and position mix. Multiple impacts have also been considered in the development and test of the air cushion. Both the driver and passenger cushion apparently remain inflated long enought to provide significant secondary impact protection during most multiple impact collisions. Details are included in Part II of this paper.

Data from these tests continue to show that the air cushion restraint system will detect accident severity and deploy the cushions as designed. In this series, of course, the results are measured in terms of injury criteria on the anthropomorphic dummy. It is difficult, however, to express complete satisfaction with the system because the dummy is not an adequate simulator of human characteristics.

RESPONSES TO PROPOSED FEDERAL SAFETY STANDARDS

Responses to the latest NHTSA notice of proposed rulemaking on passive restraints are being filed June 3, 1974. These responses will provide additional technical information on air cushion development and testing. The item of significance here is that even though the results of crash test programs utilizing dummies may be favorable, this does not mean they

CHART A
GM FIELD TRIAL AIR CUSHION DEPLOYMENTS DRIVER INJURY COMPARISONS – RESTRAINED (ACRS) VERSUS UNRESTRAINED DRIVERS IN SIMILAR ACCIDENTS

AIR CUSHION CASES		NON AIR CUSHION CASES				
Accident Location	ACRS Driver Injury Level (AIS)	Observed Unrestrained Driver Injury Levels				Number of Similar Cases
		Non-Minor	Moderate	Severe-Serious	Critical-Fatal	
Washington, DC (I)	Minor	100% (67)	0% (17)	0% (17)	0% (0)	9 (6)
Schererville, Ind.	Moderate	56% (47)	44% (34)	0% (19)	0% (0)	9 (32)
Valley Park, Mo.	Minor	50% (50)	13% (30)	31% (20)	6% (0)	16 (20)
Philadelphia, Pa.	Minor (+L/belt)	–	–	–	–	–
Homewood, Ala.	Minor	82% (86)	9% (9)	9% (5)	0% (0)	11 (22)
Detroit, Mich.	Minor	82% (100)	14% (0)	4% (0)	0% (0)	22 (13)
Washington, DC (II)	Minor	75% (79)	13% (21)	12% (0)	0% (0)	8 (19)
Gaylord, Mich.	Minor	100% (100)	0% (0)	0% (0)	0% (0)	15 (13)
Grayson, Ala.	Minor	100%	0%	0%	0%	11
Richmond, Ind.	Minor	73%	0%	27%	0%	11
Rosenburg, Texas	Minor	–	–	–	–	1
Highland, Mich.	None	92%	8%	0%	0%	12

% – indicates figures from computer selection of similar cases
( ) – indicates figures from jury selection of similar cases

indicate adequate protection of humans. There is evidence, for example, that the maximum femur loads permitted by the proposed Federal Standard are too conservative and may result in the discard of knee restraint designs that would otherwise be entirely practicable. On the other hand, the anthropomorphic dummy appears to be inadequate to evaluate rebound characteristics. The comparison of human volunteer and dummy results from identical sled tests show the results are not similar. In most cases the results measured on the dummy appear to be of higher severity than on the human.

Dummy development programs and test development programs to provide better simulation of highway conditions continue on a high priority basis.

PRODUCTION AIR CUSHION RESTRAINT SYSTEM – FIELD ACCIDENT DATA COLLECTION

General Motors is making front seat air cushion restraint systems available as production options on many Cadillacs, Oldsmobiles and Buicks. Because the predicted effectiveness of restraint systems indicated the overall protection would be greater with a combination lap belt and air cushion, these GM cars with air cushions are also provided with lap belts. The production schedule will permit 100,000 installations per year. If this rate could have been accomplished, sufficient field data could have been obtained in one to two years to determine the statistical effectiveness of air cushion restraint systems. Unfortunately, the demand for the air cushion restraint system option to date has been far below the production capability. The time for a significant statistical analyses has thus been delayed.

Field accidents involving General Motors production cars with air cushion restraint systems are being investigated by the MDAI teams of the NHTSA. To date there have been four deployment accidents. These have all been located promptly and the field data collected in depth. There has been one moderate injury, a fractured pelvis experienced by the center front seat occupant of a car involved in a two car frontal impact. The driver of this same air cushioned equipped car received only a minor injury. Occupants of the other vehicle, a 1969 ElCamino pick-up, were very seriously injured.

CONTINUING PROGRAMS

Although some work has been accomplished in the development of air cushion restraint systems for smaller vehicles, much remains to be done. Some of the major areas to be included in such programs are location of air cushion components, sensor requirements for small car impact characteristics, possible inflation hazards, and weight.

The combination of testing for compliance to the Federal Motor Vehicle Safety Standard 208, human volunteer tests, special laboratory investigations, the 1,000 Car Field Test Program, the extensive series of crash tests, is as comprehensive a program as has ever been completed for an automotive safety component. As the data from air cushion field accident program increases, the true capability of the air cushion restraint system will be established.

PART II – CRASH TESTING THE GENERAL MOTORS AIR CUSHION

R.A. WILSON
Automotive Safety Engineering
Environmental Activities Staff
General Motors Corporation

INTRODUCTION

General Motors has been a leader in the design development, and testing of the air cushion, a restraint system designed to meet the passive requirements of FMVSS 208. GM currently is offering the ACRS in 1974 as an extra-cost option on some Oldsmobiles, Buicks, and Cadillacs. The ACRS itself is a passive restraint system, which means it does not require any action by the occupant, such as buckling up lap/shoulder belts. Our ACRS vehicles for 1974 have lap belts provided as standard equipment to provide (a) additional protection against injury in accidents in which the air cushion is activated and (b) protection not afforded by air cushions in those types of accidents (such as side impacts and rollovers) where the air cushion is not designed to activate. Prior to offering the ACRS as a 1974 option, GM established a 1,000-car ACRS Field Trial Program (FTP) with several goals in mind. These goals were: (1) to gain production and field service experience with a totally passive air cushion restraint system; (2) to see experience with representative hardware in a broad range of environmental and operating conditions; and (3) to expose the passive restraint system to real-life, driver and traffic situations.

FMVSS 208 contains certain test requirements that must be met by vehicles to be certified under the passive restraint option. These include direct front, angular front, and side impacts, all using 50th percentile dummies only. This paper presents a supplemental crash test program, completed by GM on some of these Field Trial Program ACRS vehicles that goes beyond the basic tests of FMVSS 208. GM determined that this part of the FTP evaluation was necessary to study the ACRS performance in a variety of field accident conditions.

EARLY SLED TESTING

GM built the first of the 1,000 FTP ACRS vehicles in late 1972. Prior to that, the ACRS was subjected to many different laboratory development tests by GM divisions. Also, crash simulation variables were tested on the Hyge Impact Sled at the Safety Research and Development Laboratory. The types of variables studied on the sled and reasons for their evaluation are discussed below.

Environmental Conditions

The ACRS was evaluated in a dynamic sled test, using three dummies, after the system was "soaked" at 180°F and −20°F. These conditions simulated temperature exposure in the extreme hot and cold regions of the U.S. and are part of GM's qualifications series.

Sled test also were run after the system was subjected to the following cycle of environmental conditioning: 7 days at 100°F and 100% relative humidity; 24 hours of vibration testing at 70°F; 12 hours vibration at 180°F and 12 hours vibration at −20°F; 12 hours soak at 220°F; and a 12-hour soak at −40°F.

Occupant Size and Position Mix

The system was evaluated using many different dummy sizes, seating positions, and seat adjustments to determine the ACRS performance under varied loading. Many occupant size and position mix tests were considered, but because of the many combinations of these variables, only selected conditions were tested on the Hyge Impact Sled to determine the ACRS performance. Some of these tests (run at 30 mph, unless otherwise indicated) included:

- 95th percentile male, driver and right front passenger dummies* with seat in full rear.
- 50th percentile male, driver and right front passenger dummies** with seat in mid position.
- 95th percentile male, driver dummy and 5th percentile female right front passenger dummy# with seat in full rear position.
- 5th percentile female, driver dummy and 95th percentile male, right front passenger dummy with seat in full forward position.
- Three 50th percentile male, dummy occupants in panic braking position.
- 95th percentile male, driver dummy and 3-year-old male, center front passenger dummy##, and 95th percentile male, right front passenger dummy with seat in full rear position.

*Sierra 292-895
**GM Hybrid II
#Sierra 292-805
##Sierra 492-103

- 3-year-old male, right front passenger dummy standing on floor with hands on strument panel.
- 3-year-old male, right front passenger dummy seated with the seat in rear position.
- A single 95th percentile male, right front passenger dummy.
- A single 5th percentile female, right front passenger dummy.
- Three 50th percentile male, dummy occupants at 15 and 30 mph.

Child Restraint Systems

These tests were conducted with the GM Infant "Love Seat" and Child "Love Seat" to determine their performance with the ACRS. The child restraints were installed using normal lap belt hardware in the right front passenger position.

Failure Mode Analysis

These tests evaluated the restraint capability of the passenger ACRS if the inflator and/or the air cushions were inadvertently damaged. Some of these tests included:

- Release of the stored gas simultaneous with actuation of both the low and high level gas generators.
- Actuation of the low level gas generator without actuation of the detonator that is designed to rupture the diaphram. This mode causes increased inflator pressure, which eventually causes an overpressure rupture of the diaphragm.
- Intentional tears in the torso and knee cushions.

Final Qualification Tests

The final-design ACRS hardware for the FTP fleet was evaluated in a series of similar tests, including occupant size and position mix and environment conditions to determine the performance of the ACRS. Satisfactory ACRS hardware performance was required.

There were many other conditions evaluated in each of the above-mentioned areas, but they are too numerous to detail in this paper. The intent of this introduction is to show that this ACRS system was evaluated extensively and further developed for a wide range of performance before the system went into Field Trial Program production. Simply meeting FMVSS 208 was not considered sufficient.

CRASH TEST PROGRAM DEVELOPMENT

Following the successful completion of the sled test series, production of the 1,000 Field Trial Program vehicles was initiated and our full-vehicle crash test program was begun. The vehicles used in this program were 1973 Chevrolet Impala 4-door sedans, assembled at the Doraville, Georgia plant.

In determining a crash test program to evaluate the ACRS, rationales were developed to gain the most information about the performance of the passive restraint system within a reasonable number of tests. The rationales were determined after analyzing field accident data for typical, high frequency-of-occurrence accidents, and for typical occupant size and seated positions. These conditions were sought to evaluate the restraint capability of the ACRS in simulated real-world accident conditions. Other considerations in selecting the test program were the sensor performance in different accident configurations, the effect of environmental conditions and mileage accumulation, and the existence of particular conditions in real FTP field accidents. Following are the rationales used to determine the crash test program.

Impact Configuration Environment

In this rationale category, tests were developed from field accident data and FTP accident investigation results to produce important data about the performance capabilities of the sensing systems, and to assist accident investigators in their analysis of future accidents. Tests were included in this category where the probability for deployment was high, or important crash sensing data could be obtained, or it was felt that occupant injury might result in a similar field accident. In this category, the rationales were distinctly divided into three (3) areas, as follows:

1. **Field Accident Configuration Comparison.** This rationale included accident simulations of a variety of high-occurrence, real-world accidents from our GM accident files and from actual accidents involving FTP vehicles. Examples of these types of tests were: car-to-pole, frontal impact with bumper underride, two-car moving 90° intersection simulation, chain-reaction simulation, and a 40 mph frontal impact.
2. **Injury Criteria.** Several tests were conducted primarily to determine dummy injury criteria.* Several examples of these tests were: 30 mph frontal with ACRS and lap belts, 40 mph frontal barrier, and all occupant size and position mix tests.

*Although dummy injury criteria were evaluated, GM does not believe that test technology has progressed to the point where these results, in themselves, can be used as a simple measure of the acceptability of restraint systems. Values are given later in this paper that exceed the FMVSS 208 limits. This does not establish that the FTP air cushion restraint system is unacceptable for field use. A more complete discussion of current test technology can be found in GM's response to Docket 74-14, Notice 1, related to FMVSS 208.

3. **Sensor Performance.** Crash sensor and bumper impulse detector performance was an important objective of the test program, since the crash sensing system is the *heart* of the ACRS. The tests selected for evaluation of the sensing system recognized:
   - High-occurrence accidents seen in the field accident file,
   - Field results from the FTP program, and
   - Accidents in which there was a likelihood of a deployment.

   These situations were analyzed and discussed to estimate the possibility of sensor actuation under various conditions. Tests of these types included: the frontal barrier with bumper underride; 2-car, 45° side angle impact; 2-car moving 90° intersection simulation; 30° car-to-car side angle impact; and the lateral dolly rollover.

Mileage and Environmental Effects

Durability and environmental conditions were considered in order to study the effects of humidity, cold and hot weather, salt air exposure, vibration, and high mileage (45,000 miles, plus) on the performance of the ACRS. Vehicles were obtained from the FTP fleet that had been operated in the various locations of the U. S. known for these exposure conditions. The tests were conducted to confirm that no bumper impulse detector, plenum crash sensor, or ACRS component degradation had occurred. Each test used three 50th percentile male dummies in the front seat positions.

Occupant Size and Position Mix

This rationale was selected to evaluate the complete ACRS vehicle performance under different combinations of occupant size and seating position. The combinations of these variables are numerous and add to the complexity of providing a passive restraint system suitable for the real-world. As was stated earlier, many of these combinations of variables were conducted on GM's Impact Sled prior to production of air cushion restraint systems. After analyzing the sled test results, field accident data, and everyday real-life situations, the following tests were selected to be evaluated in barrier impact crashes:

- 30 mph frontal impact with two 50th percentile male, lap-belted dummies. This is the only test in which lap belts were added.
- 30 mph frontal impact with a 95th percentile male, driver dummy, 3-year-old male, center front passenger dummy (seated), and a 5th percentile female, right front passenger dummy.
- 30 mph frontal impact with a 50th percentile male, driver and right front passenger dummies.
- 30 mph frontal impact with 5th percentile female, driver dummy and 95th percentile male, right front passenger dummy with knees against ACRS cover.

TESTS AND RESULTS

Each test in the crash test program and the accident configuration it simulates is discussed in this portion of the paper. A brief summary of the results also is given that includes the plenum crash sensor and bumper impulse detector performance, the air cushion's restraint capability, dummy injury criteria data, and some vehicle performance. Refer to the Appendix for a tabular summary of the results. The summary page numbers are keyed to the following section headings. Again, the impact tests were divided into three major categories:

- Impact configuration environment
- Mileage and environmental effects
- Occupant size and position mix

Impact Configuration Environment

30 mph Rear Moving Barrier (C-3000). This test simulated an ACRS vehicle being impacted in the rear end. The purpose of the test was to determine if the plenum crash sensor or bumper impulse detector would actuate the ACRS in this type of accident configuration. As was intended and expected, neither the plenum crash sensor nor the bumper impulse detector actuated during the test. Dummy occupants were not used for this test.

60 mph Car-to-Parked Car (Front-to-Front) (C-3002). The second impact test simulated a vehicle traveling 30 mph, crossing the road centerline, and impacting an oncoming vehicle, which was also traveling 30 mph. The two vehicles hit with a 30% frontal overlap on the driver dummy's side. Both ACRSs were deployed by the bumper impulse detectors and provided good upper and lower torso restraint for the dummies. The injury criteria were well below the levels specified in FMVSS 208. The driver dummy's side glass was broken on each vehicle and the driver dummies' heads momentarily moved outside the vehicle interior during their rebound motion.

30 mph Car-to-14" Diameter Pole (C-3010). This impact simulated a vehicle impacting a tree, pole, or other narrow object where penetration is greater than with a flat barrier. The pole was positioned in line with the left frame side member and lower steering assembly. This offset was established to provide a good evaluation of the driver's system. The bumper impulse detector actuated the ACRS, which provided good upper and lower torso restraint for the two

dummy occupants. The injury criteria were below the levels specified in FMVSS 208, except for the driver dummy's left femur, which was 1910 lbs.* This femur load was attributed to the counterclockwise yaw motion of the vehicle which caused the driver dummy to move to the left. This resulted in increased load distribution to one femur. The vehicle experienced 37 inches of crush in this test.

30 mph Frontal Barrier With Bumper Underride (C-3027). The fourth test used a barrier configuration which simulated a vehicle underriding the rear end of a truck, or other high vehicle. This produces little or no bumper and frame involvement, early in the collision. The barrier and ACRS vehicle bumper height were 24 and 22 inches, respectively. In this test, the plenum crash sensor actuated the ACRS. The bumper did not contact the barrier, and the bumper impulse detector contacts did not close. The test provided a more severe demand on the ACRS than a typical frontal barrier. With no frame involvement, the upper vehicle sheet metal must experience considerable deformation before the plenum crash sensor detectors will experience sufficient deceleration to actuate the ACRS. The injury criteria were below the levels specified in FMVSS 208, except for the driver dummy's right femur, which was 2060 lbs.*

30 mph Lateral Dolly Rollover (C-3079). This test was conducted using a lateral dolly rollover fixture as specified in FMVSS 208. The ACRS was actuated by an external trigger, instead of the ACRS crash sensing system. The plenum crash sensor and bumper impulse detector circuit monitoring instrumentation indicated that the contacts of neither unit closed during the rollover sequence. The ACRS sensors are designed to actuate in a rollover only if sufficient forward deceleration is present. The reason for externally triggering the ACRS was to assure an evaluation of its restraint capability during a rollover where deployment did occur. The dummy injury criteria were well below the levels specified in FMVSS 208 and both dummies experienced lateral movement and dummy-to-dummy contact. The driver dummy momentarily experienced partial ejection* of the left shoulder and arm, and the right front passenger dummy momentarily experienced partial ejection of the right hip, arm, and knee during the roll sequence.

Two-Car Moving 90° Intersection Simulation (C-3128). Test six simulated an intersection collision in which the striking and struck vehicles were traveling 30 and 20 mph, respectively. This test was conducted using a method in which the striking vehicle was towed in at 36 mph and at an angle of 34° into a parked vehicle, which was oriented perpendicular to the striking vehicle. The moving car's front and rear wheels were angled to allow the proper alignment at impact. Vector summation shows that the simulation does represent an actual 90° intersection collision. Figure 1 gives the vector summing diagrams of the accident simulation. In the striking vehicle, the ACRS was actuated by the bumper impulse detector and the plenum crash sensor added the high level passenger stage of the system. The struck vehicle received an impact with some forward component and the ACRS was actuated by the plenum crash sensor. Both systems provided good restraint for the dummies. The injury criteria were below the levels specified in FMVSS 208. Compared to similar field accidents, the simulation produced a very similar type and extent of side and frontal sheet metal damage.

30 mph Two-Car 30° Side Angle Impact (C-3281). This test simulated a vehicle turning into the path of an oncoming vehicle traveling at 30 mph. The striking vehicle's right frame side member was aligned with the struck vehicle's front suspension. The ACRS in the struck vehicle was deployed by the plenum crash sensor. There was no bumper involvement of the struck vehicle with the striking vehicle. The system did not deploy in the striking vehicle since the deceleration forces did not reach the threshold level of the ACRS system. Threshold level was not reached by the striking vehicle due to its front end crush characteristics and the geometry of this collision. The occupant injury criteria were well below levels specified in FMVSS 208.

40 mph Frontal Impact (C-3336). This test was conducted according to FMVSS 208 test conditions, except for speed. It was run to evaluate the ACRS restraint and sensor system performance at a speed substantially above the FMVSS 30 mph requirement. In this test, the low level ACRS was deployed by the bumper impulse detector, and the high level gas generator in the passenger system was actuated by the high level detectors in the crash sensor. The bumper impulse detector and plenum crash sensor detector closure times were earlier than in typical 30 mph frontal impacts. The ACRS appeared to provide good upper and lower torso restraint for the dummies. However, the center front passenger dummy HIC and Chest SI exceeded the 30 mph limits of FMVSS 208.* Also, the driver dummy's left femur load and right front passenger dummy's right femur load exceeded the limits of FMVSS 208. There was no upper instrument panel or windshield contact, and no ACRS hardware problems during the test. The

*See note on Page 469.

*See note on Page 469.

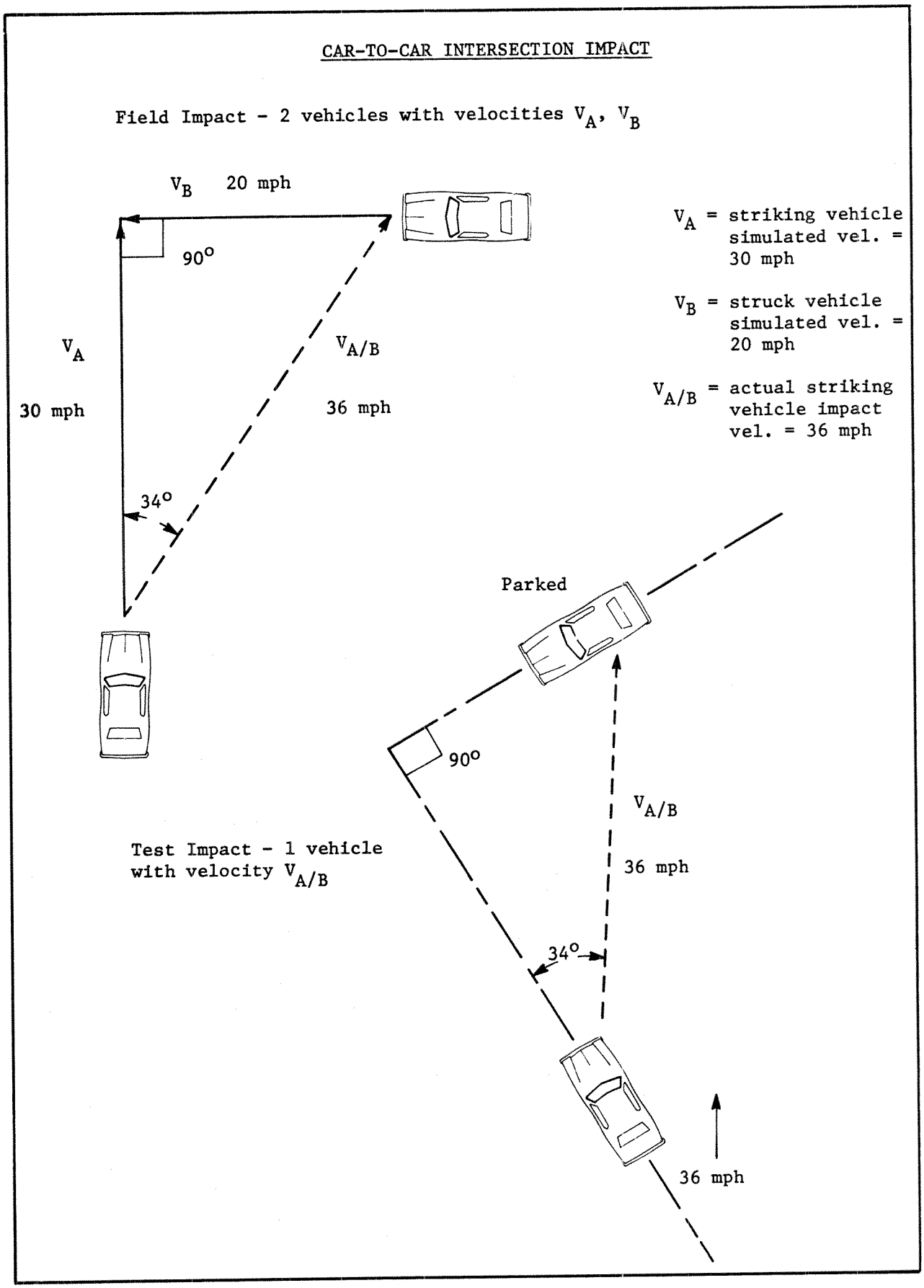

CAR-TO-CAR INTERSECTION IMPACT
Field Impact - 2 vehicles with velocities V_A, V_B
V_B 20 mph
90°
V_A
30 mph
$V_{A/B}$
36 mph
34°
V_A = striking vehicle simulated vel. = 30 mph
V_B = struck vehicle simulated vel. = 20 mph
$V_{A/B}$ = actual striking vehicle impact vel. = 36 mph
Parked
90°
$V_{A/B}$
36 mph
34°
36 mph
Test Impact - 1 vehicle with velocity $V_{A/B}$

Figure 1

driver dummy moved off the seat upon deformation of the knee restraint, and his head rebounded into the side door structure.** The center front and right front passenger dummies rebounded normally into the seat back. The vehicle experienced 36 inches of frontal crush in the test.

"Chair Reaction" Collision Simulation (C-3319). This test simulated an accident involving a stopped vehicle at an intersection, a second vehicle in a brake dive mode, and a third vehicle, or striking vehicle, traveling 40 mph. The third vehicle impacted the middle vehicle, pushing it into the lead vehicle. The striking vehicle's air cushion restraint system was deployed by the bumper impulse detector and the high level gas generator in the passenger inflator was actuated by the plenum crash sensor. The ACRS did not deploy in the middle vehicle because of the bumper underride with the lead vehicle and the impact severity being below the threshold of the plenum sensor. The test data indicated the deceleration of the middle car was not sufficient to actuate the low level crash sensor detectors. The dummy injury criteria were significantly less than the levels specified in FMVSS 208. The lead vehicle was not equipped with an air cushion restraint system.

40 mph, Two-Car 45° Left-Side Angle Side Impact (C-3340). Test ten simulated a vehicle crossing the vehicle centerline and impacting the other vehicle on the left side at a 45° angle, just behind the cowl area. This accident would evaluate the plenum crash sensor for its response to a longitudinal component from the 45° side impact. The ACRS did not deploy in the ACRS vehicle since the deceleration forces did not reach the threshold level of the plenum crash sensor detectors. The injury criteria were below the levels specified in FMVSS 208, even though the front seat dummies experienced head-to-head contact during their lateral movement.

Mileage and Environmental Effects

30 mph Frontal Impact (C-3352). This test evaluated the ACRS performance in a vehicle which had been operated for 45,000 miles in the Field Trial Program. Also, this vehicle had been subjected to two winters of cold weather in Minnesota. This test was conducted to determine if any bumper impulse detectors, plenum crash sensor, or ACRS performance degradation had occurred due to a cold and wet environment. The ACRS and vehicle hardware were visually inspected prior to the impact test. The ACRS deployed properly. The bumper impulse detector and plenum crash sensor test data indicated the crash sensing system did not experience any system performance degradation. The ACRS cushions provided good upper and lower torso restraint. The injury criteria were below the levels specified in FMVSS 208.

30 mph Frontal Impact (C-3353). This test evaluated the ACRS performance of a vehicle which also had been operated for 45,000 miles in the Field Trial Program. This vehicle had been subjected to two summers of hot weather at the General Motors Proving Ground in Mesa, Arizona. This test was conducted to determine if any bumper impulse detector, plenum crash sensor, or ACRS performance degradation occurred due to the hot and dry environment. The ACRS and vehicle hardware were visually inspected prior to the impact test. The ACRS deployed properly. The bumper impulse detector and plenum crash sensor test data indicated the crash sensor system did not experience any system performance degradation. The ACRS cushions provided good upper and lower torso restraint. The injury criteria were well below the levels specified in FMVSS 208.

Occupant Size and Position Mix

30 mph Frontal Impact With Two Lap-Belted Dummy Occupants (C-3094). This test was conducted to evaluate the performance of the ACRS with the two outboard dummies lap belted. The ACRS provided good upper torso restraint, although the right front passenger dummy experienced slightly more cushion penetration than usual for an unbelted dummy. The penetration was attributed to the higher upper torso forward velocity obtained with lap belt restraint. The injury criteria were below the levels specified in FMVSS 208.

30 mph Frontal Impact With A 95th Percentile Male Driver Dummy, 3-Year-Old Male Center Front Passenger Dummy (Seated), and A 5th Percentile Female Right Front Passenger Dummy (C-3347). This occupant size and position mix test evaluated a configuration that is likely in a family automobile. The seat was in the full rear position. The ACRS deployed properly and the cushions provided good upper and lower restraint for the driver, CF and RF passenger dummies. The 5th percentile female RF dummy experienced slightly more cushion penetration than larger dummies, but had no windshield or instrument panel involvement. The CF passenger dummy rebounded low into the seat. The injury criteria were well below the levels specified in FMVSS 208.

**In the GM response referenced by the note on Page 469, a specific recommendation is made to delete rebound contacts from the HIC calculation because of the inadequacy of the dummy in simulating human response to non-frontal and short duration impacts. The data summary for this test reflects this procedure.

30 mph Frontal Impact With Two Unrestrained Occupants (C-3321). This test is similar to the first of the occupant size and position mix tests. C-3094, except that the only restraint was the ACRS (i.e., the passenger dummies had no lap belts). The ACRS provided good upper and lower torso restraint for both dummies. The driver dummy rolled off the steering wheel cushion and rebounded into the B pillar.** The pre-test position of the driver dummy is outboard of the steering column centerline. This dummy positioning and the resulting off-center loading of the driver cushion and knee restraint, contributes to the rebound kinematics toward the B pillar. The injury criteria were acceptable.

30 mph Frontal Impact With A 5th Percentile Female Driver Dummy, and A 95th Percentile Male Right Front Passenger Dummy (C-3346). This test was conducted to evaluate the ACRS performance when the right front passenger dummy's knees are against, or close to, the passenger system vinyl cover. The seat was in the full forward position. The ACRS deployed properly and the cusions provided good upper and lower restraint for both dummies. The passenger cushion was delayed in deploying due to the 95th percentile male RF passenger dummy's knees being close to the ACRS cover, but the cushion inflated in sufficient time to prevent windshield and instrument panel involvement. The injury criteria were below the levels specified in FMVSS 208.

SUMMARY

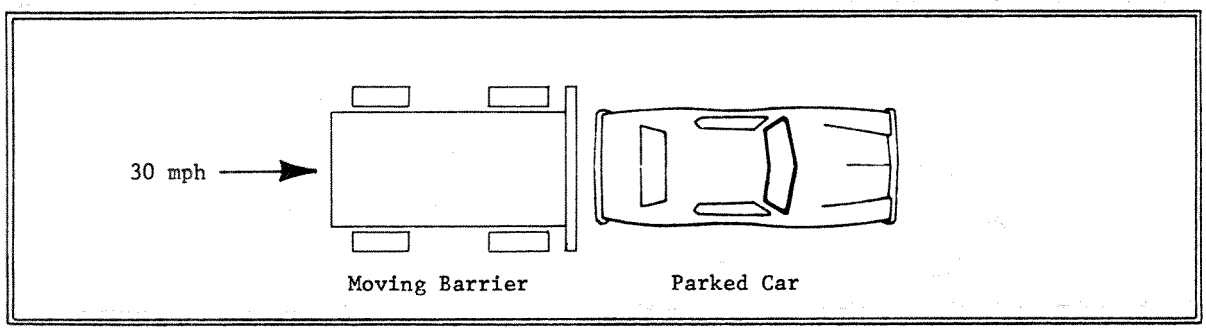
The introduction of a complex safety system such as the ACRS requires much more testing and evaluation than simply meeting FMVSS 208. The scope of the test program described in this paper is some evidence of that. Even these tests, however, are not sufficient in themselves. Additional test programs using cadavers and human volunteers have been accomplished to be more confident that the ACRS is ready for use by real people in real accidents.

**See note on Page 469.

CRASH TESTING GM'S PRODUCTION AIR CUSHION

TYPE OF TEST 30 mph Rear Moving Barrier Impact

IMPACT CONFIGURATION:



RATIONALE FOR TEST:

1. FIELD ACCIDENT CONFIGURATION COMPARISON ..... ☐
2. INJURY CRITERIA FROM DUMMIES ..... ☐
3. SENSOR PERFORMANCE ..... ☒
4. ACCIDENT CALIBRATIONS ..... ☐
5. MILEAGE AND ENVIRONMENTAL EFFECTS ..... ☐
6. OCCUPANT SIZE AND POSITION MIX ..... ☐

OTHER ..... ☐

TEST RESULTS:

SENSOR PERFORMANCE

ACTUATION DEVICE	CONTACT CLOSURE	ACTUATED ACRS
BUMPER IMP. DETECTOR	No	No
PLENUM LOW LEVEL DETECTOR	No	No
PLENUM HIGH LEVEL DETECTOR	No	No

DUMMY OCCUPANT DATA

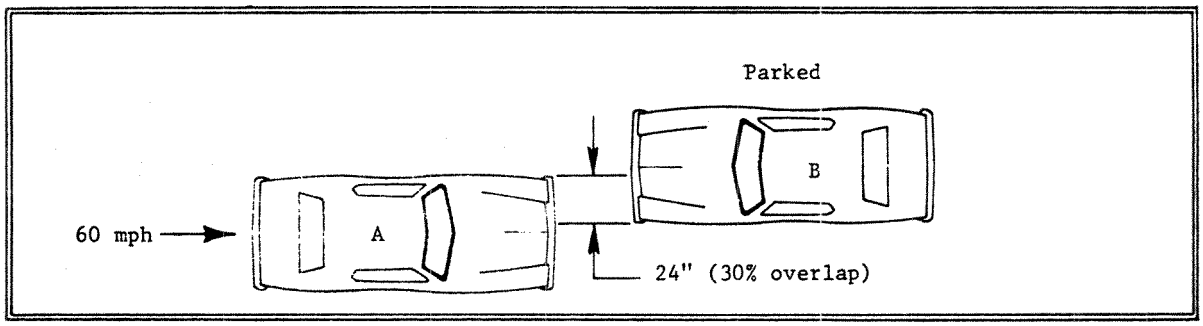
POSITION AND SIZE	HEAD HIC	CHEST SI	CHEST Gs (3Ms)	LEFT FEMUR (LBs)	RIGHT FEMUR (LBs)
		No	Dummies		

COMMENTS: The crash sensors are not designed to actuate in a rear impact collision.

CRASH TESTING GM'S PRODUCTION AIR CUSHION

TYPE OF TEST 60 mph Car-to-Parked Car, 30% Overlap (24"). Simulates Opposing Vehicles Crossing A Highway Centerline, With A Differential Velocity of 60 mph

IMPACT CONFIGURATION:



RATIONALE FOR TEST:

1. FIELD ACCIDENT CONFIGURATION COMPARISON ..... [X]
2. INJURY CRITERIA FROM DUMMIES ..... [X]
3. SENSOR PERFORMANCE ..... [X]
4. ACCIDENT CALIBRATIONS ..... [X]
5. MILEAGE AND ENVIRONMENTAL EFFECTS ..... [ ]
6. OCCUPANT SIZE AND POSITION MIX ..... [ ]

OTHER ..... [ ]

TEST RESULTS:

SENSOR PERFORMANCE

CAR A

ACTUATION DEVICE	CONTACT CLOSURE	ACTUATED ACRS
BUMPER IMP. DETECTOR	Yes	Yes
PLENUM LOW LEVEL DETECTOR	Yes	Redundant
PLENUM HIGH LEVEL DETECTOR	Yes	Yes

CAR B

ACTUATION DEVICE	CONTACT CLOSURE	ACTUATED ACRS
BUMPER IMP. DETECTOR	Yes	Yes
PLENUM LOW LEVEL DETECTOR	Yes	Redundant
PLENUM HIGH LEVEL DETECTOR	Yes	Yes

DUMMY OCCUPANT DATA

CAR A

POSITION AND SIZE	HEAD HIC	CHEST SI	CHEST Gs(3Ms)	LEFT FEMUR (LBs)	RIGHT FEMUR (LBs)
DR GM50	150	150	28	1040	630
CF	Not	Present			
RF GM50	100	90	22	610	540

CAR B

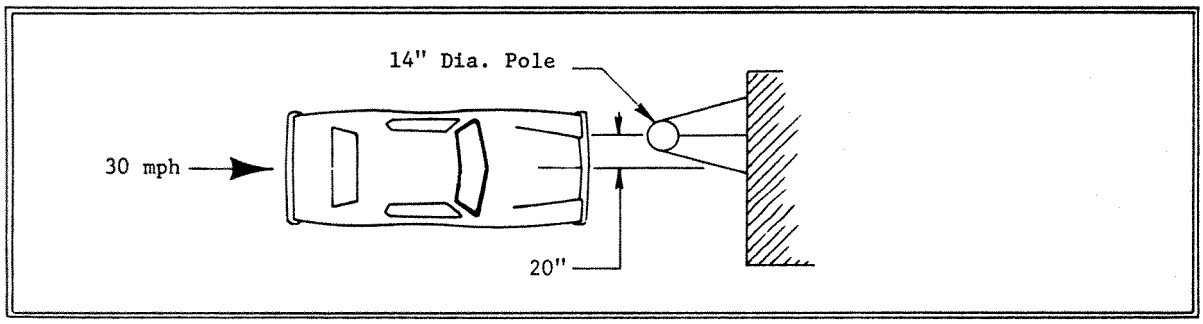
POSITION AND SIZE	HEAD HIC	CHEST SI	CHEST Gs(3Ms)	LEFT FEMUR (LBs)	RIGHT FEMUR (LBs)
DR GM50	140	210	42	1060	480
CF GM50	160	120	28	500	780
RF GM50	80	50	16	610	380

COMMENTS: (1) Vehicle B (struck) was used previously in the 30 mph rear moving barrier impact, (2) Driver side glass broken on each vehicle due to deformation, and (3) Driver dummies' heads momentarily moved outside the vehicle interior during rebound.

CRASH TESTING GM'S PRODUCTION AIR CUSHION

TYPE OF TEST 30 mph Car-to-14" Diameter Pole. Simulates An Off-Center Impact With A Tree Or Narrow Abutment

IMPACT CONFIGURATION:



RATIONALE FOR TEST:

1. FIELD ACCIDENT CONFIGURATION COMPARISON ..... [X]
2. INJURY CRITERIA FROM DUMMIES ..... [X]
3. SENSOR PERFORMANCE ..... [X]
4. ACCIDENT CALIBRATIONS ..... [X]
5. MILEAGE AND ENVIRONMENTAL EFFECTS ..... [ ]
6. OCCUPANT SIZE AND POSITION MIX ..... [ ]

OTHER ..... [ ]

TEST RESULTS:

SENSOR PERFORMANCE

ACTUATION DEVICE	CONTACT CLOSURE	ACTUATED ACRS
BUMPER IMP. DETECTOR	Yes	Yes
PLENUM LOW LEVEL DETECTOR	Yes	Redundant
PLENUM HIGH LEVEL DETECTOR	Yes	Yes

DUMMY OCCUPANT DATA

POSITION AND SIZE	HEAD HIC	CHEST SI	CHEST Gs (3Ms)	LEFT FEMUR (LBs)	RIGHT FEMUR (LBs)
DR GM50	660	400	44	1910*	620
CF	Not	Present			
RF GM50	520	530	53	1040	910

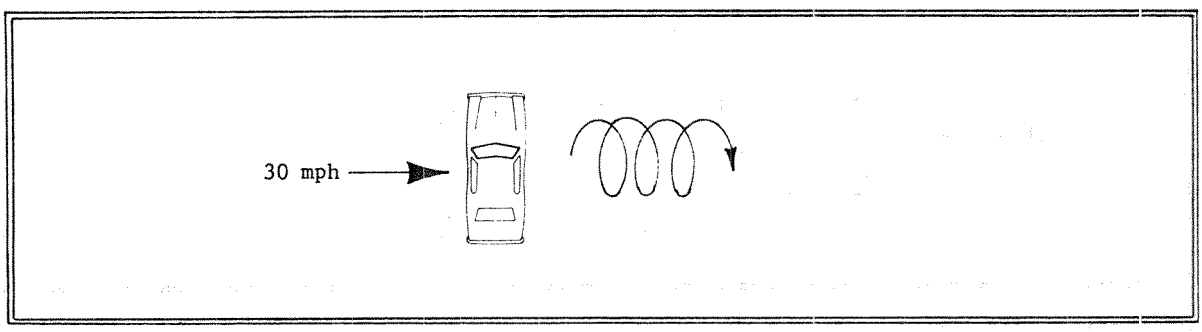
*See note on Page 4

COMMENTS: (1) Injury criteria are below the levels specified in FMVSS 208, except for the driver dummy's left femur (1910 lbs). (2) The pole was positioned directly forward of the left frame side member and lower steering assembly.

CRASH TESTING GM'S PRODUCTION AIR CUSHION

TYPE OF TEST 30 mph Lateral Dolly Rollover. The test was conducted per the requirements of FMVSS 208.

IMPACT CONFIGURATION:



RATIONALE FOR TEST:

1. FIELD ACCIDENT CONFIGURATION COMPARISON ..... ☐
2. INJURY CRITERIA FROM DUMMIES ..... ☒
3. SENSOR PERFORMANCE ..... ☒
4. ACCIDENT CALIBRATIONS ..... ☒
5. MILEAGE AND ENVIRONMENTAL EFFECTS ..... ☐
6. OCCUPANT SIZE AND POSITION MIX ..... ☐

OTHER ..... ☐

TEST RESULTS:

SENSOR PERFORMANCE

ACTUATION DEVICE	CONTACT CLOSURE	ACTUATED ACRS
BUMPER IMP. DETECTOR	No	No
PLENUM LOW LEVEL DETECTOR	No	No
PLENUM HIGH LEVEL DETECTOR	No	No

DUMMY OCCUPANT DATA

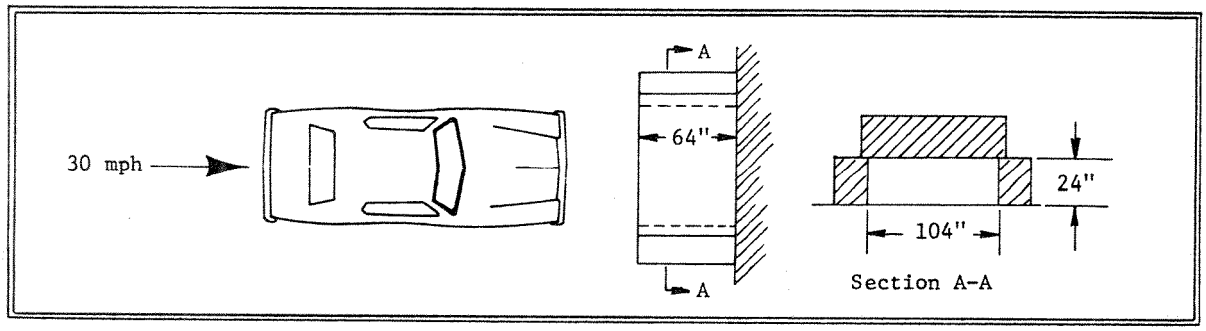
POSITION AND SIZE	HEAD HIC	CHEST SI	CHEST Gs (3Ms)	LEFT FEMUR (LBs)	RIGHT FEMUR (LBs)
DR GM50	220	180	22	370	100
CF	Not	Present			
RF GM50	370	150	29	550	270

COMMENTS: (1) ACRS was actuated from an external trigger instead of the sensing system. The BID and crash sensor circuits were monitored. (2) Driver dummy's shoulder and arm partially ejected, and (3) RF passenger dummy's hip, arm, and knee partially ejected.

CRASH TESTING GM'S PRODUCTION AIR CUSHION

TYPE OF TEST 30 mph Frontal Barrier With Bumper Underride. A Barrier With An Opening Is Used To Simulate A Car Striking The Rear End of a Truck

IMPACT CONFIGURATION:



RATIONALE FOR TEST:

1. FIELD ACCIDENT CONFIGURATION COMPARISON ..... [X]
2. INJURY CRITERIA FROM DUMMIES ..... [X]
3. SENSOR PERFORMANCE ..... [X]
4. ACCIDENT CALIBRATIONS ..... [X]
5. MILEAGE AND ENVIRONMENTAL EFFECTS ..... [ ]
6. OCCUPANT SIZE AND POSITION MIX ..... [ ]

OTHER ..... [ ]

TEST RESULTS:

SENSOR PERFORMANCE

ACTUATION DEVICE	CONTACT CLOSURE	ACTUATED ACRS
BUMPER IMP. DETECTOR	No	No
PLENUM LOW LEVEL DETECTOR	Yes	Yes
PLENUM HIGH LEVEL DETECTOR	Yes	Yes

DUMMY OCCUPANT DATA

POSITION AND SIZE	HEAD HIC	CHEST SI	CHEST Gs (3Ms)	LEFT FEMUR (LBs)	RIGHT FEMUR (LBs)
DR GM50	320**	430	57	1650	2060*
CF GM50	250	200	31	370	1140
RF GM50	970	240	40	670	1100

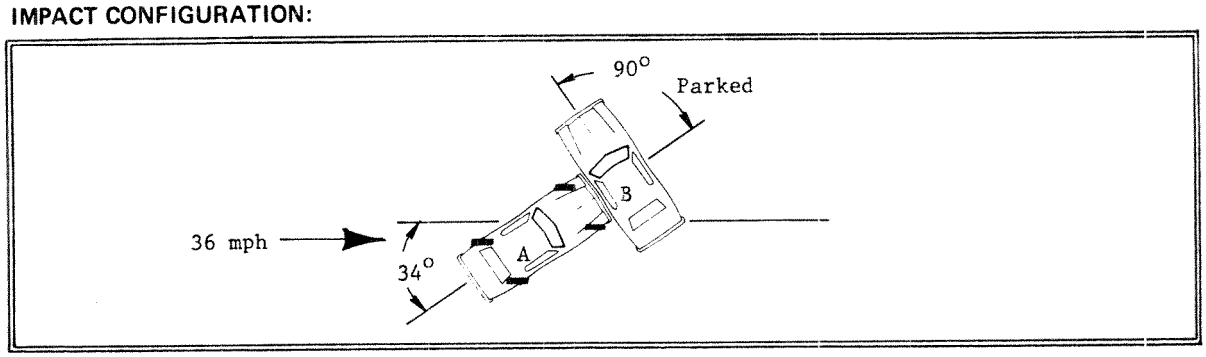
*See note on Page 4
**See note on Page 8

COMMENTS: (1) Injury criteria are below the levels specified in FMVSS 208 except for the driver dummy's right femur (2060 lbs.). (2) The front bumper height was 22".

CRASH TESTING GM'S PRODUCTION AIR CUSHION

TYPE OF TEST Two-Car 90° Intersection Collision (Simulating a 30 mph Striking Vehicle and a 20 mph Struck Vehicle)

IMPACT CONFIGURATION:



RATIONALE FOR TEST:

1. FIELD ACCIDENT CONFIGURATION COMPARISON ..... [X]
2. INJURY CRITERIA FROM DUMMIES ..... [X]
3. SENSOR PERFORMANCE ..... [X]
4. ACCIDENT CALIBRATIONS ..... [X]
5. MILEAGE AND ENVIRONMENTAL EFFECTS ..... [ ]
6. OCCUPANT SIZE AND POSITION MIX ..... [ ]

OTHER ..... [ ]

TEST RESULTS:

SENSOR PERFORMANCE

CAR A

ACTUATION DEVICE	CONTACT CLOSURE	ACTUATED ACRS
BUMPER IMP. DETECTOR	Yes	Yes
PLENUM LOW LEVEL DETECTOR	Yes	Redundant
PLENUM HIGH LEVEL DETECTOR	Yes	Yes

CAR B

ACTUATION DEVICE	CONTACT CLOSURE	ACTUATED ACRS
BUMPER IMP. DETECTOR	No	No
PLENUM LOW LEVEL DETECTOR	Yes	Yes
PLENUM HIGH LEVEL DETECTOR	No	No

DUMMY OCCUPANT DATA

CAR A

POSITION AND SIZE	HEAD HIC	CHEST SI	CHEST Gs(3Ms)	LEFT FEMUR (LBs)	RIGHT FEMUR (LBs)
DR GM50	60	40	16	660	600
CF	Not	Present			
RF GM50	30	10	9	440	560

CAR B

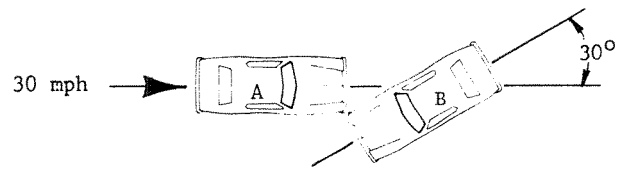
POSITION AND SIZE	HEAD HIC	CHEST SI	CHEST Gs(3Ms)	LEFT FEMUR (LBs)	RIGHT FEMUR (LBs)
DR GM50	170	180	41	50	50
RF GM50	220	50	28	200	200
LR GM50	390	850	99	300	140

COMMENTS: (1) Injury criteria for the front seat dummy occupants were below the levels specified in FMVSS 208.

CRASH TESTING GM'S PRODUCTION AIR CUSHION

TYPE OF TEST 30 mph - 30° Angle Car-to-Car Side Angle Impact Simulates A Vehicle Turning Into The Path Of An OnComing Vehicle

IMPACT CONFIGURATION:



RATIONALE FOR TEST:

1. FIELD ACCIDENT CONFIGURATION COMPARISON ..... ☐
2. INJURY CRITERIA FROM DUMMIES ..... ☒
3. SENSOR PERFORMANCE ..... ☒
4. ACCIDENT CALIBRATIONS ..... ☒
5. MILEAGE AND ENVIRONMENTAL EFFECTS ..... ☐
6. OCCUPANT SIZE AND POSITION MIX ..... ☐

OTHER ..... ☐

TEST RESULTS:

SENSOR PERFORMANCE

CAR A

ACTUATION DEVICE	CONTACT CLOSURE	ACTUATED ACRS
BUMPER IMP. DETECTOR	No	No
PLENUM LOW LEVEL DETECTOR	No	No
PLENUM HIGH LEVEL DETECTOR	No	No

CAR B

ACTUATION DEVICE	CONTACT CLOSURE	ACTUATED ACRS
BUMPER IMP. DETECTOR	No	No
PLENUM LOW LEVEL DETECTOR	Yes	Yes
PLENUM HIGH LEVEL DETECTOR	No	No

DUMMY OCCUPANT DATA

CAR A

POSITION AND SIZE	HEAD HIC	CHEST SI	CHEST Gs(3Ms)	LEFT FEMUR (LBs)	RIGHT FEMUR (LBs)
DR GM50	60	70	29	640	450
CF	Not	Present			
RF GM50	120	30	11	610	930

CAR B

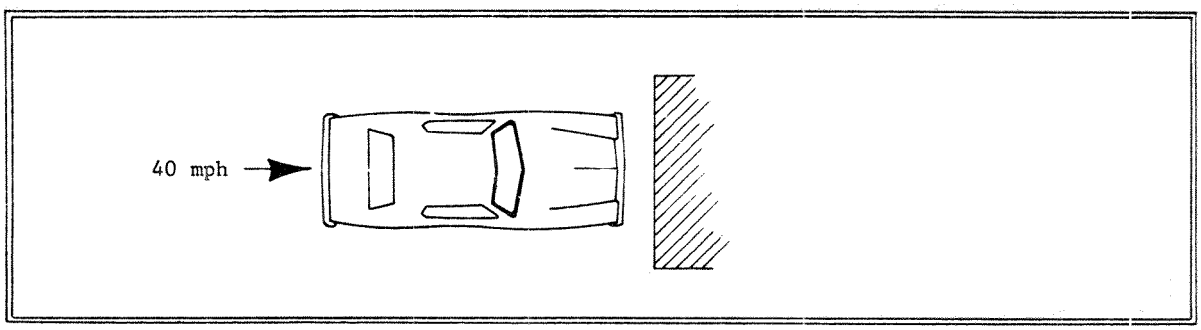
POSITION AND SIZE	HEAD HIC	CHEST SI	CHEST Gs(3Ms)	LEFT FEMUR (LBs)	RIGHT FEMUR (LBs)
DR GM50	10	10	13	520	100
CF	Not	Present			
RF GM50	80	20	9	470	150

COMMENTS: (1) The striking vehicle's ACRS did not deploy. (2) Injury criteria were well below the levels specified in FMVSS 208.

CRASH TESTING GM'S PRODUCTION AIR CUSHION

TYPE OF TEST 40 mph Frontal Barrier

IMPACT CONFIGURATION:



RATIONALE FOR TEST:

1. FIELD ACCIDENT CONFIGURATION COMPARISON ..... ☐
2. INJURY CRITERIA FROM DUMMIES ..... ☒
3. SENSOR PERFORMANCE ..... ☒
4. ACCIDENT CALIBRATIONS ..... ☒
5. MILEAGE AND ENVIRONMENTAL EFFECTS ..... ☐
6. OCCUPANT SIZE AND POSITION MIX ..... ☐

OTHER ..... ☐

TEST RESULTS:

SENSOR PERFORMANCE

ACTUATION DEVICE	CONTACT CLOSURE	ACTUATED ACRS
BUMPER IMP. DETECTOR	Yes	Yes
PLENUM LOW LEVEL DETECTOR	Yes	Redundant
PLENUM HIGH LEVEL DETECTOR	Yes	Yes

DUMMY OCCUPANT DATA

POSITION AND SIZE	HEAD HIC	CHEST SI	CHEST Gs (3Ms)	LEFT FEMUR (LBs)	RIGHT FEMUR (LBs)
DR GM50	960**	680	56	2320*	1100
CF GM50	1270	1110*	74*	350	1610
RF GM50	890	970	66*	1060	2240*

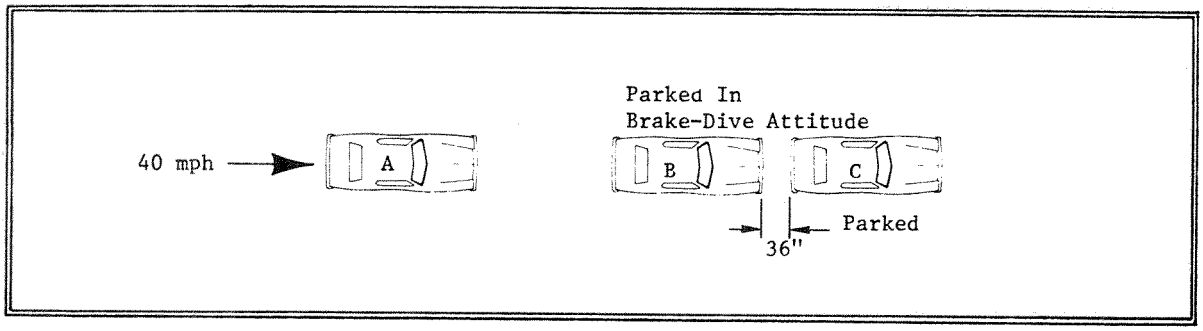
*See note on Page 4
**See note on Page 8

COMMENTS: The test was conducted according to FMVSS 208 test conditions, except for speed.

CRASH TESTING GM'S PRODUCTION AIR CUSHION

TYPE OF TEST "Chain Reaction" Collision Simulation. Simulated An Accident Involving A Stopped Vehicle At An Intersection, A Second Vehicle In A Brake Dive Mode, And A Third Vehicle Or Striking Vehicle, Traveling 40 mph

IMPACT CONFIGURATION:



RATIONALE FOR TEST:

1. FIELD ACCIDENT CONFIGURATION COMPARISON ..... [X]
2. INJURY CRITERIA FROM DUMMIES ..... [X]
3. SENSOR PERFORMANCE ..... [X]
4. ACCIDENT CALIBRATIONS ..... [X]
5. MILEAGE AND ENVIRONMENTAL EFFECTS ..... [ ]
6. OCCUPANT SIZE AND POSITION MIX ..... [ ]

OTHER ..... [ ]

TEST RESULTS:

SENSOR PERFORMANCE

CAR A

ACTUATION DEVICE	CONTACT CLOSURE	ACTUATED ACRS
BUMPER IMP. DETECTOR	Yes	Yes
PLENUM LOW LEVEL DETECTOR	Yes	Redundant
PLENUM HIGH LEVEL DETECTOR	Yes	Yes

CAR B

ACTUATION DEVICE	CONTACT CLOSURE	ACTUATED ACRS
BUMPER IMP. DETECTOR	No	No
PLENUM LOW LEVEL DETECTOR	No	No
PLENUM HIGH LEVEL DETECTOR	No	No

DUMMY OCCUPANT DATA

CAR A

POSITION AND SIZE	HEAD HIC	CHEST SI	CHEST Gs(3Ms)	LEFT FEMUR (LBs)	RIGHT FEMUR (LBs)
DR GM50	60	50	14	540	280
CF	Not	Present			
RF GM50	70	70	16	840	550

CAR B

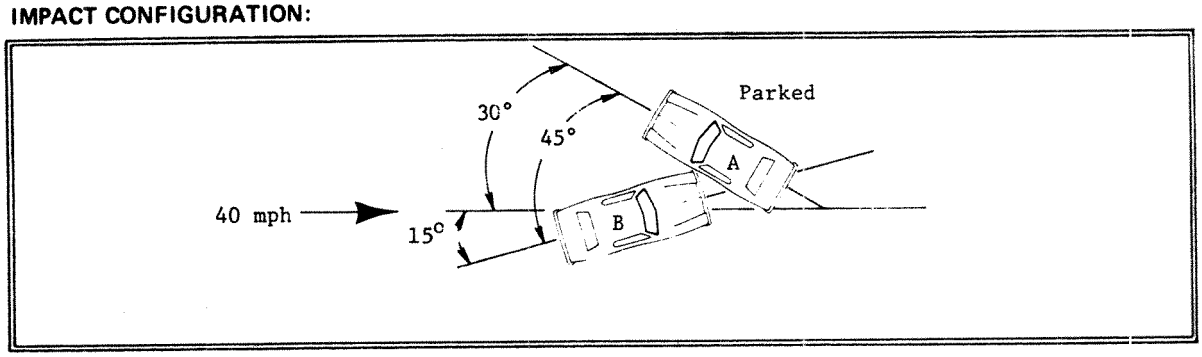
POSITION AND SIZE	HEAD HIC	CHEST SI	CHEST Gs(3Ms)	LEFT FEMUR (LBs)	RIGHT FEMUR (LBs)
DR GM95	180	50	18	1170	570
CF GM50	170	60	24	920	1200
RF GM50	130	70	22	--	--
RR GM50	70	40	18	--	--

COMMENTS: (1) No dummies in Vehicle C. (2) The injury criteria were well below the levels specified in FMVSS 208.

CRASH TESTING GM'S PRODUCTION AIR CUSHION

TYPE OF TEST 40 mph, Two-Car 45° Left-Side Angle Side Impact

IMPACT CONFIGURATION:



RATIONALE FOR TEST:

1. FIELD ACCIDENT CONFIGURATION COMPARISON ..... [X]
2. INJURY CRITERIA FROM DUMMIES ..... [X]
3. SENSOR PERFORMANCE ..... [X]
4. ACCIDENT CALIBRATIONS ..... [X]
5. MILEAGE AND ENVIRONMENTAL EFFECTS ..... [ ]
6. OCCUPANT SIZE AND POSITION MIX ..... [ ]

OTHER ..... [ ]

TEST RESULTS:

SENSOR PERFORMANCE

CAR A

ACTUATION DEVICE	CONTACT CLOSURE	ACTUATED ACRS
BUMPER IMP. DETECTOR	No	No
PLENUM LOW LEVEL DETECTOR	No	No
PLENUM HIGH LEVEL DETECTOR	No	No

CAR B

ACTUATION DEVICE	CONTACT CLOSURE	ACTUATED ACRS
BUMPER IMP. DETECTOR	(No	
PLENUM LOW LEVEL DETECTOR	ACRS)	
PLENUM HIGH LEVEL DETECTOR		

DUMMY OCCUPANT DATA

CAR A

POSITION AND SIZE	HEAD HIC	CHEST SI	CHEST Gs(3Ms)	LEFT FEMUR (LBs)	RIGHT FEMUR (LBs)
DR GM50	750	120	44	60	580
CF	Not	Present			
RF GM50	540	20	17	50	300
LR GM50	400	120	44	270	100

CAR B

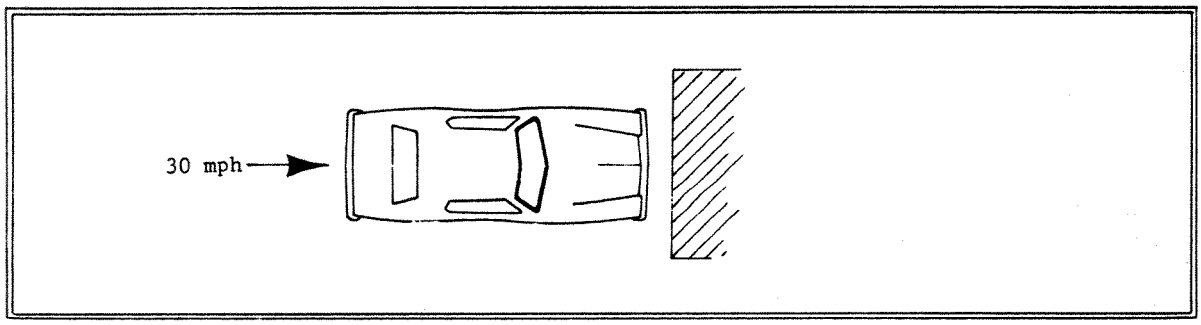
POSITION AND SIZE	HEAD HIC	CHEST SI	CHEST Gs(3Ms)	LEFT FEMUR (LBs)	RIGHT FEMUR (LBs)
DR	Not	Present			
CF	Not	Present			
RF	Not	Present			

COMMENTS: (1) The striking vehicle impacted Vehicle B, the ACRS vehicle, behind the cowl area. (2) The ACRS did not deploy.

CRASH TESTING GM'S PRODUCTION AIR CUSHION

TYPE OF TEST 30 mph Frontal Barrier (Cold Weather Exposed Vehicle)

IMPACT CONFIGURATION:



RATIONALE FOR TEST:

1. FIELD ACCIDENT CONFIGURATION COMPARISON ..... ☐
2. INJURY CRITERIA FROM DUMMIES ..... ☒
3. SENSOR PERFORMANCE ..... ☒
4. ACCIDENT CALIBRATIONS ..... ☐
5. MILEAGE AND ENVIRONMENTAL EFFECTS ..... ☒
6. OCCUPANT SIZE AND POSITION MIX ..... ☐

OTHER ..... ☐

TEST RESULTS:

SENSOR PERFORMANCE

ACTUATION DEVICE	CONTACT CLOSURE	ACTUATED ACRS
BUMPER IMP. DETECTOR	Yes	Yes
PLENUM LOW LEVEL DETECTOR	Yes	Redundant
PLENUM HIGH LEVEL DETECTOR	Yes	Yes

DUMMY OCCUPANT DATA

POSITION AND SIZE	HEAD HIC	CHEST SI	CHEST Gs (3Ms)	LEFT FEMUR (LBs)	RIGHT FEMUR (LBs)
DR GM50	340**	380	44	1420	1270
CF GM50	320	390	45	470	1210
RF GM50	360	370	44	930	1110

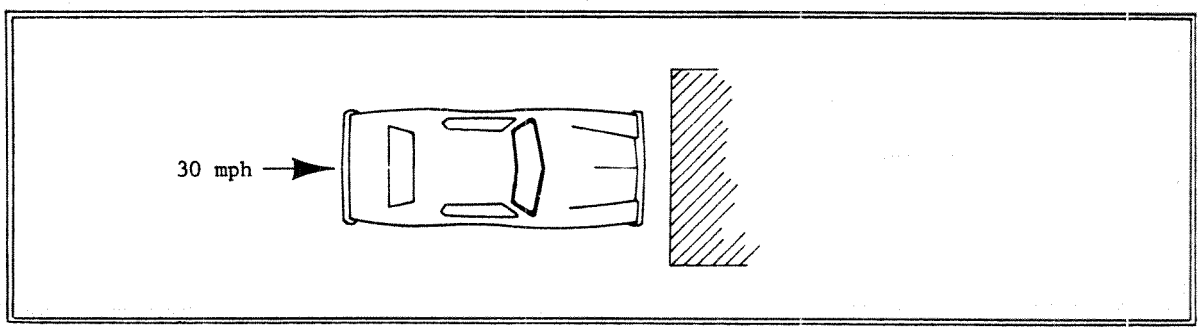
**See note on Page 8

COMMENTS: (1) The vehicle had been operated for 45,000 miles during two winters in Minnesota. (2) The injury criteria were below the limits specified in FMVSS 208.

CRASH TESTING GM'S PRODUCTION AIR CUSHION

TYPE OF TEST 30 mph Frontal Barrier (Hot Weather Exposed Vehicle)

IMPACT CONFIGURATION:



RATIONALE FOR TEST:

1. FIELD ACCIDENT CONFIGURATION COMPARISON ..... ☐
2. INJURY CRITERIA FROM DUMMIES ..... ☒
3. SENSOR PERFORMANCE ..... ☒
4. ACCIDENT CALIBRATIONS ..... ☐
5. MILEAGE AND ENVIRONMENTAL EFFECTS ..... ☒
6. OCCUPANT SIZE AND POSITION MIX ..... ☐

OTHER ..... ☐

TEST RESULTS:

SENSOR PERFORMANCE

ACTUATION DEVICE	CONTACT CLOSURE	ACTUATED ACRS
BUMPER IMP. DETECTOR	Yes	Yes
PLENUM LOW LEVEL DETECTOR	Yes	Redundant
PLENUM HIGH LEVEL DETECTOR	Yes	Yes

DUMMY OCCUPANT DATA

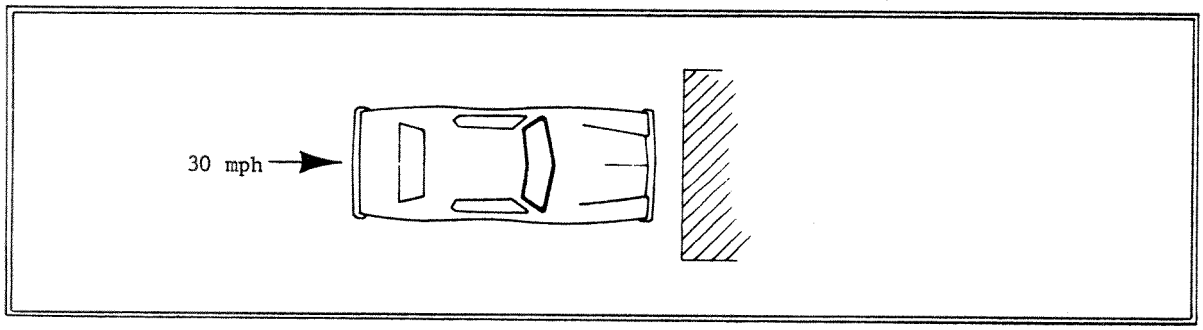
POSITION AND SIZE	HEAD HIC	CHEST SI	CHEST Gs (3Ms)	LEFT FEMUR (LBs)	RIGHT FEMUR (LBs)
DR GM50	310	300	36	1330	1180
CF GM50	370	380	44	540	1190
RF GM50	490	320	45	660	1170

COMMENTS: (1) The vehicle had been operated for 45,000 miles during two summers of hot weather at GMPG in Mesa, Arizona. (2) The injury criteria were below the levels specified in FMVSS 208. (3) The ACRS deployed properly.

CRASH TESTING GM'S PRODUCTION AIR CUSHION

TYPE OF TEST 30 mph Frontal Barrier With Two Lap-Belted Dummy Occupants

IMPACT CONFIGURATION:



RATIONALE FOR TEST:

1. FIELD ACCIDENT CONFIGURATION COMPARISON ..... ☐
2. INJURY CRITERIA FROM DUMMIES ..... ☒
3. SENSOR PERFORMANCE ..... ☐
4. ACCIDENT CALIBRATIONS ..... ☐
5. MILEAGE AND ENVIRONMENTAL EFFECTS ..... ☐
6. OCCUPANT SIZE AND POSITION MIX ..... ☐

OTHER ..... ☐

TEST RESULTS:

SENSOR PERFORMANCE

ACTUATION DEVICE	CONTACT CLOSURE	ACTUATED ACRS
BUMPER IMP. DETECTOR	Yes	Yes
PLENUM LOW LEVEL DETECTOR	Yes	Redundant
PLENUM HIGH LEVEL DETECTOR	Yes	Yes

DUMMY OCCUPANT DATA

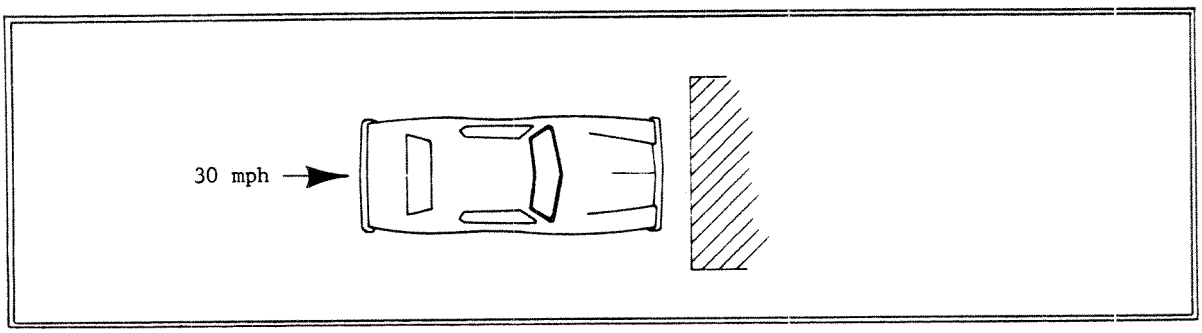
POSITION AND SIZE	HEAD HIC	CHEST SI	CHEST Gs (3Ms)	LEFT FEMUR (LBs)	RIGHT FEMUR (LBs)
DR GM50	370	390	47	1230	500
CF	Not	Present			
RF GM50	430	380	42	660	760

COMMENTS: Both dummies were restrained with lap-belts and the Air Cushion Restraint System.

CRASH TESTING GM'S PRODUCTION AIR CUSHION

TYPE OF TEST 30 mph Frontal Barrier With A 95th Percentile Male Dummy Driver, 3-Year-Old Male Center Front Passenger Dummy (Seated), and a 5th Percentile Female Right Front Passenger Dummy

IMPACT CONFIGURATION:



RATIONALE FOR TEST:

1. FIELD ACCIDENT CONFIGURATION COMPARISON ..... ☐
2. INJURY CRITERIA FROM DUMMIES ..... ☒
3. SENSOR PERFORMANCE ..... ☐
4. ACCIDENT CALIBRATIONS ..... ☐
5. MILEAGE AND ENVIRONMENTAL EFFECTS ..... ☐
6. OCCUPANT SIZE AND POSITION MIX ..... ☒

OTHER ..... ☐

TEST RESULTS:

SENSOR PERFORMANCE

ACTUATION DEVICE	CONTACT CLOSURE	ACTUATED ACRS
BUMPER IMP. DETECTOR	Yes	Yes
PLENUM LOW LEVEL DETECTOR	Yes	Redundant
PLENUM HIGH LEVEL DETECTOR	Yes	Yes

DUMMY OCCUPANT DATA

POSITION AND SIZE	HEAD HIC	CHEST SI	CHEST Gs (3Ms)	LEFT FEMUR (LBs)	RIGHT FEMUR (LBs)
DR 895	830**	490	52	1550	1010
CR 3 Yr	--	--	--	--	--
RF 805	220	350	49	600	780

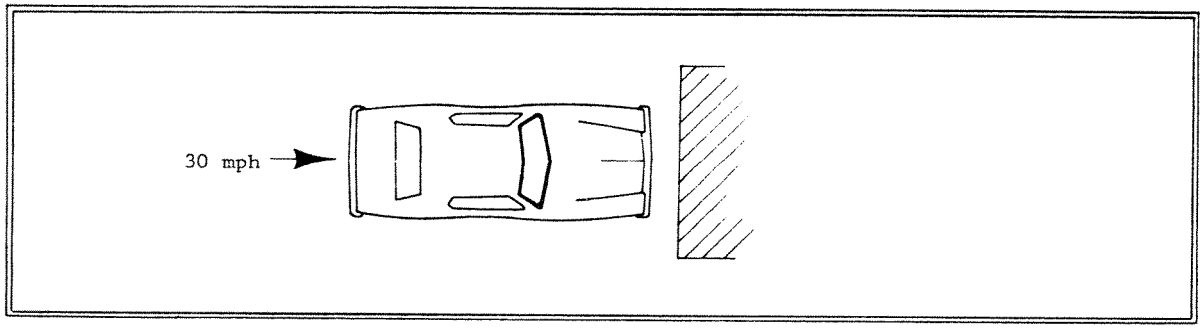
** See note on Page 8

COMMENTS: (1) The seat was in the full rear position. (2) The dummies were laterally positioned as the dummies in the FTP certification tests. (3) The 3-year-old male dummy was seated in the center front position. Injury criteria are not appropriate for child dummies.

CRASH TESTING GM'S PRODUCTION AIR CUSHION

TYPE OF TEST 30 mph Frontal Barrier with Driver and Right Front Dummies

IMPACT CONFIGURATION:



RATIONALE FOR TEST:

1. FIELD ACCIDENT CONFIGURATION COMPARISON ..... ☐
2. INJURY CRITERIA FROM DUMMIES ..... ☒
3. SENSOR PERFORMANCE ..... ☒
4. ACCIDENT CALIBRATIONS ..... ☐
5. MILEAGE AND ENVIRONMENTAL EFFECTS ..... ☐
6. OCCUPANT SIZE AND POSITION MIX ..... ☒

OTHER ..... ☐

TEST RESULTS:

SENSOR PERFORMANCE

ACTUATION DEVICE	CONTACT CLOSURE	ACTUATED ACRS
BUMPER IMP. DETECTOR	Yes	Yes
PLENUM LOW LEVEL DETECTOR	Yes	Redundant
PLENUM HIGH LEVEL DETECTOR	Yes	Yes

DUMMY OCCUPANT DATA

POSITION AND SIZE	HEAD HIC	CHEST SI	CHEST Gs (3Ms)	LEFT FEMUR (LBs)	RIGHT FEMUR (LBs)
DR GM50	590**	510	50	1250	1230
CF	Not	Present			
RF GM50	260	320	40	910	1010

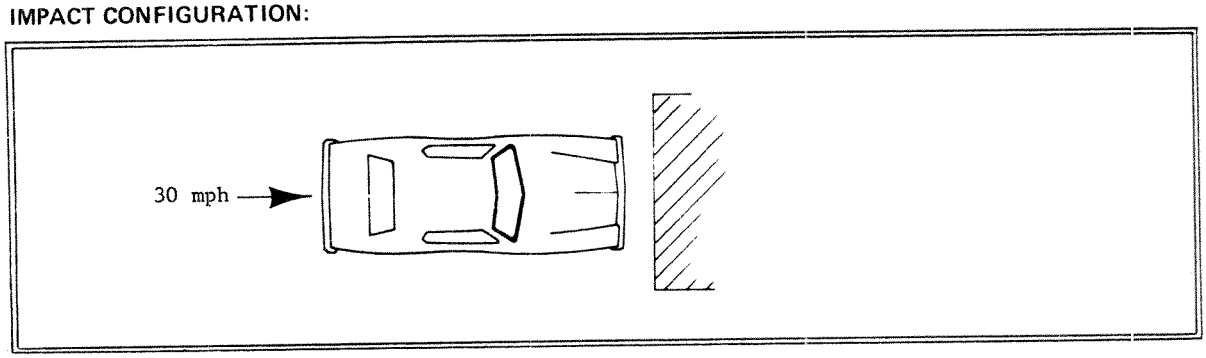
**See note on Page 8

COMMENTS: (1) The dummies were restrained by the ACRS.

CRASH TESTING GM'S PRODUCTION AIR CUSHION

TYPE OF TEST 30 mph Frontal Impact with 5th Percentile Female Driver Dummy and 95th Percentile Male Right Front Passenger Dummy

IMPACT CONFIGURATION:



RATIONALE FOR TEST:

1. FIELD ACCIDENT CONFIGURATION COMPARISON ..... ☐
2. INJURY CRITERIA FROM DUMMIES ..... ☒
3. SENSOR PERFORMANCE ..... ☐
4. ACCIDENT CALIBRATIONS ..... ☐
5. MILEAGE AND ENVIRONMENTAL EFFECTS ..... ☐
6. OCCUPANT SIZE AND POSITION MIX ..... ☒

OTHER ..... ☐

TEST RESULTS:

SENSOR PERFORMANCE

ACTUATION DEVICE	CONTACT CLOSURE	ACTUATED ACRS
BUMPER IMP. DETECTOR	Yes	Yes
PLENUM LOW LEVEL DETECTOR	Yes	Redundant
PLENUM HIGH LEVEL DETECTOR	Yes	Yes

DUMMY OCCUPANT DATA

POSITION AND SIZE	HEAD HIC	CHEST SI	CHEST Gs (3Ms)	LEFT FEMUR (LBs)	RIGHT FEMUR (LBs)
DR 805	330	280	39	710	950
CF	Not	Present			
RF 895	860	510	54	1040	970

COMMENTS: (1) The seat was in the full forward position. (2) The RF dummy's knees were approximately 1-2" from the ACRS cover prior to impact. (3) The ACRS deployed in sufficient time to restrain the RF dummy.

PART III – HUMAN VOLUNTEER TESTING

GEORGE R. SMITH
Automotive Safety Engineering
Environmental Activities Staff
General Motors Corporation

INTRODUCTION

Vehicle restraint systems have generally been evaluated with anthropomorphic dummies and defined measures of injury criteria. With a restraint as different as the air cushion, General Motors (GM) has taken an additional step in an attempt to understand the interface between the occupant and restraint system. A program was established to test the driver air cushion system in direct frontal impacts on an impact sled using human volunteers. This program, conducted by Southwest Research Institute, San Antonio, Texas, under contract to General Motors Corporation, was similar to tests on the passenger side air cushion which were done at Holloman Air Force Base for the National Highway Traffic Safety Administration. In both programs, the tests were conducted under the most safe and controlled conditions possible, and typify only what the air cushions might accomplish in direct, head-on impacts. They represent an ideal situation, a one-degree slice of reality.

The primary concern for both GM and Southwest Research Institute (SwRI) was to safeguard the rights and welfare of the human volunteers. The test program was designed to accomplish this.

This paper describes the test procedures and presents observations on the human kinematic and physical response to impacts approximately equivalent to a 30 mph (48 kph) barrier crash. Comparisons are made between the volunteers and the dummies (Hybrid II 50th percentile and 95th percentile).

APPROACH

The general approach to this test program involved the stepped-severity concept which was successfully utilized at Holloman Air Force Base. The human volunteers were exposed to increasingly severe impact environments beginning at a level approximately equivalent to a 13 mph (21 k/ph) barrier crash and culminated as planned at a 30 mph (48 k/ph) barrier crash condition. Tests were conducted at eight different impact severity levels, each used as a step toward the final 30 mph barrier equivalent. Each level contained four dummy tests (three 50th percentile and one 95th percentile) and five human tests. The 50th percentile dummy was a Hybrid II and 95th percentile was a modified Sierra 292-895. The decision to proceed from one severity level to the next was based on the volunteer's comments and information obtained from the transducers and high-speed motion pictures.

A physician was present to perform a short pre-test physical exam and a post-test exam. As a part of these exams, he reviewed the recorded physiological data.

Extensive system testing preceded the first exposure of humans to an air cushion. Validation of the system (test buck, hardware and ACRS) was conducted at the GM Safety Research and Development Laboratory. Initial SwRI tests included sled correlation (dummy), animal (baboon) and harness indoctrination (human).

GENERAL MOTORS DRIVER ACRS DESCRIPTION

The restraint system evaluated in the test program was similar to production equipment installed in the 1,000 Car GM Field Trial Program. The main elements of the driver air cushion system are an energy absorbing steering column, a steering wheel which houses a chemical gas generator and air cushion, and a knee restraint to absorb lower torso energy (Figure 1).

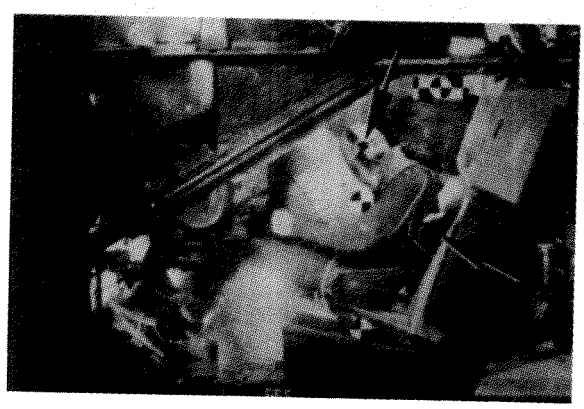


Figure 1

The steering column's energy absorption is accomplished by steel balls extruding a track between two telescoping jackets during column compression. The column has a dual stage energy absorber to obtain an improved energy absorption rate for the air cushion system. Initial low compression load is

created by pretracking the balls through the first inch of travel to reduce the initial inertia spike on the driver. The columns differed slightly from regular production air cushion equipment because they were installed without a shift lever, ignition lock, hazard warning switch, turn signal lever and were locked to prevent rotation.

The steering wheel hub supports the air cushion module and the inflated cushion is supported by the spokes and wheel rim. The air cushion module is composed of a chemical gas generator, a neoprene coated nylon air cushion, an inner container and an outer vinyl cover.

The knee restraint, mounted below the instrument panel, consists of a sheet metal retainer, flexible foam and vinyl skin. Its purpose is twofold, to control the driver kinematics during a frontal impact by restraining the lower torso and directing the upper torso and head into the steering wheel air cushion, and to mitigate or prevent injury to the legs.

The overall system operation with an anthropomorphic dummy in a crash equivalent to a 30 mph (48 k/ph) barrier test is depicted in Figures 2 through 7. Figure 2 shows the vehicle as it first contacts the barrier and is referred to as time zero. At this point no vehicle crush or deceleration exists, thus there is no crash information available.

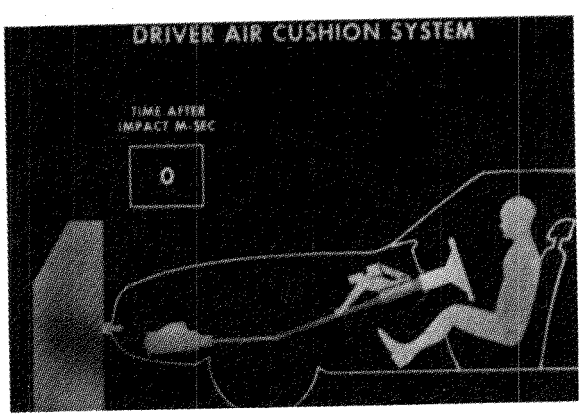


Figure 2

Figure 3 shows the vehicle at 10 milliseconds after initial contact with the barrier. The sensors in the vehicle send an electrical signal to the gas generator. The program on the impact sled with human volunteers was not intended to test the sensors, therefore, no crash sensors were employed. Instead, two limit switches mounted next to the sled rails were tripped by the test buck as it passed down the rails. The two circuits were independent and either one could deploy the air cushion. The time of air cushion deployment using these switches was similar to what is experienced using the sensors.

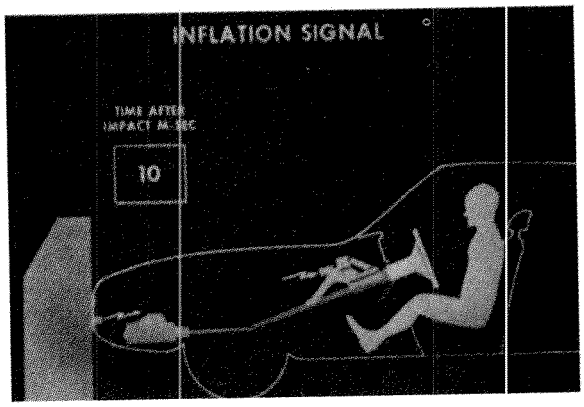


Figure 3

After 40 milliseconds, the cushion is deployed and in position to distribute and transmit the driver's upper torso load to the steering column (Figure 4).

At 50 milliseconds, the driver's knees have contacted the lower instrument panel.

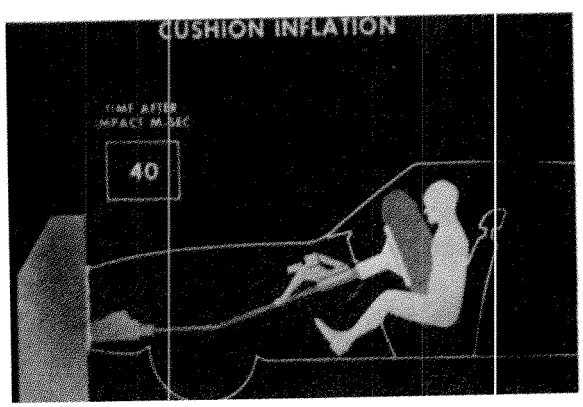


Figure 4

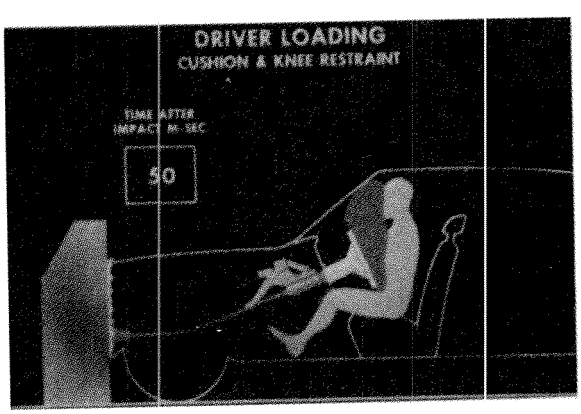


Figure 5

In Figure 6 we see the vehicle after 80 milliseconds and the driver is firmly against the cushion. The driver has moved forward in an upright position due to the restraint of the knee pad. He is firmly against the

driver cushion distributing the loads across his upper torso which creates a peak pressure in the cushion of about 10 psi. The column begins to compress and absorb energy. By design, the compression of the steering column is the principal means of absorbing energy in the driver system.

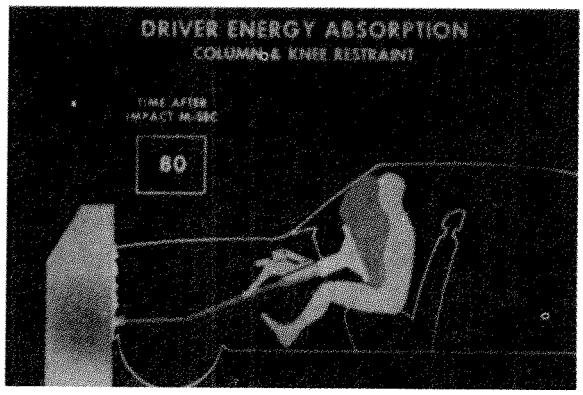


Figure 6

At 120 milliseconds, the vehicle has come to rest after rebounding away from the barrier. A major portion of the driver's energy has been absorbed by stroking of the energy absorbing steering column and plastic deformation of the sheet metal knee restraint (Figure 7). The remaining energy causes the driver to rebound rearward into the seat.

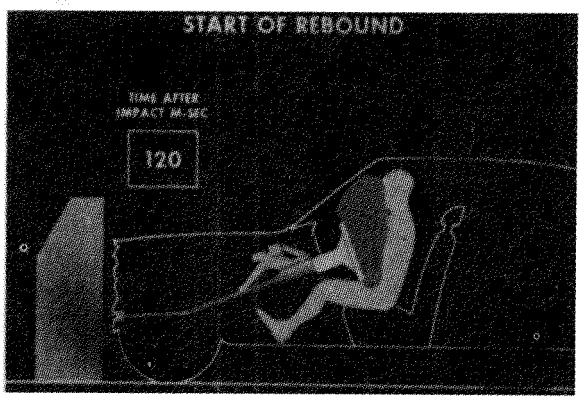


Figure 7

A more complete description of the system may be found in SAE Paper No. 730605, "General Motors Driver Air Cushion Restraint System," by T. N. Louckes et al.

TEST ENVIRONMENT

The GM designed body buck (Figure 8) resembled the buck used at Holloman and simulated the interior dimensions of a 1973 Oldsmobile Delta 88. An Oldsmobile production cloth bench seat was employed along with a special head restraint which covered the entire width of the buck. The bench seat could be adjusted fore and aft to accommodate different size volunteers.

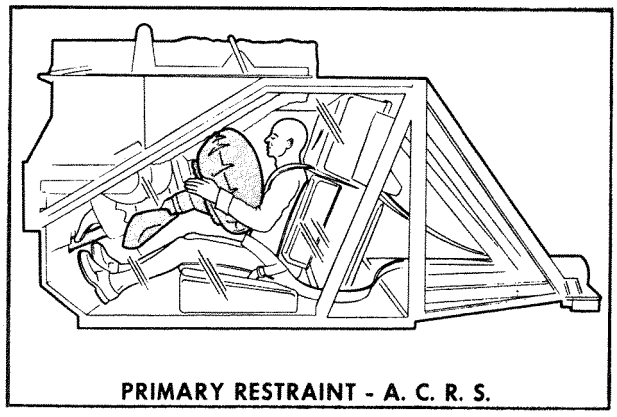


Figure 8

The simulated windshield was fabricated from styrofoam and shaped to the contours of the conventional windshield so the air cushion would interact against it in the same manner as it would in an automobile. The instrument panel above the knee restraint pad was a sheet metal mock-up covered by foam. A piece of Ensolite was inserted between the top of the knee restraint and the bottom of the instrument panel to prevent any volunteer hand or arm involvement with the bracketry.

In addition to the air cushion, which was the primary restraint, two backup restraint systems were provided as additional safety precautions. The air cushion system activated as intended in every test and the two backup systems were never needed. However, they did provide additional confidence for the volunteers in the test program.

The secondary restraint was a belt harness (Figure 9) that effectively restrained the shoulders, chest, lap, and legs. It was attached to a single bolt and

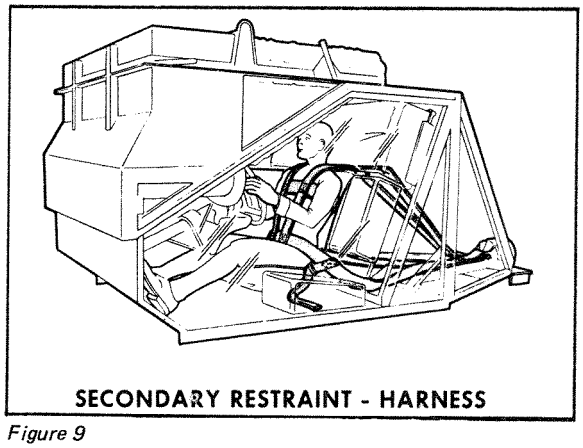


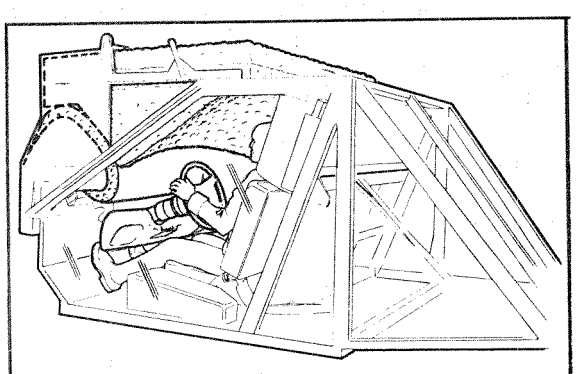
Figure 9

separation nut assembly mounted behind the seat on the body buck.

The system was controlled by proper inflation of the air cushion. As the cushion blossomed, it separated the two flaps on the inner cover of the module. Two interrupt wires were attached across these flaps. As the flaps separated, the interrupt wires were broken which caused an electric circuit to feed current to the separation nut and released the belt harness. This allowed the volunteer to interact freely with the air cushion restraint system. If the wires were not broken because the air cushion did not fire upon signal, the interrupt wires prevented any current flow to the release system and the belt harness then functioned as the primary restraint.

The belt harness system was also used in low speed volunteer indoctrination tests. Each of the volunteers was subjected to a simulated crash roughly equivalent to a 10 mph (16 k/ph) barrier crash while restrained by the belt harness system (no air cushion deployed). The purpose of this phase of tests was to acquaint the volunteers with the test procedures and instrumentation as well as building confidence in the secondary restraint. Some volunteers expressed that they received slight discomfort from the belts due to their tightness and some soreness to the neck caused from flexion during impact.

The tertiary restraint system involved the interior of the body buck (Figure 10). The windshield and instrument panel were specially designed to distribute energy in the event of failure of both the air cushion and belt restraints. The steering column's compression provides for energy absorption.



TERTIARY RESTRAINT - E. A. STRUCTURE

Figure 10

Additional protection was offered, primarily against ejection, by a clear plastic panel added across the left-hand body buck opening, and a special looping of the harness system around steel beams on the buck which limits excessive excursion. Rebound protection was provided by the slab foam head restraint which extends across the entire width of the buck.

The SwRI impact simulator is an "impact-with-rebound" type and is basically an MTS Model 858.05. The sled is accelerated up to the required speed by the use of natural rubber bungee cards. The sled impacts an SwRI designed rebound deceleration programmer. The impact of the sled forces the programmer's piston back until the sled is brought to zero velocity. The piston then returns its stroke accelerating the sled in the opposite (rebound) direction. After the sled rebounds from the programmer, it is slowed by stretching the bungee cords and is brought to a complete stop by the rebound brakes.

A 30 mph (48 k/ph) barrier crash pulse of a full size vehicle was simulated on the SwRI impact sled. The lower severity levels were scaled from this pulse.

INSTRUMENTATION

The data collected during the course of this experiment included both mechanical and physiological parameters. The data served two purposes during the tests, to assure continuous subject safety and to help evaluate the effectiveness of the driver ACRS.

Head and thorax accelerations were measured on both the volunteers and anthropomorphic dummies. The mounting techniques for the external accelerometers were the same for both humans and dummies. The triaxial head accelerometer was mounted on the left side of an adjustable plastic headband which was removed from the inside of a welder's helmet. The adjustment knob was removed from the back side of the headband and replaced by a strip of Velcro to provide infinite adjustment for head size. The accelerometer was located on the skull over the left temperoparietal region 1.5 to 2.0 inches (3.8 to 5.0 centimeters) above the Frankfort Plane and superior to the tragus. An Ensolite cup was placed over the accelerometer for the subject's protection.

The triaxial thorax accelerometer was mounted on a rubber-backed aluminum plate and covered by a 4 x 8 x 1 inch piece of Ensolite. The rubber backing was placed toward the subject and was centered over the left erector spinal muscle of the back at the level of the T6 vertebra. It was positioned off the mid sagittal plane to prevent injury to any protruding spinous process. A strip of 2-inch wide Velcro was fastened tightly around the subject's thorax as he expelled air, fixing the accelerometer to him. The chest strap portion of the belt harness was then tightened around the Velcro strap. Figure 11 shows a volunteer in the test buck with the instrumentation attached.

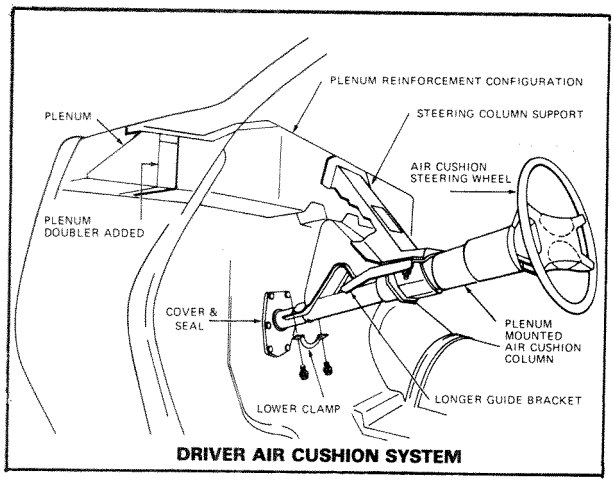


Figure 11

The output from the standard internal head, chest, and femur transducers in the anthropomorphic dummies was also recorded. The physiological data monitored included electrocardiogram, pulse rate, respirations, and blood pressure.

VOLUNTEER PREPARATIONS

Healthy young males were recruited as test subjects by SwRI. All were from the San Antonio area and had diversified backgrounds, from students to firemen. All subjects were required to take and pass a rigorous physical examination. The exam was administered at SwRI's medical facility with the individual being sent to a radiology laboratory for X-ray studies. The physical investigation included an examination of the heart, lungs, ears, nose and throat, eyes, hearing, reflexes, muscle and joint motion, pulse rate and blood pressure. The individual was also subjected to a "double masters" test to determine ECG and cardiac status during and immediately post exercise. Baseline enzyme studies including creative phosphokinase (CPK), lactic dehydroginase (LDH) and serum glutamic transaminase (SGOT) were performed on the volunteers. All individuals accepted in the test program were required to have an X-ray evaluation of the entire spine and all major bones and joints by a series of 22 views to rule out significant arthritis with spurs or lipping, marked cortical thinning, joint abnormalities with calcification or foreign bodies, ununited or poorly united fractures. As a result of the intensive examinations, nine of the candidates who applied were disqualified for various reasons.

In addition, certain liberal anthropometric size ranges were adhered to. The subjects selected were able to place their feet flat on the toe pan while harnessed in the number two seat position (one notch rear of full forward), their seated height measurement fell in the range of 34.25-37.5 inches (87-95 centimeters) and their weights did not exceed 220 pounds (100 kilograms). These requirements were established because they represent the range of the 50th and 95th percentile dummies' sizes which were utilized in the tests conducted for this program at the GM Safety Research and Development Laboratory.

A complete set of anthropometric dimensions on each volunteer was also completed. The dimensions included weight, stature, sitting height, acromion-radiale length, forearm grip distance, buttock-knee length, and sitting knee height.

Thirty-two individuals qualified for the tests. Each of these underwent an indoctrination test with the harness restraint system (no air cushion) and a full complement of instrumentation.

Twenty-six individuals participated in the air cushion tests. They ranged in age from 19 to 30 years. Their statures ranged from 67 to 74 inches (171 to 188 cm) and their weights from 138 to 208 pounds (62 to 94 kg).

Immediately before the scheduled test, a short physical exam was performed on the volunteer by the physician in attendance. The subject was then instrumented for ECG (Ag/Ag CI electrodes) and blood pressure (sphygmomanometer cuff). Clothing consisted of tight fitting cotton long sleeve "ski pajama" tops and bottoms. A cotton "thermal" undershirt was worn beneath the pajama top. Rubber soled, low top gym shoes were worn in every test.

Pliable plastic goggles, earplugs, mouthpiece, and a rubber bathing cap were worn by the volunteers. The bathing cap was worn to provide a grip for the accelerometer mount. All jewelry was removed for the test. Cold cream was applied to the volunteer's nose, chin, thumb and wrist area to minimize possible abrasions from the deploying air cushion.

In all tests the volunteer was placed inside the body buck in a forward facing position and centered laterally behind the steering wheel. The seat was adjusted to allow the volunteer to place his feet firmly on the toe pan. His hands were placed in the 9 and 3 o'clock positions on the steering wheel with the thumbs locked around the rim. His head was upright in a normal driving position. The belt harness was tightened to the point where it became painful to the volunteer. As described previously, the belt harness was always released at the start of air cushion deployment. Each volunteer was instructed to brace his whole body for an impact and to try and keep his hands on the steering wheel throughout the test.

Following the impact the volunteer was asked to relax and remain seated until the investigators had discussed the ride with him. The physician assessed and recorded the traumo, if any existed. The

volunteer was then removed from the body buck and asked to fill out several subjective reports.

Many measures were taken to insure a safe ride for the volunteers and no significant trauma was experienced in any of the tests. However, as an added precaution, the physician provided certain emergency and resuscitative measures and devices for immediate employment in the impact laboratory. These included a cardiac defibrillator, tracheotomy set, tourniquets, splints, suture set, I.V. fluids, drugs for I.V. use and morphine.

The SwRI physician involved in the test program is board certified in surgery with a specialty in the diagnosis, treatment and surgery of trauma. If a subject had been injured severely to require movement to a hospital for more definitive treatment than could be administered at SwRI, arrangements for rapid transport to a nearby hospital, as well as emergency care, had been made. Fortunately, none of the backup precautions was used.

30 MPH BARRIER EQUIVALENT TESTS (48 k/ph)

Although many tests were conducted using both dummies and humans, they all lead toward the final goal of the program, a test severity simulating a 30 mph (48 k/ph) barrier test. At this level, the total sled velocity change was nominally 34 mph (55 k/ph) with a peak deceleration nominally 26 g's. The five volunteers who participated at this test level had experienced an air cushion test at a lower severity. The following discussion pertains specifically to these last five tests.

As in all the tests, the deploying air cushion began to blossom before the subject began to translate forward. During the deployment process, as the cushion mushroomed, it wiped across the face and forearms on most of the subjects. The volunteers' knees made contact with the knee restraint pad before the upper torso and head impacted the inflated air cushion. As the internal cushion pressure began to increase from upper torso impact, loads on the toe pan and knee restraint decreased.

After initial head contact, there was little additional rotation of the head into or over the cushion, minimizing the amount of neck flexion. The ability to keep their hands on the steering wheel during the impact sequence varied from subject to subject. Most subjects had at least one hand come off the wheel. At the top severity level, four of the five subjects kept one hand on the steering wheel. When a hand came off the wheel, it generally traveled forward striking the simulated instrument panel with the back of the wrist.

The knee restraint effectively controlled the human kinematics by absorbing some lower torso impact energy through deformation and by directing the upper torso and head into the steering wheel air cushion. Additional energy was absorbed through the lower legs as they rode down the impact on the toe pan.

The steering column absorbed part of the upper torso energy and was generally accomplished in two stages. The first stage occurred when the column compressed due to the force exerted through the arms of the volunteer. The steering wheel also canted forward slightly at the hub because the resultant forces through the volunteer's braced arms were not directed axially down the steering column. The second stage of compression occurred when the subject's upper torso and head loaded the steering column through the air cushion. The air cushion was effective in distributing the forces on the upper torso and head.

The volunteers traversed essentially in the same longitudial plane of the vehicle on rebounding from the air cushion restraint system as they did during impact. The head rebound velocities of the volunteers was considerably less than the dummies tested at SwRI. It was, however, 4 mph (6 k/ph) greater than had been predicted by previous dummy tests at the GM Safety Research and Development Laboratory.

Only minor trauma was experienced by the volunteers during the test series. It would fit severity codes of 0 (no injury) and 1 (minor) of the abbreviated injury scale of the American Medical Association[1], where scale limits range from 0 (no injury) to 6 (fatal). The predominant trauma in these tests was limited to erythema, abrasions, and ecchymosis. The trauma was generally so minimal, it would not even appear in black and white photographs. The complete clinical findings for the final severity level as well as subject stature and weight may be found in Table 1.

The areas where trauma was most frequently observed were the arms, face (mouth, nose, and chin), and the chest. The only observed trauma to the legs was an abrasion to the left knee of one of the five subjects. This minimal trauma is particularly notable because no protective gear was worn on the legs, only cotton leotards. A listing of the symptoms recorded on the Physical Symptom Survey Response by each of the volunteers may be found in Table II.

Comparison of the dynamic response of dummies and humans was not the main objective of this

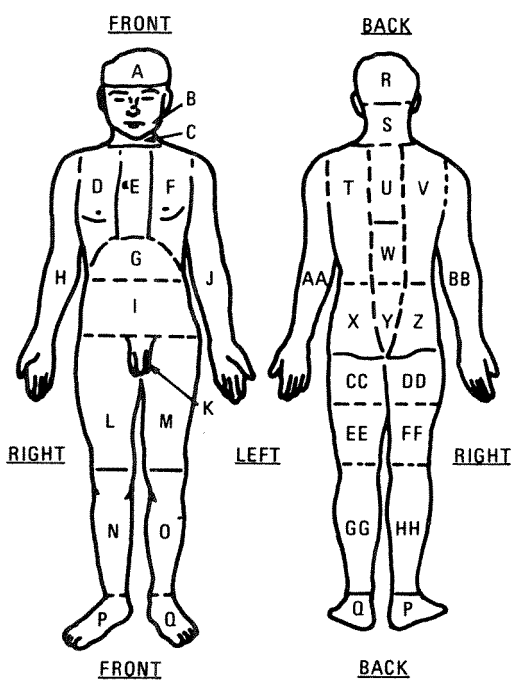
[1]Developed by the American Medical Association Committee on Medical Aspects of Automotive Safety, in cooperation with physicians representing medical specialties most involved in the diagnosis, care and treatment of crash injuries, and General Motors Corporation.

TABLE 1
CLINICAL FINDINGS AT 30 MPH BARRIER EQUIVALENT (48 kph)

Run. No.	Subject Height (Inches)	Subject Weight (Lbs.)	Sled Velocity (mph)*	Sled Deceleration (g's)	Findings
687	70.3	146	34.1	25.8	Erythema both antecubital spaces. Ecchymosis right wrist. Felt momentarily dazed.
688	69.3	150	33.8	26.5	Erythema both antecubital spaces.
689	71.3	179	34.1	25.6	Erythema both antecubital spaces. Contusions both wrists, lacerations bridge of nose from goggles.
690	70.1	162	33.9	26.0	Erythema both antecubital spaces and in epigastrium. Ecchymosis both wrists and left forearm. Minor laceration left forefinger.
691	69.7	160	33.8	25.8	Erythema both wrists and left elbow. Abrasion left knee with slight bleeding.

* In a full size GM vehicle 30 mph (48 kph) barrier crash, the collision is not totally inelastic and some rebound exists. The occupants, therefore, "see" a relative velocity change greater than 30 mph (48 kph) which is nominally 34 mph (55 kph).

TABLE 2
PHYSICAL SYMPTOM SURVEY RESPONSE



SEVERITY CODE

1. No Symptoms
2. Mild Pressure
3. Moderate Pressure
4. Slight Discomfort
5. Definite Discomfort
6. Mild Pain
7. Moderate Pain
8. Severe Pain

Test No.	Subject	Symptom Area and Severity Response
687	13	None
688	37	None
689	31	B6, K5, H4, J4, W4
690	35	None
691	9	B2, H2, J2, K2

program. Since the dummies and humans were run under identical conditions, the data obtained lends itself to some comparison. A general overview of *all* the tests in *this* program reveals that the anthropomorphic dummies' response was conservative compared to the human response. The calculated injury criteria numbers from dummy tests were generally larger than equivalent tests with humans. However, at 30 mph barrier equivalent, the head injury criteria (HIC) numbers were grouped about the same for the dummy tests as the human tests. Figure 12 charts the nine tests and includes the HIC calculated from the dummy internal accelerometers. The mean HIC for the dummy tests was 363 and for the human tests it was 338. The thorax severity index (SI) is charted in Figure 13. The mean thorax SI for dummies was 385 and was 232 for humans. The dummy femur loads are charted in Figure 14. The Hybrid II 50th percentile dummies' left femur load was about 1,600 pounds (7,100 newtons) and the right about 1,200 pounds (5,300 newtons). The modified Sierra 95th percentile dummy's left femur exceeded 2,000 pounds (8,900 newtons) and his right was about 900 pounds (4,000 newtons). It should be noted that even though the dummies' femur loads were well above 1,000 pounds (4,400 newtons), the volunteers did not record feeling even mild pressure to their legs on the Physical Symptom Survey Response which they completed within 30 minutes of the test.

The dummy's whole body response differed from the human's as evidenced in the amount of column compression. Dummies compressed the column more, averaging 5.8 inches (14.7 centimeters) at 30 mph barrier equivalent while the humans averaged 3.0 inches (7.6 centimeters).

At this severity, the humans usually compressed the column in two stages while in the dummy tests, once the column compression began, it continued without a pause. The difference can be accounted for in the strength of the humans' arms.

Because the volunteers were in a braced position, their arms were fairly rigid and could transmit a force to the steering system. Even though this force was not axial down the column, the resultant was sufficient to begin the compression. The dummies, on the other hand, did not have any muscle tone in their arms (beyond the 1 g joint setting) and column compression did not begin until their upper torso impacted the inflated air cushion. The second stage of compression in the human tests occurred when their upper torsos contacted the inflated air cushion. The time of peak impact pressures in the inflated air cushion was later in humans than the dummies (peak impact pressure occurred between 75 and 100 msec

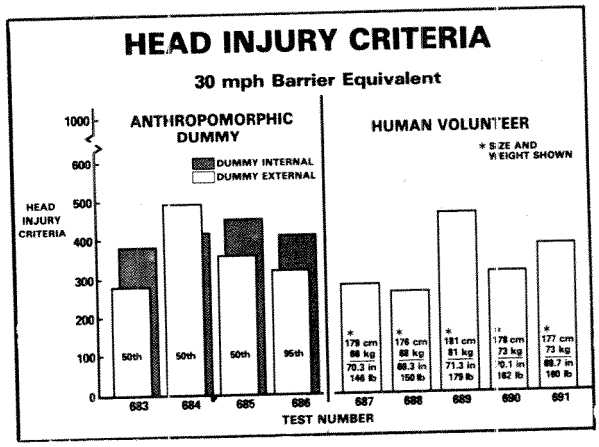


Figure 12

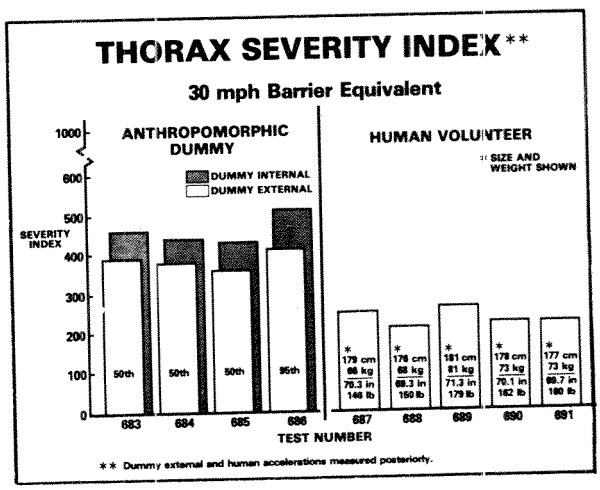


Figure 13

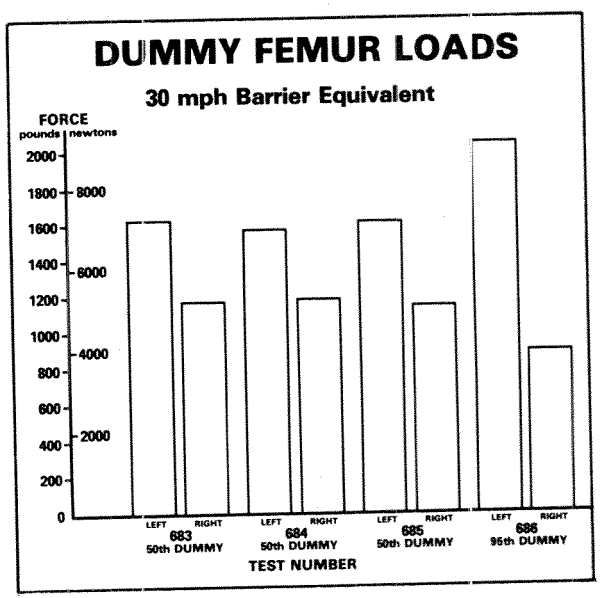


Figure 14

after onset of sled deceleration in dummy tests and between 100 and 125 msec with humans). The humans were able to "hold themselves off" the air cushion longer.

The dummies usually made contact with the inflated air cushion (or steering wheel) about the time they contacted the knee restraint, while the volunteers' knees hit before their upper torso. The time of contact with the knee restraint was monitored by contact switches mounted on the knee restraint pad. The average dummy knee contact time (52 milliseconds) occurred before that of the human's (60 milliseconds). The main reason for this difference was the muscle tone of the human's arms and legs which served to absorb some of the body's kinetic energy. It has previously been shown by Armstrong[2] in human volunteers, 3-point harness tests conducted by the National Bureau of Standards that a braced person does exert significant loads to the floor pan during sled deceleration and, as a result, reduces the loads transmitted to his body by the restraint system.

SUMMARY

The dynamic impact tests at Southwest Research Institute for the first time exposed human volunteers to production-like driver air cushion system deployments at impact levels equivalent to a 30 mph barrier crash (48 k/ph). No significant injury was produced. At no time was it necessary for the backup restraint systems to be utilized and the entire program schedule was carried out as planned.

It must be noted, however, that all tests were conducted under the most safe and controlled conditions possible, and typify only what the air cushions might accomplish in direct, head-on impacts up to 30 mph (48 k/ph). They depict an ideal situation, a one-degree slice of reality that in no way represent the total spectrum of frontal accidents occurring on our roads today.

[2] R. W. Armstrong, H. P. Waters, and J. P. Stapp, "Human Muscular Restraint During Sled Deceleration," 12th Stapp Car Crash Conference, SAE 680771, October 1971.

HUMAN INJURY TOLERANCE LEVEL DETERMINATION FROM ACCIDENT DATA USING THE OPAT DUMMY

J.G . WALL and
R.W. LOWNE
Transport and Road Research Laboratory

ABSTRACT

A technique is described for determining injury tolerance levels of motor vehicle occupants restrained by seat belts. Detailed accident injury data from accidents involving seat belt wearers are correlated with results of tests using the OPAT dummy.

Human injury tolerance levels are proposed for several parts of the body. These levels are applicable to readings from tests made using the OPAT dummy.

INTRODUCTION

Before any anthropomorphic dummy can be used to assess or develop occupant protection provisions in vehicles, it is necessary to determine the readings from the various instruments employed to measure loads on the dummy that correspond to the limits of tolerable loadings sustained by live human occupants in a crashing vehicle. It is now widely accepted that it is often inappropriate to apply directly levels of injury tolerance obtained from cadaver, animal or human volunteer tests to the transducer readings obtained with an anthropomorphic dummy. Some form of calibration or comparison determination between the dummy and the original accident-injury data is required to select dummy readings corresponding to levels of human injury tolerance.

This paper describes one of a series of determinations which are being carried out on the OPAT dummy to establish appropriate tolerance levels; it has particular reference to use of dummy to assess occupant restraint systems incorporating safety belts.

The injury tolerance technique serves in reconstructing accidents with known injury data – thus obtaining directly the injury tolerance limits in terms of dummy readings.

THE OPAT DUMMY

The OPAT dummy (Occupant Protection Assessment Test dummy) was developed as a result of a cooperative contract involving U.K. Government, Motor Industry Research Association and Ogle Design Limited. The project's object was to produce a simple, robust test dummy capable of giving repeatable results under similar test conditions. Dummy test reaction should resemble that of human in a test program targeted at developing safer road vehicles; the program should be suitable for assessing occupant protection provisions in production motor vehicles.

The construction and development testing of dummy have already been described in earlier papers References 1, 2, and 3.

ACCIDENT DATA AS A SOURCE OF HUMAN TOLERANCE LEVELS

Various methods have been used in the past to establish human impact tolerance levels. These test methods have involved: cadavers (both embalmed and unembalmed) to determine (at which loads or decelerations) skeletal or tissue injuries to the human body (albeit dead and lacking in muscle tone); animals; human volunteers to determine levels of voluntary exposure to impact loadings; and tests designed to interpret accident data for deriving tolerance levels of injured vehicle occupants in actual accidents. The relative advantages and disadvantages of these various approaches have been discussed elsewhere, Reference 4. These are the main advantages of using accident data: they refer specifically to the class of people, currently injured in accidents; and to conditions of muscle tone and tensing and seated attitude occurring in injury producing accidents.

THE ACCIDENT INJURY SAMPLE

The data utilized in probing injury tolerance levels were gathered over a period of years by a

multi-disciplinary team consisting of medical and engineer professionals. The medically qualified members of the team visited selected hospitals in a given area at frequent intervals to interview and examine patients injured in road traffic accidents.

Following the medical phase, accidents were investigated in depth with particular reference to injuries and injury causation. To date 1,600 such patients have been investigated. Of these, 140 were car occupants wearing seat belts. The data from these cases form the basis for the determinations described in this paper.

The number of seat wearers in the sample is a small proportion of the sample partly because of the protection afforded by seat belts and partly due to the wearing rate for seat belts in the area and period covered by the sample. This area covers approximately 50 km by 110 km situated west of London, and includes all types of urban, rural and motorway roads. The methods of data collection are more fully described in Reference 5.

INJURIES SUSTAINED BY CAR OCCUPANTS IN THE ACCIDENT SAMPLE

Tables 1 and 2 give breakdowns of type of injury sustained by belted occupants in the sample. This analysis was carried out at a later date than that described in the second paragraph, below, and so numbers are somewhat increased. From these tables, it can be seen that the majority of injuries (attributed to excessive loading by the seat belt) occurred to the clavicle, the rib cage and the abdomen. It should be noted that although certain injuries received by car occupants may be ascribed to loadings applied by the seat belt, this should not be taken to mean that the injuries have been "caused" by the seat belt in the popular sense of the word. This latter statement might imply that the occupant would have been better off without the belt whereas, in fact, he might have suffered the same or, as the overall accident statistics indicate, far more serious injuries from impacts with the interior of the vehicle.

CORRELATION OF THE ACCIDENT DATA WITH INPUT PARAMETERS FOR TESTS USING THE OPAT DUMMY

In order to be able to relate transducer readings obtained during OPAT dummy tests with injuries to car occupants, it is necessary to have some means of correlating accident data with dummy test readings. In earlier tests (Reference 4) that used a kneeform impactor to establish hip joint tolerance level to loading caused by knee impact with vehicle fascias in accidents, depth and type of dent produced on similar fascias were used to correlate loadings recorded on the kneeform with injury data from accidents.

In the case of wearers restrained by seat belts, there is seldom any such indicator of load levels attained in impact. Permanent stretch of the webbing is an unreliable indicator because such stretch recovers slowly with time after the accident.

TABLE 1

Injuries to Belted Occupants

Regions of the body with injury more severe than minor by mechanism of injury amongst 133 seriously injured belted front seat occupants (Grattan and Clegg – Unpublished data)

Region of body \ Mechanism of Injury	Excessive Load	Excessive Load and/or Excessive Forward Movement	Forward Movement and Insufficient Space	Forward Movement and Intrusion	Intrusion	Deceleration	All Mechanisms
Face and Scalp			32	4	9		45
Brain (Concussion			28	3	11		42
VIth nerve (palsy)						1	1
Cervical Spine	1			1		3	5
Thoraco-Lumbar Spine	2						2
Clavicle	7						7
Chest cage and/or contents	25	5	1	2	2		35
Abdomen	4	1			1		6
Pelvis	1				5		6
Hip Joint			4	2	1		7
Thigh			6	1	7		14
Knee			22	3	3		28
Lower leg, foot and ankle			8	4	24		36

TABLE 2

Injuries to Belted Occupants

Regions of the body with injury more severe than minor by mechanism of injury amongst 18 fatally injured belted front seat occupants (Grattan and Clegg – Unpublished data)

Mechanism of Injury / Region of Body	Excessive Load	Excessive Load and/or Excessive Forward Movement	Forward Movement and Insufficient Space	Forward Movement and Intrusion	Intrusion	Deceleration	All Mechanisms
Face and Scalp			2	1	5		8
Skull and Brain					5 (5)		5 (5)
Cervical Spine							
Thoraco-Lumber Spine							
Clavicle	3				1		4
Chest cage and/or contents	8 (4)	1			7 (2)		16 (6)
Abdomen	6 (5)	1			5 (5)		12 (10)
Pelvis	1				1		2
Hip Joint							
Thigh			1	2	1		4
Knee			1	2	2		5
Lower leg, foot and ankle			2	1	5		8

Numbers in brackets indicate fatal injuries included in the totals.

Deformation of the anchorages occurs very infrequently, but when it does it can be used as an indicator. Rupture of the seat belt occurs very rarely, but such rupture is a good indication of the peak load exerted by seat belt on wearer during impact.

For the investigation described in this paper, we used equivalent barrier speed estimated from damage to the vehicle as an indicator (using results from impact tests on similar vehicles at various speeds for comparison purposes) in order to make maximum use of available injury data. Only impacts with a predominantly frontal component were considered. Tests were then carried out using the OPAT dummy restrained by typical safety belts on a dynamic test rig at various impact speeds with typical car deceleration pulses to decelerate the rig. The transducer data from these tests were correlated with the injury data from the accident investigations.

ANALYSIS OF THE ACCIDENT INJURY DATA

Only injuries attributed to seat belt loadings in impacts with a predominantly frontal component were considered. Three specific types of injury were selected: fracture of the clavicle, fractured ribs, and internal abdominal injuries. We decided to use rib fracture as an indication of tolerance level because the fracture of even one rib is an indication that seat belt loadings are approaching dangerous limits. This decision was made even though the consequences of such rib-fracture injuries can range from *discomfort* (if only one rib is fractured) to *death* (for multiple rib fractures).

Each of these injuries was associated with the equivalent barrier speed estimated for the vehicle in which it occurred. Table 3 lists these data together with the total number of seat belt wearers in the sample who were involved in impacts at any given equivalent barrier speed. The purpose of the test program was to determine loads applied by safety belts at the various equivalent barrier speeds during dummy test for load correlation with injury data from accident sample.

TABLE 3

Incidence of injuries more severe than minor attributable to seat belt loadings in impacts with a predominantly frontal component

Equivalent Barrier Impact Speed Ranges km/h	Total No. of Seat Belt Wearers in Sample	No. of People With Injuries Attributable to Seat Belt Loadings			
		Clavicle	Ribs	Abdomen	All Areas
12 - 20	2	0	0	0	0
20 - 28	6	0	0	0	0
28 - 36	9	0	3	0	3
36 - 44	16	0	5	2	5
44 - 52	20	3	6	4	9
52 - 60	23	5	6	2	7
60 - 68	3	1	1	0	2

THE TEST PROGRAM

Testing on the TRRL impact rig involved various speeds between 13 km/h and 62 km/h. The OPAT

dummy was seated on a car seat – one replaced by an identical copy after test. These seats were selected as typical of popular British cars. The seat belt anchorage locations, shown in Figure 1, were also chosen as typical of such vehicles. Transducers were attached to the ends of the safety belts and allowances made for their length in the location of the anchorages. Identical belts were used for each test, type of belt being representative of lap and diagonal belt types in general use in accidents involving seat belt wearers in the test sample.

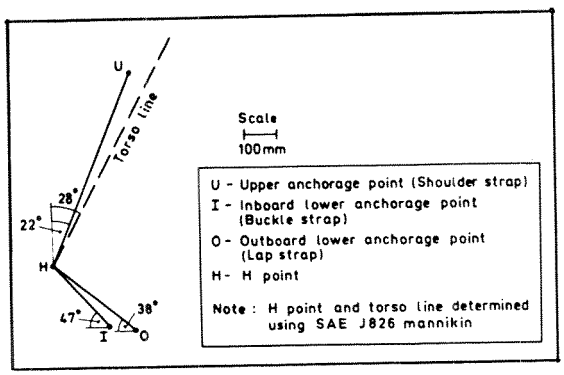


Figure 1. Location of Effective Anchorage Positions Used in Tests with Opat Dummy

Accelerometers were located in the head of the dummy and on the spine some 425 mm below the top of the head. The skeletal deflection of the rib cage was measured with a potentiometer located at the same level. The outputs from these transducers, together with the output from an accelerometer mounted on the test trolley, were registered on a photographic galvanometer recorder using channels of class 1,000 for head deceleration, class 400 for seat belt loading, class 100 for trolley deceleration, class 400 for chest skeletal deflection, and class 400 for spine deceleration. See Reference 6 for definitions of these channel classes.

Deceleration pattern for the trolley, Figure 2, was chosen to be representative of the decelerations thought to have been experienced by the vehicles in the accident sample. Rate of onset of deceleration, and deceleration level required for each test speed were determined by plotting data from a large number of barrier impact tests.

RESULTS OF THE DUMMY TESTING PROGRAM

Plots of the various transducer outputs against impact speed are presented in Figures 3-6. Decrease in chest skeletal deflection at higher impact speeds, indicated in Figure 4 may have been due to a change in the attitude of the dummy at maximum belt extension in the higher speeds tests. Inspection of high speed film records of the tests indicated that dummy torso was more flexed in the higher speed tests, and with the more horizontal chest cage. Further investigation of dummy performance under these conditions is planned.

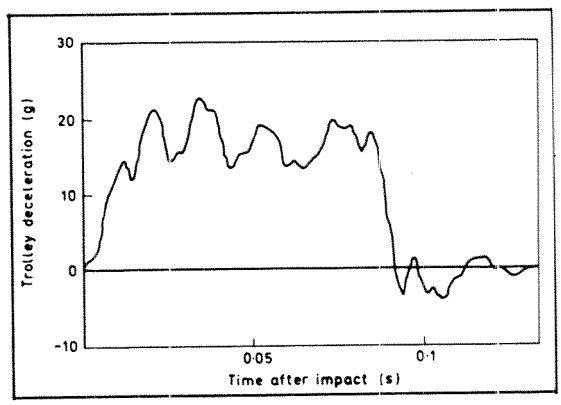


Figure 2. Typical Trolley Deceleration Pattern (Equivalent Barrier Speed 45 km/h)

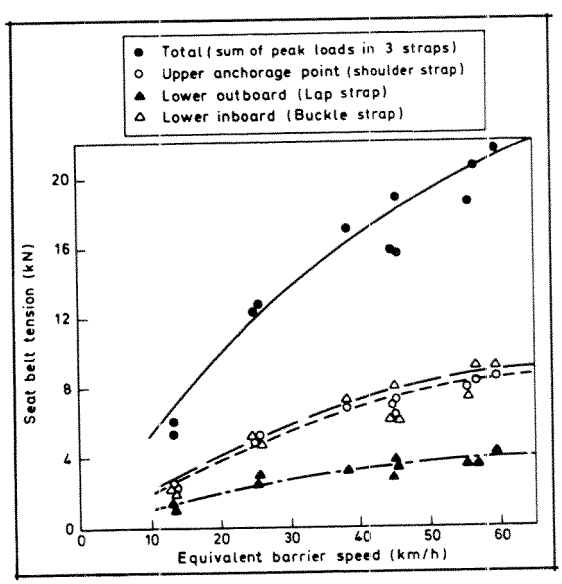


Figure 3. Seat Belt Tension Plotted Against Equivalent Barrier Impact Speed

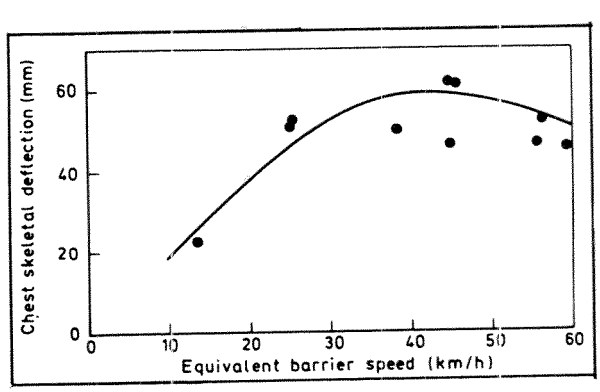


Figure 4. Chest Skeletal Deflection Plotted Against Equivalent Barrier Impact Speed

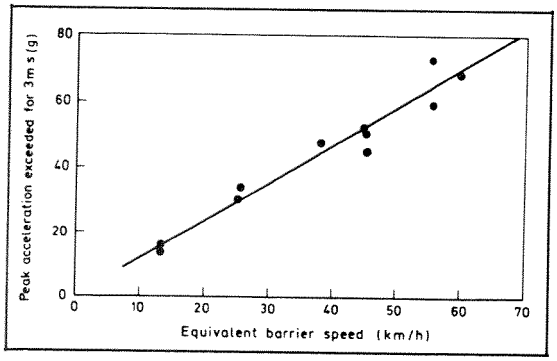


Figure 5. Peak Head Acceleration Plotted Against Equivalent Barrier Impact Speed

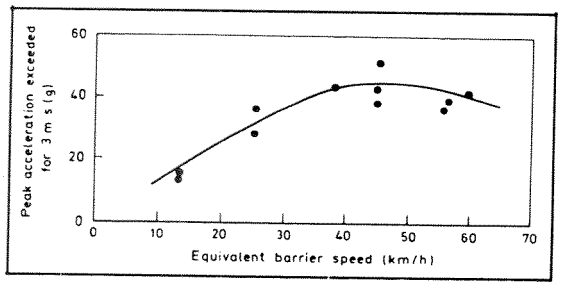


Figure 6. Peak Chest Acceleration Plotted Against Equivalent Barrier Impact Speed

CORRELATION OF THE OPAT TEST RESULTS AND ACCIDENT INJURY DATA

Distribution of the three classes of injury against the relevant transducer outputs from the OPAT dummy are plotted in Figures 7-11. These distributions are presented as percentage of people in sample who would be expected to receive injury when subjected to accident conditions corresponding to the transducer reading from OPAT dummy. If sample subjects can be considered to be representative of population as a whole, then these graphs can be used to give predictions of tolerance levels for total population. If, as is more likely, sample subjects are not representative of total population but only of those who receive injuries in accidents, the data is still valuable as it provides estimates for tolerance levels in reducing such injuries.

A statistical technique known as probit analysis was used to derive distributions from the data, Reference 7. This technique also permits estimations of probable accuracy of analysis. The 95 percent confidence limits are shown in Figures 7-12. OPAT dummy test results indicate shoulder belt loading to be a good predictor of clavicle injuries but not of fractured ribs – the incidence of fractured ribs registering little sensitivity to variations in seat belt load. Further, in clavicle injury prediction, peak deceleration measured at the spine is ineffective, but chest skeletal deflection appears to be particularly sensitive to variable belt load. Abdominal injuries reflect insensitivity to variations in belt loading and detailed study of such injuries indicates that they result mainly from incorrect location of the lap belt – the belt not bearing on the strong bony iliac crests of the pelvis but directly on the abdomen during impact. It is probable that the tolerance to such loading is both relatively and severely low so as to produce injuries to any part of the body. Injuries to the abdomen commonly occur from this loading. Hence, design of safety belt installation should require a minimizing loading risk.

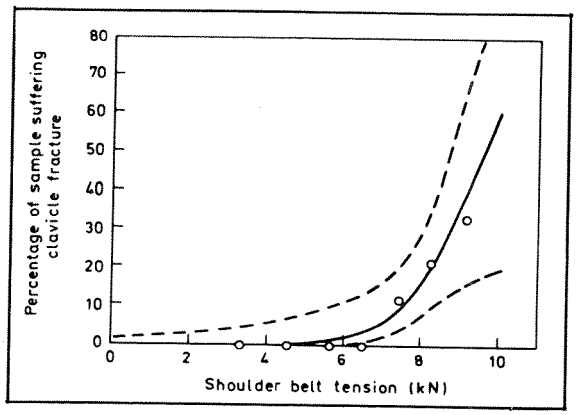


Figure 7. Incidence of Fractured Clavicle Plotted Against Shoulder Belt Tension (at Upper Anchorage Point)

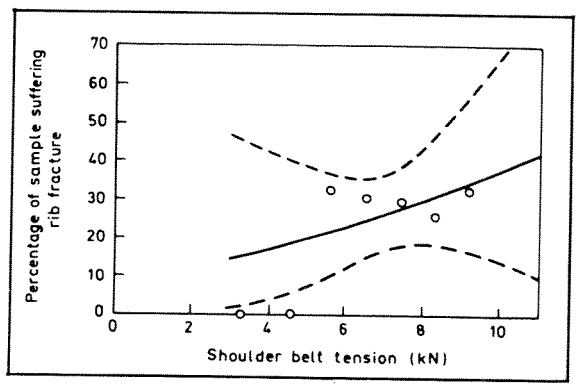


Figure 8. Incidence of Fractured Ribs Plotted Against Shoulder Belt Tension (at Upper Anchorage Point)

It should be noted that in the absense of impact between human and vehicle structure, only one case of brain injury occurred in the sample. It is probable that peak head decelerations recorded in these OPAT dummy tests are below the human injury tolerance level for this type of restraint.

Figure 12 indicates that a human injury tolerance level for the sum of total tension in the seat belt

system might be set at about 18kN if a significant proportion of the injuries attributable to seat belt loadings are to be prevented.

Table 4 lists proposed injury tolerance criteria applicable to OPAT dummy when used to test occupant restraint systems incorporating safety belts.

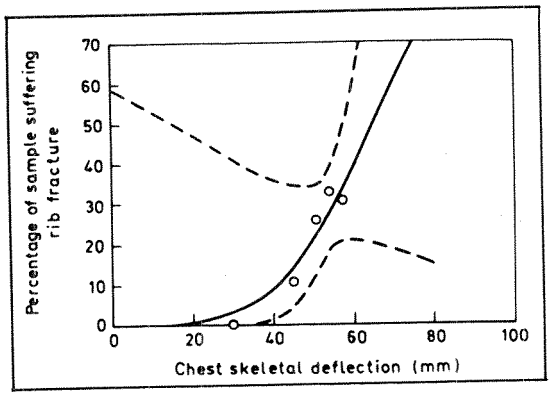


Figure 9. Incidence of Fractured Ribs Plotted Against Chest Skeletal Deflection

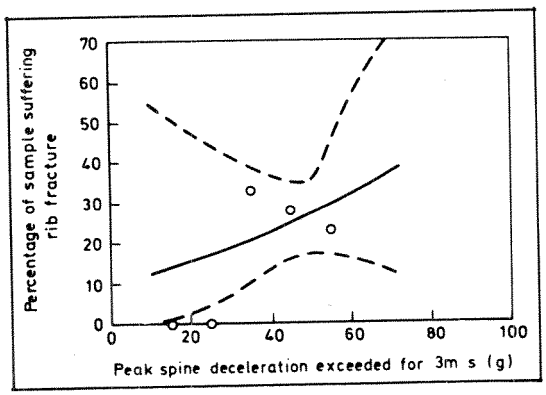


Figure 10. Incidence of Fractured Ribs Plotted Against Peak Spine Deceleration

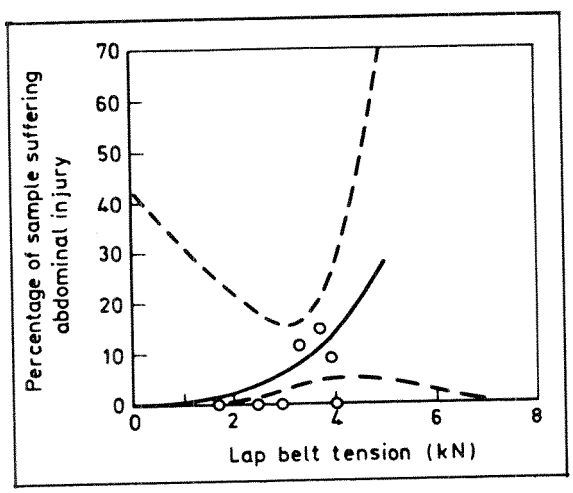


Figure 11. Incidence of Abdominal Injury Plotted Against Lap Belt Tension (at Lower Outer Anchorage Point)

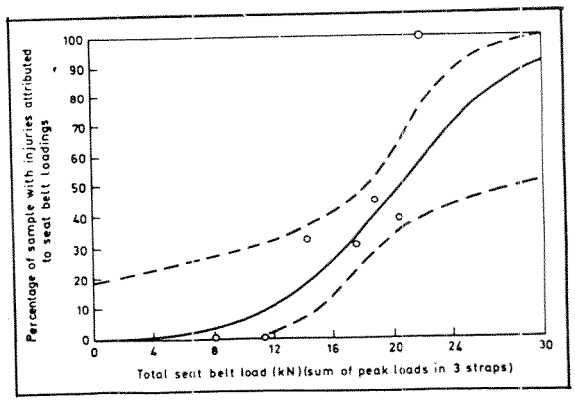


Figure 12. Incidence of Injury Attributed to Seat Belt Loading Plotted Against Total Seat Belt Load

TABLE 4

Injury tolerance criteria for use with the OPAT dummy when testing restraint systems incorporating safety belts

Injury	Criterion	Tolerance Level	Injury Rate at Limit	
			Expected	95% Confidence Limits
Clavicle fracture	Shoulder belt tension (measured at upper anchorage position)	8 kN	15%	10-30%
Rib fracture	Chest skeletal deflection (between 3rd and 4th rib)	45 mm	15%	5-35%
Abdominal injuries	Lap belt to remain located on pelvis of dummy			
All injuries attributed to seat belt loadings	Sum of seat belt tensions	18 kN	35%	25-50%

Specifications for loading level will depend on loading situation. Thus, consider a design of restraint system such that body loading is dependent on accident severity, as in the case of standard seat belts. Then, a high tolerance level culminating in injury to some 25 percent of the people involved in the relatively rare severe accident, might be justified ot improve protection for the remaining 75 percent, *at high speeds,* and of the whole population *at lower speeds*. On the other hand, if loading is applied in all accidents as is the case with pre-tensioned seat belt systems, presently under investigation, then an injury rate of 25 percent, would also apply to minor accidents as well and this would not be acceptable.

CONCLUSION

The method described in this paper has shown promise for estimating human injury tolerance levels for use in OPAT dummy tests. We intend to develop this method in providing distribution estimates for human tolerance levels through use of accident data. In analyzing injury patterns, we plan to provide

information on design options in appraising the various tolerance levels.

The tests may be viewed as part of a continuing program in which the OPAT dummy will be used in tests for reproducing both actual accidents for which injury data is available, and published cadaver tests to improve the definition of injury tolerance levels, and to improve the accuracy of estimated levels.

ACKNOWLEDGEMENTS

The work described in this paper forms part of the program of the Transport and Road Research Laboratory; the paper is published by permission of the Director. The accident-injury data on which the work was based was supplied by the Traffic Medicine and Injury Section and advice on the statistical treatment of the data was provided by the Accident Analysis Division. Assistance in ascribing equivalent barrier impact speeds to the damaged cars involved in the accidents was provided by the vehicle manufacturers.

REFERENCES

1 Warner, P. The U.K. Approach to Dummy Specification. Paper presented to the 4th International Technical Conference on Experimental Safety Vehicles, Kyoto, Japan, March 1973.

2 Macauley, M. Performance Measurements on the OPAT Dummy. Paper presented to the 5th International Technical Conference on Experimental Safety Vehicles, London, UK, June 1974.

3 Warner, P. The Development of the OPAT Dummy. Paper presented to the 5th International Technical Conference on Experimental Safety Vehicles, London, UK, June 1974.

4 Lister, R.D. and J.G. Wall. Determination of Injury Threshold Levels of Car Occupants Involved in Road Accidents. New York: Society of Automotive Engineers Inc. Paper No. 7,000,402, 1970.

5 Hobbs, J.A. The Work of the Road Accident Injury Group. Crowthorne: Ministry of Transport RRL Report LR 108., 1967. (Road Research Laboratory).

6 S.A.E. Recommended Practice J211a. New York: Society of Automotive Engineers Inc., 1972.

7 Finney, P.J. Probit Analysis, 3rd Edition, Cambridge University Press, 1971.

C. TARRIÈRE, G. STCHERBATCHEFF,
P. DUCLOS, A. FAYON
Laboratoire de Physiologie et de Biomécanique
Association Peugeot-Renault

SUMMARY

Over half the people killed on European roads are not in motor vehicles: 28% are pedestrians and 24% are on two-wheelers.[1]

IS IT POSSIBLE TO REDUCE THE NUMBER OF THESE VICTIMS?

After two years research in a field where pessimism remains the general rule, the Peugeot-Renault Association

- presents:
  - a synthesis of its works,
  - proposals for equipping the vehicles,
- proves the efficiency of certain solutions for speeds up to 40 kmh,
- proposes to calculate the "advantages/cost" ratio.

PEDESTRIAN ACCIDENTOLOGY

The Victim

The average height of the adult victim is 1.64 m. Over 62% of fatal injuries are located at the head.[2]

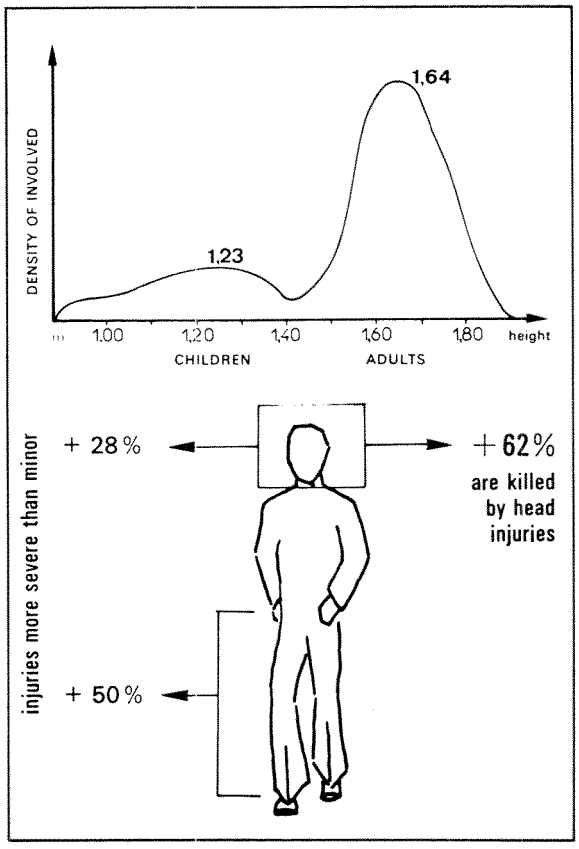
The Motor Vehicle

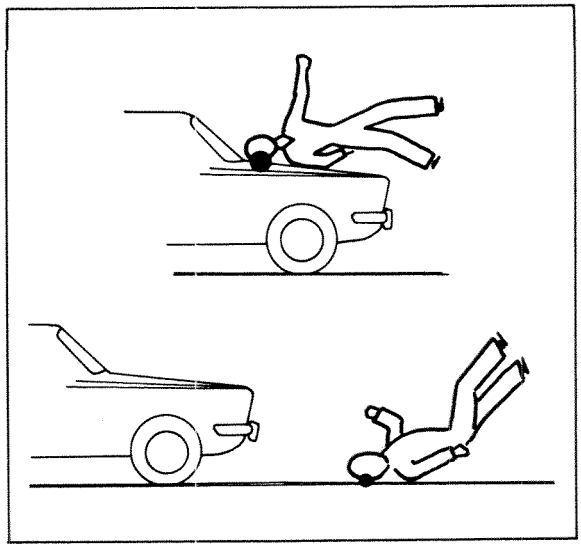
The motor vehicle, rather than the ground, is more frequently the cause of grievous or fatal injuries,[3] at least for the class of impact speeds most frequently encountered: 20 to 40 kmh.[4,5]

About 30% of these grievous or fatal injuries are caused by the hood, the windshield and its framework.[3] This percentage seems to be higher for fatal injuries in 50 cases of fatal accidents: 44% of skull fractures and 44% of thoracic fractures are in relation with more than 50% of vehicle deformations on the windshield and its framework and 50% on the hood.[6]

The Speed

Because 90% of the accidents happen in built up areas, and in most cases after braking, the average impact speed is relatively low.[7]





EXPERIMENTAL COLLISIONS

Fifty experimental collisions, involving approximately 100 adults and children, were carried out. Simultaneously, a mathematical simulation with bidimensional models of the pedestrian was carried out; it is operational.

Reconstitutions of real accidents were carried out:

- some partial, regarding the impact of the head with certain areas of the vehicle,
- some total, by special biomechanical experimentations entrusted to European Medical Institutes.

Synthesis of the Results

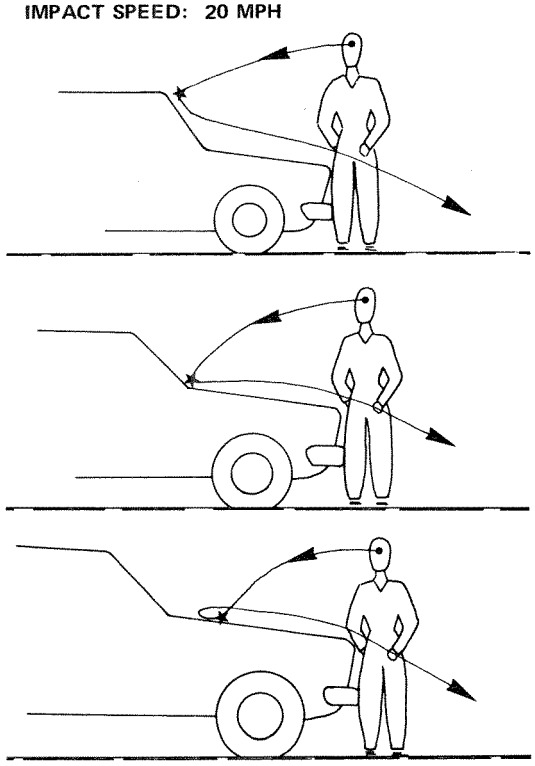
For a given speed the height of the bumper, the shape, the height, and the rigidity of the front section of the vehicle, and the length of the hood, command the overall kinematics of the pedestrian. These, in particular determine:

- the zone hit by the head, and thus the severity of this impact,
- the shape of the trajectory and the projection speed of the pedestrian hit by the vehicle, and thus the severity of the impacts on the ground.

The essential of these results appears in Figure 1. One can see that the higher the front edge of the hood:

- the greater the chances for head impact on the hood, which is a more favorable area and is one which is easier to arrange,
- the flatter the secondary trajectory of the pedestrian, which is favorable since ground impact is less severe.

PEDESTRIAN KENEMATICS FOR THREE DIFFERENT VEHICLE FRONT-END PROFILES. IMPACT SPEED: 20 MPH



THE OBJECTIVES FOR IMPROVED VEHICLES

1. Control the trajectory of the pedestrian so as to avoid head impacts against the most rigid and the most difficult to arrange areas of the vehicle.
2. Arrange the impacted areas of the vehicle.
3. Cut down the severity of impact against the ground.

THE SOLUTIONS

The First Solution

An original idea, the efficiency of which has been demonstrated by full scale experimentation:

- "A front section of the vehicle having a variable geometry."

The Principle (Figure 2) – The first contact with the pedestrian produces the tipping of a shield, which, by optimizing the shape of the front face of the vehicle, makes it possible to:

- modify the kinematics of the pedestrian, and consequently, the trajectory of the head so that its impact, though unavoidable, is made with the arranged hood of the vehicle,
- to keep the pedestrian on the vehicle, or at least to delay as much as possible his fall on the ground, so as to cut down its severity.

The Tests – The biomechanical tests carried out by experts in a specialized European Medical Institute demonstrate the efficiency of such a solution. For example, at 32 kmh, a pedestrian would only have had slight injuries, and only one lesion; a simple fracture of the fibula.

Advantages – This "variable geometry" has the great advantage of reaching such a degree of efficiency without imposing a particular shape to the hood area, and thus, without introducing any supplementary important restraints regarding the styling of the vehicle, nor regarding its characteristics (aerodynamics, cooling, etc.).

This solution remains valid for most of the motor vehicle park but loses some of its utility regarding very short vehicles with short hoods.

Protection in the Case of a Child – In the great majority of cases during an impact, the head of a child is higher than the upper edge of the hood located below 0.80 m. It follows that the child will sustain a violent lateral flexion of the neck and an impact of the head against the hood.

The device described above has not been tested with a child. However, it can be expected that by the impact of the head against the tipped shield (taking into account the damping properties of the shield) the hyperflexion of the neck and the danger of severe head injuries could be avoided or reduced at least for some children of low height.

The Other Solutions

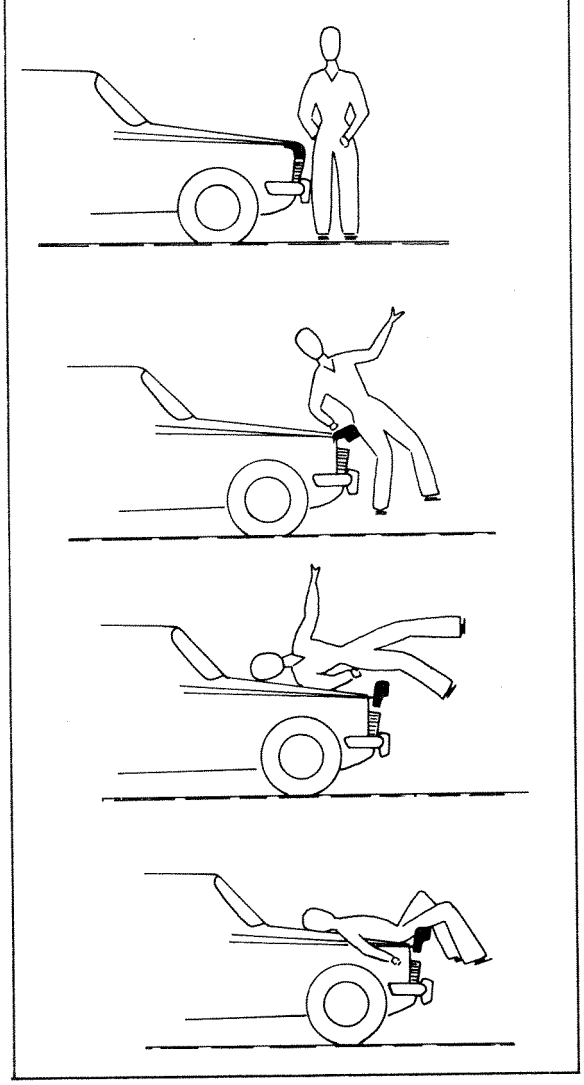
Multiple arrangements of areas usually hit by the pedestrian can, within a range of speeds reaching 32 kmh, decrease the risk of severe or fatal injuries.

How? By optimizing the shape and the damping properties of the areas frequently hit by the pedestrian.

1. *The bumper*

- the height of the bumper should be reduced (40 cm would be a maximum) so as to avoid articular lesions of the knee to the biggest percentage of the population and to optimize the kinematics of the pedestrian;
- the surface in contact with the leg should be larger, when possible, and covered with a shock absorbing material.

EXPERIMENTAL VEHICLE WITH A VARIABLE FRONT-END GEOMETRY



2. *The front edge of the hood*, the headlamps and the fenders should have a shape and a rigidity as least aggressive as possible.
3. *The windshield* should be made of laminated glass and its *frame*, which is the most dangerous of the areas hit by the pedestrian, should be:

- masked by the hood at the lower crossmember,
- made less likely to be hit by means of various designs (for example withdrawn regarding the main plane of the windshield which can have large curves at its upper and side edges).

NEXT STEP

1. The defining of specification sheets in terms of Protection Criteria for the various body segments.
2. Evaluation of the "advantages/cost" ratio for the solutions put forward (including the advantages that they have regarding improvement of the protection of the "two-wheeler" drivers).

BIBLIOGRAPHY

1. François Hartemann and C. Tarriere, "Synthesis of statistical data on traffic accidents in France, West-Germany, Italy and United Kingdom." Report to the Fifth International Technical Conference on Experimental Safety Vehicles, June 1974.
2. B. Fabricius, J. Niklas, and E. Fiala, "Pedestrian accident tests with catapult." Research Report No. 40. Motor Vehicle Institute, Technical University, Berlin, 1968.
3. Transport and Road Research Laboratory, "Pedestrian Injuries." Department of the Environment, Crowthorne, Berkshire, RG 11 6 AU.
4. G. Murray Mackay and C.P. de Fonsecka, "Some aspects of traffic injury in urban road accidents." In Proceedings 11th STAPP Car Crash Conference. S.A.E., 485 Lexington Ave., New York, 10017, 1967.
5. Paul M. Culowski et al., "Research in impact protection for pedestrians and cyclists." CALSPAN, P.B. 201260, May 1971. National Technical Information Service, U.S. Department of Commerce.
6. S. Kamiyama and G. Schmidt, "Relations between collision speed, car damage, fractures and distance of projection in fifty cases of pedestrian-car accidents."
7. G. Murray Mackay, "Injury to pedestrians." A report to the Committee on Pedestrian Safety, Committee on the Challenges of Modern Society, NATO, Brussels, Belgium 1972:8.

BIOMECHANICAL STUDY OF SIDE IMPACT ACCIDENTS

D. CESARI
M. RAMET
Organisme National de Securite Routiere

INTRODUCTION

A technical and medical investigation on road accidents has been conducted by the "Laboratoire des Chocs et de Biomécanique" of ONSER in Bron since November 1969. As a result of this investigation, the lateral impact is seen to be the most frequent configuration of accidents after the frontal impact.

When considering a sample of 650 vehicles, the different impacts are distributed as follows:

- frontal impact : 66.5%
- lateral impact : 18.5%
- rear impact : 5.5%
- rollover : 9.5%

The relative frequency of side impacts is similar to data of other countries, especially in European countries and in Australia.[2,3] This fact is the main reason for studying more deeply the severity of side impacts. Average total overall severity index (OSI)[5] of vehicles' occupants subject to a side impact = 2.90. Average OSI of occupants subject to a front impact = 2.34.

The severity of injuries becomes more serious if we consider the occupants seated on the same side as the impact location. Average OSI of vehicles' occupants involved in a side impact accident and seated on the same side as the impact location = 3.40.

Two types of side impacts can be considered:

- collisions vehicle to vehicle
- collisions against fixed objects

An intersection accident is representative of the first type vehicle-to-vehicle where the directions of impact are mostly orthogonal. This type is 83.3% of all side impacts in our investigation.

Impacts against fixed objects occur when the car sideswipes. 16.7% in our investigation are side impacts against fixed objects.

The history of injury occurrence is quite different according to the types of side impacts:

- when against fixed objects, the role of lateral structures of the vehicle is practically insignificant. The occupant seated on the side of the impact location strikes the object as the vehicle is being blocked against it.
- during a collision vehicle-to-vehicle, the impacted vehicle is accelerated by the striking vehicle; the values of Δv and of the intrusion can be changed by better designing the front of the striking car, the lateral structures of the struck car and/or both. This study will review only this type of impact.

RESULTS OF THE INVESTIGATION

General Data

This investigation will review only crashes between small vehicles where impact directions are distributed between 2 and 4 o'clock (right side impact) and between 7 and 9 o'clock (left side impact).

Not considered in this investigation:

- ejected occupants
- occupants whose vehicle was overturned after the first impact
- in case of several impacts, all occupants when a correlation between injuries and impacts was not clear.

Corresponding to these conditions, 53 accidents are reported. 87 front passengers* and 9 rear passengers were seated in the 53 struck vehicles with the following distribution:

- 12 left side struck vehicles with 1 occupant
- 10 right side struck vehicles with 1 occupant
- 17 left side struck vehicles with 2 or more occupants
- 14 right side struck vehicles with 2 or more occupants.

The distribution between right side impacts and left side impacts is approximately similar, but there are, however, a little more left side impacts.

Distribution of Impact Directions

Side impacts are those where the main impact force is directed towards 2, 3 and 4 o'clock (right side impact) or towards 8, 9 and 10 o'clock (left side impact). Among these six directions, the distribution of impacts for the 53 cars is as follows (Figure 1):

- 2 o'clock: 3 vehicles
- 3 o'clock: 20 vehicles } Right side impacts: 24
- 4 o'clock: 1 vehicle

- 8 o'clock: 2 vehicles
- 9 o'clock: 26 vehicles } Left side impacts: 29
- 10 o'clock: 1 vehicle

The study of this distribution shows that there are a little more left side impacts (29) than right side ones (24). This can be explained because only corporal accidents are enclosed in our sample. Another reason is that, in case of a vehicle with only one occupant, the risk for him to be injured is higher when it is a left side impact (on the side of the occupant) than when it

*Six of them wore safety belts.

is a right side impact (on the opposite side of the occupant, see Severity Depending on Deformation).

It may be noted that most impacts show a direction perpendicular to the axis of the struck vehicle (46-53), these accidents happening mostly at intersections. However, the determination of the accurate direction of the impact for the struck vehicle is rather imprecise. The study of a crashed vehicle does not always allow the determination of the vehicle position at the beginning of the impact. Sometimes, the struck vehicle shows a partial rotation during the deformation period as can be seen in the tests described later.

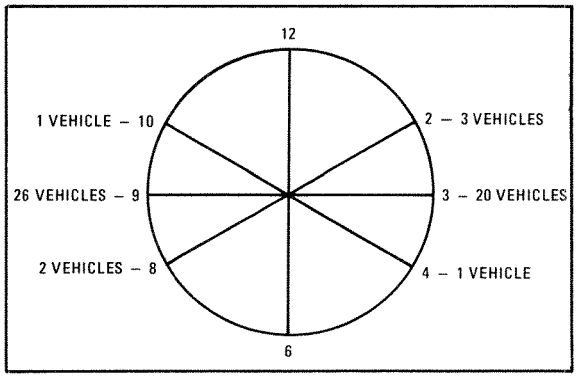


Figure 1. Distribution of Impact Directions (O'Clock Position)

Distribution of Areas of Impact

To study the distribution of the impacted areas, the side of the vehicle is divided into four parts: the front fender, front part of the passenger compartment (area of the front door of a four door vehicle), rear part of passenger compartment (area of the rear door), rear fender. Each of these parts can be struck, either alone or together with an attached part. Among the 53 impacted vehicles, the following distribution can be observed (Figure 2):

- front fender : 6 vehicles
- front fender plus front part of passenger compartment : 9 vehicles
- front part of passenger compartment : 2 vehicles
- whole passenger compartment : 1 vehicle
- rear part of passenger compartment : 1 vehicle
- rear part of passenger compartment plus rear fender : 6 vehicles
- rear fender : 1 vehicle

The distribution shows that :

- the front half of the vehicle is more frequently struck than the rear half (47 times against 36)
- in most cases, the deformation of the passenger compartment can be observed (46 cases among 53)
- mostly, two parts of the vehicle near each other show a deformation (43-53). In three cases, the passenger compartment was only slightly affected by the impact. These three cases are rear directed impacts and of low intensity (4 and 8 o'clock impact).

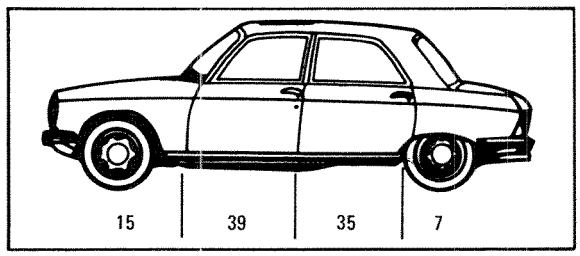


Figure 2. Impact Frequency of the Lateral Parts of the Car

Distribution of the Δv

The determination of a value of speed change during the period of deformation (Δv) allows us to estimate the impact intensity (Figure 3).

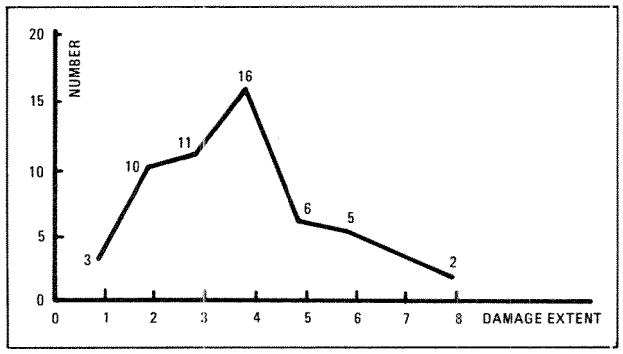


Figure 3. Distribution of Damage Extents

The Structures Department of the RNUR worked out a method to calculate the values of Δv for the damaged vehicles.[6,7]

The values of Δv are determined by comparison with impact tests of standard design. For every damaged vehicle, either a speed equivalent to a considered test or a value of average acceleration is calculated. These data are related to the value of the deformation of every car (Δe) and allow the energetic balance and the approach speed of vehicles to be estimated. The weight of the vehicles being known, the values of the Δv for each vehicle can be calculated. This method is for the moment rather imprecise, because the impact tests conducted do not correspond to the deformations observed from in-the-field crashes. That is why the starting point of the calculation is the determination of an intuitive value for speed or acceleration depending on the subject knowledge the analyst has about impact tests.

France has promoted a program of research in which every car manufacturer takes part. This

program should allow us to determine more precisely the values of Δv with the help of a great number of crash tests for the main types of vehicles in different configurations and depending on three ranges of speed.

The values of Δv have been calculated for 47 of the 53 vehicles of this sample of lateral impacts. These values have been calculated together with technical departments of Régie Renault. The calculated values of Δv have been grouped (0-15 kph; 15-25 kph, Figure 4).

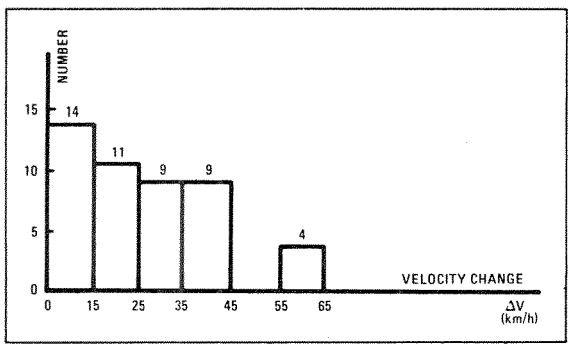


Figure 4. Distribution of Δv Values

The distribution of Δv decreases, the most represented group being the category 0-15 kph with 14 vehicles (i.e. 30% of the sample). More than half of the side struck vehicles studied have a Δv lower than 25 kph.

In our sample there is no vehicle in category Δv 45-55 kph; there are four vehicles in category 55-65 kph; these are serious crashes occurring in the countryside.

There is no obvious connection between the distribution of Δv and the intrusion. This can be explained by the fact that the values of Δv are an increasing function of the rigidity of the struck vehicle, while the values of intrusion are a decreasing function of this rigidity. The rigidity varies according to the impacted areas (see Analysis of the Struck Vehicle: Kinematics) and is a function of the height of the rigid areas of the striking vehicle.

Severity Depending on Impact Characteristics

In 47 of the 53 crashes, the impact is characterized by the determination of the velocity change on the striking vehicle during its deformation (i.e. Δv of the struck vehicle). Data obtained have been arranged in different groups of Δv.

For each group, the average value of the total OSI has been calculated for each occupant, the value 6 being given for all dead people, however great the number of fatal injuries may be.

The average variation of OSI in the deformation groups is shown in Figure 5. The group 45-55 kph is not represented and the average value of OSI for this group has been calculated through a linear interpolation between groups 35-45 and 55-65. It may be noted that this variation is practically linear, therefore the slope of the curve is a little greater for low values of Δv.

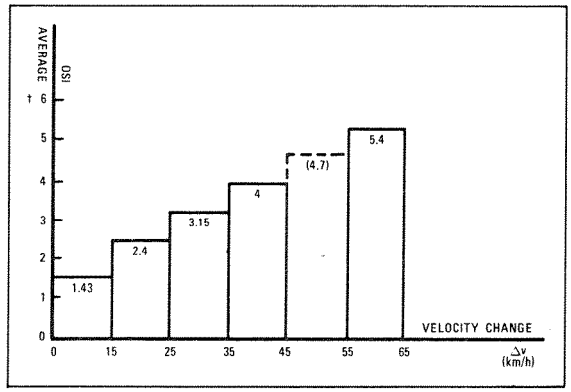


Figure 5. Severity Depending on Δv Values

This histogram shows that for values of Δv equal or higher than group 25-35 the risk of serious injuries for an occupant is great (OSI 3). For the highest group of Δv in this sample, the probability of survival is low, the average value of OSI varying between 5 (uncertain survival) and 6 (death within 24 hours).

Severity Depending on Deformation

For the same 53 vehicles, the average severity of injuries to people (average total OSI) depending on intrusion has been considered.

The deformation related to the width of the vehicle is represented by the value of the last digit of the deformation index (level of deformation).

The value 1 is representative of an intrusion equal to the width of the door, the value 5 of an intrusion reaching half the width of the vehicle.

The variation of the average total OSI depending on the intrusion has been made by differentiating occupants seated on the side of the impact and occupants seated on the opposite side of the impact. The study of these two variations shows that the severity of injuries to the occupant seated on the side of the impact is always greater than that affecting the occupant seated on the opposite side of the impact. This difference exists in all types of deformation.

Severity of Occupants Seated on the Side of the Impact – The curve of the average total OSI of occupants seated on the side of the impact depending

on intrusion shows that the severity decreases when the intrusion varies from 1 to 2, then increases regularly when the intrusion exceeds 2 (Figure 6).

This diminution can be due to a coupling phenomenon, the deformation 2 corresponding to a position of the internal lateral side of the vehicle

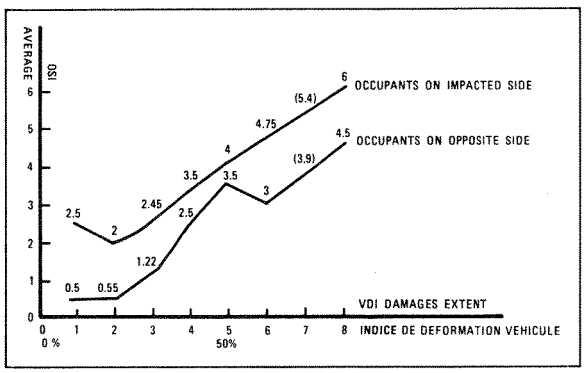


Figure 6. Severity Depending on Deformation

reaching the side of the occupant, which would prevent the occupant from being thrown against the walls of passenger compartment. Each time the deformation reached the level 8, all the occupants seated on the side of the impact died.

Severity of Occupants Seated on the Opposite Side of the Impact – The variation curve of the average total OSI of the occupant seated on the opposite side of the impact is similar to the one concerning the occupants seated on the side of the impact; however, the first one is always lower than the second one (Figure 6).

The diminution of severity when the deformation varies from 5 to 6 can be due to the coupling of the occupant seated on the opposite side of the impact with the lateral wall deformed up to a position located near the occupant in concern. However, this assumption cannot be confirmed. Because of the small size of sample, the average values may be far from the value calculated from the statistical variation.

Interaction of Occupants

The interaction between occupants can be seen by comparing severity of injuries received by drivers and severity of injuries received by two front passengers of vehicles with at least two passengers.

This interaction may have two effects:

- the increase in frequency and the severity of injuries received by the occupant seated on the side of impact, this occupant being pushed by the occupant seated beside him.
- the protection of the occupant seated on the opposite side of impact, keeping him from being thrown against the walls of passenger compartment, the occupant seated beside acting as padding.

The first effect is difficult to be seen, because in some cases, injuries are found in areas far from the impact point (see Analysis of the Dummy). However, it may be noted that, for a given value of Δv, the occupants seated on the opposite side of impact are worse injured when alone than when there is a passenger beside them.

Frequency and Different Kinds of Lesions

The human body may be divided into seven main parts: head, cervical spine, chest, upper extremities, thoracic and lumbar spine, pelvis with abdominal viscera, lower extremities. The frequency of injuries to these different parts is shown in Figure 7.

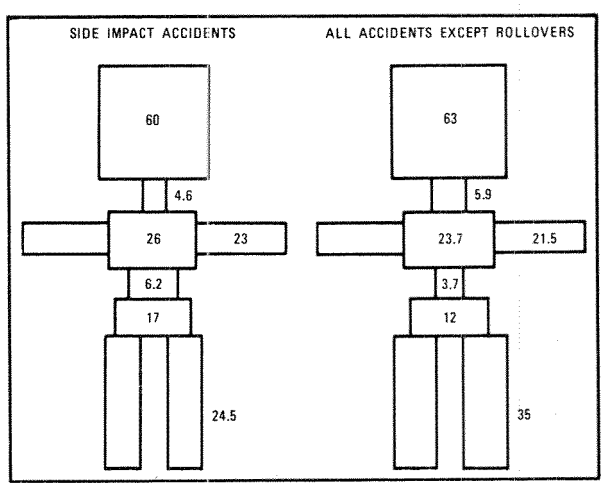


Figure 7. Frequency of Injuries (in Per Cent of Injured People)

It may be noted that the head is the most frequently injured part (60%), this result being found with a higher percentage for all corporal accidents together. These lesions are generally cerebral commotions without fracture of the skull.

With regard to all accidents together, differences are mostly injuries to pelvis and abdomen which are more frequent in case of side impacts. These injuries are fractures of pelvis through lateral compression and are often together with lesions of viscera due to the depression of pelvis. These lesions increase the risk and, in some cases, may lead to death. These are spleen ruptures, lesions to liver, pancreas, kidneys or genital and urinary organs.

On the other hand, in case of side impacts, injuries to lower extremities are very few. These lesions are generally fractures through bending of femur or tibia

and occur when the knee remains fixed during the impact (by the steering-column, for instance). This fact is the reason for the low rate of these injuries.

Thoracic lesions are single or multiple fractures of ribs with a great number of fly chests. These fly chests are found mostly in occupants seated on the side of impact and in the case of the most serious accidents.

EXPERIMENTAL COLLISIONS

The main existing standard test procedure for side impacts consists of crashing a stationary vehicle, either with another vehicle or with a moving barrier or a pendel facing the H point of the driver.

The results of our investigation show that all parts of the side of the vehicle may be crashed, but in case of corporal accident, the more frequent configuration is the impact of the passenger compartment.

Important experimental work has been carried out in this crash configuration.[3]

Description of Crash Tests

Four tests with moving vehicles were conducted in order to estimate the influence of location of impact point on the crash severity (Figure 8).

In the four tests, the path angle between the two vehicles was 90°. For each test, the speed of both vehicles is the same. The range of speed varies from 30 to 41 kph. A three point belted dummy is seated as driver of the struck vehicle. The impact points were: front fender, windshield pillar, lock pillar, rear fender.

Analysis of the Struck Vehicle

Kinematics – The kinematics of the vehicles is different depending on the location of the impact point.

In test No. 1, the two vehicles rotated about 45° and were found, after the deformation period, on parallel paths.

In test No. 2, the struck vehicle kept the same direction during all the crash period, but swept at the same time as it was moving ahead and involved the striking vehicle which rotated 45°.

In tests No. 3 and 4, the striking vehicle made the struck vehicle rotate while translating and spinning it like a top. The rotation of the struck vehicle was, in both cases, approximately 270°. On the other hand, in these two tests, the striking vehicle moved in a different manner.

In test No. 4, the path of the striking vehicle was not modified, while in test No. 3, the latter involved the struck vehicle in an angular movement of 90° and then, both vehicles were separated from one another. In the same test, the struck side rose and the struck vehicle began to roll, then fell down again, its center of gravity staying within its standing axis.

Deformations and Intrusion – Three parameters have a direct effect on the importance of the deformation of the struck vehicle:

- the speed of impact: we have to remember that in reality, the vehicle moving at 41 kph shows 65% more energy than a vehicle at 32 kph.
- the rigidity of structures: this parameter becomes effective when the struck car crashes into both a

Test Configuration	Speed Kph	Impact Duration	Vehicle Acceleration (Struck Vehicle)				Dummy Acceleration								Stricking Vehicle Accelerations		
							Head				Chest						
			γx max	γy max	γz max	γy γx	γx max	γy max	γz max	γ max	γx max	γy max	γz max	γ max	γx max	γy max	γ max
1. Front Fender	38	105 ms	11 g	14 g	6 g	1.27	28 g		16 g		15 g	18 g	7 g	24.4	28 g		16.g
2. A. Pillar	32	95 ms	7 g	12 g	4 g	1.71		52 g	95 g		25 g	18 g	33 g	46.6			
3. B. Pillar	41	70 ms	18 g	12 g	16 g	2.50	20 g	80 g	40 g		13 g	57 g	18 g	61 g	14 g	9 g	
4. Rear Fender	30	65 ms	5 g	13 g	4.4 g	2.69		18 g	19 g		25 g	25 g	29 g				

rigid and a crushable structure. In test No. 2, for instance, the left side of the struck car crashes into the front wheel (rigid structure) whereas the right side crashes into the front door which is not very strong. The result is a deformation of a few centimeters of the front fender of the impacted vehicle, its wheel not being depressed on its axis and a deformation of 38 centimeters of its front door. This impact location determines a corner intrusion (front right corner) of the striking vehicle although the directions of the two vehicles were strictly perpendicular at the beginning of the impact. The low rigidity of the passenger compartment explains that the impact against this compartment gives a very great intrusion, i.e., an intrusion whose depth varies from 22 to 27 centimeters and whose surface is equal to the front part of the striking vehicle. In the two other tests, rigid areas were struck (front and rear wheels) and the deformations obtained were lower.

- the distance between the center of gravity and the impact point: when the impact point is located near the longitudinal position of the center of gravity, a great amount of energy is absorbed by deformation, the vehicles having, during the period of deformation, a common direction approximately situated along the bisectrix of their direction before impact. They diverge and stop after a few meters (test No. 2). On the other hand, when the center of gravity is further, the deformations are not so great, the striking vehicle changing the direction of the struck vehicle. After the deformation, the remaining energy allows the vehicle to keep on moving.

Accelerations of the vehicle depending on the location of the impact point – The duration of the acceleration period in the struck vehicle is a function of the location of the impact point. When the lateral impact point is located at the front of the struck vehicle, the deformation period is higher than when the impact point is located at the rear part of the vehicle. When the impact point on the struck vehicle is situated at the front part, the two vehicles move together for a long time (about 100 ms), each vehicle deviating the other from its path. On the other hand, when the impact point is located more at the rear, the struck vehicle is submitted to a rotation couple and leaves the striking vehicle faster.

The struck vehicle is submitted to an important Z acceleration (18.8 g in test No. 3). This acceleration appears very late in the crash and corresponds to a rolling movement of the vehicle, the struck side-part raising itself at the end of the crash. After the deformation period, the struck vehicle oscillates, its movement damped through the suspension, giving a sinus wave on the recording of the Z acceleration.

The Y acceleration (γy) measured on the struck vehicle is always higher than the X acceleration and the ratio $\gamma y/\gamma x$ is an increasing function of the distance between the impact point from the front of the struck vehicle. In other words, the acceleration force acting on the occupant is determined by two components, a transversal one and a longitudinal one and this last component is more effective when impact point moves away from the front of the vehicle.

The transversal acceleration grows faster than the longitudinal acceleration and the highest value of γy is obtained before the maximal value of γx; because of this, the path of the occupant is almost lateral, the movement being directed forwards after support on the lateral wall of the passenger compartment.

Analysis of the Dummy

Kinematics – The kinematics of the dummy placed in the impacted vehicle is different whether or not the deformation reaches its sitting position in the passenger compartment.

In test No. 3, the struck door quickly pushes the dummy towards the right before it even begins to move normally towards the left. This pushing movement is so great that the dummy chest is released from the safety belt and lies down on the passenger seat. The highest force to be registered at that moment on the buckle is 8400 N.

In the other tests, the passenger compartment did not directly push against the dummy and, because of this, about 50 ms after the beginning of impact, it begins to move to the left and forward. The movement ahead is decelerated by the restraint system and the dummy goes on moving sideways and crashes into the door about 100 meters after the beginning of the impact.

In test No. 2, the movement of the dummy has been restrained by the deformation existing in front of it, which shows that the impact against the wall of the compartment happens earlier in the time history of impact.

In test No. 1, the front door opened itself at the beginning of the impact; the dummy was restrained by the three point belt, which prevented it from being ejected.

Accelerations of the Dummy –

Thoracic Accelerations. Considering the four tests, it can be noted that in tests 1 and 4 – impacts where there is no intrusion in the passenger compartment – the curves remain flat during 60 ms;

then the accelerations γx and γy slowly increase to give a maximum 100 ms after the beginning of the impact. On the other hand, in tests in which there are deformations of the passenger compartment, the variations of acceleration are very often near the impact point, i.e. 20 ms after this one for test No. 3 and 25 ms for test No. 2.

These variations are very fast (11,400 gs) and the peaks correspond to successive impacts of the thorax.

The highest thoracic acceleration has been recorded in test No. 3. The maximum resultant value of this test is 61 g. This value corresponds to the first impact against the lateral wall of passenger compartment. The registered peak is very short and, if we admit that the value 60 g can be exceeded during added periods lower than 3 ms, it can be considered that the thoracic acceleration of this test is below the tolerance presently admitted.

In the same test, a second peak situated 19 ms after the first one and corresponding to a second impact with the wall of passenger compartment is found on the records. Then, the accelerations γx and γy change their directions and show a lower maximum (13 g for γx and 9 g for γy) at 85 ms after the beginning of the impact; this acceleration is due to the safety belt; the dummy, pushed by the intrusion, is maintained by the shoulder and lap strap. At the same time, the values of the forces applied to the anchorage points of the safety belt are at their maximum, the highest force being applied on the buckle strap.

In test No. 2, the deformation area has happened in front of the chest of the driver, which explains that the maximum value of γy is lower than the one in test No. 3. On the records of the acceleration γy, a maximum 25.5 g is to be found. It corresponds to the impact of the front part of the torso against the crushed area; this impact occurs at a low speed because the dummy is restrained by a three point belt which has decelerated its movement.

In test No. 1, the front door opened and the low registered decelerations were due to the restraint system of the dummy.

Accelerations of the Head. The films representing the four tests show that the head has never impacted the passenger compartment. However, although the maximal values of accelerations of the vehicle are approximately the same values, the values of accelerations of the head are not alike in all tests. They can be classified in two categories:

- low values (less than 30 g) are found in tests without intrusion in the passenger compartment.
- high values (more than 40 g) recorded in tests 2 and 3 where intrusion appeared and where the chest and/or the pelvis of the dummy impacted the door during its deformation period. In both cases, the maximal values are very short peaks corresponding to the acceleration peaks of the chest and occur later on in the time history of impact.

The HSI value (601) was calculated only in test No. 3. Records of a component of acceleration during the three other tests were not available. Therefore, the highest values were obtained in this test and it may be supposed that the values of HSI in the other tests were lower than those calculated for test No. 3.

The HSI value of the head in test No. 3 (HSI_3) is lower than 1000 corresponding value of the cerebral concussion in a frontal impact. However, the HSI value of the cerebral concussion in a lateral impact is unknown.

COMPARISON BETWEEN IN-THE-FIELD ACCIDENT AND CRASH TESTS

This comparison can be made when looking at deformations of vehicles and comparing the dummy behavior to the injuries received by occupants.

Vehicles

The study of distribution of impact areas shows that the more frequent impact point is an impact against the whole passenger compartment following an orthogonal direction to the struck vehicle.

This configuration was simulated in test No. 3. The static deformation observed corresponds to the value 3 of the VDI scale. In in-the-field accidents, a great number of vehicles are found with a more important deformation, although the speed of test No. 3 (40 kph) is not very high.

In the conducted test, the striking vehicle is supported by the beam of the frame of the struck vehicle and for this reason, the intrusion is limited.

On the other hand, in in-the-field accidents, a penetration of the striking vehicle can be found when the front of this one goes under the beam of the struck vehicle and does not meet a rigid structure which could stop it.

The configurations of tests 1 and 4 are not very often found in in-the-field accidents because they practically involve a risk of ejection. This has been confirmed by those two tests in which one of the doors situated on the side of the impact opened itself during the impact.

In in-the-field accidents, the struck vehicle sometimes turns over after the impact; this phenomenon has been shown for tests in which, either the passenger compartment or the rear fender was struck.

Dummies and Occupants

It is difficult to compare the dummy, placed at the driver's seat in a crash test, with the occupant who sits on the side of impact during in-the-field accidents because the dummy does not show injuries that could have been produced by the impact and because interpretation of the sustained accelerations is impossible in terms of human tolerance, the tolerance levels in side impacts being unknown.

However, most in-the-field accidents (see Results of the Investigation) involve deformations of passenger compartment and the severity increases with the greatness of intrusion.

The highest head and chest accelerations have been recorded for the crash test in which the intrusion was directly onto the dummy although the acceleration of the vehicle was nearly the same in every test.

CONCLUSION

This study has shown that in-the-field side impact accidents generally occur involving important deformations of the passenger compartment. These deformations are often greater than those created by crash tests conducted with two moving vehicles, the speed of each one being 40 kph.

The severity of injuries is a little higher for occupants seated on the side of the impact than for the others. This severity increases with the deformations of the passenger compartment. The crash tests give acceleration values of the dummy which are higher when the intrusion reaches the dummy during the impact. However, it is not possible to foresee the effect of stiffened structures on the passenger because the human tolerance to side impact is still little known.

REFERENCES

1. Gadd, C. W. (1966); "Use of weighted impulse criterion for estimation injury hazard." Proceedings of the 10th Stapp Car Crash Conference.

2. Mackay, G. M. (1969); "Causes and effects of road accidents." The University of Birmingham. Vol. 3.

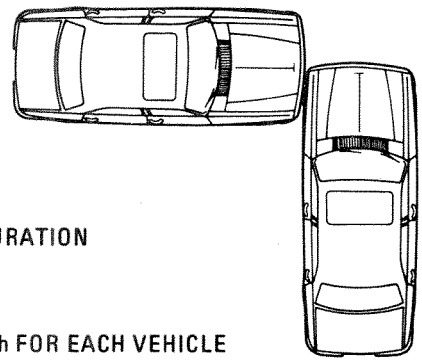
3. Ryan, A. (1967); "Injuries in urban and rural traffic accidents. A comparison of two studies." Proceedings of the 11th Stapp Car Crash Conference.

4. Severy, D. M. and Coll. (1952); "Automobile Side Impact Collisions Série II." SAE Paper No. SP-232.

5. States (1969); "The abbreviated and comprehensive research injury scales." SAE Transactions paper No. 690810.

6. Ventre (1972); "Proposition de méthode d'analyse des vitesses de collisions en accidents réels." Régie Nationale des Usines Renault.

7. Ventre, P., Provensal, J. (1973); "Proposition d'une méthode d'analyse et de classification des sévérités de collision en accidents réels." IRCOBI Conference Compendium.

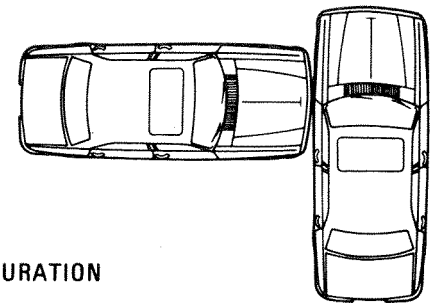


CRASH n° 1

TEST CONFIGURATION

ANGLE : 90°

SPEED : 38 kph FOR EACH VEHICLE



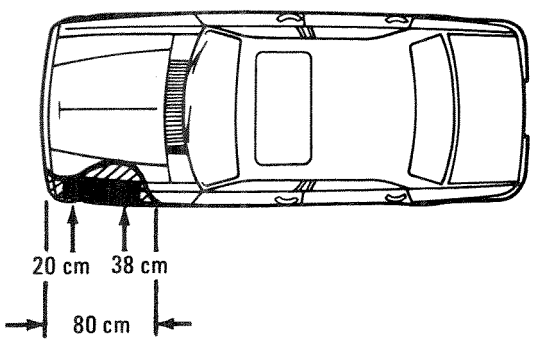
CRASH n° 2

TEST CONFIGURATION

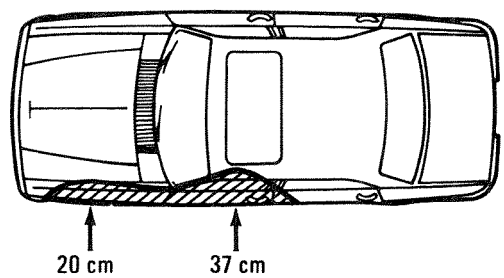
ANGLE : 90°

SPEED : 32 kph FOR EACH VEHICLE

STATIC DEFORMATIONS



STATIC DEFORMATIONS



Accelerations

Vehicle	Dummy	
Duration – 105 ms	Head	Chest
γx – 11 g	γx – 28 g	γx – 15 g
γy – 14 g	γz – 16 g	γy – 18 g
γz – 6 g		γz – 7 g
$\frac{\gamma y}{\gamma x}$ – 1.27		γ max – 24.4 g

Decelerations

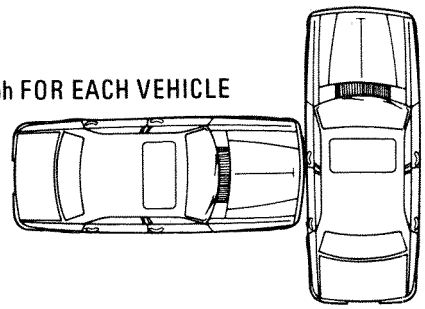
Vehicle	Dummy	
Duration – 95 ms	Head	Chest
γx – 7 g	γy – 52.5 g	γx – 25 g
γy – 14 g	γz = 95 g	γy – 18 g
γz – 6 g		γz – 33 g
$\frac{\gamma y}{\gamma x}$ – 1.71		γ max – 46.6 g

CRASH n° 3

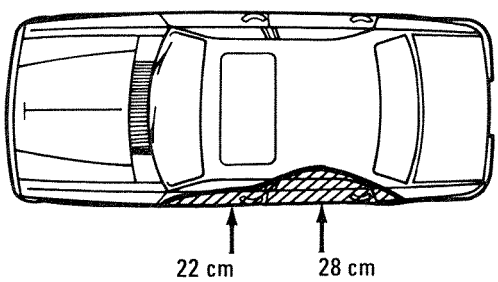
TEST CONFIGURATION

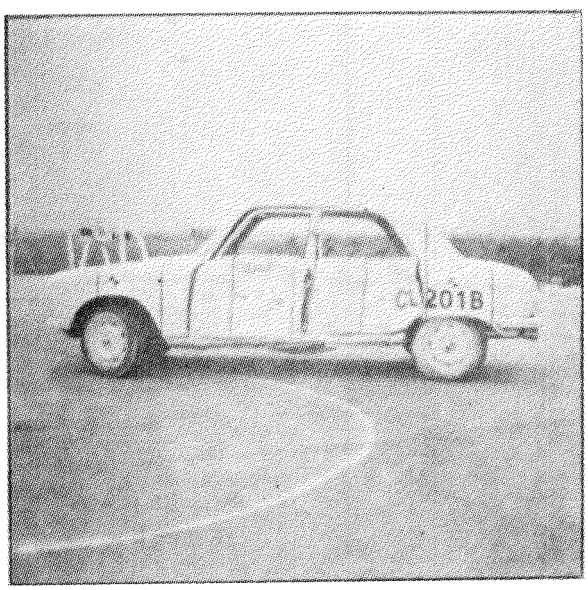
ANGLE : 90°

SPEED : 41 kph FOR EACH VEHICLE



STATIC DEFORMATIONS





Accelerations

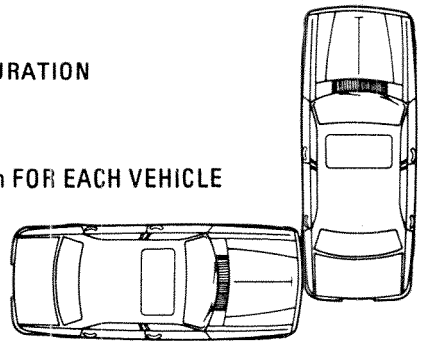
Vehicle	Dummy	
Duration – 70 ms	Head	Chest
γx – 18.5 g	γx – 20 g	γx – 13 g
γy – 12 g	γy – 80 g	γy – 57 g
γz – 16 g	γz – 40 g	γz – 18 g
$\frac{\gamma y}{\gamma x}$ – 2.5	SI – 601	γ max – 61 g SI – 196

CRASH n° 4

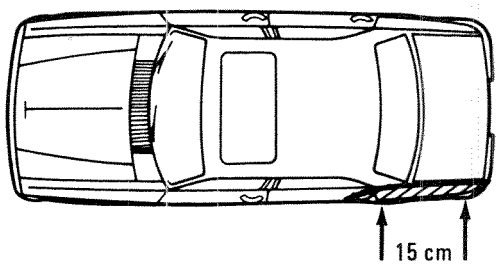
TEST CONFIGURATION

ANGLE : 90°

SPEED : 30 kph FOR EACH VEHICLE



STATIC DEFORMATIONS



Vehicle	Dummy	
Duration – 65 ms	Head	Chest
γx – 5 g	γy – 8.0 g	γy – 25 g
γy – 13.3 g	γz – 9.2 g	γz – 29 g
γz – 4.4 g		
$\frac{\gamma y}{\gamma x}$ – 2.69		

THE DEVELOPMENT OF THE OPAT DUMMY

PETER WARNER
David Ogle Limited

THE NEED FOR NEW DUMMY SPECIFICATION

As part of UK research program to develop improved safety for vehicle occupants, a series of performance requirements have been drawn up, both for primary safety or injury reducing features. For the latter requirement it soon became apparent that it would be necessary to specify closely the dummy to be used in the various impact tests called up and also that no current dummy was suitable for such an application. At this time there were only two specifications for anthropomorphic dummies in existence, the SAE J963 and the DOT purchase specification for compliance test dummies.

So that there would be a useful exchange of data, the UK Government decided that they would take as a base for their own specification both these others and expand upon them and elaborate where necessary.

DESIGN CRITERION

The criterion on which these additions and modifications to the existing specifications were based incorporated the main points of the USA purchase specification, but with certain additions and modifications in terms of form, performance, and construction and test procedures.

The modified specification was based upon the surmise that dummies should, as far as possible, be:

- humanlike in their behavior,
- able to produce repeatable results under similar test conditions,
- robust and suitable for repeat testing,
- economical in use and supply,
- built to comply with a performance specification,
- easy to adjust and set up for testing,
- calibrated against known real life accident data.

As a desirable addition to these criteria it was requested that the dummy should be able to stand upright for pedestrian work.

As a result of consideration of the implications of the above requirements, it was decided that it would be necessary to produce a design standard for the dummy in addition to the performance standard. This is in accord with current international standards practice where, although performance standards are generally favored for safety features, the actual design of test devices used to assess the performance is usually specified.

The design specification may take the form of specifying precise ranges of motion and preferred construction of the structure. For instance, it may say that the forearm shall be 9.9 inches ±.2 inches between elbow and wrist pivots and made of single round section tube, and that the radius of the elbow joint shall be a minimum of .75 inches and a maximum 1.125 and that the hand shall rotate on the axis of the forearm. But actual joint structure and configuration may be unspecified, provided that it meets all other aspects of the standard.

Each dummy used for testing will be calibrated and certified as conforming to this specification.

This type of specification will allow users a choice of supplier and will allow dummy manufacturers to be able to further develop and refine dummies within the scope of the standard specification. This specification will, of course, be updated from time to time as new information becomes available.

DEVELOPMENT PROGRAM

The Head

The prime objective with regard to the development of the head was repeatability. With this in mind a featureless head form, as that shown in Figure 1, has been adopted.

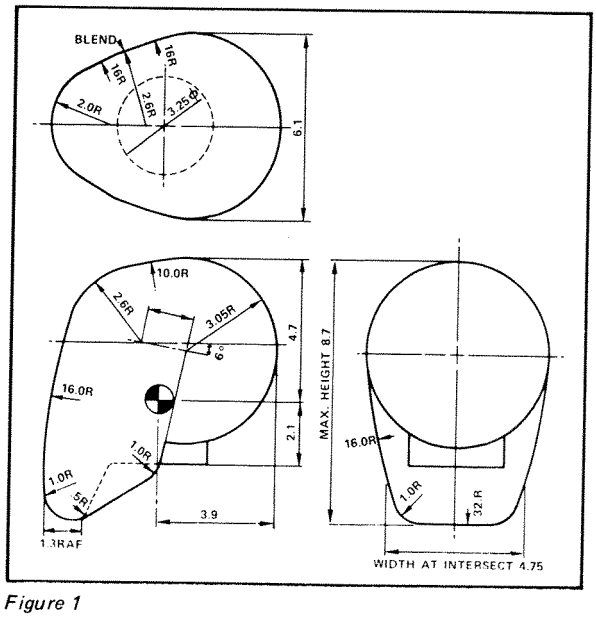


Figure 1

This head form was selected because its regular geometric form, while adequately representing the human head form, was not so sensitive to exact orientation to impacts, as current dummies which

reproduce human facial features. Such features, apart from varying from individual to individual, render the test device performance very susceptible to setting up, and may make repeats of tests virtually impossible. The skull construction is essentially a 5mm thick high grade aluminum shell which is reinforced at its base for the neck attachment, the rear of the skull is removable to obtain access for neck fitting, and the mounting of instrumentation. The skull design creates a light stiff structure but at the same time reduces resonance. This light skull is covered with an integrally moulded vinyl skin (this to help control resonance) of uniform thickness which has been specially blended to obtain the desired test results on the two specified drop tests of 10 and 35 inches.

The frontal form of the head form approximates with the SAE windscreen test block so that there is a closer degree of compatibility with screen-only tests.

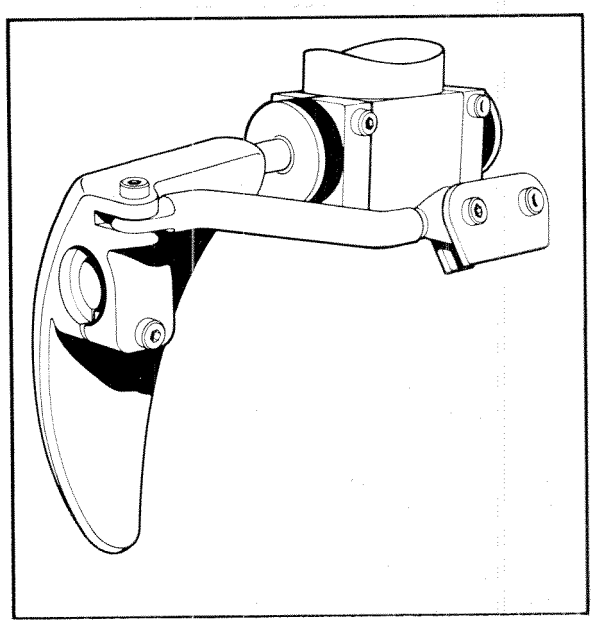


Figure 2

The Neck

The neck is of an entirely conventional tubular form fitted with a steel cable through its center, with a special mix of rubber which, to date, has been found to give the most lifelike and repeatable results of the alternative designs that have been tried. There is at present much research being carried out in Britain into neck performance and as new data becomes available it will be incorporated into the design.

The head and neck were tested as a unit on the end of a rigid pendulum, the mid-sagital plane being coincident with that of the pendulum.

The Shoulders

The shoulders and clavicle are, as far as possible, accurate representations of their human counterparts. The shoulder blades are represented so that their mass and that of the arms can be transmitted to the thorax in a necessarily representative way (ref MIRA Bulletin No 5 September 1970). The clavicles are represented by 12mm diameter steel bars formed to represent human clavicles, to give correct chest loads and shoulder belt retention. It can be seen from Figure 2 that the metal to metal sliding has been eliminated, and that a more definite control of motion is achieved by the use of representation of the trapezius muscle area. Motion of the component is controlled by a rubber bump stop. The shoulder pivot is of similar construction to all those throughout the arm and consists of a split threaded plastic bush which retains the turned steel pivot. Friction is controlled by tightening a pinch bolt. This bolt also controls the range of motion of the joint Figure 3. It can be seen that this type of joint eliminates metal to metal contact (a major source of spurious acceleration signals), is easy to adjust and eliminates end and side float at the joint (a problem which is evident in similar, less sophisticated versions of these joints found in Hybrid II dummies), and is extremely robust in operation as well as reliable and easy to set to the prerequisite 'G' level.

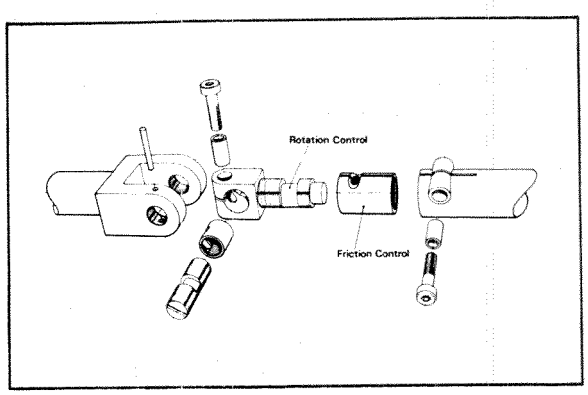


Figure 3

The Chest

The chest unit is based upon the design principles outlined in the MIRA Bulletin No. 5 September 1970, with the performance matched to the latest cadaver data. The basic design features are:

- The form of the human rib cage is followed as closely as possible so that when the cage is struck by an external force the locus of

movement of the front wall is as close to human as possible. The upsweep from front to back on each rib form is 10°. It is also surmised that by following close to the human form that response to impacts other than those directly frontal may approximate to those found in humans.

- The static stiffness of the ribs is matched very closely to that found in cadavers. The ribs are located at the spine in such a manner as to allow the rib cage as a whole to hinge forwards and downwards in a humanlike manner. The static stiffness of the chest was measured by placing the dummy with its back against a flat rigid surface and deflecting the front of the chest in regular increments of load, both on the center line and off center at the front. A maximum deflection of 3.5 inches was achieved, the average stiffness is approximately 170 lbf/in.
- To enable the effective stiffness of the chest to increase during impact, simulations of the lungs and internal organs are allowed to float freely in close proximity to the wall of the chest. Additional dynamic stiffening comes from the effects of the mass of the shoulder blades moving forwards under impact and supporting the upper rear part of the rib cage. See Figure 4.

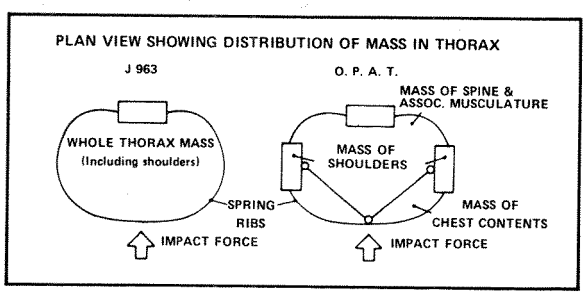


Figure 4

- So that the performance of the chest could be compared with cadaver data, a similar test to that used for cadavers was used.

  The dummy was sat upright with the arms extended horizontally, the chest was then impacted with a 6 inch diameter probe of 51.5 lbs mass at velocities of 14 and 22 ft/sec. Chest accelerations and deflections were measured.

  A series of 30 mph sled tests were also carried out. Early tests showed that high transitional peaks occurred at the onset of loading, these were more closely matched to cadaver data by the fitting of a plastic sternum. This feature gave an added by-product advantage in that with the original all steel sternum, rib fracture at the sternum was at an unacceptably high level of occurrence for a repeatable test device, and the fitting of the plastic sternum reduced the high load concentration and consequent failures at this point, at the same time reducing overall noise levels.

  A further spike has appeared on traces later during the forward loading phase. This was found to be due to the bottoming of the top sternum on the thoracic spine. This has been reduced by the fitting of a large die rubber cushion above and ahead of the thorax accelerometer box.

  It is thought that the overall slightly higher loads on the force time history may be due to higher weight of the dummy of 164 lbs as to the average 139 lbs of the cadavers.
- Originally the lead polyether lung pack deformed under constant usage, in consequence it was necessary to develop a specially formulated plastic lung which gives similar results to those of the lead polyether pack without noticeable deterioration over a series of at least 50 chest impacts.

The Abdomen

The abdominal sac is manufactured from the same material as the lungs and then tested by applying a load through a 6 inch diameter x 18 inch long cylinder positioned 26.5 inches from the top of head perpendicular to the spine at deflection increments of 0.1 inches so that it can be compared with the American Request For Purchase Specification.

The Lumbar Spine

The human lumbar spine is an area which can be compared superficially at least with the neck structure in terms of construction and action. The dummy spine has evolved therefore, as an adaptation of the neck structure but with stiffer rubber compounds. It has been designed to the static performance requirements of the American Purchase Specification. The main advantage in use is that the lumbar spine requires no setting between tests, and it helps to isolate spurious noise created by the lower torso from the chest accelerometers.

The Pelvis

The form of the pelvis conforms with FMVSS 208 and has been adapted to suit a more robust

overcenterball, hip joint representation differs from earlier Ogle dummies in construction by a pinch mechanism operating on the external diameter of a ground ball joint which has proved simple to adjust and maintain, and robust in operation. The femur is attached via a 'T' head bolt system which allows the ball to be precision ground and a close tolerance maintained on the fit of the femur to the ball.

The Femur/Knee

The femur is shaped in a lifelike manner as specified below. The ballast weight which has to be carried to obtain the correct weight for this limb has been specially shaped so that if the femur strikes and slides under a dashboard or along the underside of a steering wheel, the 'bone' structure does not offer unnaturally sharp corners which may affect the movement of the dummy.

Provision has been allowed for the mounting of a standard load cell 4.25 inches back from the pivot of the knee joint. (To comply with FMVSS 208) see Figure 5.

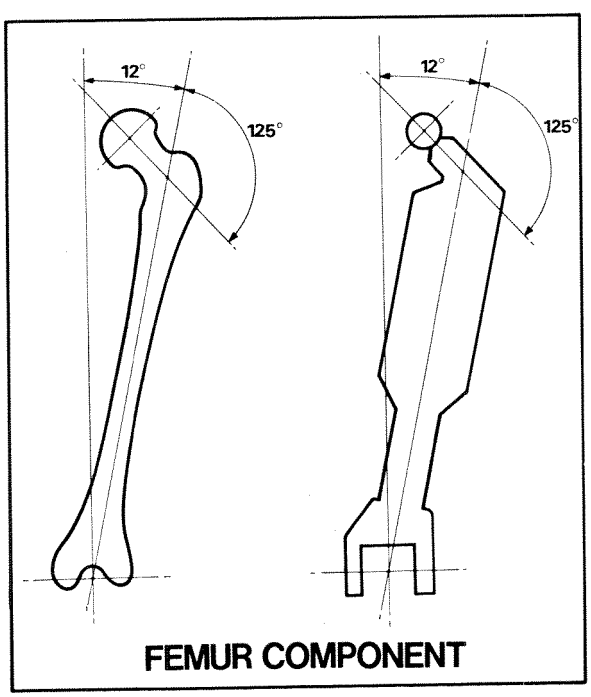


Figure 5

The lower leg incorporates at its upper end a steel hemisphere of 2¾ inches diameter which represents the knee cap. This was thought to be an important innovation as again minor variations in the angle at which a rectilinear type knee form strikes a dash or any other object within a car can affect the repeatability of the reading at this joint. See Figure 6.

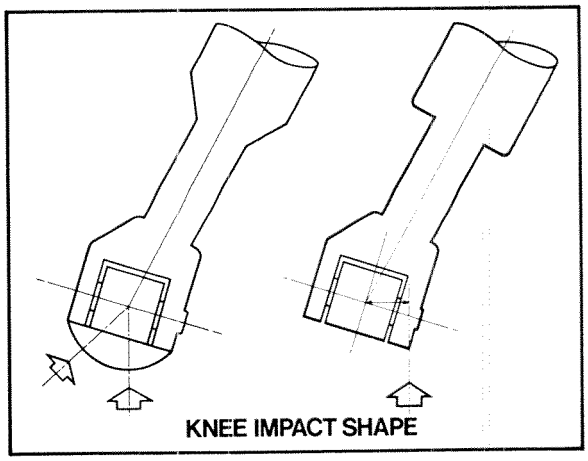


Figure 6

Flesh System

A vinyl covered foam flesh system which, with the exception of the chest and knee and pelvis areas is permanently bonded to the structure of the dummy.

All relevant test results concerned with this development are shown in MIRA paper presented at this 5th ESV Conference.

General Constructional Details

The whole skeletal structure follows as far as possible a human skeleton; however, the major parts of body weight are concentrated in the dummy 'bone' structure rather than in the flesh as on humans. The main reasons for this are:

- robustness of skeletal components
- repeatability of mass distribution
- ease of manufacture

In general the skeleton is of steel, notable exceptions are the skull and shoulder blades. The prime reasons for the selection of steel as the major structural material are:

- ease of supply
- ease of weldability
- robustness
- quality control

However, whenever practical, plastic or rubber bump stops and bearing surfaces are used to help eliminate noise from the structure.

SUMMARY

The coefficients of variation for the dummy have been calculated and are compared with Hybrid II dummies tested at Calspan. It should be noted that the larger variability on the chest results of the OPAT dummy can be attributed to noise generated by the sternum striking the thoracic spine. This area has

since been redesigned to eliminate this source of noise.

In the UK, the ESV program has shown a need for a fully specified, repeatable, robust and humanlike test dummy. At the same time as producing a dummy specification, the Transport and Road Research Laboratory are keeping abreast of other dummy research work so that the final specification will be suitable for testing to existing and future Occupant Protection standards.

The development work to date has proved that it is possible to attain practical standards of humanlike performance combined with relative simplicity, robustness and economy of manufacture. In particular, the concept of producing a more simple and robust dummy which is less sensitive to minor variation in setting up procedures and test parameters have produced benefits in terms of repeatability of test results.

When the final specification has been drawn up it will be available from Transport and Road Research Laboratory, Department of the Environment, Crowthorne, Berkshire, RG11 6AU, England.

All test data is quoted from Performance Measurements on the OPAT Dummy January 1974 prepared by the Motor Industry Research Association, Watling Street, Nuneaton, Warwickshire, VC10,OTU, England.

PERFORMANCE MEASUREMENTS ON THE OPAT DUMMY

C.M. HASLEGRAVE
and
M. A. MACAULAY
The Motor Industry Research Association

INTRODUCTION

A series of performance measurements were made at MIRA on four components of the OPAT dummy to assess the compliance with the "Specification for Occupant Protection Assessment Test Dummy"[1] which was the basis for the design of this dummy. This specification is based on the NHTSA. Request for Proposal Specification of July 1972, but includes various modifications in test conditions and in performance requirements. The requirements include performance tests for the head, neck, thorax and abdomen. In addition, two further series of tests were carried out which were thought to be relevant even though they are not included in the specification. The thorax was tested by impact with a 52 lb weight under similar conditions to those used in published cadaver tests and an assessment was made of the repeatability of head and chest acceleration measurements, which would be expected from use of the dummy in crash tests. For this latter series several sled tests were carried out with the dummy restrained by lap and diagonal belts.

COMPONENT TESTS

Head Response

The group of tests of the dummy components began with drop tests on the head. The head was suspended above a steel plate and allowed to fall freely from heights of 254 mm (10.0 inch) and 890mm (35.0 inch). The acceleration curves obtained are shown in Figures 1 and 2. At both levels, the responses were substantially unimodal as required by the specification and the peak resultant accelerations were 235 g and 539 g against required values of not greater than 250 g and 700 g.

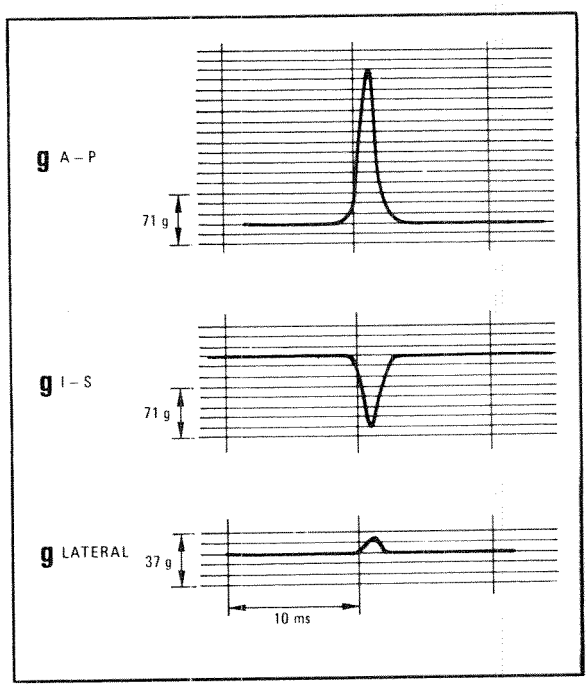


Figure 1

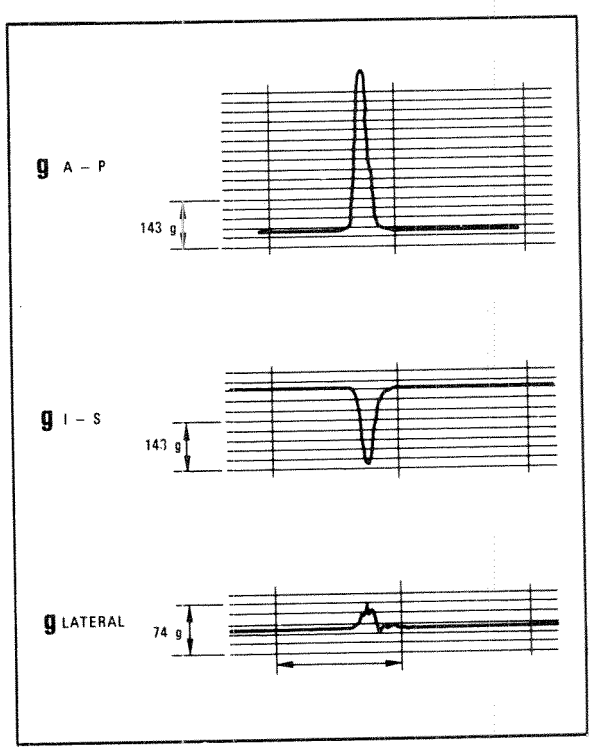


Figure 2

Neck Response

The performance of the neck was tested by mounting the head and neck assembly on a rigid pendulum. The pendulum was allowed to fall from the chosen height to give impact speed of 7.16 ms

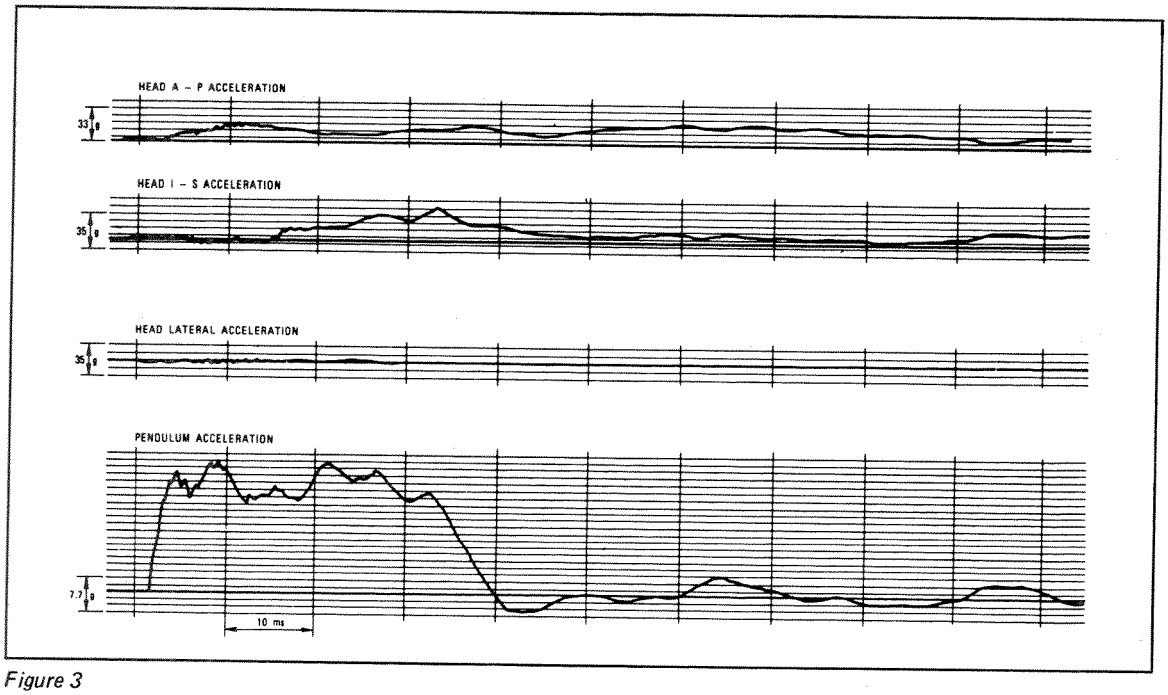


Figure 3

then flexed freely and measurements were made of the angular rotation of the head and the chordal displacement of its center of gravity. These were obtained from high speed film records of the test. In addition a triaxial accelerometer was mounted in the head. The acceleration response curves are given in Figure 3. The peak resultant acceleration measured was 34.4 g against the required value of not exceeding 40 g and the maximum angles of rotation were 88° in flexion and 56° in hyperextension against the required values of not exceeding 90° and 60°.

Chest Response

The deflection characteristics of the chest were measured with the variable speed impact. The probe, which was constrained to move longitudinally, was fired at a known impact speed at the dummy's chest and the resulting deflection of the rib cage was measured with a potentiometer attached to the sternum. The load developed in the chest was recorded simultaneously with the deflection. The same system was used for static measurements, but in this case the probe was wound forward manually to load the chest.

This equipment was used first with the dummy positioned with its back against a vertical flat rigid surface to obtain the static loading characteristics at three points on the chest. The load/deflection curves are given in Figure 4. The deflection is approximately linear and is 3 x 10^4 N/m (170 lbf/in) against the required value of not exceeding 5.25 x 10^4 N/m (300 lbf/in).

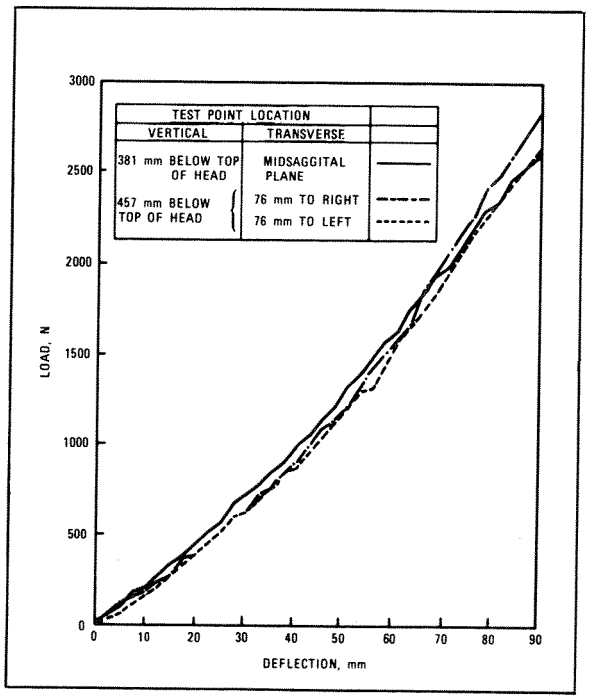


Figure 4

For measurement of dynamic load/deflection curves the dummy was seated in an upright position unsupported except for the arms which were extended horizontally and rested upon a support. Two impact speeds of 4.27 m/s (14.0 ft/s) and 6.71 m/s (22.0 ft/s) were used to test the chest and the load/deflection characteristics are given in Figures 5(a) and (b). The maximum forces were 3536 N (795 lbf) and 5431 N (1221 lbf) against not exceeding 4 x 10^3

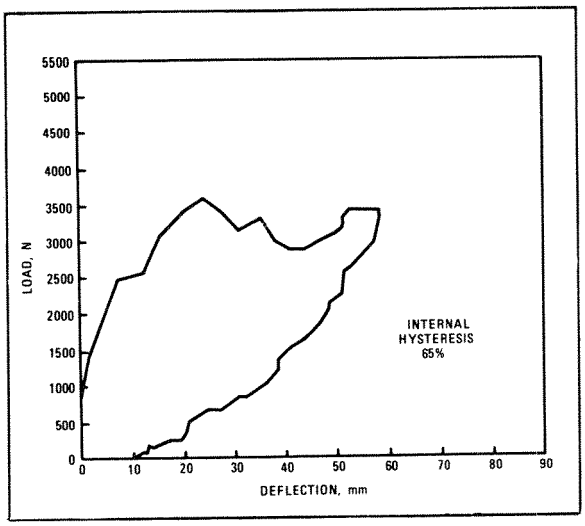


Figure 5(a)

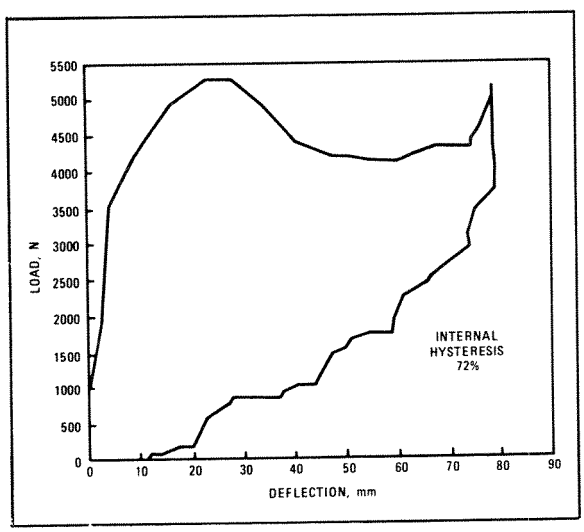


Figure 5(b)

N (900 lbf) and 6 x 10^3 N (1350 lbf). The maximum deflections were 58 mm (2.3 in) and 79 mm (3.1 in) against 38 to 64 mm (1.5 to 2.5 in) and 64 to 89 mm (2.5 to 3.5 in). The values of internal hysteresis were 65% and 72% against not less than 50%.

Abdomen Response

The force developed on compression of the abdomen was measured with the dummy supine on a flat rigid surface. The load was applied vertically through a rigid half cylinder and the characteristic measured is given in Figure 6, where it is compared with the characteristic required in the specification.

SIMULATION OF HUMAN DATA

The thorax had been designed to simulate available impact data on cadavers. Tests were therefore carried

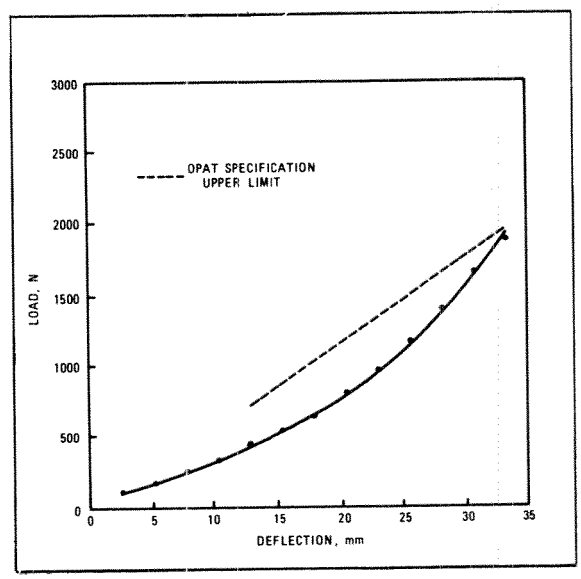


Figure 6

out which duplicated as closely as possible those described by Kroell et al[2]. In their paper they describe the results of five tests carried out on male cadavers at impact speeds of approximately 6.7 m/s (15 mph) with an impactor of weight 23.59 kg (52.0 lb). It was possible to repeat fairly closely their test procedures using the MIRA impactor. Figure 7 shows the force/time curves obtained together with the cadaver results. The heavy solid line represents the results for the OPAT dummy. As can be seen, the OPAT dummy represents very closely the general form of the curves. The overall loads are higher than those observed for the cadavers. This may well be due to the greater weight of the dummy, which is 74.8 kg (165 lb), when compared with the low weights of the cadavers, which averaged 63.0 kg (139 lb). Kroell et al also measured the total chest deflections in the cadaver tests from high speed film records. The peak deflections ranged from 74 to 91 mm (2.4 to 3.6 in), which may be compared with the peak skeletal deflection of 79 mm (3.1 in) measured in the OPAT dummy. Even allowing for the additional skin compression, this value falls well within the range for cadavers.

Dynamic Performance in Sled Tests

A series of nine sled trials were carried out on the dummy to assess the repeatability of the head and chest accelerations in a crash test situation. The dummy was tightly restrained by separate lap and diagonal straps, which were replaced after each test. The diagonal strap was used to restrain either the right or the left shoulder in alternate tests. The sled was decelerated by crushing of aluminum honeycomb, with a typical deceleration trace as shown in Figure 8.

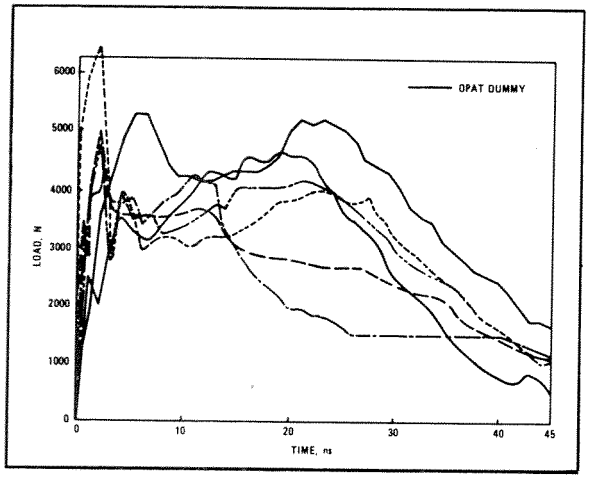


Figure 7

Measurements were made of accelerations in three axes in the dummy's head and two axes in the chest. A typical response trace is shown in Figure 8. The performance of the dummy was assessed in terms of the peak accelerations in head and chest, the Head Injury Criterion, Head Severity Index and Chest Severity Index, the results being shown in Table 1. Although the dummy was alternately restrained by right or left shoulders, no statistically significant difference was found between the two groups and the results have been pooled to give the best estimates for the population means and standard deviations. When considering the levels of the injury indices and accelerations, it should be noted that the sled deceleration pulses were of the order of 40 g, which is approximately 80% higher than commonly encountered in 13.5 m/s (30 mph) barrier impact tests because the durability of the dummies was also being studied.

The coefficient of variation was also calculated for each performance measure in order to give an indication of repeatability and defined as the ratio of the standard deviation to the mean value expressed in percentage terms. In addition to these, Table 1 gives the coefficients of variation which have been quoted for the GM Hybrid II and modified Alderson dummies[3]. If the three dummies are compared, it may be seen that in Head Injury Criterion and Head Severity Index, the OPAT dummy gives a percentage scatter one quarter or one third that of the GM Hybrid II. For Chest Severity Index, the OPAT dummy had a coefficient of variation up to 50% greater than the GM Hybrid II. The peak resultant acceleration was also less repeatable in the OPAT dummy chest.

CONCLUSIONS

Measurements of performance of the OPAT dummy head, neck, thorax and abdomen showed that

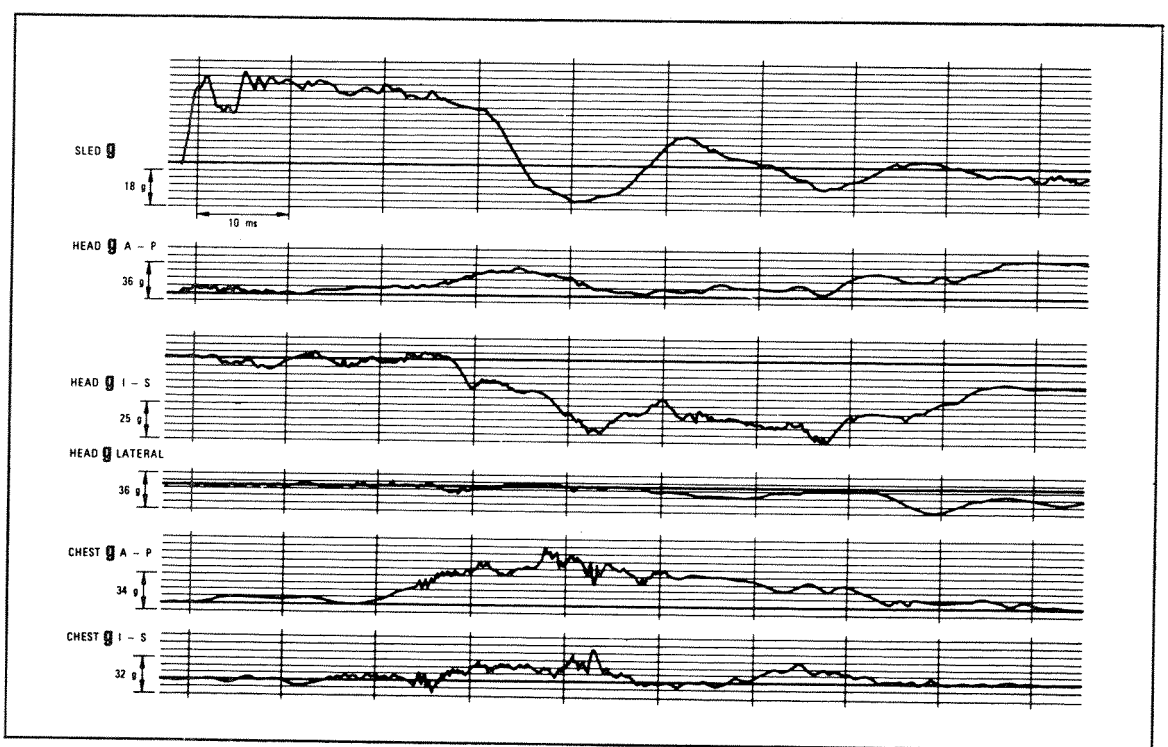


Figure 8

Table 1.
Results of Dummy Sled Tests

		HIC	HSI	Peak Head Resultant Acceleration (g)	Chest SI	Peak Chest Resultant Acceleration (g)
OPAT	Mean ($\bar{x}$)	762.3	892.7	58.28	328.2	53.70
	Standard Deviation (σ)	41.0	67.0	6.6	62.0	10.0
	Coefficient of Variation $\left(\frac{100\,\hat{\sigma}}{x}\right)$	5.4%	7.5%	11.3%	18.9%	18.7%
GM Hybrid II -36*		19.3%	22.3%	12.0%	13.2%	4.3%
GM Hybrid II -37*		21.2%	21.4%	6.2%	16.7%	6.7%
Alderson VIP50a with modified chest and neck -02*		10.0%	19.5%	7.5%	12.6%	25.9%

*Reference: J.S. Miller. Performance Evaluation of the General Motors Hybrid II Anthropomorphic Test Dummy. CAL Report No. ZS-5272-V-1.

Table 2.
Comparison of Dummy Specifications

	Component	NHTSA Purchase Specification	OPAT	FMVSS 208 Part 572	Comments
Head	254 mm (10 in) drop 890 mm (35 in) drop	250 g max 790 g max	250 g max 700 g max	210–260 g –	OPAT dummy tested complies with Part 572 specification.
Neck	Impact Speed	7.92 m/s (26 ft/s)	7.16 m/s (23.5 ft/s)	7.16 m/s (23.5 ft/s)	OPAT dummy tested does not comply with Part 572 specification. Neck is much less stiff.
	Head Resultant Acceleration	30 g max	40 g max	26 g max	
	Peak Flexion	70°	90°	63°-73°	
	Peak Hyperextension	60°	60°	–	
Thorax	Static Stiffness	<10.5 X 10^4 N/m (600 lbf/in)	<5.3 X 10^4 N/m (300 lbf/in)	–	OPAT dummy tested complies with Part 572 specification except that the deflections are much greater.
	Impact Speed	4.27 m/s (14 ft/s) 6.71 m/s (22 ft/s)	4.27 m/s (14 ft/s) 6.71 m/s (22 ft/s)	4.27 m/s (14 ft/s) 6.71 m/s (22 ft/s)	
	Peak Load	<4000 N (900 lbf) <7100 N (1600 lbf)	<4000 N (900 lbf) <6000 N (1350 lbf)	<6200 N (1400 lbf) <9300 N (2100 lbf)	
	Peak Deflection	38-64 mm (1.5-2.5 in) 64-76 mm (2.5-3.0 in)	38-64 mm (1.5-2.5 in) 64-89 mm (2.5-3.5 in)	<25 mm (1.0 in) <41 mm (1.6 in)	
	Internal Hysteresis	>50%	>50%	>50%	
Abdomen		710 N (160 lbf) at 13 mm (0.5 in) <156 N (35 lbf) for each increment of 2.5 mm (0.1 in)	710 N (160 lbf) at 13 mm (0.5 in) <156 N (35 lbf) for each increment of 2.5 mm (0.1 in)	5250-7180 N (30-41 lbf/in) up to 25 mm (1.0 in) 9280-11730 N (53-67 lbf/in) up to 33 mm (1.3 in)	OPAT dummy tested does not comply with 572 specification. Abdomen is much stiffer.

these were comfortably within the specification produced by TRRL. As this specification differs from that issued by the NHTSA in July 1972 and the current U.S. Federal Standard FMVSS 208 Part 572 also differs from the NHTSA specification, the situation is summarized in Table 2.

The head of the OPAT complies with all three specifications. There are differences in the requirements for the neck and abdomen which are beyond the scope of this paper. The OPAT dummy achieves a very close simulation of available cadaver data on chest response, as can be seen from the figures given earlier, and consequently it deflects more than the Part 572 specification allows. This appears to be a major shortcoming in the FMVSS 208 specification.

In considering the problem of repeatability of measurements, a compliance test device giving a scatter in the region of 5% would appear to be likely to be acceptable. For comparison purposes, the scatter has been defined by the coefficient of variation which is the standard deviation divided by the mean value. The scatter in the measures of head injury for the OPAT dummy is 5% to 7.5%, which is better than the other dummies listed in Table 1 and is probably good enough for a test device. The coefficient of variation for the Chest Severity Index of the OPAT dummy is about 19%, which does not appear satisfactory for a test device. The Chest Severity Indices for the two G.M. Hybrid II dummies give coefficients of variation of 13% and 17%, but these are still not adequate for compliance testing. Metal to metal contact in the OPAT dummy chest is one cause of the lower repeatability in the Chest Severity Index and also contributes to the high scatter in the peak chest acceleration. David Ogle Limited have modified the dummy to rectify this, but it has not yet been retested.

REFERENCES

1. Transport and Road Research Laboratory. "Specification for Occupant Protection Assessment Test Dummy." 1974.

2. Kroell, C.K., Schneider, D.C. and Nahum, A.M. "Impact tolerance and response of the human thorax." Proceedings of Fifteenth Stapp Car Crash Conference, November 1971, 84-134.

3. Miller, J.S. "Performance Evaluation of the General Motors Hybrid II Anthropomorphic Test Dummy." CAL Report No. ZS-5272-V-1.

APPENDIX
TECHNICAL DETAILS OF TESTS

HEAD DROP TEST

- Orientation of I.S. axis of head relative to horizontal: 30°
- Accelerometer: Endevco Model 2267C-750 Triaxial (±750 g)
- Frequency response: SAE channel class 1000

NECK PENDULUM IMPACT TEST

- Pendulum length: 1.834 m
- Pendulum moment of inertia about pivot: 26.3 kgm^2
- Distance of accelerometer from pivot: 1.656 m
- Impact speed: 6.83 m/s
- Typical deceleration pulse – duration over 20 g level: 31 ms
  - rise time: 2 ms
  - decay time: 6 ms
- Pendulum Accelerometer: Either type UA1 (±100 g)
- Frequency response: SAE channel class 60
- Head accelerometer: Endevco Model 2267C-750 Triaxial (± 750 g)
- Frequency response: SAE channel class 1000
- Film speed: 800 frames/sec

THORAX RESPONSE TEST

- Probe weight: 23.36 kg
- Loading face: 152 mm diameter, edge radius 13 mm
- Static deflection – loading rate: 0.25 mm/s
  - pre-load: 90 N
- Dynamic deflection – impact point: 457 mm below top of head
  - accelerometer: Bell & Howell 09384/4-202-0012
  - frequency response: SAE channel class 180

ABDOMEN RESPONSE TEST

- Loading cylinder – length: 457 mm
  - diameter: 152 mm
- Loading point: 673 mm inferior to top of head.
- Loading rate: 0.2 mm/s
- Pre-load: 45 N

SLED TEST

- Details of sled – The sled seat was a rigid construction with a 13 mm covering of rubber. The seat pan was horizontal and the seat back angle 15° to the vertical.
- Lap belt angle to horizontal: 70°
- Shoulder belt angle from shoulder: horizontal
- Impact speed 12.2 m/s
- Head accelerometer: Endevco Model 2267C-750 Triaxial (± 750 g)

- Frequency response: SAE channel class 1000
- Chest accelerometers: Bell & Howell 09384/4-202-0012 (± 250 g)
- Frequency response: SAE channel class 1000
- Sled accelerometer: Either type UA1 (± 100 g)
- Frequency response: SAE channel class 60

TOWARDS PEDESTRIAN SAFETY

V. J. JEHU
Transport and Road Research Laboratory
Department of the Environment

INTRODUCTION

On the roads in Great Britain in 1971 injuries to pedestrians amounted to 38 percent of the total road fatalities and 26 percent of all seriously injured casualties. Children (0-14 years) accounted for 22 percent of the pedestrian fatalities and 44 percent of the seriously injured pedestrians.

A detailed investigation at the Transport and Road Research Laboratory of 45 pedestrian fatalities has shown that fatal injuries most frequently derive from impact with the ground. Next most frequent is impact with the windshield. Impacts with the windshield generally do not cause fatal injuries.

Whereas it should be possible by suitable design to reduce the severity of pedestrian impacts with various parts of the vehicle, nothing can be done to make the road less unyielding. Hence it seems logical to attempt to pick up all pedestrians on the hood rather than just knock them down. Support for this approach comes from a study by McLean[1] who examined a large sample of pedestrian accidents and concluded that the low front profile of the Volkswagen "Beetle" car was less injurious to pedestrians than the high front of the Cadillac car.

TRAJECTORY STUDY

The trajectories of anthropomorphic dummies struck by passenger cars have been studied with a view to determining a front-end configuration which will alleviate pedestrain injury. Both adult and child dummies were struck by cars simulating the situation of a pedestrian walking across the road in front of the car.

The limbs of the dummies were fitted with shear pins to introduce flexibility of movement after impact, and the dummies were supported in the standing position by means of a prop which was dislodged at impact. The cars were fitted with a protective steel grille over the windshields and were manually driven at nominal speeds of 8, 16, 24 and 32 km/h. Tests at higher speeds were precluded by the violence of the impacts. Each event was recorded on film.

Initial tests with large, medium and small cars showed that the leading edge of the hood must be below the height of the center of gravity of the pedestrian dummy in order to prevent it from being suddenly accelerated and thrown to the ground ahead of the vehicle.

Provided that the leading edge of the hood was sufficiently low, impact of the car against the legs of the pedestrian caused the torso to fold over and the head to strike the hood or windshield. For a brief period of time vehicle and pedestrian then travelled with the same forward velocity. However the pedestrian rarely remained upon the vehicle, but fell off either in front of the vehicle or to one side, striking the ground with a severity which increased with the speed of the vehicle. The worst situation arose when the vehicle brakes were not applied until after the impact had occurred; the pedestrian was then projected forward of the vehicle to strike the ground with considerable force.

POSSIBLE FRONT-END MODIFICATIONS

The medium-size car which picked up the 50th percentile adult dummy was modified by the addition to the hood of various high friction surfaces in an attempt to retain the pedestrian on the vehicle after impact. Since no noticeable improvement over the unmodified car was observed it was concluded that a positive catcher device which altered the front-end profile of the car was necessary to prevent the pedestrian being thrown on to the road.

The effect of rounding the front of the vehicle was then studied. With a sufficiently rounded front-end and a low enough leading edge the pedestrian was projected at an angle across the hood for off-center impacts rather than straight back towards the windshield. Impact of the head with the windshield was either obviated or reduced in severity dependent upon the speed of impact and the length of the hood. It was considered that the angled trajectory induced by the rounded front-end also reduced the severity of head impacts with the hood of the vehicle.

The spatial relationship of the leading edge of the hood and the front bumper was also investigated. In order to pick up the 6-year old child dummy the leading edge of the hood must be considerably lower than the height which is typical of current large volume production cars. Bumper height should also be lower than the current practice in order that the first contact with the legs of an adult is below rather than at the knee. The bumper should also be below the pelvis of a 6-year old child. Some extension of the bumper forward of the vertical plane through the leading edge of the hood was beneficial in that the pedestrian was thereby rolled less violently about the leading edge.

Figure 1. Modified Front End of Car and Pedestrian Catcher Device

MODIFIED CAR

The modifications described above were built into a medium-size car together with a pneumatically operated pedestrian catcher-device which consisted of a metal bar shaped to conform to the periphery of the vehicle. In its rest position the bar itself forms the leading edge of the hood. Impact loading from a pedestrian raises the bar to a height sufficient to prevent the pedestrian from sliding off the vehicle (Plate 1). In practice, the front bumper and the catcher bar would be covered with energy absorbing material to reduce pedestrian injury.

Details of the modified front-end are as follows:

Overall height of front bumper	375 mm
Overall height of leading edge/catcher bar	625 mm
Horizontal distance between bumper and catcher bar	100 mm
Extension to existing hood at center line (reducing to zero at the front corners)	300 mm
Lift of catcher bar	300 mm

This vehicle successfully picked up and retained the adult dummy at impact speeds of 8, 16 and 24 km/h, and the child dummy at impact speeds of 16 and 24 km/h. At the low impact velocity of 8 km/h pick up of the child dummy was unreliable.

FURTHER DEVELOPMENTS

On most current vehicles the junction of the scuttle and the base of the windshield has considerable stiffness. It could be made more flexible by continuing the shield well below the scuttle leaving a gap between the two. This treatment would also have the advantage that the pedestrian was prevented from striking the lower edge of the screen thereby reducing facial injuries from any residual fragments of a broken windshield.

In order to reduce occupant injury and structural damage in vehicle-to-vehicle collisions the energy absorbing bumper must be positioned further forward of the leading edge of the hood than the 100 mm of the modified car. Extending the front bumper in this way has the effect that the center of gravity of the 6-year old child falls just below the leading edge at impact so that the child is no longer picked up on the hood.

Thus to cater to the child the height of the leading edge needs to be reduced still further, or eliminated altogether as in the front end profile of the Volkswagen Beetle car. The child would then probably be picked up for any practicable forward extension of the front bumper, but at the expense of a more violent trajectory of the adult. A major difficulty created by this approach would be finding sufficient height both to house the engine at the front of the car, and to position the headlamps high enough to provide an acceptable lower beam. Redesign of the catcher device is being pursued in an attempt to pick up and retain both child and adult dummies without the penalty of a drastic reduction in hood height, the catcher bar being located on top of the extended bumper in its rest position.

CONCLUSION

The influence of changes in the front-end shape of cars on the trajectories of adult and child pedestrians has been demonstrated. No drastic changes are needed to pick up an adult on the hood of a front-engined car, but if the young child pedestrian is not simply to be knocked down by the car, more radical changes are needed.

One way of retaining the pedestrian on the hood of the car has been demonstrated. It is possible that redesign of the catcher device may enable both adult and child to be picked up and retained on the hood without radical changes in the front-end configuration of the car.

ACKNOWLEDGEMENT

The work described in this paper forms part of the program of the Transport and Road Research Laboratory and the paper is published by permission of the Director.

REFERENCE

1. A. J. McLEAN, "Car Shape and Pedestrian Injury," Proceedings of Symposium on Road Safety, Expert Group of Department of Shipping and Transport, Canberra, Commonwealth of Australia, March 1972.

A REVIEW OF DEVELOPMENT OF PASSIVE RESTRAINT SYSTEMS

Auto Restraint Systems, Ltd.

INTRODUCTION

This is a review of the principle aspects of work being carried out, in conjunction with the Transport and Road Research Laboratory, on the subject of vehicle occupant safety. Reports have been presented at the previous ESV International Technical Conferences dealing with current developments and this paper covers, mainly, the work program from early 1973 to the present date.

This resume of progress covers activity on three development contracts with the Department of the Environment as:

- Development of Passive Seat Belts.
- Passive Seat Belt System for TRRL/British Leyland Safety Car.
- Research into crash deployed/activated restraint systems.

IMPACT TESTING WITH SLEDS, BODY SHELLS AND VEHICLES

When considering occupant protection, a passive seat belt is essentially the same as a conventional seat belt. The detailed layout and position of anchorages may differ slightly, but the same order of protection will be provided to the vehicle occupant in an energency. Work has been devoted to further improvements in the already proven effectiveness of the seat belts in accident situations. The improved performance is being monitored by means of standard barrier impact tests.

Some aspects of passive seat belt systems dictate such requirements as anchorage positions and webbing lengths. Consequently performance improvements have been achieved mainly by the specific type of webbing used and the incorporation of load limiting, energy absorbing devices.

Seat belt webbings in general use have extensions in the range of 18% to 26% at 2,250 lbf. These extensions combined with any slack in the belt system can lead to insufficiently controlled occupant movement and possible injurious contact with the vehicle interior. In addition to this forward movement, the energy stored due to the elasticity of the webbing results in a degree of occupant rebound with its attendant risk of injury. Webbing having extensions in the range of 4% to 6% have been substituted reducing the excursion of the occupant. They also link the occupant to the vehicle more rigidly gaining advantage from any vehicle ride down that may be present.

The inclusion of an energy absorber at some position in the belt system enables the peak load in that belt to be limited and also provides controlled belt extension, thereby ensuring optimum occupant movement within the vehicle. The peak belt load can be chosen to eliminate belt induced injuries, and the occupant movement controlled to allow deceleration within the confines of the particular vehicle. In practice the modifications which were made to the belt systems and proved by impact tests, related to variation of the position and yield load of included energy absorbers.

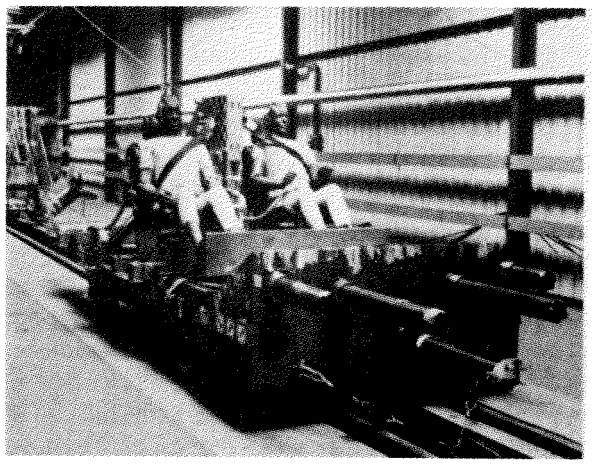


Figure 1. Crash Test Sled

The impact test work took the form of three basic types: open sleds, bodyshells and vehicles. The sled and bodyshell deceleration was made to simulate that of the vehicle by means of collapsible crush tubes. All tests used 50th percentile dummies, fully instrumented, and high speed film to provide complete data.

Tests with an open structure mounted on a wheeled trolley were performed at the early stages of development to provide basic information. A rigid structure (illustrated by photograph Figure 1) supporting two fixed seats and seat belt anchorages, was employed. Forward movement of the occupants was not restricted by any vehicle structure, and dummy movement could be observed without obstruction.

The next stage of work was to use vehicle bodyshells, mounted on a wheeled trolley as shown in Figure 2. This provided a realistic environment in terms of anchorage position and strength, the

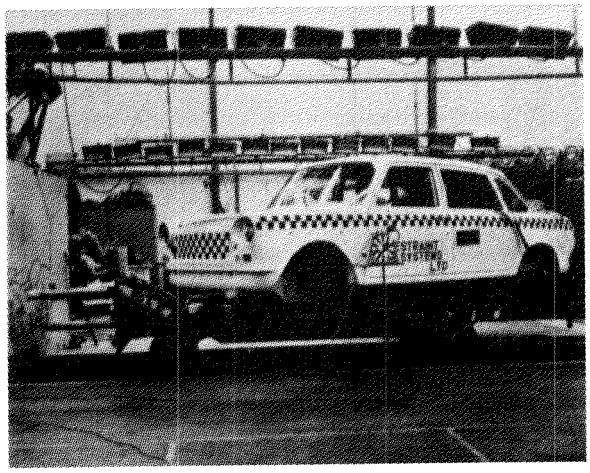


Figure 2. Crash Test Bodyshell Rig

Figure 3. Vehicle Crash Test

influence of vehicle seating, and the limitations imposed on occupant movement by steering wheel, facia, etc.

The final stage was to impact test an actual vehicle. This provided the further factors of vehicle deformation and intrusion combined with movement of anchorages and seats.

The majority of work performed has been of the sled and bodyshell type, but in addition some vehicle tests have been carried out. One such vehicle test is illustrated in Figure 3.

Most of the bodyshell test were performed at 30 mph perpendicular to the barrier. However, one requirement was to extablish the performance of the restraint system in other modes and so the bodyshell was mounted to the wheeled trolley via an adjustable platform. This enabled side and oblique impacts to be simulated, tests normally associated with vehicles. Deceleration was achieved by collapsible tubes in the normal manner but varied to simulate the vehicle behavior in the required mode.

If a restraint system is modified to give improved performance and this performance is gauged in a particular set of conditions it is possible that the modification made may have a detrimental effect in other circumstances. It was decided therefore that after establishing the belt system giving optimum performance in a 30 mph barrier impact, this identical belt system should be subjected to impact testing in different modes. The results achieved with this common restraint system in one particular body shell are shown in Table 1 below.

These results were obtained using identical seat belts within the confines of a body shell, and they indicate that such a restraint has a high degree of overall effectiveness even with an increase in speed to 40 mph. Optimizing the performance for one particular condition has not resulted in unacceptable performance in other circumstances. In order to allow for even higher impact speeds all energy absorbers used have a fixed limit of travel which, when achieved, prevents further extension while still maintaining an integral restraint system. Thus, should the impact speed be very high with correspondingly high energy levels the occupant movement is restricted within the confines of the vehicle.

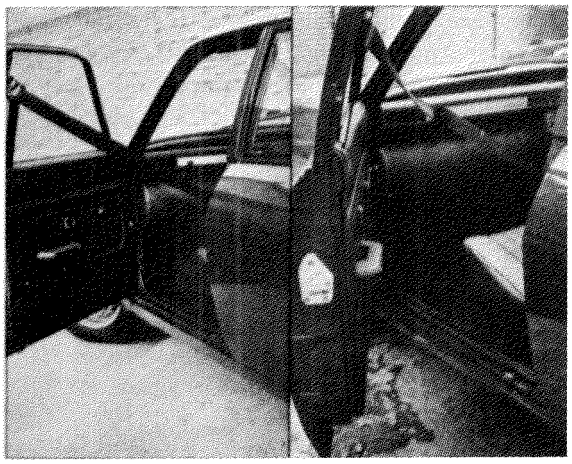
In addition a vehicle rollover test has been performed and in common with any properly adjusted lap and diagonal belt system provided very low levels of occupant deceleration. The occupant was held in his seat throughout the duration of the test.

PASSIVE SEAT BELT SYSTEMS – VEHICLE INSTALLATIONS AND EVALUATION

Development work has continued on three different systems aimed at providing viable fully passive schemes for typical British saloon cars. Two of the cars previously equipped have been crash tested and new vehicles similarly equipped have now been prepared for further demonstration, evaluation and longer term testing. It is appropriate to outline the principles and up to date details of the Passive Seat Belt Systems now installed in three cars and currently available for viewing in the exhibition area at TRRL Crowthorne.

TABLE 1

TEST MODE	30 mile/h Frontal	40 mile/h Frontal	30 mile/h 30°oblique
H.I.C. Head S.I	478	717	299
Peak Head G	53	133	39
Peak Chest G	39	57	25



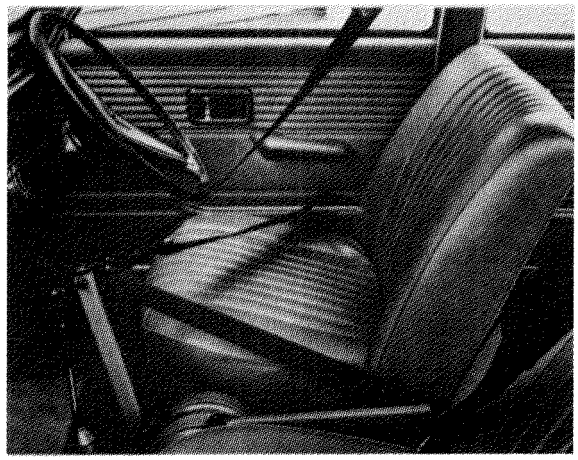


Figure 4. *Figure 5.*
Diagonal Seat Belt and Knee Restraint System

Figures 4 and 5 are photographs of an Austin Allegro on which a diagonal seat belt and knee restraint system has been installed. This scheme was originally proposed by TRRL Crowthorne and the concept is maximum user convenience and simplicity of a single diagonal seat belt, with an energy absorbing knee restraint designed to limit forward movement of the occupants' lower torso and limbs. This knee restraint takes the form of a padded, steel semi-cylinder which on impact will collapse to ensure that compressive femur loads do not exceed a prescribed maximum value.

The seat belt is anchored inboard via an emergency release buckle which could, if required, incorporate a switch to actuate a warning or stater inhibit in case of release. The seat belt reel incorporating retractor spring and vehicle sensitive inertia locking mechanism is installed in the door with the belt passing through a guide loop attached at the door mounted shoulder anchorage position. To ensure maximum restraint effectiveness and occupant comfort, the height of the door mounted guide loop is made adjustable. The attachment plate also interlocks with the 'B' pillar to provide adequate strength in the event of an impact.

Installation of the reel in the door gives addid user convenience in that the belt remains stationary across the occupant when the door is opened and closed. It means, however, that a door actuated reel locking inhibitor is required otherwise vigorous door opening would operate the vehicle sensitive inertia lock. This inhibitor is built into the reel and actuated by a small plate attached to the door sill.

Figure 5 illustrates the use of the belt parking hook on the vehicle facia. The use is optional but does provide additional convenience when leaving and entering the car. These parking hooks are connected to the parking brake lever to ensure that the belts cannot be retained while the car is in motion.

Figure 6. Lifting Arm System

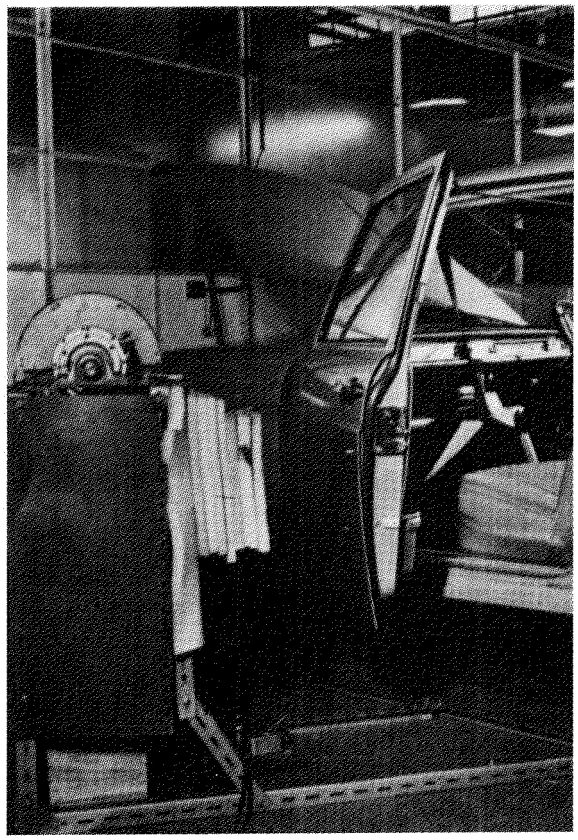


Figure 7. Endurance Test Rig

The British Leyland '1800' featured in Figures 6 and 7 has an improved installation of the motorized, centrally mounted lifting arm system. The experimental version of this equipment was originally installed in a similar vehicle which has subsequently been subjected to a frontal impact crash test.

Detailed improvements have been made on the first experimental scheme among which is modification to the slot openings, along which the arm travels, to keep it fully closed and prevent risk to fingers. The shoulder anchorage on the passenger side

incorporates both a height adjustment and an emergency release buckle. Improvements to the lifting arm mechanism have resulted in a decrease in the electric current required to operate the system, and more reliable door switches have been used.

Figure 7 illustrates the endurance test rig used in the laboratory to open and close the car door repeatedly to simulate longer term usage and enable a study of webbing wear characteristics and mechanical reliability of the system to be made. A total of 30,000 operating cycles has been carried out to date and opportunity taken to test different webbing materials and weaves to establish the optimum for this duty. At this stage none has proved entirely satisfactory and it may be that further design work is necessary on the guide loop mounted on the moving arm, to reduce friction effect. The use of rollers may eventually prove necessary.

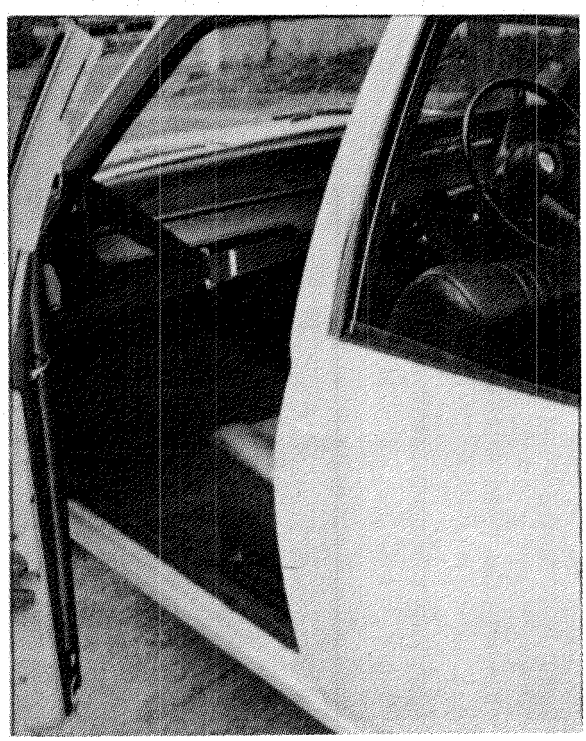


Figure 8. Cord Actuated System

Photographs of the Hillman Avenger car with the cord actuated passive seat belt installation are shown in Figures 8 and 9. As in the case of the lifting arm system, this is a similar scheme to that installed experimentally on a previous vehicle which has since been crash tested to prove restraint effectiveness. One of the principal aims of this system is to evaluate the additional user convenience achieved by the use of door tracks to withdraw the seat belts from both inboard and outboard anchorages. The use of cords enables one motor gearbox unit to actuate both inboard and outboard belt movements, and location of the mechanism in the engine compartment may be more convenient in some vehicles. For purposes of comparison, the door tracks can be eliminated and the system then viewed as an alternative to the lifting arm type previously discussed.

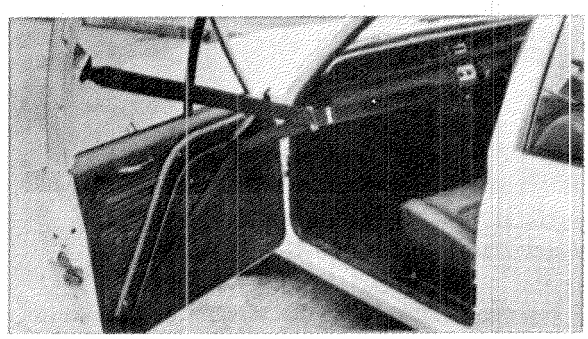


Figure 9. Cord Actuated System

DEVELOPMENT OF PASSIVE SEAT BELT SYSTEM FOR TRRL/BRITISH LEYLAND SAFETY CAR

The original concept of this project involved the application of one of the passive seat belt systems already developed to the above vehicle. Preliminary investigations therefore included design layouts and mock up schemes of lifting arm and cord actuated systems in the car. It became apparent that several special features of this vehicle were going to influence the choice of passive belt system, not least the fact that available space inside an already small car was decreased substantially by the energy absorbing padding installed to ensure compliance with other important occupant protection criteria.

A further special feature of the vehicle involves the design of the front seats, which offer sufficient strength to accept loads imposed by seat belt anchorage points under crash impact conditions. This feature meant that serious consideration was given to a seat belt system with belt layout using the seat for the shoulder anchorage of the diagonal section of webbing. The retractor reel could then be mounted in the door and belt deployment achieved by use of a lifting arm or a cord actuated arrangement.

It was finally decided that a centrally mounted lifting arm scheme would be adopted with the retractor reels installed on the transmission tunnel and the lap and diagonal anchorages attached to the rear edge of the front door frames. Specific features of the scheme supporting this decision included the following. From the user aspect, this particular belt layout ensures that when the vehicle door is closed, the belts are deployed correctly around the occupant

without need to re-arrange webbing in relation to the occupants' arms. In other words, the act of closing the door by the seated occupant places the arm in the correct position for seat belt deployment.

This is not the case for all systems originally considered. A second feature relates to the shoulder anchorage of the diagonal belt. Since the original use of an adjustable height anchorage on earlier vehicle installations it has become apparent that this is regarded as important not only for added convenience for occupants of differing size, but to ensure maximum restraint effectiveness. This feature can readily be incorporated in a door mounted anchorage.

Having decided on the system in principle it was necessary to design the individual components, in particular, the centrally mounted lifting arm unit. It became immediately apparent that the design previously used was unsuitable, as the space was very limited compared with previous installations on larger vehicles such as the British Leyland 1800 and Triumph 2.5.

It was decided to design an entirely new mechanism to ensure the most compact construction to suit this type of vehicle. Figures 10 and 11 are photographs of the mechanism on laboratory test before installation on the vehicle on demonstration at the ESV Exhibition currently taking place at TRRL Crowthorne.

The assembly comprises a main pivoting arm with built in reduction gearing and limit switches, and an electric motor connected by means of a flexible drive.

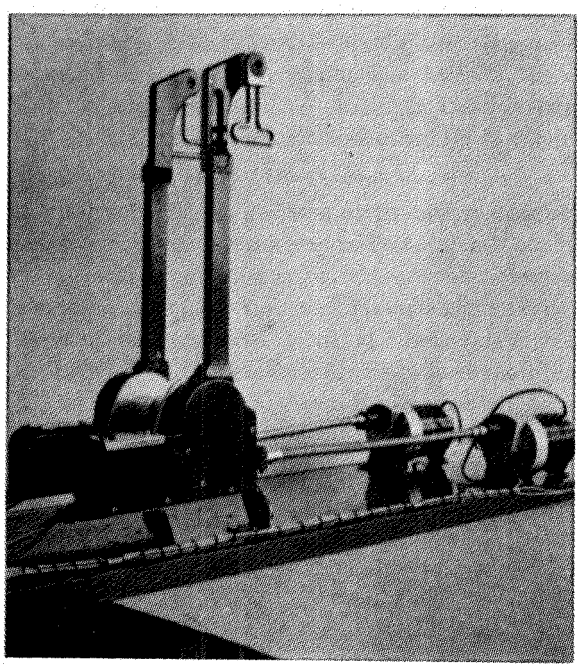


Figure 10. Laboratory Test

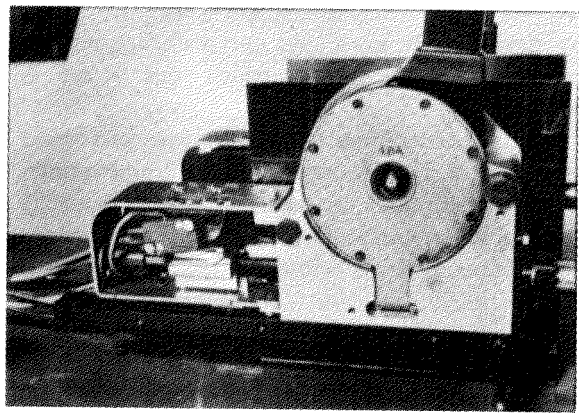


Figure 11. Detail of Arm Mechanism

To ensure a compact arm unit, some of the reduction gearing is incorporated in the motor unit and use made of an appropriate heavy duty flexible drive. The arms are shown before covering with suitable padding and matching trim material and terminate in a pivoting guide loop through which the seat belt webbing is passed.

It can be seen that with the mechanism cover fitted, no slot or other openings is present that might permit trapping of articles or fingers during actuation. Figure 11 illustrates the main gear and damper unit, and also the detail of limit switch operation and adjustment. A separate box contains the control relays, overload cutout switches and resistors to provide reduced speed for operation when the belts are being deployed around the occupant. The control system including manual operation buttons and seat switch, is similar to that previously used on earlier vehicles with this type of passive seat belt system.

EMERGENCY PRE-TENSIONING OF SEAT BELTS

Initial studies of a very simple single mass model of a restrained occupant showed that an occupant's forward movement might be reduced by up to 70% with no associated increase in peak accelerations if the belt loads could be generated before significant forward movement occurs. Further consideration of the more complex problem of preventing the high head deceleration which result from violent angular movements of the head, suggested that these could be alleviated by applying tension to the belts early in the crash.

Pyrotechnic retractors were chosen to provide the pretensioning force required, as their small size, low weight and rapid actuation were favorable factors when compared with hydraulic or mechanical methods of storing energy. Early examples of pyrotechnic devices produced forces which were very dependent on the amount of resistance experienced,

but development work by the manufacturer has produced a retractor in which this effect is controlled.

Frontal barrier impact tests have been performed using a test sled, as shown in Figure 12, and in order to make the results as representative as possible, dynamic torque measurements were made of various types of dummy necks before the tests.

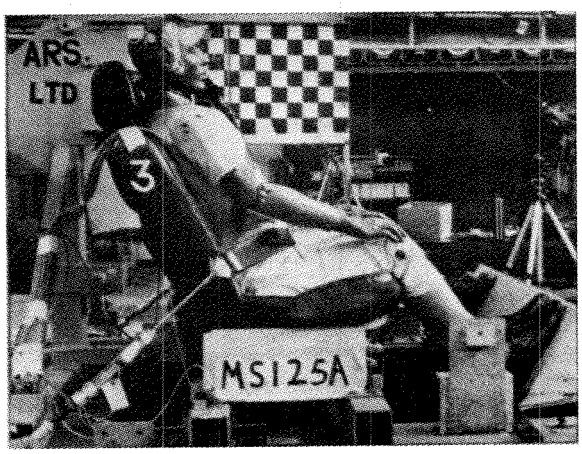


Figure 12. Sled for Pre-Tensioning Tests

The one finally selected was an Alderson neck with both cables removed, as this had a stiffness, at all angles of deflection, which was either within or below the "Mertz Corridors". Alderson VIP 50A dummies were used, and head angular accelerations were measured by means of two linear accelerometers mounted horizontally, one above the other. In all the tests at 30 mph the same type of belt was used, with stiff webbing (6% extension) and an energy absorbing force limiter (ply-tear webbing) in the shoulder strap. The seat belt anchorages were positioned to represent a 4 door car fitted with a passive seat belt system. The variables investigated were: applied forces (800 lbf and 1600 lbf), ignition time (to represent an inertia sensor and a bumper sensor), tensioner location and seat position.

During a datum test, with no applied tension, higher accelerations than before were recorded due to the more flexible neck, and dummy movements were such that they would have resulted in the dummy's head striking the steering wheel hub.

In the tests with a tensioner in the center strap the head and chest decelerations were reduced such that in one test the Head H.I.C. was 294 and the Chest S.I.103. Movements were also reduced and if a steering wheel had been present, contact with the hub would either have been prevented or the speed of impact reduced to less then 5 mph. Head angular accelerations varied considerable but the best result was +1760 rads/sec^2 and -1696 rads/sec^2. Further tests at 40 mph impact speed are scheduled.

While the study of performance is progressing, consideration has been given to possible modes of installation in motor cars. Although tensioners are applicable to conventional seat belts, passive seat belt systems only are reviewed in this study and as a typical case the lifting arm system installed in the British Leyland 1800 car is considered. Several different schemes have been examined, most applying pretension to the webbing strap retracting on to the reel but tensioning of the diagonal belt has also been considered. One scheme has been selected and detailed design work undertaken on a body shell for performance testing prior to installation on the vehicle.

COMPUTER SIMULATION OF DUMMY KINEMATICS

As part of the research and development work in conjunction with TRRL a study is being made of occupant movement related to pre-tensioned seat belts. Initially a very simple mathematical model was used for preliminary study of the basic aspects, followed by a more complex computer model for more advanced investigation. Preliminary work confirmed the potential importance of a moderately complex computer model to assist the study and as a preferred alternative to repeated sled tests. Investigations indicated that purchase of a suitable program, if such existed, would be expensive, as would modification and usage of an existing one. A base model was obtained from an associate company in the U.S.A. and modified to make the content appropriate for the duty and ensure suitability for operation on the available computer terminal.

Figure 13 is a diagram of the mathematical model and shows the three masses and four degrees of freedom considered appropriate. Other conventional features of this model are the tensions and angles of the three main webbing sections of a lap and diagonal seat belt system. In some areas modifications and additions have been made which are required by the particular nature of the investigation.

Calculation of results is critically dependent on neck characteristics and the derivation of suitably representative input data is difficult. Due to problems at very low head angular velocities a modification has been made to prevent hunting between the head loading and unloading conditions.

Belt characteristics have had to include not only the hysteresis resulting during extension and contraction of webbing, but also the effect of

incorporating load limiter, energy absorbers. Ply tear webbing has been used in the majority of the seat belt performance testing and consequently the appropriate characteristics have been incorporated. One of the difficult areas was that of balancing belt tensions at all stages of the impact. It is important to ensure true simulation of the tension distribution in the three sections of the webbing during the varying conditions throughout the impact, and also to take account of the friction effect ot the webbing in contact with the occupant.

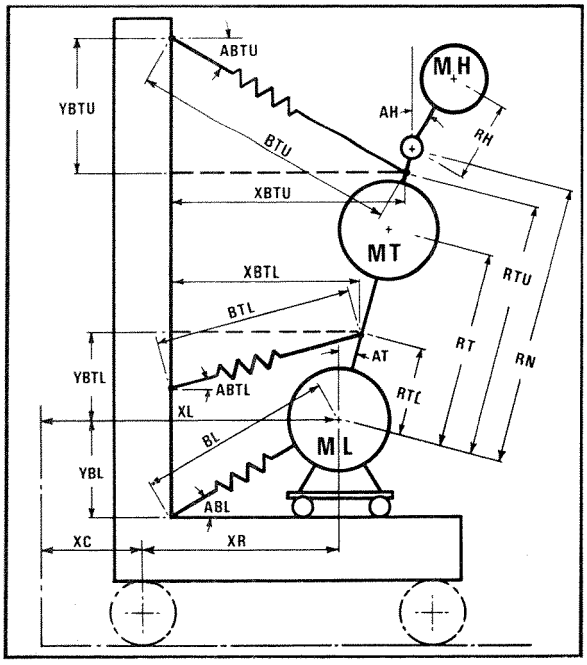


Figure 13. Three-Mass Mathmatical Model

A principal addition to the basic model has been the webbing pretensioning devices. At this stage of development work, these take the form of pyrotechnic actuators to retract and tension the webbing with a predetermined force, firing taking place at a time to simulate a crash sensor.

Factors which have had to be considered include the pressure/time history of the gas generated in the actuator and the locking of the actuator after completion of stroke. Allowance has also been made for the stiffness and depth of the seat cushion as this influences occupant movement, particularly when the lap belt anchorage location results in a steep belt angle.

Referring to the upper anchorage, the effect of location is likely to be of particular significance in passive systems. The tension in the shoulder belt is determined by the strain in the belt, as is the forward movement of the shoulder. The belt extension per unit movement of the shoulder changes with anchorage position, the strain per unit movement increasing as the webbing run from the shoulder approaches the horizontal. This effect is influenced by anchorage position in three dimensions and consequently an input has been introduced for the sideways location of the anchorage.

Work on validation of the model has been carried out and studies are being made related to pre-tensioned seat belts and associated problems. One example is an investigation with a specific impact condition, of the effect of upper belt anchorage location on dummy movement belt loads and injury criteria related to head and chest.

SUBJECTIVE EFFECTS ON OCCUPANTS

This investigation has been carried out with considerable assistance from specialist consultants and has included much research into published data from various sources to extract the information relating specifically to the topics being studied. A resume is given of work carried out in two principal areas.

Assessment of Hearing Damage Due to Air Bag Deployment

Research on air bags, particularly in the U.S. had indicated a potential hearing hazard to car occupants. Although several noise damage risk criteria existed, it was considered that a consistent approach to the acceptability of air bags was needed. It was hoped that once an agreed assessment method became available, research could be carried out to determine the relationship between various air bag system parameters and the deployment noise.

A critical review was made of the methods of correlating short duration noise with hearing damage. This showed, fairly conclusively, that damage to the ear drum or ossicles is unlikely unless the sound pressure level reaches 185 decibels. This figure related to 95% of the population and was beyond that reported so far during any air bag deployment test. Damage to the ears was thus likely to be limited to the cochlea and result in temporary deafness that might persist to a lesser extent. It seemed that this damage would be related to the energy transmitted to the ear as result of the air bag deployment.

It was assumed that one person would only be exposed to the consequences of air bag deployment in a vehicle very infrequently, and there are good grounds for thinking that the damage done by one burst of loud short duration noise can be related, in terms of total energy, to a lesser noise lasting for a normal working day.

Evidence from the field of occupational health indicated the amount of hearing damage to be

expected from various levels of noise, expressed in dBA values, that persist for periods of eight hours. This suggested that continuous noise of this duration at a level of 90 dBA, would not cause significant hearing damage in 95% and perhaps 99% of the population. Reports by other researchers, on sound levels found to accompany air bag deployment, revealed that the energy impinging on the ears of occupants during an individual incident did not constitute a significant hazard. Similarly, experiments using various kinds of air bags, inflation pressures and volumes, support the results from elsewhere.

Experimental work confirmed that noise and hence energy to the ear are related to speed of deployment of the air bag, the effect of the explosive detonator being insignificant. For effective crash protection, deployment time was stipulated as a maximum of 30 ms. This resulted in an energy dose equivalent to an exposure of eight hours duration to a noise of between 79 and 85 dBA in the series of experiments.

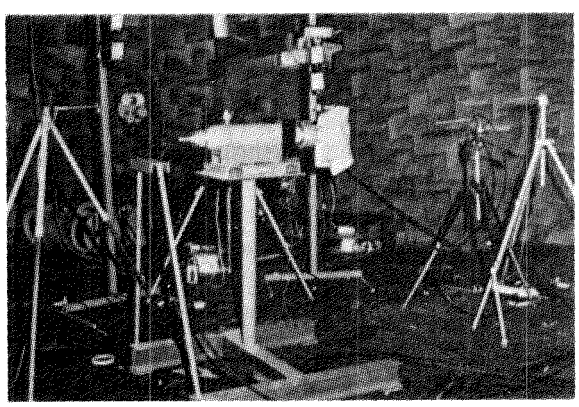


Figure 14. Air Bag in Anechoic Chamber

Figure 14 shows these tests being carried out in an anechoic chamber. It is likely that in cars, reflections would somewhat augment the noise values. However, the indications were that, even so, permanent damage to the hearing of vehicle occupants is only a remote possibility to some specially susceptible individuals. Even these are unlikely to suffer from more than a slight impairment of their hearing. It was therefore concluded that with the modifications that can already be envisaged, air bag restraint systems could be made effective, without constituting a threat to the hearing of vehicle occupants.

Limitation on Loads Imposed by Belt Tensioners

An important aspect of the investigation into crash activated seat belt tightening is the study of the likely "side effects" on occupants. The beneficial effects would appear as improved restraint effectiveness but there are circumstances in which consideration must be given to possible limitations on forces and loads to ensure no adverse effect on the vehicle occupant. To evaluate these effects, a study has been made involving data published by other researchers on chest loading experiments and closed chest cardiac massage. Tests have also been carried out with simulated conditions on a dummy.

Comparisons of real accidents with the results of laboratory impact tests, indicate that many persons have been subjected to shoulder belt tensions in excess of 1500 to 2000 lbf without suffering any chest injuries other than bruising. The tension in a seat belt is usually maintained in the belt for less than 100 ms in an impact.

However it would be wrong to assume, as a result of the above, that it would be safe to apply this load to a belt by an external means. There are two reasons for this. First, the loads might be applied for a considerable longer period. This might happen if the force limiter is not deployed in a moderate impact with a lightweight occupant, or in the case of an inadvertent firing. Second, in a conventional belt the load applied to the chest is divided between the force required to deflect the rib cage and force required to decelerate the chest contents which have been thrown forward by the gradually increasing chest deceleration. In the case of a force applied by a tensioner, over a period of a few milliseconds, before the chest has experienced any deceleration, nearly all the applied force may be used to deflect the rib cage.

In order to estimate the load which might be safely applied the following have been considered:

- Reports of chest impacts performed on cadavers.
- The frequency of chest injury during external cardiac massage.
- The comparison of the deflection of the chest of an OPAT dummy during simulated external cardiac massage with that which occurs when tension is applied to a belt being worn by a dummy.

This work has so far suggested that the limit on applied tension in the shoulder strap might be as low as 1000 lbf but further work is being performed on this subject.

Another very important factor is the amount of residual belt tension which might be acceptable for periods of more than a few seconds. Human volunteer tests have indicated that shoulder strap loads in excess of 70 lbf are very uncomfortable and cause difficulty with breathing. If seat squab cushion stiffness is too great to allow the load to drop below

this value then it will be necessary to have some device which releases the load a few seconds after the tension is applied.

CONCLUSIONS

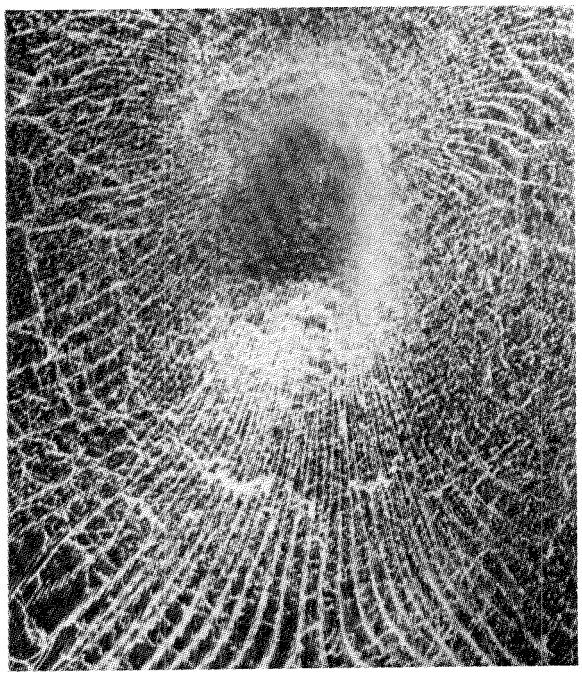
It has been established that improvements in the occupant protection capability of seat belt systems can be made, mainly by selection of webbing and use of load limiters in the belt layout. Further improvements, including an increase in the impact speed at which such systems are effective, can be made, including the use, where appropriate, of crash activated belt tensioning devices.

Vehicle installations of three different systems of passive seat belts have been completed for longer term user evaluation. A further developed version, achieving a more compact layout of one system, has been installed for evaluation in the TRRL/British Leyland Safety Car.

Further work is planned on pre-tensioning of seat belts with reference to performance at increased impact speed, and study of problems related to inadvertent actuation. Activity is also scheduled on the rapid response crash sensor, associated with energy absorbing bumpers, and crash deployed knee restraints for possible use in future safety cars.

REDUCED LACERATION FROM A NEW LAMINATED WINDSHIELD

JOHN PICKARD
Research and Development Manager
Triplex Safety Glass Co., Ltd.



SUMMARY

A new car windshield has been developed by the Triplex Safety Glass Co., Ltd. of England which promises a substantial reduction in facial laceration without adverse consequences in other respects. The new windshield is called Triplex Ten Twenty.

Of laminated construction, using two pieces of 2.3 mm thick float glass and a standard HPR polyvinyl butyral interlayer of 0.76 mm thickness, the two glass components are thermally stressed to different levels. The outer glass is lightly stressed to about 7 $MN.m^{-2}$ for optimum stone impact resistance and the inner glass to 47 ± 3 $MN.m^{-2}$ (center tensile stress) for major reduction in laceration.

Test programs have been carried out at Triplex using a dropping head form and using a skull impactor sled and by Wayne State University in Detroit, U.S.A., using a 50th percentile anthropomorphic dummy on their Wham III crash simulator.

The results of these evaluations are given in terms of Head Injury Criterion, Gadd Index and various laceration scales including a newly developed objective scale – the Triplex Laceration Index, or TLI.

The TLI is a formula based on the number and the measured length and depth of cuts in the dummy head coverings. When compared with conventional laminated windshields Triplex Ten Twenty has been demonstrated to give a reduction in laceration of 2 units on this logarithmic scale.

A reduction of TLI units is equivalent to a reduction of:

99% in the number of cuts of constant size,
or 90% in the length of cuts of constant depth and number,
or 78% in depth of cuts of constant length and number,
if the differently weighted attributes (depth, length and number) are considered separately.

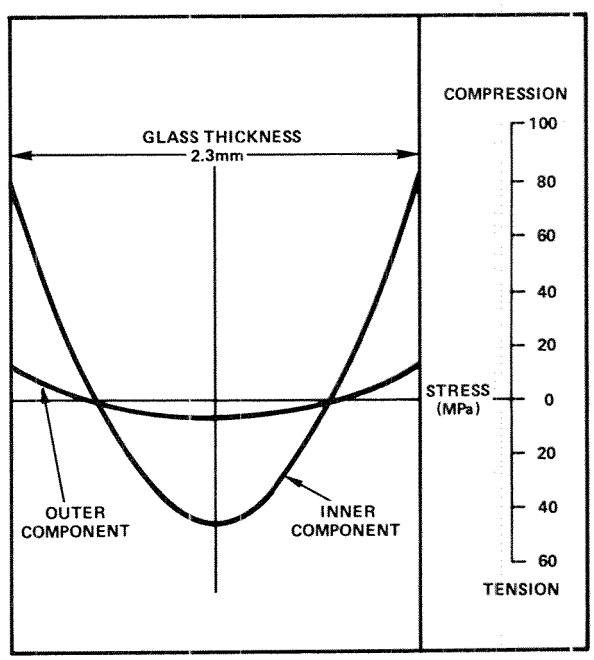


Figure 1. Stress Profiles for Two Components of Ten Twenty Windshield

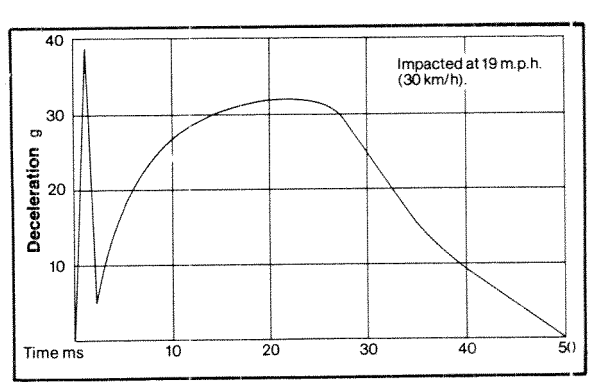


Figure 2. Typical Deceleration Trace for Ten Twenty Windshield

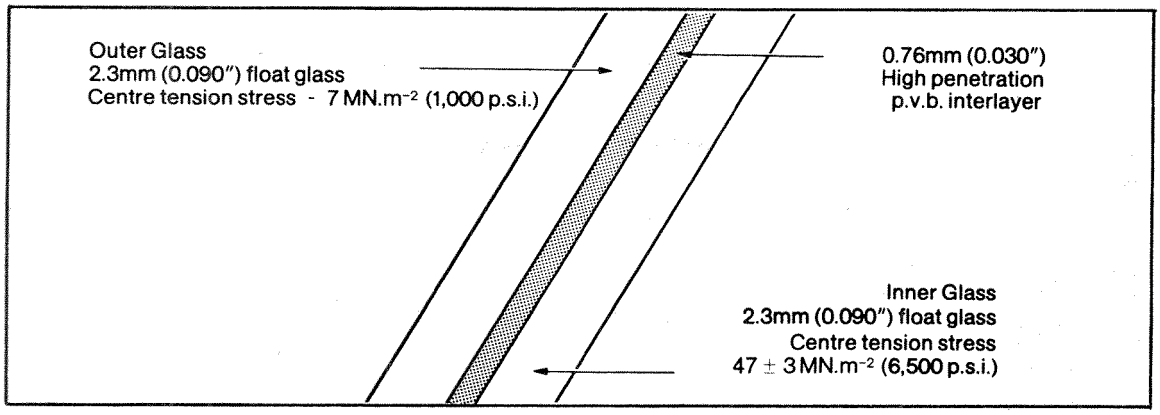


Figure 3. Construction of the Ten Twenty Windshield

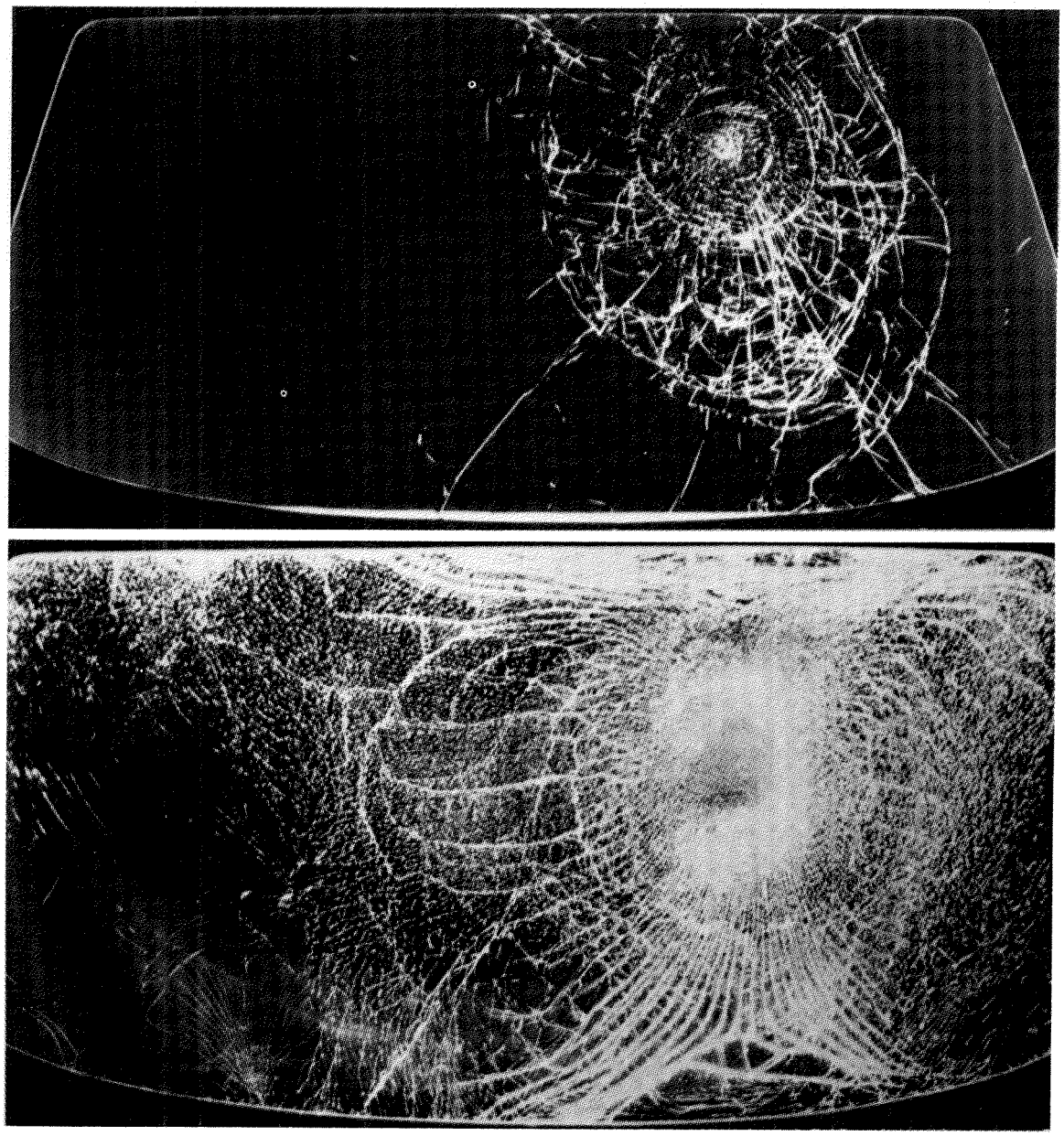
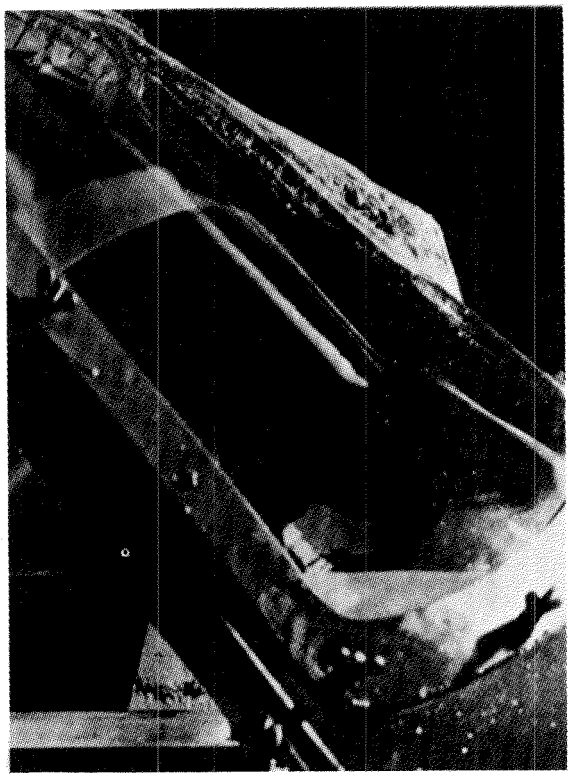


Figure 4. 3 mm + 3 mm Laminated Glass and Ten Twenty after 19 mph (30 km/h) Sled Impact.



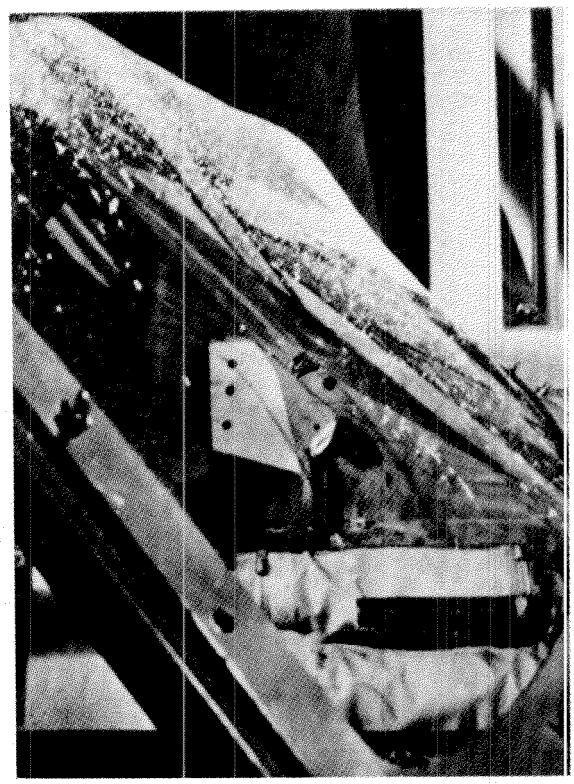


Figure 5. 3 mm + 3 mm Laminated and Ten Twenty Side View after 19 mph (30 km/h) Sled Impact

In practice all three attributes contribute simultaneously and a case where half the average length, half the average depth and one third of the number of cuts occurred would rate 2 units TLI.

THE OBJECTIVE

A car windshield is required ideally to possess a number of performance characteristics apart from its normal transparency. It is expected:

1. To retain its transparency after normal road stone impact.
2. To retain weather protection after such an impact.
3. To retain an occupant thrown against the glass at an acceptable deceleration rate to avoid permanent brain damage.
4. To retain the occupant within the car.
5. To retain the occupant while causing a *minimum of lacerative injury*

The test program described briefly here (Reference 1) shows that the first four features are met by Triplex Ten Twenty in the same way as by conventional HPR laminated windshields, while providing the important additional advantage of a major reduction in laceration.

TRIPLEX TEST PROGRAM

For deceleration measurement a 10 kg dropping headform was used falling onto windshields mounted normally to the direction of impact. Measurements were made in terms of Gadd Severity Index (SI) and Head Injury Criterion (HIC) at speeds up to 38 mph (60 km/h).

The results are shown in Table 1 comparing Triplex Ten Twenty windshields with conventional 3 mm + 3 mm and with conventional 2.5 mm + 2.5 mm laminated windshields at 20°C.

The results are plotted on Figure 6 and show superior performance of Triplex Ten Twenty in terms of Gadd Severity Index. No significant difference was found in terms of HIC (Figure 7) nor in penetration velocities between the three types.

For laceration measurement a skull impactor, similar to that developed by Corning and shown in Figures 8 and 9, was used.

Windshields were clamped in an aperture set at the usual car mounting angle. The headform was covered first with a 6 mm thick cap of PVC to represent flesh and to enable the depth of "flesh" lacerations to be measured. Over this were placed two 1 mm layers of moist chamois leather to represent skin.

Table 1

RESULTS OF 10 kg HEADFORM NORMAL DROP-TEST ON TEN TWENTY 2.5 mm AND 3 mm WINDSHIELD

Test No.	Windscreen Composition	Velocity, km/h	Deceleration: Max Deceleration Due to Glass, g	Deceleration: Duration of Deceleration Due to Glass, ms	Deceleration: Max Deceleration Due to Interlayer, g	Deceleration: Total Duration of Impact, ms	Gadd SI	HIC	Total Interlayer Tears, mm	Pene-tration
1965	3mm	38.9	56	2.2	45	111	240	187	230	No
1966	3mm	38.9	114	3.9	41	82	293	173	–	Yes
1967	2.5mm	38.9	55	2.2	39	57	234	193	180	No
1968	2.5mm	41.0	59	1.9	38	119	213	179	254	No
1969	2.5mm	43.0	63	1.2	41	114	212	172	252	No
1970	2.5mm	46.0	98	1.5	40	43	181	132	–	Yes
1971	Ten Twenty	38.9	37	1.4	40	55	233	198	130	No
1972	Ten Twenty	41.0	40	1.2	40	60	209	176	150	No
1973	Ten Twenty	43.0	42	0.9	38	148	172	141	–	Yes
1974	3mm	34.9	53	2.2	40	51	210	175	95	No
1975	2.5mm	34.9	58	1.7	35	61	193	168	105	No
1976	2.5mm	34.9	75	1.4	38	51	220	181	100	No
1977	2.5mm	34.9	38	1.9	35	120	167	140	262	No
1978	3mm	32.0	61	1.7	38	51	183	153	0	No
1979	2.5mm	32.0	57	1.2	40	51	234	192	100	No
1980	Ten Twenty	32.0	40	0.9	34	55	175	153	0	No
1981	3mm	27.8	45	1.7	31	51	115	95	15	No
1982	2.5mm	27.8	85	2.4	34	51	207	163	5	No
1983	Ten Twenty	27.8	39	1.2	33	50	131	113	55	No
1984	3mm	24.0	60	2.4	28	51	104	84	0	No
1985	2.5mm	24.0	53	1.9	30	51	117	98	30	No
1987	Ten Twenty	24.0	51	1.2	28	50	98	85	40	No
1990	2.5mm	20.0	32	1.4	23	51	64	57	0	No
1991	3mm	20.0	94	3.9	21	53	124	78	0	No
1992	Ten Twenty	20.0	49	2.2	24	51	68	59	0	No
1993	3mm	43.0	52	2.7	43	52	161	137	–	Yes
1994	Ten Twenty	43.0	45	1.4	41	60	232	193	–	No
1995	2.5mm	43.0	51	2.9	45	71	227	170	–	Yes
1996	3mm	41.0	43	2.2	43	71	160	141	–	Yes
1998	Ten Twenty	29.9	39	1.4	33	50	138	121	50	No
1999	2.5mm	29.9	86	2.4	36	49	176	138	50	No
2000	3mm	29.9	84	2.2	32	61	185	140	0	No
2001	2.5mm	20.0	77	2.9	23	52	95	56	0	No
2002	2.5mm	24.0	63	1.4	30	55	112	89	0	No
2003	2.5mm	27.8	52	1.4	33	48	123	103	50	No
2004	2.5mm	32.0	43	1.4	36	50	166	143	80	No
2005	2.5mm	38.9	65	2.2	45	56	273	215	190	No
2006	2.5mm	43.0	46	1.2	43	109	208	154	–	Yes
2007	3mm	20.0	112	3.9	18	51	172	122	0	No
2008	3mm	24.0	90	3.2	33	43	208	136	0	No
2009	3mm	27.8	88	2.9	34	47	179	136	0	No
2010	3mm	32.0	96	2.2	43	46	264	202	0	No
2011	3mm	38.9	65	2.4	48	47	292	244	100	No
2015	3mm	20.0	90	3.7	21	53	116	57	0	No
2017	Ten Twenty	20.0	31	1.7	27	52	76	63	0	No
2018	Ten Twenty	27.8	66	1.0	34	51	175	145	0	No
2019	Ten Twenty	24.0	29	1.7	29	50	91	78	0	No
2020	Ten Twenty	32.0	37	1.0	35	75	153	128	150	No
2021	Ten Twenty	38.9	36	1.2	38	42	93	72	–	Yes
2022	3mm	43.0	81	2.0	38	43	139	97	–	Yes

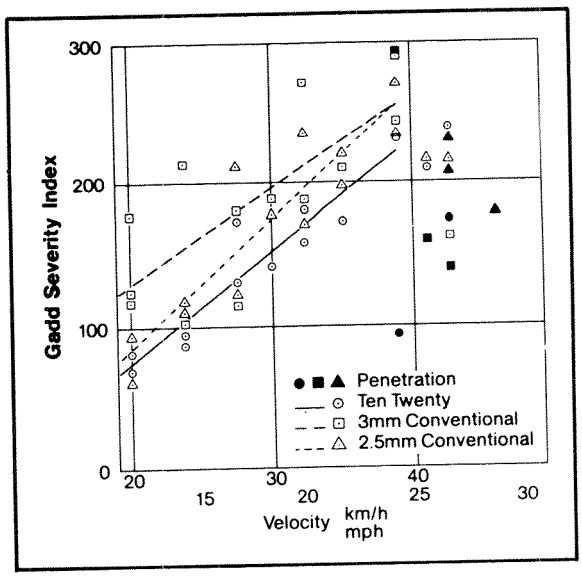


Figure 6. Gadd SI as Function of Impact Velocity by Dropping Headform Test

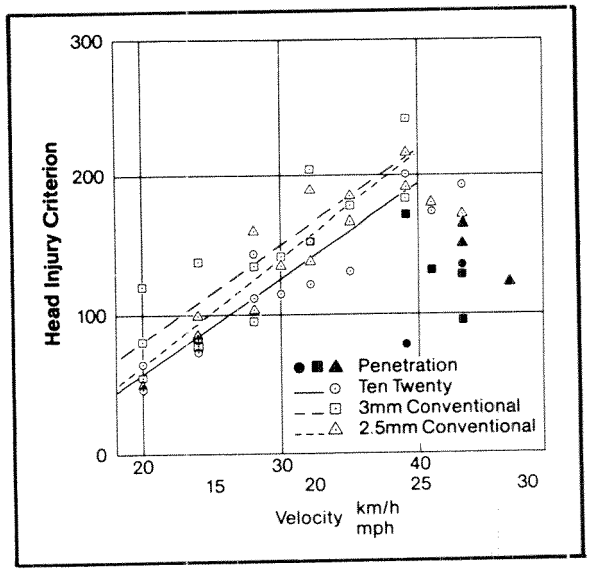


Figure 7. H.I.C. as Function of Impact Velocity by Dropping Headform Test

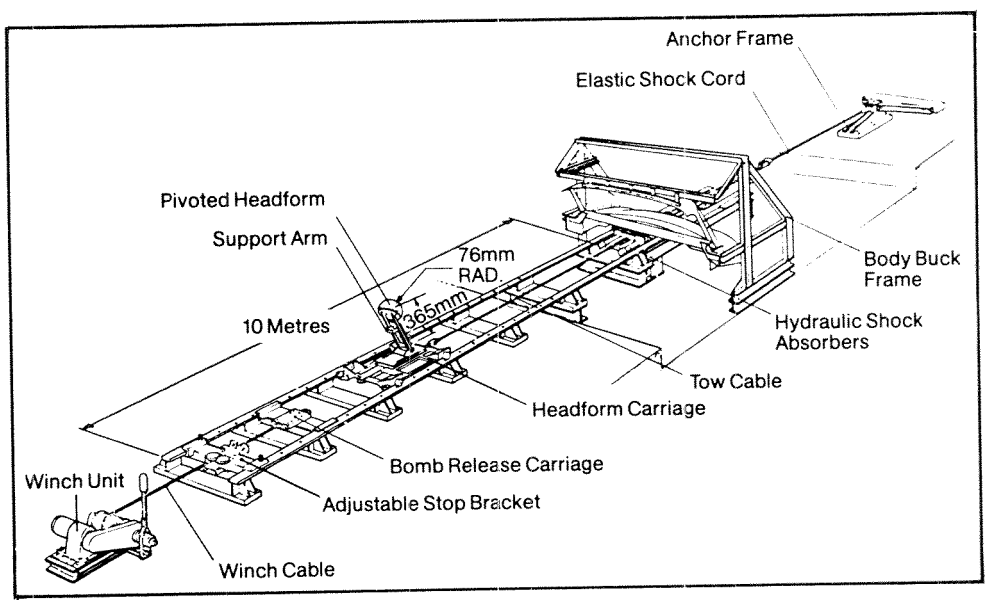


Figure 8. Skull Impact Test Rig

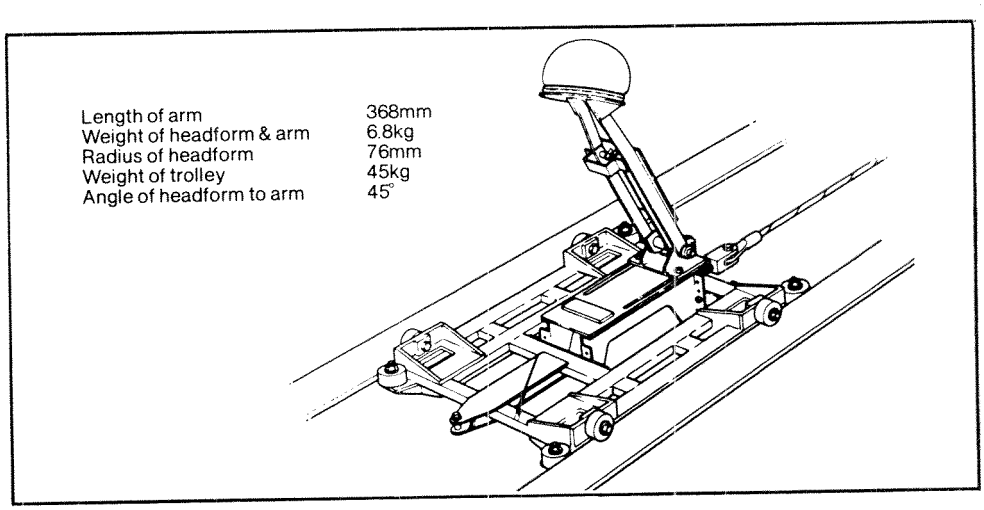


Figure 9. Carriage and Headform Assembly of Skill Impact Test Rig

A high degree of subjective assessment was found to be the weakness of existing laceration indices. A new objective index was therefore evolved (The Triplex Laceration Index, or TLI) based solely on measurements of cuts produced in the leathers and PVC facemasks. Full details of this index were the subject of a separate paper presented to the American Association of Automotive Medicine (Reference 2).

Mathematical analysis of experimental results led to the logarithmic formula:

$$TLI = 2 + \log_{10}(1 + 1.16\,D_1 + 50.8\,D_2 + 16{,}500\,D_3).$$

Where $D_1 D_2$ and D_3 are the lacerative damage in the two layers of chamois and PVC cap respectively. The lacerative damage in each layer is computed as a function of the length, depth and number of cuts.

The results given in Table 2 and plotted on Figure 10 show a 2.7 (logarithmic) unit improvement over conventional 3 mm + 3 mm annealed laminated and a 2.0 (logarithmic) unit improvement in TLI over conventional 2.5 mm + 2.5 mm.

These results indicate reduced laceration by Triplex Ten Twenty of 5 X 100 and of 100 times respectively without penalty in other directions.

Table 2

DATA FROM SKULL IMPACT RIG AT 20°C (TRIPLEX SAFETY GLASS)

Test No.	Type of Windscreen	Velocity of Impact, km/h	TLI	Total Interlayer Tears, mm
80	2.5mm	57.1	10.3	250
81	Ten Twenty	55.3	9.8	290
82	3mm	33.8	8.9	20
84	3mm	50.8	12.1	250
87	3mm	27.4	5.7	–
88	Ten Twenty	52.9	8.7	30
89	2.5mm	27.5	5.9	–
90	3mm	51.5	12.5	390
91	2.5mm	41.5	9.4	112
93	2.5mm	28.3	4.8	–
94	Ten Twenty	28.0	2.9	–
95	2.5mm	50.8	10.8	250
97	3mm	41.5	9.7	230
98	3mm	40.7	10	–
100	3mm	33.6	9.8	–
102	2.5mm	33.8	9.8	–
104	Ten Twenty	27.4	0	–
106	3mm	26.7	8.4	–
107	2.5mm	58.1	12.9	350
108	Ten Twenty	57.1	11.9	207
109	2.5mm	33.1	9.0	3
110	Ten Twenty	40.4	7.6	134
113	Ten Twenty	33.1	6	–
114	2.5mm	50.7	12.7	235
115	Ten Twenty	40.4	8.1	14
120	3mm	54.5	12.3	190
122	Ten Twenty	31.5	9.2	–
123	2.5mm	37.8	10.1	52
124	Ten Twenty	46.8	8.2	50
125	3mm	54.5	12.1	500
127	Ten Twenty	26.1	6	–
134	Ten Twenty	48.8	10	148
142	2.5mm	58.1	12.2	221
143	Ten Twenty	26.1	5.4	–
145	Ten Twenty	48.8	10.5	61
147	2.5mm	49.4	12.4	400
148	Ten Twenty	54.2	12.8	–
149	Ten Twenty	32.7	3.1	22
151	Ten Twenty	55.0	8.7	161
152	Ten Twenty	32.3	5.5	–
153	Ten Twenty	41.0	5.8	12
154	2.5mm	54.9	12.2	190
157	2.5mm	26.1	5.7	–
159	Ten Twenty	39.3	5.7	–

WAYNE STATE UNIVERSITY TESTS

The Deceleration and Lacerative performance of Triplex Ten Twenty and conventional annealed 2.5 mm + 2.5 mm laminates was compared on the Wham III crash simulator at Wayne State University using a Sierra 1050 anthropomorphic dummy in a modified Volvo car.

Comparisons were made at speeds between 16 and 56 km/h head impact speeds and the results are shown in Table 3 and on Figures 11 (TLI) and 13 (HIC).

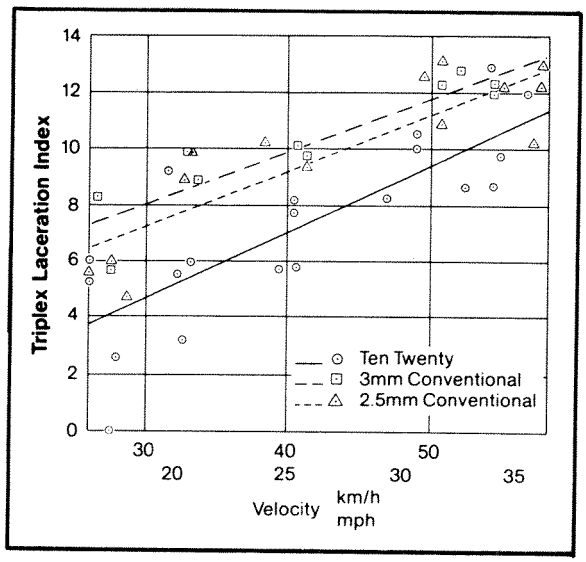


Figure 10. TLI as Function of Velocity by Skull Impact Test at 20°C

Though the Wham III and Triplex laceration tests showed a difference in absolute levels of TLI the Wayne Test program confirmed the improvement of 2 logarithmic units TLI for Triplex Ten Twenty compared with 2.5 mm + 2.5 mm annealed laminate.

At the same time Wayne found that HIC was not significantly increased and remained well below the accepted danger level of 1,000 (U.S. MVSS 208).

MECHANISM

The very large reduction in laceration is attributable to the fine fragmentation on head impact of the highly stressed inner glass of Triplex Ten Twenty.

When the head strikes a Triplex Ten Twenty windshield the energy of the impact plus the stored stress in the inner glass component causes the inner glass to break into a toughened type of fragmentation over the depth of the glass down which the head moves after initial impact. With the characteristic finer and more blunted fragments this causes much less laceration damage than when the head has to pass over the coarser and sharper fragments of a conventional annealed laminated windshield (see Figures 4 and, particularly, 5).

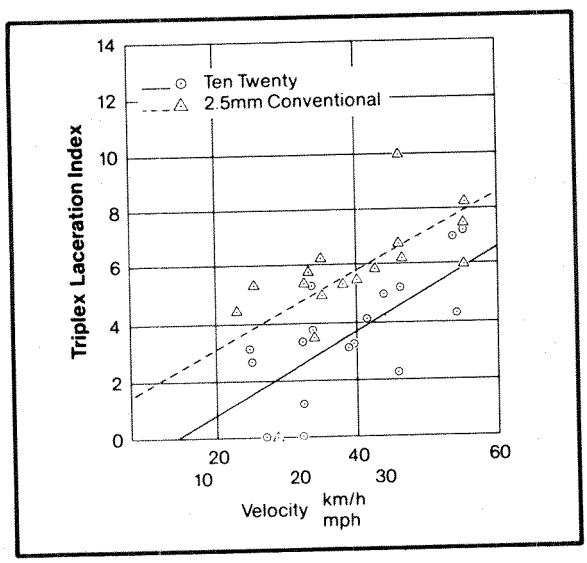


Figure 11. TLI as Function of Velocity by WHAM III Test Rig

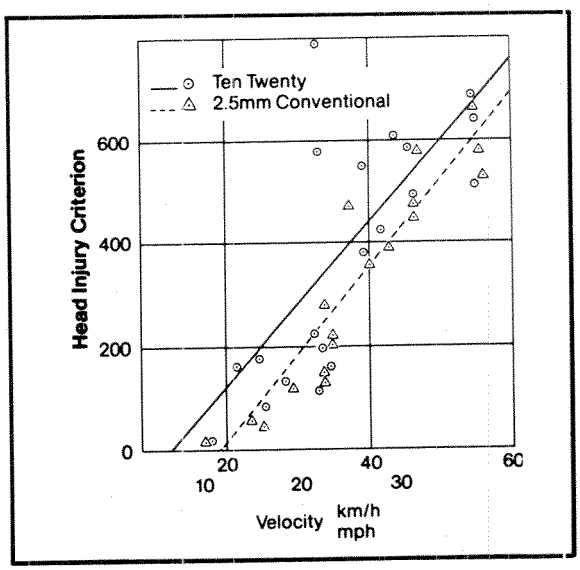


Figure 13. HIC as Function of Impact Velocity by WHAM III Test Rig

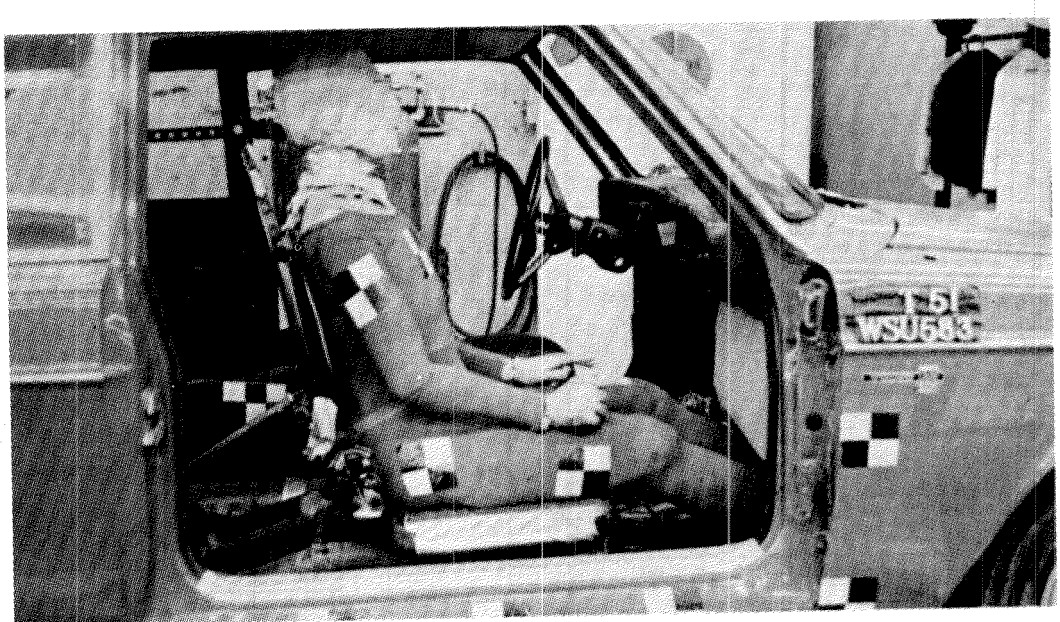


Figure 12. Vehicle with Dummy Installed on WHAM III

Table 3

SUMMARY OF DATA ANALYSIS FOR TRIPLEX WINDSHIELD LACERATION STUDY AT WSU

Run No.	Type of Windscreen	Velocity, km/h	Stopping Distance, mm	Head Index Gadd	Head Index HIC	Laceration Index WSU	Laceration Index Corning Scale	T.L.I.	Pummel	Remarks
T- 4	2.5mm	34.1	410	253	220	3	4	4.9	3	Abrasion on forehead
T- 5	Ten Twenty	33.5	410	200	164	1	4	5.3	3	Light abrasion on forehead
T- 6	2.5mm	34.1	410	248	202	3	5	6.2	4	Windshield slightly pulled out, abrasion on forehead
T- 7	Ten Twenty	32.8	410	249	196	1	3	3.8	5	Abrasion on forehead, one cut through outer layer on nose
T-10	2.5mm	16.3	300	11	9	0	0	0	5	Head broke outer glass, no cuts
T-11	Ten Twenty	16.9	200	13	11	0	0	0	5	Head broke outer glass, no cuts
T-14	2.5mm	24.8	300	67	55	1	4	5.2	4	Light abrasion and few small cuts on forehead and nose
T-15	Ten Twenty	24.9	300	131	82	1	3	3.2	3	Abrasion on forehead
T-16	2.5mm	36.7	500	576	482	3	4	5.4	3	Severe abrasion and a few cuts on forehead, buttocks slid off seat, legs jammed under instrument panel
T-17	Ten Twenty	41.5	500	677	423	1	3	4.1	4	Severe abrasion on forehead and nose, no SI acceleration
T-18	2.5mm	42.2	500	465	393	3	4	5.9	3	Lacerations and severe abrasion on forehead, abrasions on nose
T-19	Ten Twenty	27.5	330	371	121	0	0	0	5	–
T-20	2.5mm	28.3	330	172	113	0	0	0	3	–
T-21	2.5mm	33.1	410	174	143	3	4	5.4	3	Abrasions and cuts on forehead
T-22	Ten Twenty	32.0	410	278	222	1	2	3.4	4	Scratches on forehead
T-23	2.5mm	33.0	410	229	161	1	3	3.6	4	Scratches on forehead
T-24	Ten Twenty	32.3	410	175	131	1	1	1	4	Scratches on forehead
T-25	Ten Twenty	43.9	610	787	606	3	4	4.9	3	Severe abrasions on forehead, dummy slid off seat
T-27	Ten Twenty	45.9	610	582	494	3	4	5.2	3	127mm tear (vertical), severe abrasions with few cuts on forehead and nose
T-28	2.5mm	46.0	610	570	482	3	5	6.2	5	50mm tear, several cuts on forehead and near right eye
T-29	2.5mm	45.9	610	593	463	6	6	6.8	3	More severe abrasions on forehead, cuts on forehead, eyes, and nose
T-30	Ten Twenty	54.1	710	871	523	6	6	7.1	3	267mm jagged tear, extensive lacerations through outer chamois and couple of cuts through inner chamois
T-31	2.5mm	55.0	710	747	584	6	7	8.3	3	50 x 152mm tear, severe lacerations through both layers of chamois, 11mm cut on chin in rubber flesh
T-32	Ten Twenty	55.0	710	798	643	6	6	7.3	3	216mm jagged tear, severe lacerations through both layers of chamois, no cuts in rubber flesh
T-33	2.5mm	55.0	710	905	673	6	6	7.4	3	Severe lacerations through layers of chamois, no cut in rubber flesh
T-35	Ten Twenty	22.5	330	258	167	0	0	0	3	No fracture
T-36	Ten Twenty	32.2	410	1380	811	0	0	0	3	No fracture
T-37	Ten Twenty	39.3	500	836	543	1	2	3.1	3	–
T-38	2.5mm	23.3	330	114	68	1/3	3	4.4	4	–
T-45	Ten Twenty	32.2	410	1069	568	0	0	0	4	No lacerations
T-47	2.5mm	33.1	410	329	268	3/6	5	5.8	4	–
T-48	Ten Twenty	25.4	330	330	185	1/3	3	2.7	4	–
T-50	2.5mm	40.1	500	487	369	3	4	5.4	4	–
T-51	Ten Twenty	39.4	500	534	381	1	2	3.1	4	–
T-54	2.5mm	47.0	610	689	581	10	9	10.0	3	Severe lacerations, cuts (15 and 4mm) in flesh of dummy
T-55	Ten Twenty	45.9	610	1054	585	1	2	2.3	3	–
T-58	2.5mm	55.0	710	629	528	6	5	6.0	4	63 and 140mm tears
T-59	Ten Twenty	54.9	710	1166	689	3	3	4.5	4	–

REFERENCES

1. Improved Laminated Windshield with Reduced Laceration Properties, S. E. Kay and J. Pickard (Triplex) and Professor L. M. Patrick (Wayne State University), presented at 17th Stapp Conference, Oklahoma, Nov. 1973.

2. An Objective Method of Assessing Laceration Damage to Simulated Facial Tissues, The Triplex Laceration Index, J. Pickard, P. Brereton and A. Hewson (Triplex), presented at AAAM Conference, Oklahoma, Nov. 1973.

EVALUATION OF SEAT BELT SYSTEM AND DUMMY CHARACTERISTICS

YOSHIO SERIZAWA
Manager, Safety Research Department
Nissan Motor Co., Inc.

INTRODUCTION

In recent years, many accident investigations and analyses have been reported in various countries (References 1, 2, 3, 4, 5 and 6).

Much data on biomechanical loads such as g-load and femur force produced on the dummy have been obtained in collision tests. No such data, however, have been available for actual collisions.

Consequently, occupant injury levels in actual collisions are not well agreed with the occupant injury criteria of FMVSS no. 208 and ESV.

In this paper, we wish to recognize the effectiveness of seat belts, to discuss the problems involved in seat belt usage, and to examine the restraint capability of seat belts, through our experience in the Nissan ESV E_2 development.

We wish to clarify questionable points regarding dummies and head injury criteria.

PROBLEMS WITH CURRENT SEAT BELT SYSTEMS

Effects of Seat Belt Usage

The use of seat belts is effective in reducing almost all types of injuries and also in preventing occupant ejection from the vehicle. The seat belt reduces the frequency rate of occupant injury and lowers the maximum damage degree (References 1, 2 and 3).

According to the investigation in West Germany (Reference 3), the level of injuries sustained by belted occupants is the same as that sustained by unbelted occupants in a relative collision speed approximately 30 km/h lower.

Further, according to investigations (Reference 1) by the AB Volvo Company, non-fatal injuries in the case of drivers wearing 3-point belts were reduced by 57% in low-speed collisions and by 48% in high-speed collisions; in the case of front-seat passengers wearing 3-point belts, the injury reduction rate was slightly higher.

The same company reports that 3-point safety belts provided "full protection" for occupants against fatal injuries in collisions at speeds of up to about 60 mph.

Problems with Seat Belts

Occupants frequently wear the belts loosely (References 1, 3, 4 and 5), and are thus liable to be injured in secondary collisions with such objects in the car interior as the steering wheel and column, the windshield, the roof over the windshield, or the dashboard (Reference 1).

When the lap belt is not worn properly, occupants sometimes are injured in the internal organs in high-speed collisions. Furthermore, seat belts cause rib fractures, chest pains, collarbone fractures, abrasions from the webbing and other non-fatal injuries.

PERFORMANCE IMPROVEMENTS AND THEIR EVALUATION

Effectiveness of Reduction in Belt Slack

A reduction in the slack of the seat belt assembly can greatly improve the effectiveness of the belt (References 7, 8 and 9).

A reduction in belt slack will make possible the effective utilization of ride-down effect, and will result in the following effects on occupant injuries:

- The occupant's relative forward displacement is decreased, and the level of injury in secondary collisions is lowered.
- It is possible to lower the deceleration produced on the occupant.

To reduce the belt slack, a preloader was devised during the development of the Nissan ESV E2. For the lap and shoulder belts, two preloaders were installed inside the front seatback of the ESV E2 (References 7 and 8).

The operation principle of the preloader is that 75 milliseconds prior to the collision, when the electric signal is transmitted from the radar sensor to the squib type trigger, high-pressure gas in the inner tube is released into the atmosphere and high pressure gas in the outer tube gives the belt a load of more than 400 kg.

Protection of Occupant from Secondary Collision with Steering Wheel and Column

It is frequently impossible for the driver to avoid a secondary collision against the steering wheel and column especially in a high-speed collision.

In fact, the results of traffic accident investigations disclose that drivers equipped with 3-point seat belts have sustained 36% fewer injuries with the steering wheel and column than drivers wearing no seat belts (Reference 1).

It is considered necessary to reduce the level of driver injury caused by the steering wheel and steering column. An auto-retractable steering system (in which the steering wheel and column are retracted immediately after the collision) was adopted in the Nissan ESV E2 (References 7 and 8), so that the driver's head could bend completely downward without touching any object.

The installation of a retractable steering system, in addition to the preloader mentioned in 3-1 above, would significantly increase the driver's survival space (Figure 1).

Force Limiting and Energy Absorbing Belt

In a vehicle collision, various biomechanical loads such as g-load and femur load are rapidly produced on the occupant.

These loads can be maintained below the human tolerance level by limiting the maximum belt force with a force limiter (References 7,8, 9 and 10).

Further, as a result of the belt force limitation, the occupant's forward displacement is effectively increased to absorb the occupant's kinetic energy.

Figure 2 shows a sample of the webbing force limiter utilized in the Nissan ESV E2 (References 7 and 8).

The load-elongation characteristic of this webbing force limiter is of two stages. The limiting force in the second stage, (B portion in Figure 2) is lower than that in the first stage (A portion). Since the motion of the occupant's knees is checked by the knee protector, the occupant's effective mass contributing to the motion decreases during the process of the collision. Therefore, the second state with the lower limiting force compensates for the above-mentioned decrease in the effective occupant mass.

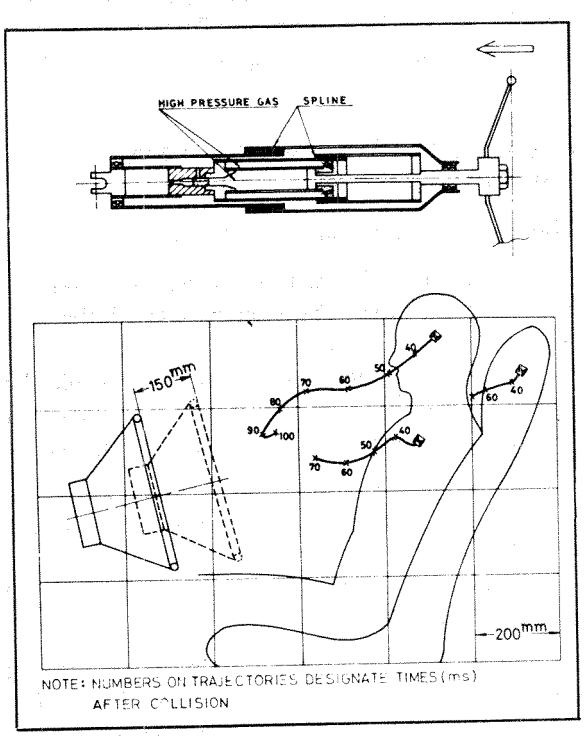


Figure 1. Motions of the driver dummy in the 3-point seat belt restraint system with the auto-retractable steeling and pre-loaders at 50 mph

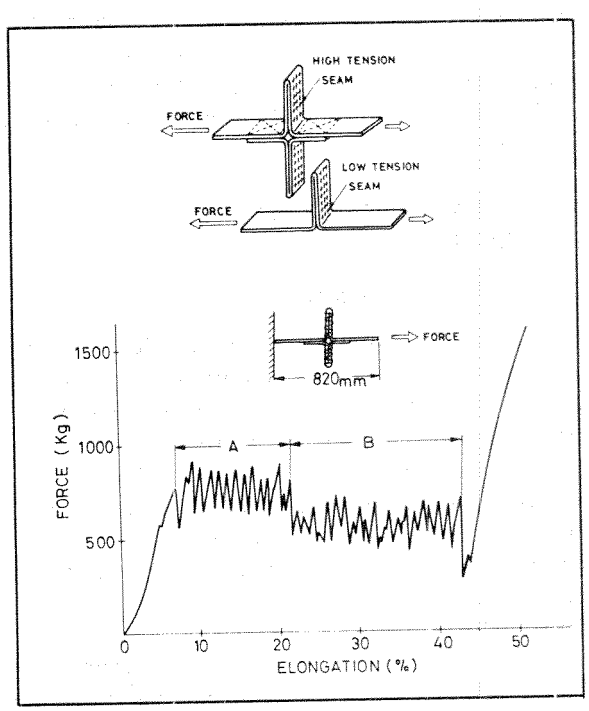


Figure 2. Webbing force limiter

Position of Anchor Points for Seat Belts

In high-speed collisions, the anchor point of the shoulder belt is especially important in occupant restraint performance.

If the anchor point is not appropriate, the shoulder belt may get out of place in a collision.

Therefore, both for ease in belt handling as well as improvement in occupant restraint performance, it is considered most effective for all the anchors of the 3-point belt for the front seat to be installed on the seat (Figure 3 and References 7 and 8).

The torso restraint performance of the shoulder belt in the rear seat can be improved by installing its anchor points on the parcel shelf (Figure 3).

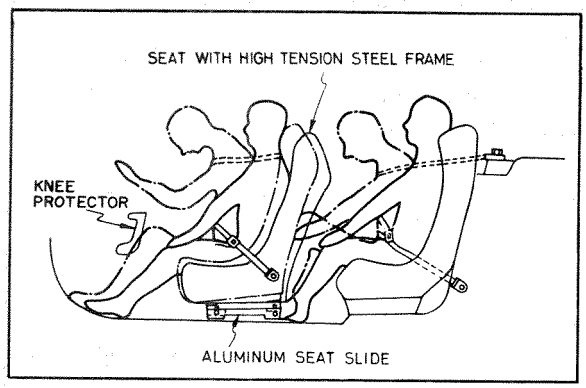


Figure 3. 3-point seat belt restraint system for high speed collision

Knee Protection in Seat Belt System

The purpose of knee protection is two-fold:

- To maintain the femur load at less than the critical load.
- To reduce the severity of injury to the occupant's abdomen by decreasing the load on the lap belt.

The Nissan ESV E2 utilized a knee protector composed of a steel plate with a urethane pad attached thereon.

Vehicle Crashworthiness

According to the results of traffic accident investigations, it is most essential that in a collision the passenger compartment be not so totally deformed as to leave no occupant survival space.

To improve occupant protection performance, however, the vehicle deceleration time history, as well as an increase in the front-end crash stroke, will become key points.

In fixed barrier collisions, the most ideal vehicle deceleration time history for the occupant is the one in which deceleration increases sharply at the initial stage of the collision and then decreases gradually. But a vehicle construction that would effectively realize this is practicably impossible. Thus a square wave becomes the second best deceleration time history.

However, when, in addition to fixed barrier collisions, consideration is given to medium-speed collisions and the problems of compatibility in car-to-car collisions, a square wave cannot necessarily be said to be the next best.

Summary

As previously discussed, there are various ways to improve the performance of seat belts experimentally.

Of these, improvement in belt anchor points, force limiting belts, and knee pads shows little practicable possibilities.

However, a pre-tension device with a radar-sensor and a retractable steering wheel and column with a G sensor are not considered practicable at this time.

At the present time, in consideration of cost benefit, it is most desirable for the occupant to wear the convenient 3-point seat belt properly.

In other words, the experimental method tried in the Nissan ESV E2 has left many practical problems to be solved, in respect to cost, reliability and weight.

INFLUENCE OF DUMMY CHARACTERISTICS, AND THEIR VARIATION

Results of Vehicle Collision Tests

After installing the above-mentioned devices such as the preloader, the auto-retractable steering system, and the 3-point seat belt system with a force limiter, we conducted various tests in order to evaluate 50 mph crashworthiness (References 7 and 8).

For that purpose, two types of dummies, Alderson VIP-50A and ITOH AM50-71, were utilized (Figure 4).

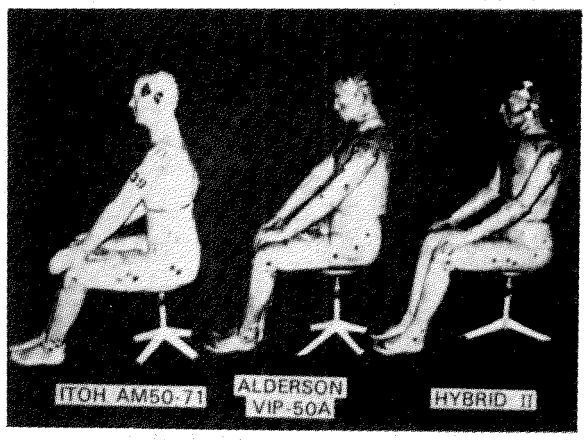


Figure 4. Dummies (except Hybrid II) used in vehicle crash tests

Table 1 indicates the value of deceleration produced on each dummy.

TABLE 1
50 mph FRONTAL IMPACT TEST RESULT

OCCUPANT POSITION / INJURY CRITERIA / UNIT / CAR NO.	DRIVER		ASSISTANT		REAR RIGHT		REAR LEFT	
	HEAD 3ms-G (G)	CHEST 3ms-G (G)	HEAD 3ms-G (G)	CHEST 3ms-G (G)	HEAD 3ms-G (G)	CHEST 3ms-G (G)	HEAD 3ms-G (G)	CHEST 3ms-G (G)
# 1	45	Ⓘ 66	62	Ⓘ 71	81	Ⓐ 52	67	Ⓘ 65
# 2	90	Ⓐ 85	91	Ⓐ 77	63	Ⓐ 40	122	Ⓐ 48

KEY TO SYMBOLS: Ⓐ ALDERSON VIP 50 A
Ⓘ ITOH AM50-71

Figure 5 shows the typical time history of deceleration produced on each dummy with an identical-specification belt. The figure indicates that different types of dummies produce great differences in deceleration value and time history. Even among dummies of the same type, there is such great variation that it is considered difficult to always satisfy the occupant injury criteria in 50 mph head-on barrier collisions.

Further, it can be said that in comparison with the ITOH AM50-71, the Alderson VIP-50A generally tends to show a small value of deceleration at the chest and a large one at the head. Of particular note, the A-P deceleration of the Alderson VIP-50A is substantially increased due to the hyperflexion of the neck.

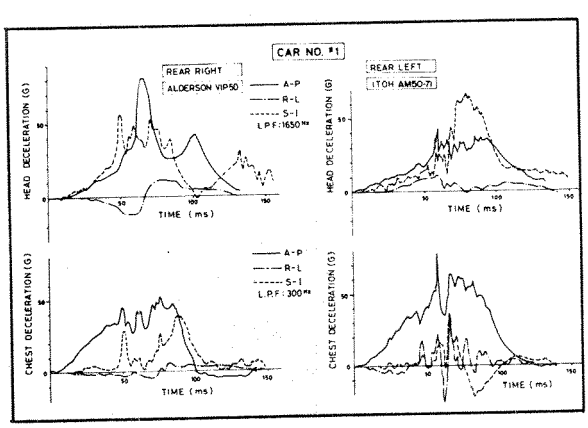


Figure 5. Dummy deceleration time histories

From the above results, it is obvious that any evaluation of the seat belt system to be used in 50 mph collisions is influenced by the characteristics of the dummies themselves. Of the various characteristics of dummies, the following two are the most essential:

- The resistance characteristics of the neck to motion of the head during loading in flexion, especially in hyperflexion with chin-chest contact.
- The dynamic force-deflection characteristic of the dummy chest. The above item will be specifically discussed in the following subsection.

Resistance of Necks in Flexion

Three types of dummies are utilized for a comparative measurement of the resistance of the necks: Alderson VIP-50A, ITOH AM50-71, and the recently obtained Hybrid II.

In hyperflexion causing chin-chest contact, as shown in Figure 6, each type of dummy reveals its own particular characteristic, which is extremely different from that of the others. Particularly in the case of the Alderson VIP-50A dummy, a very sudden increase in torque is caused in hyperflexion, and this fact is considered to greatly affect the A-P deceleration profile of the head.

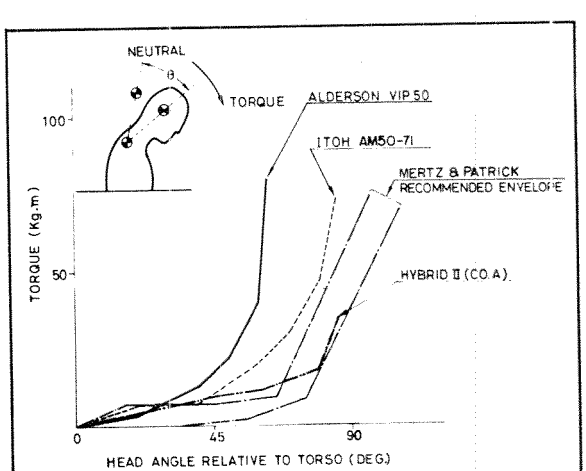


Figure 6. Flexion response characteristics of necks (dynamic)

On the other hand, the resistance characteristic of the Hybrid II dummy is close to the envelope recommended by Patric and Mertz as a result of their tests conducted with volunteers and cadavers (References 11 and 12); it does not show the sudden increase in stopper torque in hyperflexion that the Alderson VIP-50A dummy does.

Comparison of Dummy Head Motions

The purpose of the dynamic tests we mention herein is to measure the value of deceleration produced on each dummy's head and its time history, so as to determine the difference in dynamic head response among dummies. Figure 7 illustrates the test method.

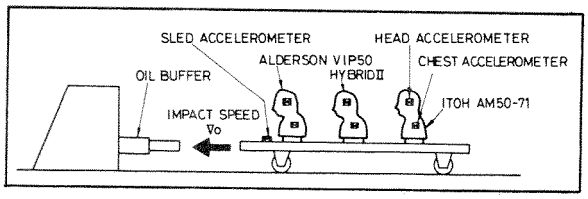


Figure 7. The test method for the dummy head motion analysis

Four types of dummies are used: Alderson VIP-50A, ITOH AM50-71, and two types of Hybrid II (manufactured by Company A and Company B).

In order to examine the influence of chin-chest contact in hyperflexion, the upper thorax of each dummy was mounted directly on the sled table for the impact tester. Owing to this fixed mounting, each upper thorax remained upright, without rotating, while the sled impact took place.

The tests were carried out at various impact speeds ranging from 21 to 52 mph.

Figure 8 indicates the resultant head deceleration, A-P G, S-I G and head angle of each dummy at various impact speeds. The deceleration time history of each dummy will be referred to later.

As can be seen from this figure, the head deceleration of any dummy increases in direct proportion to the velocity in collisions at speeds of up to 30 mph, and increases in a parabolic curve in collisions at speeds over 30 mph.

In the Alderson VIP-50A, sharp A-P G peak begins to appear even at 30 mph because of the influence of chin-chest contact; at 40 mph or more this tendency becomes extremely remarkable.

On the other hand, in the Hybrid II dummy, the A-P G value is small so that S-I G is consistently the major element of the head resultant G; and since there is not such a sudden increase in stopper torque as caused by chin-chest contact in hyperflexion, the maximum head angle is greater than in any other dummy.

The maximum A-P G and S-I G of the ITOH AM50-71 dummy are almost the same in level. In impact response characteristic, this dummy is between the Alderson VIP-50A and Hybrid II dummy.

With regard to head motions, the Hybrid II dummy notably exceeds other dummies in maximum forward displacement. In fact, the Hybrid II dummy's maximum forward displacement, reaching the value of 150 mm, is about 2.5 times as great as that of the Alderson VIP-50A dummy (Figure 9).

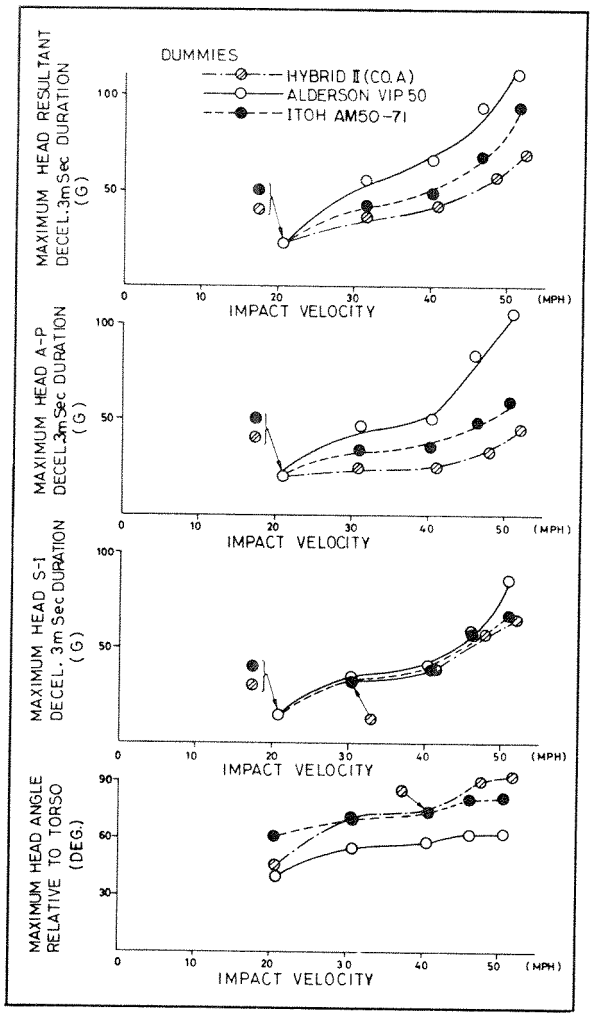


Figure 8. Maximum head deceleration & head angles at various impact speeds

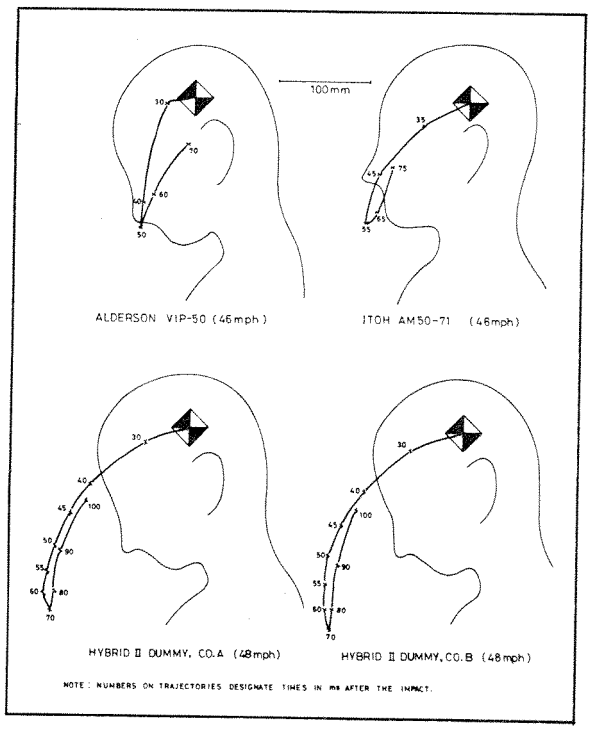


Figure 9. Dummy head motions

Analysis of Computed and Measured Head Motions

Simulation Model. Figure 10 shows a mathematical model for dummy head motions in the dynamic tests mentioned previously. This model is simplified to the utmost in order to make analysis easier and to discover what most contributes towards dummy head deceleration and the time history.

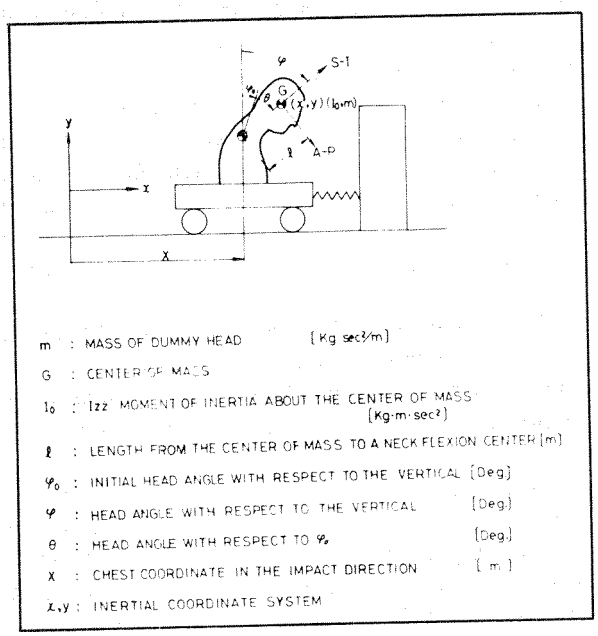


Figure 10. Math-model for dummy head motion

The equations for the motion, the A-P G and S-I G of the head are as follows:

$$\ddot{\theta} = \frac{m\ell \ddot{X} \cos\varphi + T1(\theta) + T2(\dot{\theta})}{m\ell^2 + Io}$$

$$\text{A-P} = \frac{Io \cos\varphi}{m\ell^2 + Io} \cdot \ddot{X} - \frac{\ell}{m\ell^2 + Io}\left\{T1(\theta) + T2(\dot{\theta})\right.$$

$$\text{S-I} = \ddot{X} \sin\varphi - \ell\ddot{\theta}^2$$

Comparison of Computed and Measured Head Motions. The main input data for the mathematical model shown in Figure 10 are the resistance characteristics of the neck in flexion (Figure 6), and the translational deceleration in the X direction ($\ddot{X}$) of each dummy's thoracic vertebra (Figure 11).

The computation of the mathematical model using a simple viscoelastic model T_2 (θ) for pivot torque is essentially in good agreement with the measured data (Figures 12a, 12b and 12c).

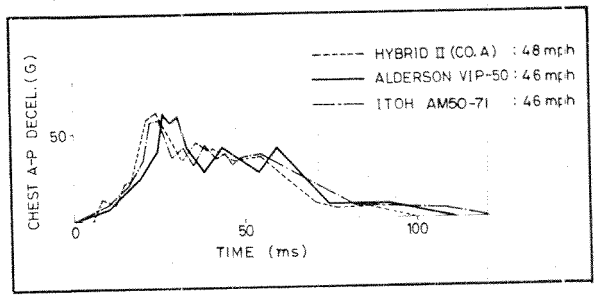


Figure 11. Input data (translational chest deceleration $\ddot{X}$)

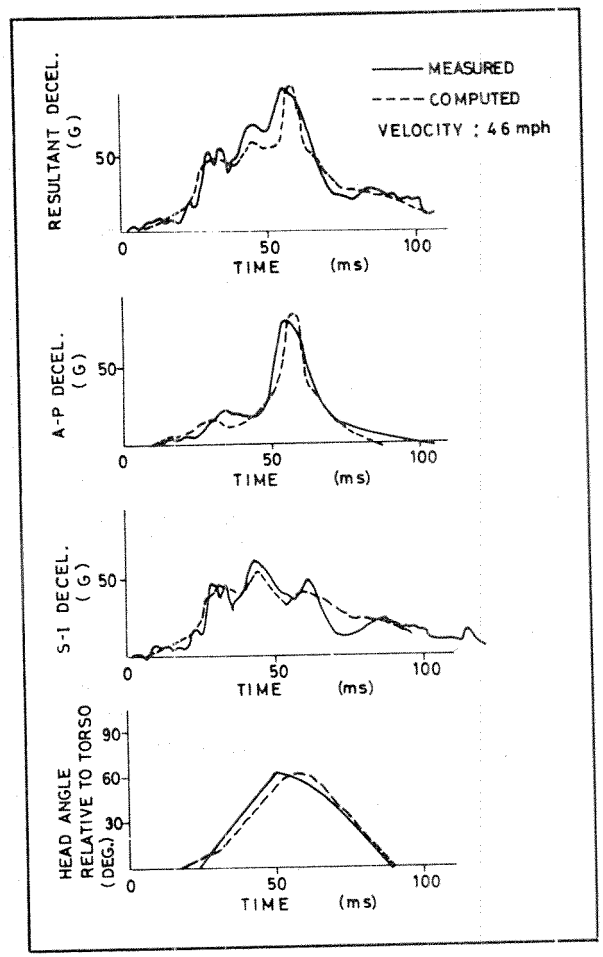


Figure 12a. Comparison of computed and measured head motion (Alderson VIP-50)

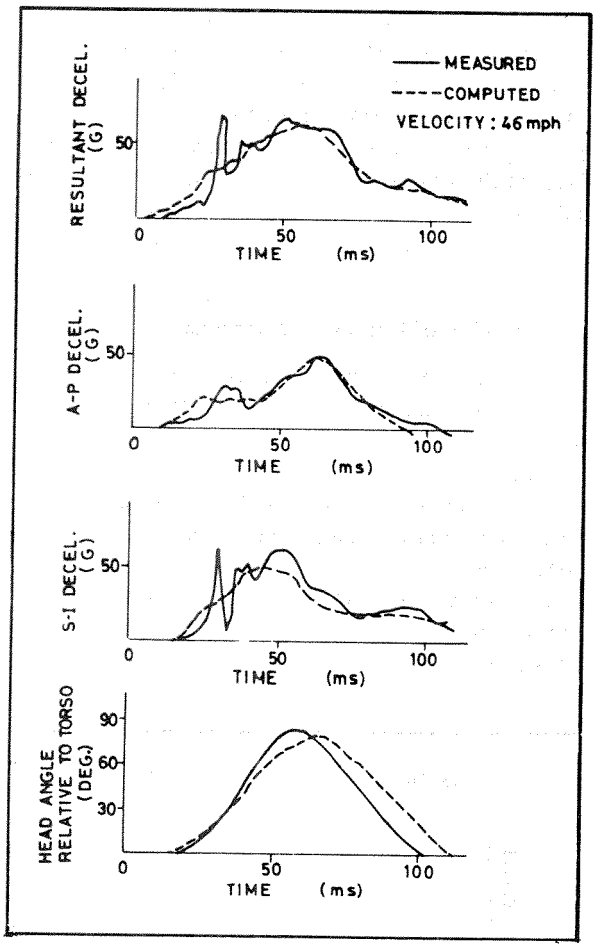


Figure 12b. Comparison of computed and measured head motion. (ITOH AM50-71)

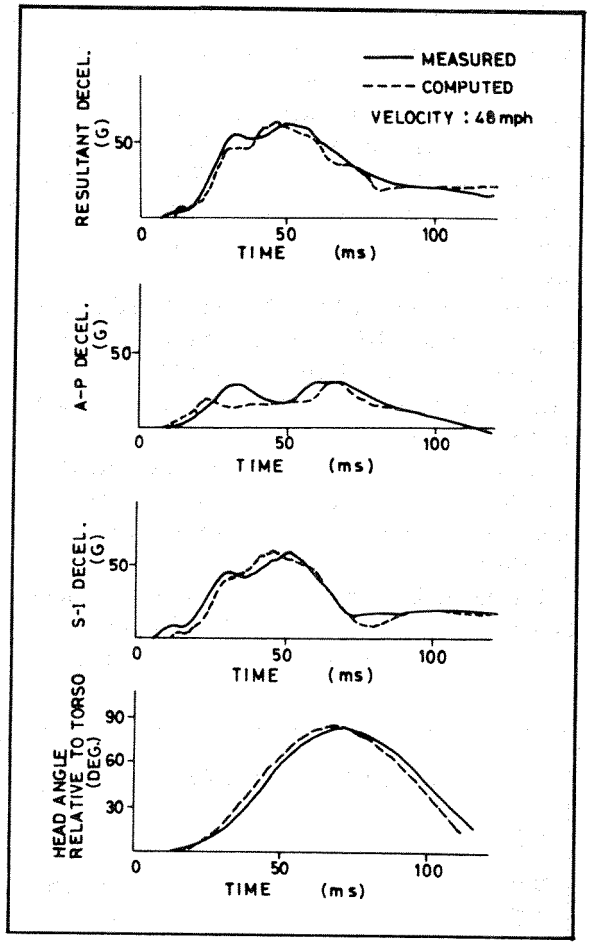


Figure 12c. Comparison of computed and measured head motion. (Hybrid II, Co.A)

Summary. From the previous equations for the dynamic model illustrated in Figure 10, factors causing deceleration on the dummy's head are analyzed as indicated in Table 2.

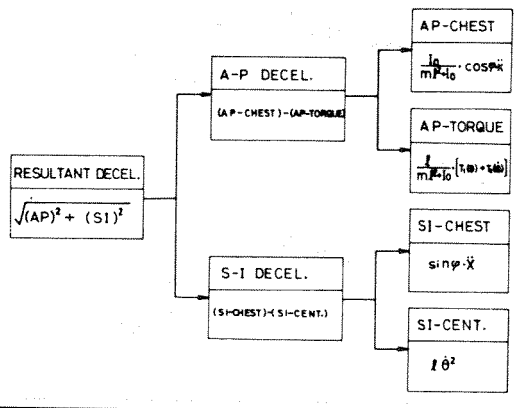
The head A-P G is composed of the AP-chest affected by translational chest deceleration $\ddot{X}$, and the AP-torque affected by the torque resistance of the neck in flexion.

The head S-I G is composed of the SI-chest affected by translational chest deceleration $\ddot{X}$, and the SI-centrifugal force.

In the conventional 3-point seat belt system, which has no special head protector, the four elements of deceleration mentioned above are all influenced by the chest G directly or indirectly.

However, of the four elements of deceleration, the AP-chest and the SI-chest which are directly affected by translational deceleration $\ddot{X}$ will equally increase

TABLE 2
RESULTANT DECELERATION OF THE HEAD AND ITS COMPONENTS



in proportion to an increase in translational chest deceleration. On the other hand, the AP-torque and SI-centrifugal force which are indirectly affected by translational chest deceleration $\ddot{X}$ will not always equally increase in proportion to an increase in translational chest deceleration $\ddot{X}$. They will possibly increase to an extreme, depending upon the torque resistance of the neck in flexion, including viscoelastic torque.

In fact, as Figures 13a, 13b and 13c indicate, the maximum head A-P G in a high-speed collision, especially in the Alderson VIP-50A dummy, is determined by AP-torque, owing to the great torque resistance of the neck in hyperflexion.

In the Hybrid II dummy the maximum AP-torque is smaller than in any other dummy, since there is no such extremely violent chin-chest contact in hyperflexion.

Therefore, differences in head deceleration among dummies are chiefly the result of differences in torque resistance of the dummies' necks (Figure 6).

On the other hand, in the maximum head S-I G of any dummy, the SI-chest is more important than SI-centrifugal force (Figures 13a, 13b and 13c). Accordingly, in the sled tests above, the translational chest deceleration, X, of any dummy whose chest is fixed on the sled table, is almost the same (Figure 11), and therefore the maximum head S-I G of any dummy is also almost the same (Figure 8).

Repeatability of Dummy Performance

So far we have discussed differences in performance among dummies made by different manufacturers.

Apart from this, however, it is most important that the response of a dummy should be reproducible when repeatedly tested under the same impact conditions and with the same restraint system.

As pointed out (References 13 and 14), two aspects of dummy response repeatability must be considered:

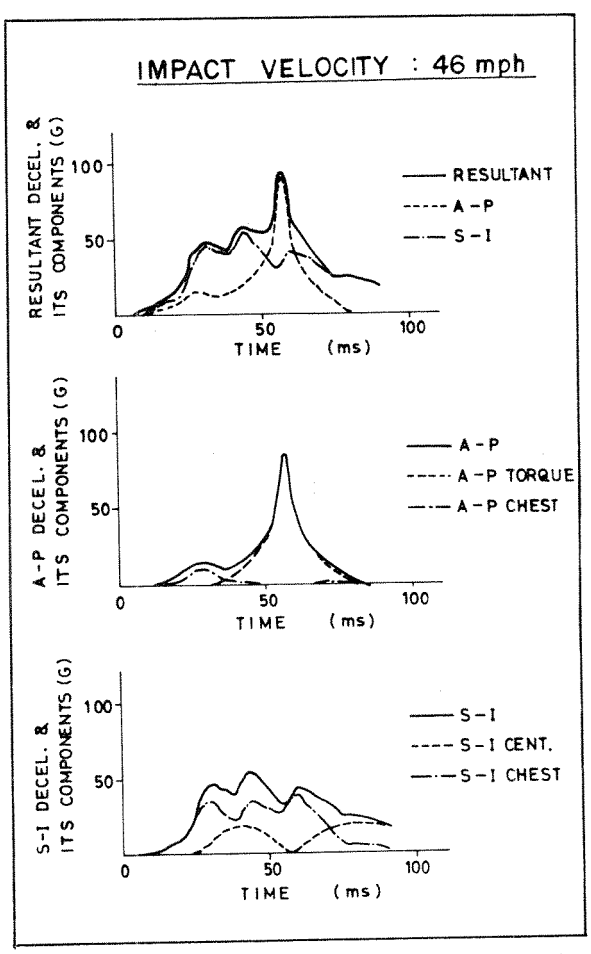


Figure 13a. Time histories of head decelerations and their components (Alderson VIP-50)

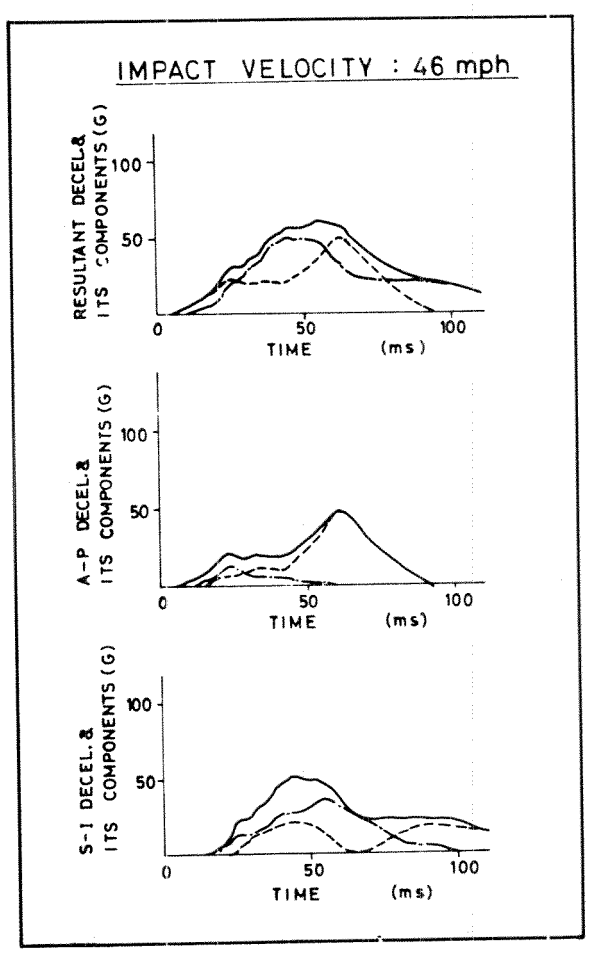


Figure 13b. Time histories of head deceleration and their components (ITCH AM50-71)

- The run-to-run repeatability of a particular dummy.
- The dummy-to-dummy repeatability of two or more dummies of the same manufacture.

Using the sled test method illustrated in Figure 14, we tested the Hybrid II dummy (manufactured by Co. A) and the ITOH AM50-71 dummy, both equipped with the same 3-point belt systems, to examine the run-to-run repeatability at various impact speeds.

For the definition of repeatability, we used the standard deviation (one-sigma) for the maximum resultant deceleration.

At a 40 mph impact speed, one-sigmas for the maximum head resultant decelerations of the Hybrid II and ITOH AM50-71 dummies were 4.9 g and 8.0 g respectively (Figure 15).

At the same impact speed, one-sigmas for the maximum chest resultant deceleration of the Hybrid II and ITOH AM50-71 dummies were 2.9 g and 6.0 g respectively (Figure 15).

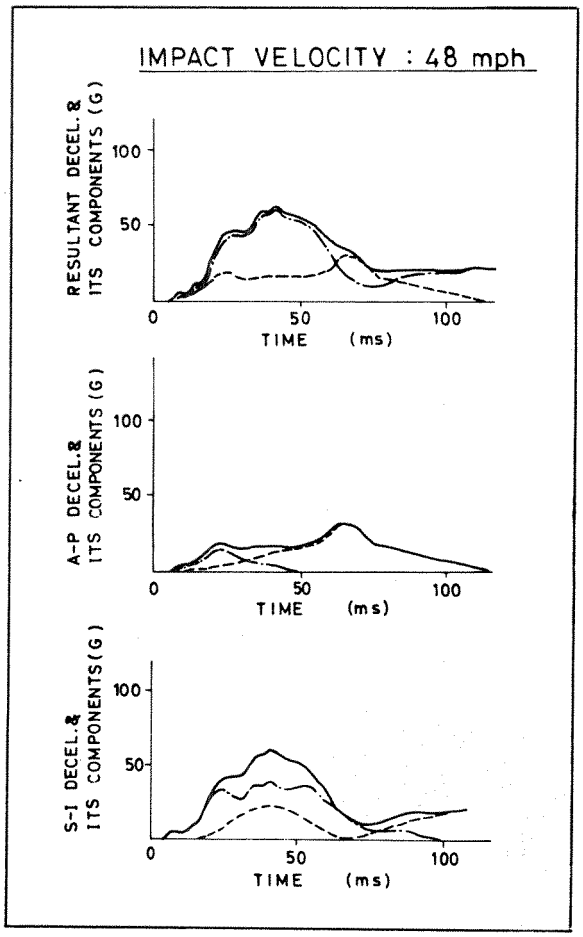


Figure 13c. Time histories of head decelerations and their components (Hybrid II, Co.A)

In general, the repeatability of the Hybrid II dummy is better than that of the ITOH AM50-71, though the repeatability of its head deceleration should be further improved in some respects.

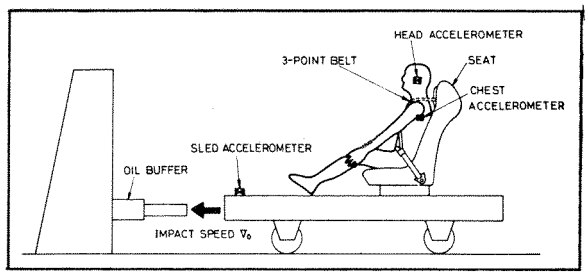


Figure 14. Sled test method for repeatability (3-point belt restraint.)

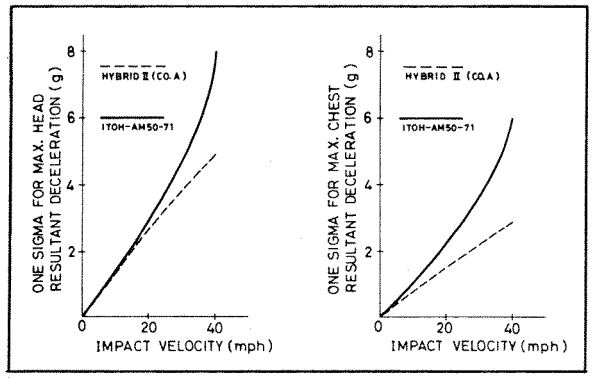


Figure 15. One sigmas for maximum head and chest resultant decelerations at various impact speeds

DIFFERENCE BETWEEN TEST AND ACTUAL COLLISIONS

Consideration of Data Obtained So Far

A great deal of literature (References 12, 15, 16, 17 and 18) has dealt with comparisons in response between dummies and cadavers of human volunteers, both for the whole body and for specific parts such as head, neck, chest, and knee. Satisfactory results, however, have yet to be obtained. There remain several questionable points:

- In our tests of a 3-point seat belt system provided with no particular head protection device, it was found that the dummy's head motions, tending to be affected by the chest motions, etc., vary greatly. It is, therefore, not appropriate to use a dummy to measure the behavior and g-load of the human head.
- As shown clearly in traffic accident investigation results, most occupants wearing seat belts properly are saved from fatal injury in collisions. It follows that if the frequency rate of injury in secondary collisions can be reduced, there will be very few

cases of fatal injury. Therefore, as far as the head is concerned, the g-load is considered to be of no import if the head bends down without hitting any object in collisions.

- Traffic accident investigations reveal that almost all occupants survive in spite of head impact on the front glass in collisions at an equivalent barrier speed of 30 mph (Figure 16 and Reference 19). On the other hand, our 30 mph collision tests with a Hybrid II dummy showed that when the unrestrained dummy's head hit the front glass, the HIC (the proposed SI in MVSS No. 208) reached approximately 1,500. From this fact, the present head injury criteria are considered to be too severe. Therefore, further extensive study is required to determine to what extent dummy injury criteria correspond to fatal human injury and to ascertain the real human tolerance limit.
- Traffic accident investigations conducted in various countries reveal considerable differences in relations between collision speeds and injury levels. For an accurate reading of information on actual accidents, further investigations are considered necessary.
- The responses of the dummies in current use are very much different from those of cadavers and human volunteers (References 12, 15 and 17). However, much is not known in this respect about the Hybrid II dummy.
- As mentioned earlier, the Hybrid II dummy has a larger forward displacement of the head than any dummy in current use. In this respect, it remains unknown whether the Hybrid II dummy resembles cadavers or human volunteers.
- At present, in collision tests with dummies, we evaluate all injuries except femur loads basically by g-load. It is better to evaluate chest injuries by force, deflection or pressure.
- In collisions, human beings are considered to take a defensive posture instinctively to lessen the injury. Of course, dummies do not take such a defensive posture.

The above-discussed difference in response between the dummy and the human being represents a rather desirable tendency for us, when checked against traffic accident investigations. This fact is considered to be an encouragement for continuing our safety research.

Theory of Head Impacts and Human Tolerance

The HIC formula adopted in the proposal in FMVSS No. 208 is based upon the tolerance curve referred to in SAE J855a. That tolerance curve is a modification of the experimentally derived Wayne State Tolerance Curve.

On the other hand, the establishment of a theory of the human tolerance limit to impacts on the head has been attempted by T. Hayashi (Reference 20). His paper deals with a case of linear head impacts where positive pressure and cavitation caused by negative pressure produced in the skull result in fatal brain contusion, bleeding and hematoma.

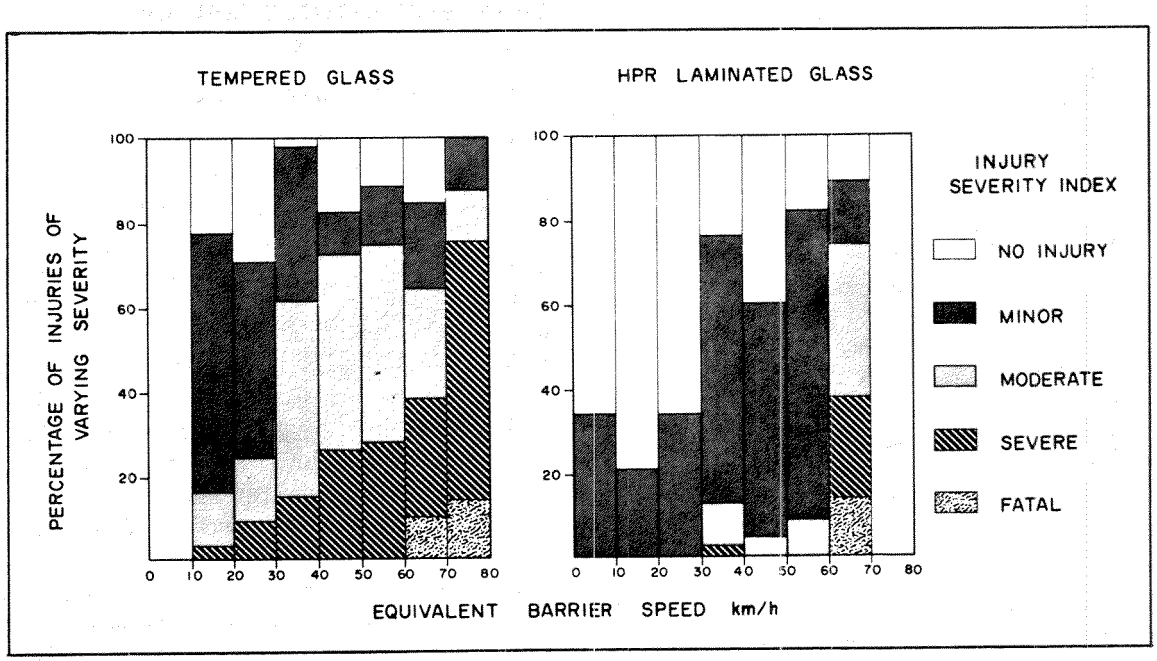


Figure 16. Relative frequency of occupant injury severities from glass and associated mechanisms (by G. M. Mackey, et al.)

In both cases, the frontal impact and the occipital impact shown in Figure 17, the pressure at the forehead is greatest, involving the greatest danger.

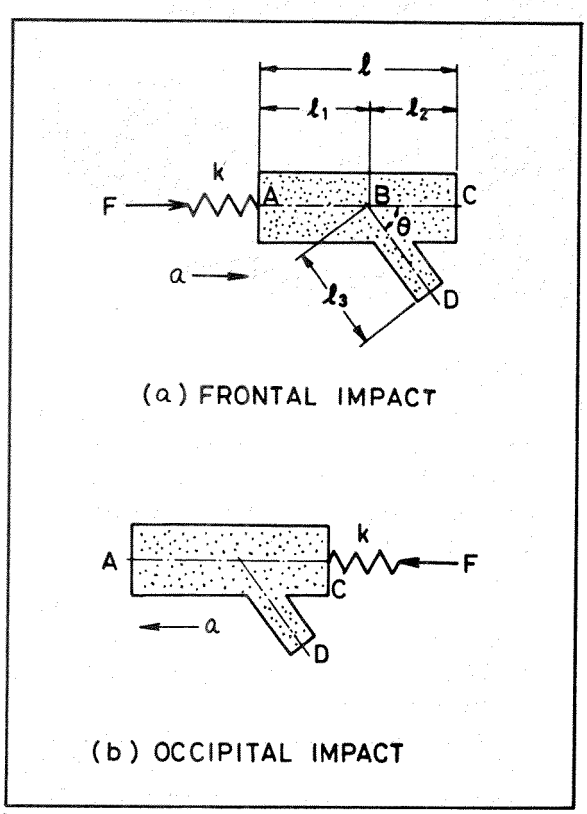


Figure 17. Head model in linear impact theory by T. Hayashi

It is supposed that when the absolute value of that pressure reaches a certain value P*, the fatal limit has been reached.

$$\frac{1}{2}\, a\rho\ell\,(1+a) = P^* \qquad (1)$$

where,

- P* = Critical pressure limit set at 1 kg/cm^2
- a = Linear head acceleration
- ρ = Brain material density set at 1.04 gr/cm^3 (Reference 21)
- ℓ = Total length of skull set at 16 cm (Reference 21)
- a = Coefficient revised for existence of branch tube BD

If we take $a = 0.104$ according to the other reference, then "a" becomes 109 g. In both cases of Figures 17a and 17b, the duration time †D is expressed in the following equation:

$$\dagger D = 2\sqrt{\frac{m}{k}} \qquad (2)$$

where,

- m = Effective mass of head set at 5,000 gr
- k = Elastic spring constant of a unit composed of skull, head skin, and cushioning materials.

From the above two equations, (1) and (2), obtained by T. Hayashi, we can obtain the HIC by the following equation:

$$\begin{aligned} HIC &= a^{2.5} \cdot \dagger D \\ &= 2\left[\frac{2P^*}{\rho\ell\,(1+a)}\right]^{2.5} \cdot \sqrt{\frac{m}{k}} \qquad (3) \\ &\cong \frac{17667}{\sqrt{k}} \end{aligned}$$

The HIC-versus-k curve shown in Figure 18 is the result of plotting equation (3).

This figure clarifies that when the HIC is 1,000, k is 312 kg/cm, which means that the impacted place is considerably hard.

In an ordinary vehicle, the HIC of up to about 1,800 can be tolerated since the spring constant of the instrument panel pad is about 100 kg/cm.

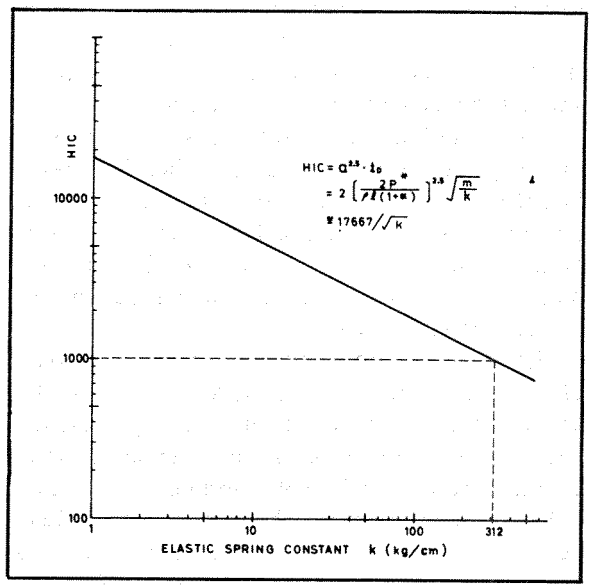


Figure 18. HIC versus k curve in head linear impact theory by T. Hayashi

As can be seen from the above, HIC = 1,000 proposed in FMVSS No. 208 is too severe for application to actual collisions.

PROBLEMS REMAINING UNSOLVED

To meet the requirements in 50 mph frontal barrier collisions, a seat belt system requires a pretension device with a radar sensor, an

auto-retractable steering system with a G sensor, and other devices. When cost benefit is taken into account, it is difficult to manufacture such a seat belt system for practical use.

In evaluation, we find great variation in injury level by dummy type. Even in the same type of dummy, variation becomes greater with increasing speed. The injury level of the head, affected by the dynamic motion effect of the chest and so on, shows especially great variation.

In consideration of secondary collisions, the great forward displacement of the Hybrid II dummy's head presents more problems than that of other dummies. However, we don't know the real displacement of human beings. If it is the same as that of Hybrid II, we shall have to accept it.

It should be noted that if the injury level in secondary collisions is reduced, the occupants will be saved in actual collisions. The present injury criteria are considered excessively severe. Because the design concept changes in accordance with injury criteria, inappropriate injury criteria may cause vehicle design to proceed in a wrong direction as in the ESV project.

A fixed limit to injury criteria presents a problem in itself. Even if injury criteria are fixed and the vehicle is designed in response to that injury criteria, still, in various instances, not all occupants will be saved. Thus, the injury criteria should not mark the limit but should represent one aiming index.

Tests have been conducted at Nissan not only on the 50 percentile dummy but on other dummies to investigate the influence of their sizes and postures. As a result, the 50 percentile dummy was found to be average so far. However, it is considered necessary to more carefully examine to what extent dummy size and posture influence results.

With all the above taken into account, it becomes obvious that the injury criteria obtained by the employment of dummies are not appropriately applicable to the seat belt system even in low-speed collisions, still less in 50 mph collisions. In the future, it is considered necessary that proper injury criteria be set up from comparison of data obtained with dummies and with human beings. Moreover, continuous efforts should be made towards improvement in test method repeatability.

Finally, when new injury criteria are to be established in the future, consideration should be given to the fact that we are compelled to use dummies which are less than adequate replicas of the human body.

Therefore, injury criteria which better reflect those of the human being should be established. We hope that those concerned all over the world will cooperate in making efforts towards this end.

REFERENCES

1. N. I. Bohlin, "A Statistical Analysis of 28,000 Accident Cases with Emphasis on Occupant Restraint Value," Paper 670925, Proceedings of Eleventh Stapp Car Crash Conference, New York: Society of Automotive Engineers, Inc., 1967.
2. W. D. Nelson, "Lap-Shoulder Restraint Effectiveness in the United States," Paper 710077, New York: Society of Automotive Engineers, Inc., 1971.
3. K. Langwieder, "Passenger Injuries in Collisions and Their Relation to General Speed Scale," Paper 730963, Proceedings of Seventeenth Stapp Car Crash Conference, New York: Society of Automotive Engineers, Inc., 1973.
4. J. Michael Henderson and James M. Wyllie, "Seat Belts – Limits of Protection: A Study of Fatal Injuries Among Belt Wearers," Paper 730964, Proceedings of Seventeenth Stapp Car Crash Conference, New York: Society of Automotive Engineers, Inc., 1973.
5. G. Anthony Ryan, "A Study of Seat Belts and Injuries, "Paper 730965, Proceedings of Seventeenth Stapp Car Crash Conference, New York: Society of Automotive Engineers, Inc., 1973.
6. Dr. Claude Tarriere, "Efficiency of the 3-point Belt in Real Accident," Report On the Fourth International Technical Conference On Experimental Safety Vehicles.
7. Technical Presentation from Nissan Motor Co., Fourth International Experimental Safety Vehicle Conference Proceedings, Kyoto, March 13, 1973.
8. Y. Fujiwara, H. Murakami, K. Hirashima, and Y. Ogiwara, "Development of the Belt System of the Nissan ESV," J. SAE Proceedings, 1973 Autumn Conference.

9. Willumeit, H. P., "Passive Preloaded Energy-Absorbing Seat Belt System (Improved Belt Systems)" SAE-Paper 720433.

10. Serge Bohers, "Progress Report on the Second Phase of Research and Experimentation Activities," Report On the Fourth International Technical Conference On Experimental Safety Vehicles.

11. H. J. Mertz and L. M. Patrick, "Strength and Response of the Human Neck," Paper 710855, Proceedings of Fifteenth Stapp Car Crash Conference, New York: Society of Automotive Engineers, Inc., 1971.

12. H. J. Mertz, R. F. Neathery and C. C. Culver, "Performance Requirements and Characteristics of Mechanical Necks," Human Impact Response Measurement and Simulation, Plenum Press, New York-London, 1973.

13. H. T. McAdams, "The Repeatability of Dummy Performance," Human Impact Response Measurement and Simulation, Plenum Press, New York-London, 1973.

14. J. Sam Miller, "Performance Evaluation of the General Motors Hybrid II Anthropomorphic Test Dummy," Contract No. DOT-HS-053-3-603.

15. L. M. Patrick, "Comparison of Dynamic Response of Humans and Test Devices (Dummies)," Human Impact Response Measurement and Simulation, Plenum Press, New York-London, 1973.

16. Voigt R. Hodgson and L. M. Thomas, "Comparison of Head Acceleration Injury Indices in Cadaver Skull Fracture," Paper 710854, Proceedings of Fifteenth Stapp Car Crash Conference, New York: Society of Automotive Engineers, Inc., 1971.

17. T. E. Lobdell, C. K. Kroell, D. C. Schneider, W. E. Hering and A. M. Nahum, M. D., "Impact Response of the Human Thorax," Human Impact Response Measurement and Simulation, Plenum Press, New York-London, 1973.

18. John J. King, William R. S. Fan, and Raymond J. Vargovick, "Femur Load Injury Criteria – A Realistic Approach," Paper 730984, Proceedings of Seventeenth Stapp Car Crash Conference., New York: Society of Automotive Engineers, Inc., 1973.

19. G. M. Mackay, A. W. Siegel and P. V. Hight, "Tempered versus HPR Laminated Windshields: A Comparative Study of United Kingdom and United States Collisions," Paper 700911, Proceedings of Fourteenth Stapp Car Crash Conference, New York: Society of Automotive Engineers, Inc., 1970.

20. T. Hayashi, "Theory of Human Tolerance Limit for Head Impact, Part 3 Dynamical Study of Head Impact and its Cushioning Method," Jour., Japan Soc. Auto. Engrs., Vol. 24, No. 7 (January, 1970) P. 109-112.

21. Dr. T. Ogawa and Dr. H. Hosokawa, "Japanese Brains," Kinbara Publishing Co.

SECTION 3

TECHNICAL SEMINARS

PART THREE – VEHICLE STRUCTURAL PROPERTIES

Chairman: EDWARD M. CHANDLER, Senior Staff Member
Office of ESV Programs
National Highway Traffic Safety Administration
United States Department of Transportation

Themes: Design for Occupant Protection;
Protecting Other Road Users;
Energy Management

EDWARD M. CHANDLER
Senior Staff Member
Office of ESV Programs
National Highway Traffic Safety Administration

Ladies and Gentlemen, it is a pleasure for me to present an overview of activities underway in the United States which are related to efforts to develop better vehicle structures. The subject logically covers four major topics as follows:

1. Structural materials.
2. Car-to-car compatibility.
3. Structure/restraint compatibility.
4. Computer simulation.

Since only ten minutes has been allocated for each overview, I shall attempt to only touch on each of these topics.

STRUCTURAL MATERIALS

From the standpoint of safety, many current structures studies in the United States are aimed at meeting existing and near-term requirements with lighter weight materials or in exploring ways of satisfying more stringent requirements which are anticipated in the next decade. While the advantages of reducing weight have long been recognized, the increased emphasis on fuel economy, in addition to making small cars more popular, has accentuated the search for lighter weight construction in all sizes of vehicles. I am not aware of any startling breakthrough in this field which has been reported in the literature. Instead, there appears to have been an intensification of research and development work in the application of alternate materials.

Aluminum continues to receive attention with the earliest applications being hang-on components which includes many safety related items. The construction of an all aluminum auto body on an experimental basis has recently been reported. It is interesting to note that the introduction to this report refers to the use of aluminum in many European vehicles.

Other alternate materials are also receiving attention. Research is underway on composite materials for possible application in areas where their unique properties may be beneficial. There is also a continuing interest in plastics. However, it appears that the plastics industry may be hard pressed to supply the demand for conventional uses without expanding into new applications.

CAR-TO-CAR COMPATIBILITY

Structures emphasis in the United States is also shifting to include consideration of car-to-car crashes to a greater extent than heretofore. This is a natural and encouraging outgrowth of the ESV program, both at the international level and in the United States. This trend has been hastened by the increased interest in small cars in our country which was generated by the energy crisis of last winter and the threat of a continuing fuel shortage. Therefore, the need for compatibility between all sizes of cars is no longer widely regarded as a problem of only academic interest. In fact, there is a growing recognition that this is a real life practical problem which will become increasingly serious in the United States for an extended period. In addition, there is a growing realization that cars of equal, or near equal, mass are not necessarily compatible.

Concomitant with the shift toward seeking compatibility is the attendant question which always arises in one form or another in the world of engineering and which has been raised in previous ESV conferences. That is, "How can we economically determine with confidence how two or more vehicles will interact in a multitude of collision modes without destroying two or more vehicles in the investigation of each mode?" No simple answer to this question appears to be immediately available. Interest in the United States is high in several approaches. One of these is obviously computer simulation which will be discussed later. Second, or possibly equal priority at the moment seems to be directed toward the development of deformable barriers. Two preliminary investigations have been completed. One of these studied deformable fixed barriers while the other was concerned with a deformable moving barrier. The results of these studies seem to indicate that further work on the moving barrier is more promising. However, it is well recognized that much more work must be done in this area before we can hope to even approximately

mechanically simulate the dynamics of many of the important and complex crash modes. Scale modeling has also been proposed as a potential method of mechanically simulating automobile crashes. Useful results have been reported in determining the performance of components using this technique. To date, however, all of the techniques discussed are supplements to full scale crash testing rather than replacements.

STRUCTURE/RESTRAINT COMPATIBILITY

With the overall goal of reducing occupant injuries and fatalities in automobile accidents, there is a limit to the progress that can be made by dealing with structure alone. It is necessary at some point to introduce the interaction of the restraint with the energy absorbing structure on the one hand and of the restraint with the occupant on the other. The problems related to the interface between the restraint and occupant were included in one of the seminars this morning. Therefore, this seminar will generally confine itself to the compatibility between restraints and energy absorbing structure.

It was not uncommon in the early days of crashworthiness studies to hear that the structural crush distance in the vehicle had been added directly to the interior distance available for occupant travel to determine occupant deceleration distance. The first published attempt to account for the free flight distance lost by the occupant before engagement with the restraint was presented at Stuttgart in 1971 by one of the international participants. Several investigators from the United States have carried this reasoning forward to include consideration of reasonable reductions in vehicle structure efficiency. One such paper will be distributed today by the United States.

As a result of the foregoing considerations, it is anticipated that much more detailed studies will be forthcoming in the United States on the complex dynamic interactions which exist between restraints and structure in both car-to-car and rigid barrier collisions.

COMPUTER SIMULATION

There has been considerable activity in the United States in the development of computer simulation programs to model structural response during vehicle impacts. The advantages of developing such programs are well recognized by this group. I am sure that the group is equally as aware of the computational complexity involved in obtaining detailed quantitative results.

In the area of lumped mass models for the simulation of collinear or right angle impacts, programs are available which give qualitative results that are reasonable. Valuable insight is provided by these programs concerning gross impact behavior.

The usefulness of the lumped mass computer models has been extended by using empirical data for the connecting resistances. In this case, the connecting resistances in the model are identified as specific components or subassemblies which crush under impact. Force deformation data are obtained for the simulated hardware in static crush tests. A dynamic correction factor is then applied to the static crush data. These programs are also limited to collinear or right angle applications. Good results are reported for relative deformations and total crush. Discrete energy dissipation as well as total energy absorption are computed. Reasonable agreement is obtained between computed and empirical average accelerations although the fidelity on the higher frequencies usually is not good.

While the models discussed up to this point are of considerable assistance in estimating or predicting the performance of structures under specific impact conditions, they are not capable of dealing with most of the complex situations that occur in real world collisions. In the accidents that are observed in traffic a large number involve various combinations of partial engagement and angles of impact that are drastically different than the unidirectional translational assumptions upon which the foregoing models are based. For emphasis, let me repeat that in spite of the apparent negative conclusions which I have just reached, these models have been, and will continue to be, very useful in providing engineering insight into many of the fundamental force deformation phenomena that occur during vehicle impacts.

Attempts have been reported to extend the lumped mass models to three dimensions with appropriate changes in the interconnecting resistances. These changes, in essence, are a replacement of the structure by an equivalent three dimensional frame. Without belaboring the details of the assumptions used, this approach at its current state of development has shown promise for only restricted types of impacts under special well known loading conditions.

Planar frame models have been reported and efforts are continuing to refine these programs and to extend them to three dimensions. Also, efforts are continuing on the development of three dimensional finite element programs using nonlinear geometry and providing plastic deformation capability. These programs show considerable potential for use in advanced and detailed simulations. They are

reportedly being used for component and, in some cases, subassembly design at present.

In summary, attention in the United States is centering on investigations of lighter materials, more efficient structures, impact compatibility between cars, as well as between structure and restraints, and simulation of collisions using both mechanical and computer techniques.

RSV, CRASH HAZARDS AND PUBLIC SUPPORT

A. B. KELLEY
Senior Vice President
Insurance Institute for Highway Safety

Few health research programs are more profoundly public than the Research Safety Vehicle program. It embodies a commitment to directly influence the lives and health of millions of people throughout the world.

For the hundreds of thousands of people who may be expected to die in motor vehicle crashes each year under present automotive design conditions, and for the many millions who likewise will be injured and often permanently crippled in those crashes, a useful Research Safety Vehicle program is not a blue sky, other world exercise. It is an urgent practicality – a down-to-earth attempt to find and apply solutions to automotive design deficiencies that needlessly are initiating crashes, aggravating human damage once a crash takes place, and further aggravating the damage after the crash subsides.

That is the description of a realistic RSV program. But it probably is not what the motoring public perceives.

Few of the world's motorists – the millions upon millions of people who drive motor vehicles or ride in them – probably have heard of RSV. Even fewer may have a clear idea of its purpose. And still fewer would view the RSV as having a bearing on their own or their loved ones' lives and future health; they would see it as a futuristic adventure ("The target date for implementation [is] 1985"*) with no influence on *their* chances for surviving *their* crashes next year, or the year after, or even ten years hence.

For the RSV program to be perceived by the public as something more than a remote technological exercise with little near-term practical payoff for motorists, it is essential that the program's goals and results be publicly linked with real-world problems confronting the average motorist in a crash.

This would represent, of course, a dramatic departure from past practice. With a paucity of data and test results available to it, the public historically has had little alternative but to misperceive crash death and injury as being caused by driver behavior that initiated the crash, rather than by vehicle (and roadway) hazards that actually *produced the damage* once the crash was initiated.

As a result the driving population has often been an unwitting accomplice, through its tolerance, in the perpetuation of many needless vehicle design threats to life and limb. One wonders, for instance, whether auto manufacturers would have been permitted for decades to equip their cars with rigid steering columns, designed so that they stabbed and bludgeoned countless thousands of drivers into fatal or disabling injury in front-end crashes, had the motoring public been told how many of its members were being needlessly damaged by these spear-like structures in crashes, and how simple it would be to design steering columns to break or compress under crash-force loadings in ways that help, not hurt, drivers.**

With adequate information about the nature of such automotive threats and the frequency of damage they produce in crashes, the public can be expected to stop tolerating them and start demanding their eradication, just as it has demanded the eradication of public health hazards it perceives in other commonplace components of our environment – air, water, food, and the like.

Within the broad scope of its research activity, the Insurance Institute for Highway Safety has conducted a variety of crash and related test work to analyze structural deficiencies in passenger automobiles, to

*Page 8, Experimental Safety Vehicle Program Quarterly Progress Report, 4th Quarter 1973, U.S. Department of Transportation.

**As it happened, the motoring public – at least in the United States – was not able to count on having energy-managing steering columns in new cars until the U.S. government required them in the late 1960s under Federal Motor Vehicle Safety Standard (FMVSS) 203.

determine the nature of the hazards to human health caused by those deficiencies, and to give the public insights into these hazards as a means of providing it with a baseline for measuring future practical payoff from such programs as the U.S. motor vehicle standards making activity, RSV, and private industry research.

We are not unique in conducting such research; there is every indication that auto manufacturers and, to a less comprehensive extent, government agencies have been carrying out very similar tests, on a much wider scale and for a much longer number of years. (See Appendix A.) A difference is that we place a great deal of importance on making the public aware, through exposure of our test results, that the hazards exist *and* that alternatives for minimizing or eliminating them also exist.

Three of our test series that bear directly on RSV's goals have involved small car crashworthiness, fuel tank system crashworthiness, and general vehicle crashworthiness in very low-speed crashes.

SMALL CARS

The disparity between the occupant protection capabilities of smaller cars and larger cars in comparable crashes is hugh and frightening – and, it must be stressed, not nearly well known enough to the general public.

In head-on collisions between standard-size U.S. sedans and smaller, subcompact-size models, at relatively moderate traveling speeds, the passenger compartments of the latter performed in ways that would provide no protection to occupants. On the other hand, the standard-sized sedan compartments remained virtually intact in the crashes, suggesting a reasonably high level of passenger protection. (See Appendix B for details.)

At the heart of the current RSV program is the stated goal of making smaller cars more crashworthy for their occupants – a recognition, in other words, of real-world data and test results that demonstrate the great differences in crash protection afforded by larger and smaller vehicles. (See Appendix C.) Yet the world's motoring public, including people who are routinely riding and crashing in smaller vehicles, has not been made aware either of the problem's severity or the urgency of the need for RSV to find and apply solutions to it.

FUEL SYSTEM CRASHWORTHINESS

Here we sought to determine the extent of fuel system vulnerability, in new-model cars, to rupture and potentially fire in relatively low-speed rear-end crashes.

In each of our six rear-end impacts, a fuel tank rupture occurred as a result of vehicle design characteristics; in two, fires followed the rupture, one initiated during the crash and one initiated by a roadside flare of the type commonly used by motorists during emergency stops along the highway. (For details, see Appendix D.)

The release of these filmed test results in the U.S. stimulated considerable attention from both the general press and the national legislature, sufficient to prompt adoption of a federal standard prohibiting designs that permit such hazardous fuel system failures (see Appendix E). Although the standard is under attack by some auto manufacturing interests, the National Highway Traffic Safety Administration so far has remained firm in its decision to have it take effect for 1977 model automobiles. Continued public interest in the fuel tank rupture hazard and the new federal standard has been underscored by a recent announcement that the American Broadcasting Company plans to show, on June 12, a one-hour television documentary containing an analysis of both.*

LOW-SPEED CRASHWORTHINESS

The economic hazard of needless material damage in very low-speed crashes has been the subject of a crash test series we initiated in 1969. We have tested popular new cars, of both U.S. and foreign manufacture, to determine the nature and cost of the damage they are designed to sustain in crashes at or less than 15 miles per hour front into barrier. Our latest tests involved cars of the 1974 model year, which performed this way in 15 mile front into barrier and 10 mile per hour intervehicular impacts. (See Appendix F for details.)

One result of the widespread publication of our low-speed test work in the press and on television has been adoption by the U.S. government of standards limiting the amount of damage that front- and rear-end auto designs may be designed to sustain in such minor impacts.** In addition to the existing "no damage to safety parts" standard, the government is proposing a "no damage to any parts" standard for the future.***

In each of the three crashworthiness areas that I have described – areas in which we have done work to identify a needless, designed-in hazard – RSV has

*"ABC Close-Up on Fire: An Update," June 12, 1974, 10:00 p.m. EDT.

**FMVSS 215.

***Proposed standard under Title II of the Motor Vehicle Information and Cost Savings Act of 1972.

staked out goals for solutions, to be reflected in its prototype vehicles. But while the hazards and the solutions are technological in character, it is a matter of policy, rather than technological prowess, whether the public is to be actively invited to understand and support the RSV program on the basis of a clear understanding of the threats to public health that it proposes to eliminate.

A central question is this: *What are the benefits to be gained by seeking a broad base of public support for the RSV program?*

One benefit is that on such bases are large, well-funded, intellectually enriched programs built. Whether to cure cancer or crime, heart disease or airplane crash deaths, no venture into researching and solving the problems of public health can become very ambitious without well informed and reasonably aggressive public support. Governments and sympathetic private sources aren't inclined to invest heavily in indifferently regarded problem solving projects. (Why should they be, when their own voters and board members aren't even aware that the problem needs to be or can be solved?) Nor can top-flight research institutes or the handful of universities that house first-rate scientists be expected to lend their best brains to work on the problem.

That's not to suggest that vast improvements in motor vehicle crashworthiness can't be accomplished by those already at work on the problem – they can, using basic engineering approaches many of which have been well known for years and routinely applied in many nonautomotive situations requiring protection of people and property in impacts. However, a steady infusion of high-quality scientific brains into the vehicle design field might well produce additional, unplanned payoffs not now envisioned by the RSV's prospectus.

Another benefit of strong public support is that it greases the skids for fast translation of research results into applied benefits. Faced with an informed and perhaps impatient public, it's no easy matter for an auto manufacturer or a sluggish government official to delay the introduction on marketplace automobiles of an RSV design innovation. The old shibboleths about "years of lead time needed" and "further research required" don't wash as well with a buyer population that has been told both about the new design *and* the hazard it will eliminate.

Those are benefits that could flow to an RSV program from strong public support. There are, of course, trade-offs.

One is that to cultivate public awareness, the program would find itself much of the time in a goldfish bowl. Each of its failures as well as successes, each of its unmet as well as achieved deadlines, would be the subject of careful public scrutiny. The public at large would have a right to full answers, even before it asked, to such questions as:

- Are current RSV goals realistic, or are they being outstripped by real-world technology? Are "lead times" too short, as some auto makers complain? Or, are they in fact longer than public health needs deserve and the pace of engineering accomplishment justifies? (In this respect, note the considerable recent evidence that passive restraint technology already is outpacing the U.S. government's tentative plans for passive restraint standards for 1980-model cars. See Appendix G.)
- Did every ESV and every RSV, paid for by the government or promised by a manufacturer, meet the criteria set forth for it? If not, why not? Did some meet the standards, and did others fail? Why the difference?
- Were ESV and RSV results translated into design changes in the new-car marketplace, to benefit motor vehicle drivers and passengers? Which ones were so translated? How long after research was completed? Which ones weren't? Why not?

Difficult as it sometimes is for large programs to expose their failings as well as their strong points to public inspection, few good ones have been hurt as an outcome. As for the public, it is always in a better position to support public health ventures such as RSV when it is told about the problems to be solved in no-nonsense, straightforward terms.

CONCLUSION

If the public at large is encouraged, by exposure to public and private research that exposes vehicle crash causation and crashworthiness deficiencies, to adopt high expectations and demands for RSV program payoff, the program will stand a much greater chance of sustained public support for necessary funding and resources, *and* a much greater chance for active public demand for translating RSV results into real-world motor vehicle standards making and design decisions.

the highway
loss reduction

STATUS REPORT

APPENDIX A

INSURANCE INSTITUTE
for Highway Safety

Vol. 8, No. 23 EXCERPT December 20, 1973

NHTSA Urged To Require Auto Maker Crash Data

The Center for Auto Safety has urged the National Highway Traffic Safety Administration to require all crash test data from auto makers on a regular basis.

Auto makers seldom release their crash test data to NHTSA or the public. Such data contain information on, among other things, occupant protection in crash situations.

In a formal request that NHTSA require the data from auto makers, the center said that the agency could use it "in assessing technological capabilities for upgrading of present standards, in providing direction for the development of new standards, and in evaluating manufacturer performance which relates to standards compliance and defect investigations."

Along with its request, the center sent "an 'intra-company' memorandum which categorizes information on Ford Motor Co. tests through June 1968. This document lists 900 separate vehicle crash tests and includes such data as speed, dummy location, type of crash and purpose of test." The Ford memorandum shows that the auto maker also conducted crash tests on cars made by General Motors, Chrysler, American Motors and several foreign manufacturers. The memo is dated June 10, 1968. It deals with crashes conducted prior to that date. A Washington spokesman for Ford refused a *Status Report* request for an updated list of crash tests. The lists are "strictly for internal purposes," he said.

The center also sent a General Motors Corp. report that claims, "In 1970, General Motors conducted 273 full-scale crashes into the barrier. These tests were supplemented by over 900 simulated crash runs on two impact sleds."

The National Traffic and Motor Vehicle Safety Act of 1966 gives DOT authority to require that auto makers provide "performance data and other technical data related to performance and safety as may be required to carry out the purposes of this Act."

Crash test data "would be among the most valuable information that the secretary could request from the manufacturers for the purposes of enforcing safety standards and investigation defects," the center asserted. Auto makers should be required to supply the agency with crash test data on a quarterly basis, the center said.

Earlier this year the government's General Accounting Office recommended that NHTSA make "expanded and systematic use" of manufacturers' certification information. In many cases those data would include crash test performance information. At that time the agency said it would be "naive of us to expect a manufacturer to submit certification data that showed him to be in non-compliance." (See *Status Report*, Vol. 8, No. 10, May 7, 1973.)

The Insurance Institute for Highway Safety is an independent, nonprofit, scientific and educational organization. It is dedicated to reducing the losses—deaths, injuries and property damage—resulting from crashes on the nation's highways. The Institute is supported by the American Insurance Association, the National Association of Automotive Mutual Insurance Companies, the National Association of Independent Insurers and several individual insurance companies.

Document Shows Extensive Fuel Tank Tests

Between 1966 and 1968, Ford Motor Co. conducted at least 250 crash tests involving fuel tanks, according to Ford's "intra-company" memorandum on crash tests. None of these crash test results have been made public. After seeing only six crash tests involving 1973 models, conducted and made public by the Insurance Institute for Highway Safety, the chairman of the House Commerce and Finance Subcommittee, which has jurisdiction over vehicle safety standards, said the tests "glaringly spotlighted" the deficiencies in auto fuel system design.

The 1968 Ford "intra-company" memorandum supplied to NHTSA by the Center for Auto Safety shows that during the same two-year period, the auto maker also conducted crash tests involving child restraints, bumpers, air bags, truck underride, passenger compartment intrusion and other areas of vehicle crashworthiness that were subjects of federal standard-setting or defect investigations activity.

Notations in the Ford memorandum indicate that some of the tests were conducted for "public relations" purposes. The fuel tank tests were not among them.

APPENDIX B

Vol. 6, No. 21 November 16, 1971

Tests Show Small-Car Dangers

Occupants of so-called "economy cars" face dangers in crashes that are "far greater" than those faced by occupants of larger-size cars, according to filmed results of an exploratory crash test program made public today by the Insurance Institute for Highway Safety.

Dr. William Haddon, Jr., president of the Institute, said in a statement accompanying release of the test results that the discrepancy between crash protection afforded by smaller cars and that offered by larger, family sedan-type cars is a "tragically widespread" kind of intervehicular incompatability that "should be of particular concern to society because of the sizable human damage it is producing."

The test series consisted of six two-car head-on crashes, with each car traveling at speeds between 40 and 50 miles per hour — speeds much lower than those allowed and often driven on newer highways today. Each test involved the crash of a small car with a sedan-size car. Seven of the cars were 1972 models; the other five were 1971's, four of which were not instrumented with dummies. Both cars in each crash test were made or marketed by the same company — General Motors, Ford Motor Company, Chrysler or American Motors.

Haddon termed the tests "particularly timely" since "domestic manufacturers, in an effort to compete with the sales of small imported cars, have vigorously entered the field of small car production and marketing . . . ."

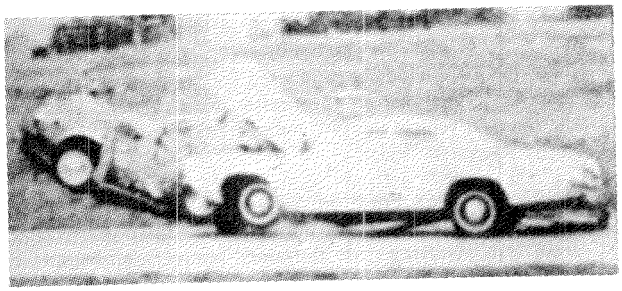
"The public is being heavily exposed to advertising information extolling the purchase-price and operating economies claimed for these cars, but it is not being given vital information about their distinct safety hazards," Haddon noted. He cited estimates that small car sales will represent 50 per cent of all new car sales before 1980.

"As the share of these small cars in the total vehicle population continues to expand on U.S. highways, they can be expected to become involved in a commensu-

rately increasing share of collisions, including collisions with such larger cars as the traditional family sedan," he said.

Head-On

1972 Ford Pinto (left) and 1972 Ford Galaxie at actual moment of impact in a medium-speed crash test. More photos, crash result highlights inside.



The relative lack of crashworthiness of smaller cars in their collisions with larger ones must be viewed, Haddon said, in the context of a "modern day highway environment that mixes large, intermediate and small vehicles at speeds comparable to and often considerably higher than those in the 40 to 50 mile per hour tests" conducted by the Institute.

The problem would not be solved simply by removing larger vehicles from the highway, he said. The "vulnerability of small-car passenger compartments and the amounts of exposure to hazard they permit their occupants . . . cannot be explained away by the size and weight of larger vehicles. We believe, rather, that the relative lack of crashworthiness of small cars is substantially inherent in their design — in the amounts and kinds of spaces and structures they use to shield, or not to shield, their occupants from injury and death," he said.

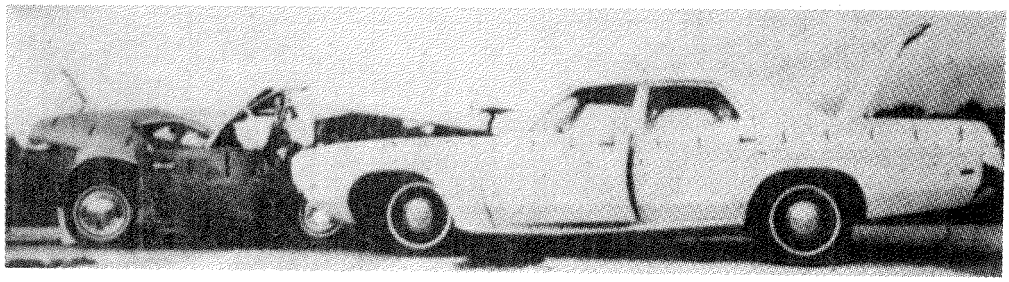
In Brief

How The Tests Were Run

Each test in the medium-speed crash test program involved a head-on crash of a small-size car and a sedan-size car, each traveling between 40 and 50 miles per hour, with a speed differential never larger than four miles per hour in any given test. The test cars were "driven" into each other by test drivers using remote control drive units in nearby "command" cars. In the initial tests — of 1971 Chevrolets and 1971 Fords — the cars were not equipped with dummies; anthropometric dummies were installed in the outboard front seating position of each of the cars in all other tests. Following are the models tested, their weights and their speeds:

SMALL CARS				SEDAN CARS		
MODEL	WEIGHT LBS.*	MPH		MODEL	WEIGHT LBS.*	MPH
1971 Chevrolet Vega	2,300	43.0	← →	1971 Chevrolet Impala	4,107	43.2
1971 Ford Pinto	2,108	46.5	← →	1971 Ford Galaxie	3,973	43.0
1972 Chevrolet Vega	2,526	47.2	← →	1972 Chevrolet Impala	4,362	48.4
1972 Ford Pinto	2,390	46.5	← →	1972 Ford Galaxie	4,188	44.7
1971 Dodge Colt	2,368	47.4	← →	1972 Plymouth Fury	4,208	49.4
1972 American Cremlin	2,885	43.4	← →	1972 American Ambassador	3,914	44.7

*All weights are curb weights as advertised, with the weight of remote control drive units (between 90 and 100 pounds each) added for all cars, and the weight of dummies (215 pounds each) also added for all 1972 cars and the 1971 Dodge Colt.



SMALLER CAR 1972 Ford Pinto	LARGER CAR 1972 Ford Galaxie
OCCUPANT PACKAGE	
Roof caved in; right **door** latch broke, door opened; **windshield** tore loose, fell into passenger compartment onto dummy; **rear window** popped out.	
COMPARTMENT INTERIOR	
Lap and shoulder belts broke; non-adjustable **head restraint** would not position fully behind dummy's head.	
TEST DUMMY	
Head impacted roof, sun visor, windshield, dashboard; lacerations on forehead, right side of face, nose, left hand, right thigh; glass splinters in face; legs pinned under dashboard.	Abrasions on right forearm; lacerations on right palm.
OTHER	
Battery smashed; **fuel tank** ruptured, leaked.	**Battery** smashed.

Haddon noted that "smaller cars increasingly common on today's highways are characterized by sizes, weights and designs that are much less adequate even than currently produced larger cars in terms of:

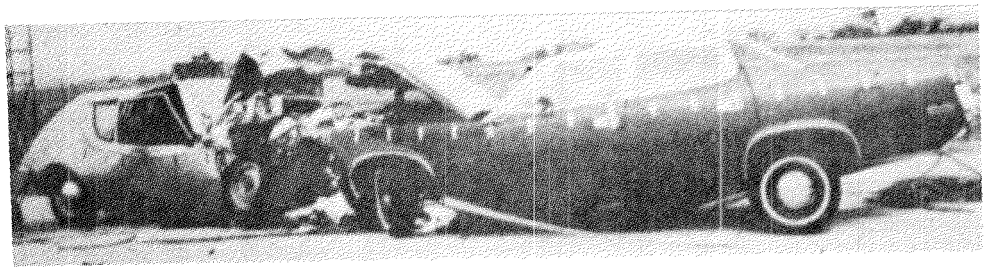
- "Provision of sufficient energy-absorbing structure, external to the passenger compartment itself, to guarantee maximum attenuation of crash forces by means of adequately planned compression or crushing.

- "Maintenance of the integrity of the passenger compartment itself — the package within which the human cargo is located.

- "Protection of the passenger compartment from intrusion by outside objects, such as hoods.

- "Provision of sufficient space within the passenger compartment to keep belted occupants from impacting damaging structure — roof pillars and beams, for example — and also to allow enough spatial depth for padding and other passive restraints."

The Institute's test results are consistent with findings in a New York state study, carried out in 1968 for the U. S. Department of Transportation, that compared injuries from crashes of 420,000 various-size automobiles. The study found a "strong association" between "the weight of a car and the per cent of accidents in which there was a fatality or serious injury in that type of car."



SMALLER CAR 1972 American Gremlin	LARGER CAR 1972 American Ambassador
COMPARTMENT INTERIOR	
Shoulder belt latch broke; nonadjustable **head restraint** would not fully position behind dummy's head.	**Shoulder belt anchor** tore out of roof; adjustable **head restraint** would not fully position behind dummy's head.
TEST DUMMY	
Head impacted roof, sun visor, windshield, dashboard; face severely lacerated; legs pinned under dashboard.	Head impacted sun visor, dashboard.
OTHER	
Battery smashed.	**Battery** smashed.

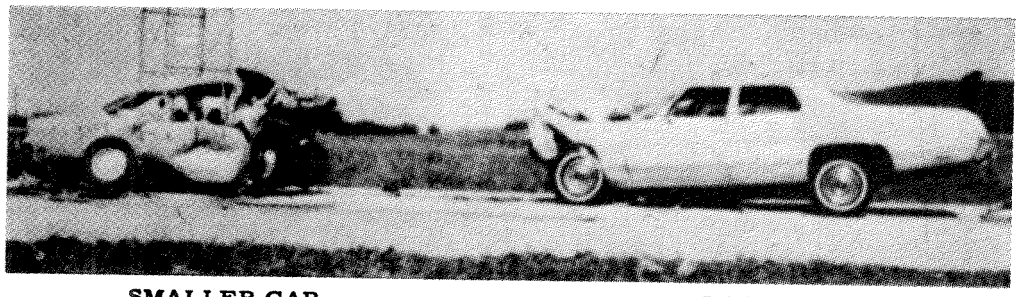
A preliminary report to the Department of Transportation on the New York state study included the following table:

Crash-Involved Size Group	Average Weight	Per Cent Occurrence of Serious or Fatal Injury
1. Domestic "luxury" regular	4,800	3.1
2. Domestic "intermediate"	3,700	4.0
3. Domestic "economy" regular	3,400	5.2
4. Domestic compact	2,800	6.4
5. Foreign compact	1,900	9.6
Combined	-	5.5

In describing the Institute's medium-speed crash test series Haddon pointed out that, although "the tests were not meant to show whether or not a car's safety components were in compliance with the (federal motor vehicle safety) standards," they "may well indicate where standards require toughening."

The film and Haddon's statement identified, for example, head restraints that could not be positioned fully behind the test dummy's head, one head restraint that flew out of the car in a crash, shoulder and lap belts and latches that broke or otherwise failed, three windshields that fell into the passenger compartments of their small cars, and one windshield — that of the 1971 Chevrolet Vega — that was penetrated by the car's hood "like a horizontal meat cleaver."

Data and films from the crash tests have been supplied to the National Highway Traffic Safety Administration, National Transportation Safety Board, Federal Trade Commission and the manufacturers of the vehicles tested.



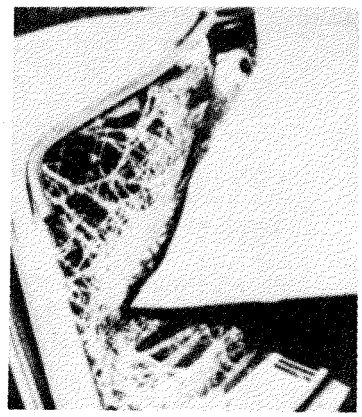
SMALLER CAR 1972 Chevrolet Vega	LARGER CAR 1972 Chevrolet Impala
COMPARTMENT INTERIOR	
Nonadjustable **head restraint** would not position fully behind dummy's head.	Adjustable **head restraint** would not position fully behind dummy's head.
TEST DUMMY	
Head impacted windshield, dashboard; dummy beheaded; legs pinned under dashboard; glass splinters in face; chin, forehead lacerated; arm cut; battery acid on shirt.	Head impacted dashboard, heel of right hand lacerated.
OTHER	
Battery smashed, acid entered passenger compartment.	**Battery** smashed.

'Meat Axe Effect'

Test Shows Need For Rule

The National Highway Traffic Safety Administration will find in the outcome of the crash test of the 1971 Chevrolet Vega and Chevrolet Impala strong support for moving forward with its plan, now in the Advance Notice of Proposed Rulemaking stage (Docket No. 69-17), to set a standard prohibiting penetration of automobile windshields by their hoods in crashes.

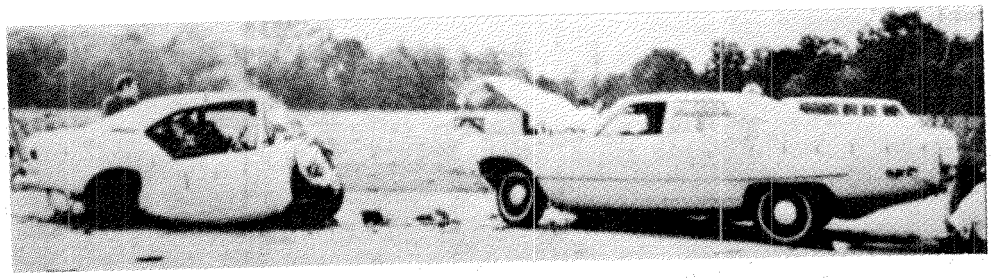
As in this crash, hood penetration of windshields in guillotine fashion exacerbated in small cars, which give the front-seat occupants less room between their heads and the windshield. A person riding in the crash-tested 1971 Vega would have been impacted transversely by the hood edge.



Since November 1969, the NHTSA has kept its advance notice docket open. Both foreign and domestic auto manufacturers' submissions to the docket indicate that the auto industry supports adoption of such a standard. The next step — issuance by NHTSA of a Notice of Proposed Rulemaking — has not yet been taken.

Meanwhile, the agency's accident investigation division has submitted a report of an analysis of 716 serious, real-world crashes looked at by NHTSA multi-disciplinary accident investigation teams. In 467 of these, damage was sustained by the fronts of vehicles.

From this sample the investigation division discovered that in 33 of the



SMALLER CAR 1971 Dodge Colt	LARGER CAR 1972 Plymouth Fury
OCCUPANT PACKAGE	
Windshield partially tore loose, draped into passenger compartment.	
COMPARTMENT INTERIOR	
Lap belt buckle unlatched; **shoulder harness** strap broke; adjustable **head restraint** would not position fully behind dummy's head, flew out of car in crash.	**Shoulder harness** strap broke; adjustable **head restraint** would not position fully behind dummy's head, pushed into "down" position in crash.
TEST DUMMY	
Head impacted sun visor, header, windshield; head lacerated; glass splinters in face; legs jammed under dashboard, broke at hip joint.	Head impacted dashboard.
OTHER	
Battery smashed.	**Battery** smashed.

crashes the rear edge of the hood penetrated the windshield — 4.6 per cent of all the crashes and 7.1 per cent of those with frontal damage. It also found that 44 of the 46 occupants in those cars involved with hood penetration of the windshield were front-seat occupants — that is, those most exposed to the lethal meat-axe effect.

Test Report Available

Single copies of the report detailing the Institute's medium-speed crash test program may be obtained by writing, "Medium Speed Tests," Communications Department, Insurance Institute for Highway Safety, Watergate 600, Washington, D.C. 20037.

Inquiries as to availability and cost of the crash test film should be addressed to "Test Films," Insurance Institute for Highway Safety, at the above address.

Vol. 9, No. 2 EXCERPT January 28, 1974

New Studies Show Small Car Hazards

Two studies recently completed at the University of North Carolina have added to growing evidence that small car occupants face much greater risk of severe injury and death than those faced by occupants of larger cars.

In one study, highway loss researchers documented for the first time that the risk of severe injury and death is about twice as great when two smaller cars crash than when two larger cars crash.

In the second study, they found that belted drivers in sub-compact cars fare as badly in crashes as unbelted drivers in full-size cars.

These and other findings come from two significant studies conducted by researchers at the university's Highway Safety Research Center. The year-long study that yielded the injury data for crashes of like-sized cars, supported by the Insurance Institute for Highway Safety, involved data on 162,000 cars – 1966 through 1970 models – that were involved in crashes in North Carolina in 1966 and 1968 through 1971. That study analyzes injuries sustained by unbelted drivers in cars of various weight classes, both in car-to-car and single vehicle ran-off-road crashes.

The second study compares injury experience of more than 17,000 belted and unbelted drivers in 1970 through 1972 model full-size and sub-compact cars.

WEIGHT AND INJURY

In the first study, entitled "Relationship Between Driver Crash Injury and Passenger Car Weight," the results show that chances of injury and death increase with decreasing vehicle weight. The data tend to show that the chances of severe injury and death increase especially rapidly in cars weighing less than 3,000 pounds.

Similarly, in front-to-rear and front-to-side crashes between vehicles of equal weight the chances of severe injury and death increase more rapidly in cars weighing less than 3,000 pounds than in cars weighing more than 3,000 pounds. In head-on crashes between vehicles of equal weight, however, the relationship between vehicle weight and chance of death and injury is more constant throughout the complete range of vehicle weights.

(The Department of Transportation recently awarded five contracts for development of a "Research Safety Vehicle" in the 3,000 pound range. This is an extension of DOT's "Experimental Safety Vehicle" program that was widely criticized, in part, because the ESV's, weighing about 6,000 pounds, were considered by many to be too large for practical use.)

The researchers report that the association between vehicle weight and occupant injury "seems to decline with newer model cars. This probably reflects the progressive increase in energy management systems in newer cars," the study suggests.

(See charts on pages 3-6 illustrating injury severity by weight class in various types of crashes.)

In the same study, the researchers found, in four of the five model years analyzed, a weak negative relationship between vehicle weight and serious injury in single vehicle ran-off-road crashes. The weak relationship led the researchers to note that "from these data it cannot be concluded that there is a negative relationship between vehicle weight and serious injury in single ran-off-road crashes." Consequently, "pure weight does not seem to be an overriding factor in the single vehicle crash. . . ," the authors concluded.

BELTED AND UNBELTED

In the other study, entitled "Safety, Small Cars and the Gasoline Shortage," comparing belted and unbelted drivers of more recent model sub-compact and full-size cars, the researchers found:

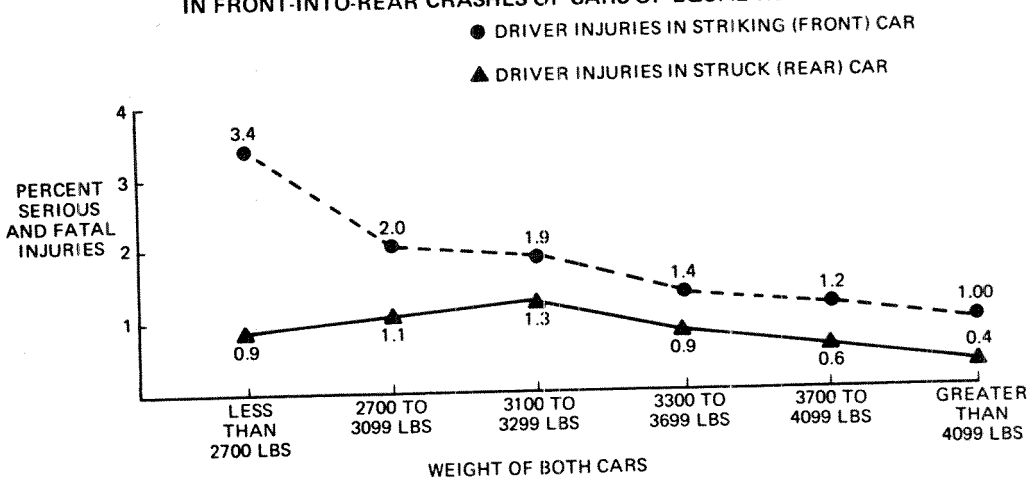
- "Overall the injury picture for belted drivers in sub-compacts is almost identical to the situation for unbelted drivers of standard sized cars."

- In both full-size and sub-compact cars, "drivers who use a belt fare much better than those who do not."

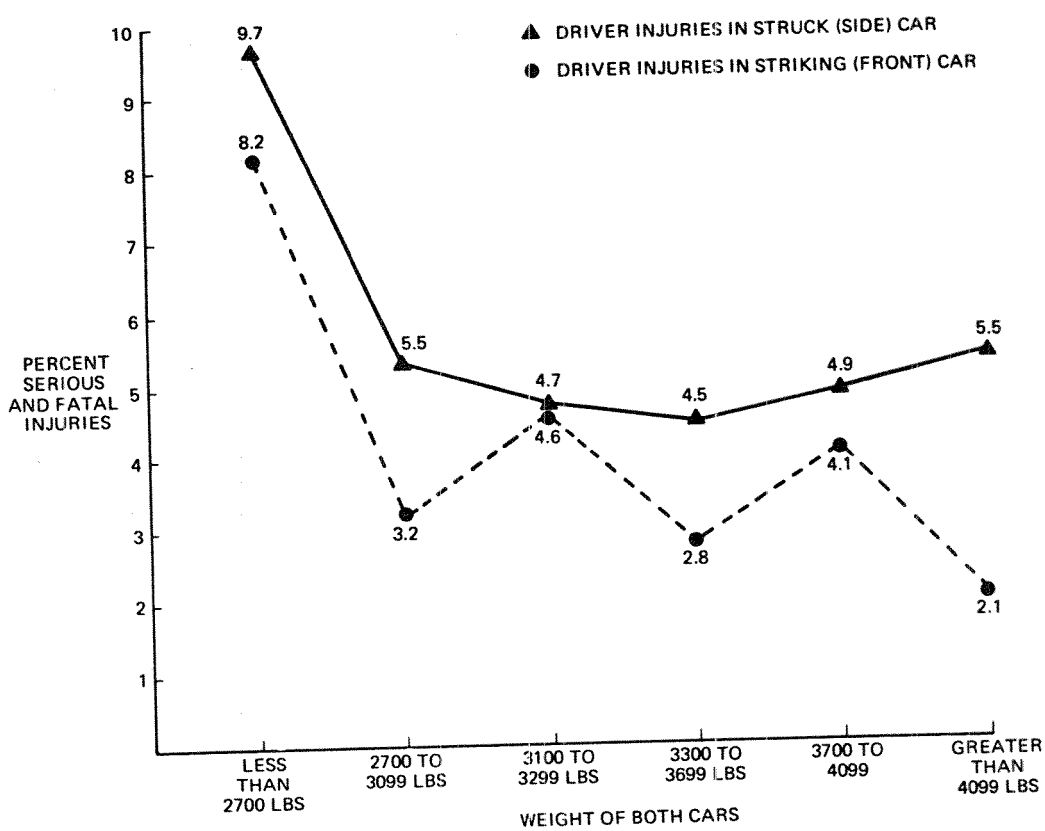
- "For both belted and non-belted drivers, serious injuries in sub-compacts are one and one-half times more frequent" than they are in standard size cars.

Using the injury data developed in the study, the researchers suggest that "if a hypothetical fleet of current standard sized cars were being operated but the drivers rarely used belts, and this fleet were converted to sub-compacts, and then somehow universal belt use were achieved, it would be possible to 'hold the line' on injury. However, if belt wearing rates stayed the same, the resulting injuries would increase by half or more."

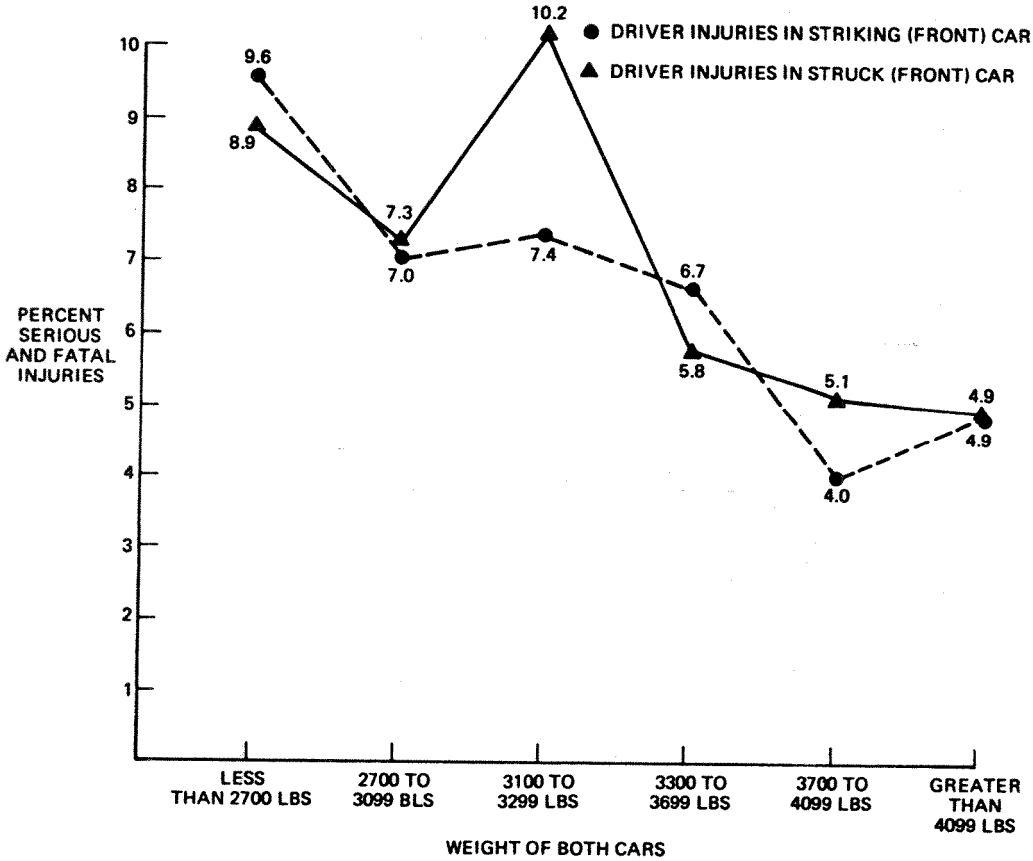
SEVERE AND FATAL INJURIES TO UNBELTED DRIVERS IN FRONT-INTO-REAR CRASHES OF CARS OF EQUAL WEIGHT



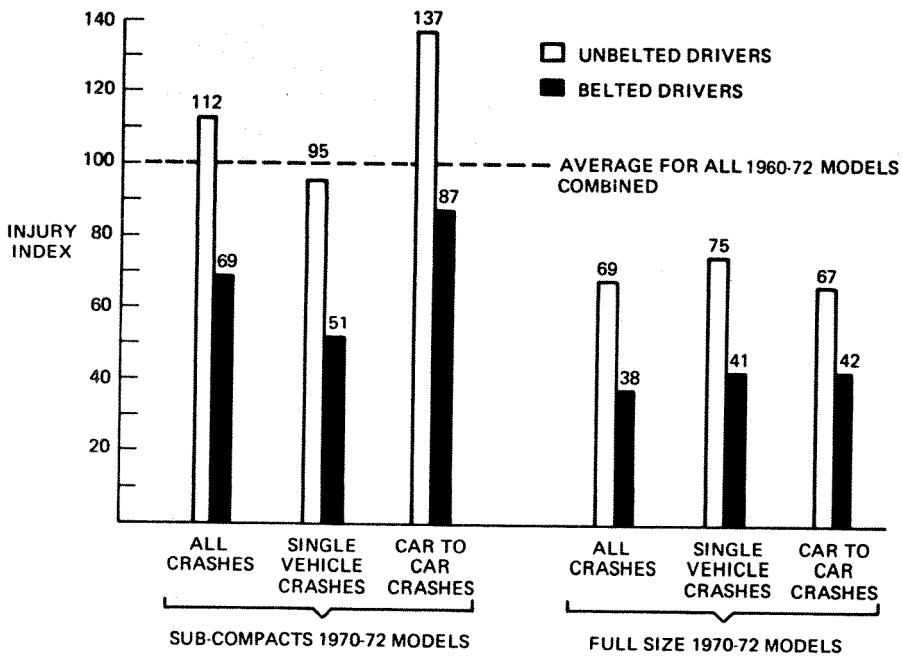
SEVERE AND FATAL INJURIES TO UNBELTED DRIVERS IN FRONT-INTO-SIDE CRASHES OF CARS OF EQUAL WEIGHT



SEVERE AND FATAL INJURIES TO UNBELTED DRIVERS IN FRONT-INTO-FRONT CRASHES OF CARS OF EQUAL WEIGHT



SEVERE AND FATAL DRIVER INJURIES IN FULL SIZE AND SUB-COMPACT AUTOMOBILES



"Projected onto a national scale these additional injuries will too soon mount up to hundreds of thousands." (Injury experience of belted and unbelted drivers in large and small cars is compared in a chart on page 6.)

Information on obtaining copies of both studies is available from the University of North Carolina Highway Safety Research Center, Craige Trailer Park, Chapel Hill, North Carolina 27514.

APPENDIX D

Vol. 8, No. 11 May 29, 1973

Test Results Released

Car Designs Allow Crash Fire Hazard

Fuel tank design deficiencies in even the latest model cars are needlessly exposing drivers and passengers to the possibility of death by burning in moderate-speed rear-end crashes – the kind of crash from which occupants otherwise would emerge with little or no injury.

Reporting on the results of a new Institute test series involving six moderate-speed rear-end crashes of 1973 cars, William Haddon, Jr., M.D., president of the Insurance Institute for Highway Safety, told a subcommittee of the House Commerce Committee that in every test, the impacted car's fuel system failed. "All six of the struck cars had gasoline pouring, gushing or leaking out," he said.

In one of the six crashes, he said, fire erupted spontaneously when "the crash itself provided the spark." In another, fire occurred when technicians provided a source of ignition – "a typical roadside flare."

The Institute decided to undertake the tests, Haddon said, "because we were alarmed by reports, cropping up all the time and in all parts of the country, suggesting that people are frequently, needlessly dying in fires that burst out during car crashes that otherwise would have resulted in little or no injury to those people."

Mute testimony to the hazard posed by inadequate fuel tank design, these manikins were charred by fire that spontaneously engulfed the 1973 Toyota Corona, in which they were riding, when it was crash tested.

The tests, he said, were run with six pairs of new 1973 model cars "typical of those on the highway. The cars' occupants were human-form manikins – four in each car. Prior to testing, each car's fuel tank was filled with regular automotive gasoline and its engine, headlights and accessories were turned on. In each test, the moving car struck the parked one at a speed between 36 and 40 miles per hour, considerably lower than commonplace highway travel speeds. The target cars were parked with their foot brakes on, as if waiting at a stop light."

Haddon showed filmed results of the six crash tests, in which the fuel tank failed in each struck car:

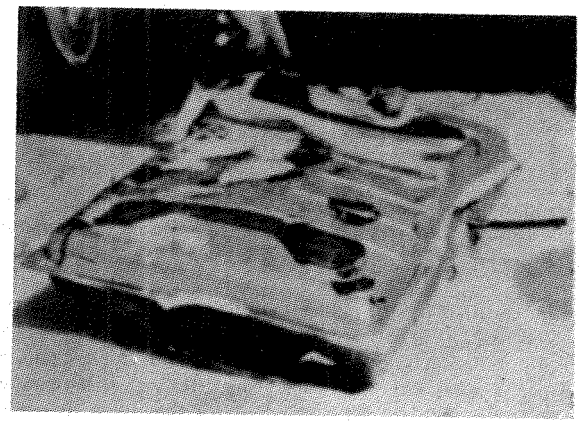
1973 Plymouth Fury III sedan into a 1973 Chevrolet Vega: "During postcrash inspection, we found that in this impact, the Vega's design had allowed its own shock absorber bracket to penetrate and rupture its gas tank . . . (and) we found that the Vega's right front door was jammed – the sort of hazard that could have lethal implications for occupants in a crash fire."

1973 Datsun 610 sedan into a 1973 Ford Pinto: ". . . because of the struck Pinto's design, a corner of its fuel tank was torn by its own differential housing structure in this crash – torn so severely that it leaked gas at the rate of more than a quart each minute."

INSURANCE INSTITUTE FOR HIGHWAY SAFETY
SUMMARY OF RESULTS
1973 MODERATE SPEED FRONT-INTO-REAR CRASH TESTS

MOVING CAR	PARKED CAR	SPEED	GAS LEAKAGE	FIRE
1973 Plymouth Fury III	1973 Chevrolet Vega	39.8 mph	Yes Vega	Potential
1973 Datsun 610	1973 Ford Pinto	38.5 mph	Yes Pinto	Potential
1973 Ford Galaxie 500	1973 AMC Ambassador	37.2 mph	Yes Ambassador	Potential
1973 Volkswagen Beetle	1973 Plymouth Fury III	38.8 mph	Yes Fury	Initiated
1973 Chevrolet Impala	1973 GM Opel 1900	36.4 mph	Yes Opel	Potential
1973 AMC Gremlin	1973 Toyota Corona	39.8 mph	Yes Corona	Spontaneous

NOTE: In addition to the moderate speed front-into-rear crashes of 1973 vehicles tabulated above, in an earlier pilot test a 1959 Oldsmobile 98 was crashed into the rear end of a 1964 Mercury Comet at 39.2 miles per hour. Spontaneous ignition occurred.



1973 Ford Galaxie 500 into a 1973 AMC Ambassador: "In this crash, the struck Ambassador's filler tube was allowed by the car's design to be torn from the hose that connects it to the fuel tank, and the tank itself was exposed to view . . . . A spark would have sent these cars up in flame literally in a split second. (See photo lower right.)

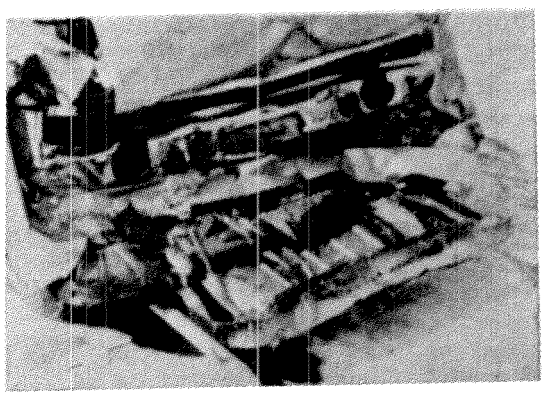
"Like all the test manikins, the ones in the Ambassador were safety belted. Had they been human, their belts probably would have kept them conscious and alert enough during the crash to escape right afterwards . . . only to find, if flames had suddenly engulfed the car, that the doors were jammed shut."

1973 Volkswagen Beetle into a 1973 Plymouth Fury III sedan: "The VW's tank survived its frontal impact into the Plymouth Fury with *no* fuel spillage. The struck Fury's tank, however, failed in the crash. Punctured by two bolts on the VW's license plate bracket, it spewed gas. . . .

"To find out how occupants in this fuel-spilling crash might have fared, we introduced an ignition source – a typical roadside emergency flare – less than a minute after impact. . . . Less than two minutes after the fire started, the fire fighters moved in to put it out. By then it was raging so fiercely that they needed minutes to bring the flames completely under control.

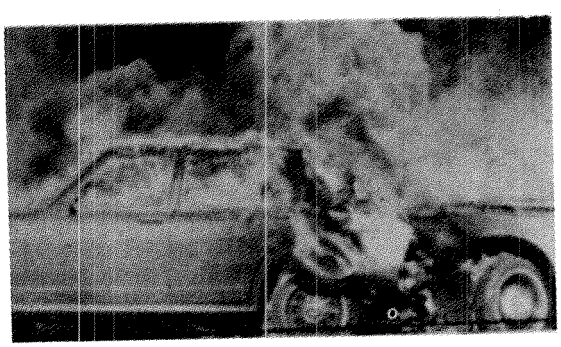
". . . The Volkswagen's gas tank was located in the front of the car, behind the spare tire. This tank was so designed that it successfully resisted damage. It remained intact during the crash, and even survived the fire without rupturing." (See photo, page 5.)

1973 Chevrolet Impala into a 1973 GM Opel 1900: "The Opel's tank, held in place by only one metal strap, was literally torn from the car and hurled to the ground, where it *spewed* gas from a number of failure points. (See upper left photo, page 3.) The tank (was) losing fuel at a rate of nearly a quart per minute. Both the striking Impala and the struck Opel are made by the same manufacturer, General Motors. In a company sponsored study of gas tank ruptures in 1968, GM engineers concluded that 'fuel tanks of General Motors cars are highly satisfactory."



"The inadequacies of its design allowed the Opel tank and its attachments to fail in a number of ways, any one of them a potential fire hazard: a severed fuel gauge line, severed fuel return and vent lines, the severed fuel filler, the broken tank strap and its bracket, and a hole in the tank itself, punched there by the Opel's own rear suspension mounting brackets. . . .

". . . both (the Opel's) right doors were jammed shut, greatly increasing the danger to occupants had the leaking gasoline erupted in flame."



1973 AMC Gremlin into a 1973 Toyota Corona: "Here's the Toyota tank. Design permitted it to rupture . . . a 3/8 inch hole that spilled enough fuel to start and feed a raging inferno. Its fuel cap was also torn off in the crash, creating another escape route for gasoline. . . .

"We did not initiate this fire – it started somewhere in the process of the crash itself, just as fires start in crashes on highways, unplanned, when a fuel system rupture combines with a momentary spark from electrical components, crash friction, road flares or some other source." (See photo, upper right.)

JAMMED DOORS

Recapping the results of the tests, Haddon pointed out that "the design and structure of five of those (struck) cars was so inadequate that one or more doors jammed on impact – doors that otherwise could have been escape routes for occupants. . . ."

He also pointed out that the cars were certified by their manufacturers as meeting the current federal safety standard for fuel tanks. "That standard (FMVSS No. 301) was issued in 1967. It covers front-end crashes, by requiring that in a front-into-barrier impact at 30 miles an hour, a car's tank and system lose no more than an ounce of fuel per minute. But today, six years after issuing its front-end standard, the U.S. Department of Transportation has still not set a standard for fuel tanks in rear-end crashes – the kind, as we've seen, in which these tanks were so vulnerable – not to speak of side, corner and roll-over impacts, in which fuel system failures also kill people.

"All the *striking* cars in our tests emerged without damage to the fuel tanks, including the one whose fuel tank is up front – the Volkswagen."

COUNTERMEASURES DESCRIBED

Haddon told the subcommittee that countermeasure approaches to the fuel tank design problem underscored by the Institute tests "are well known and straightforward. . . . There are many problems in the world for which we do not know solutions, but here is a problem that for many decades we have known how to handle." He gave these examples:

". . . the first approach is to build structure around the tank in such a way that impact forces are less likely to reach it with violence sufficient to cause rupturing, tearing and dislocation. In our tests, the structure behind and around the gas tanks proved in crash after crash, in vehicle after vehicle, to be so flimsy that it provided little protection for the tank itself. We also know that tanks can be made substantially more rugged, so that even when these forces reach them, they do not spill gas.

"And, there need not and should not be automobile designs that place hostile hardware, such as sharp bolts and sharp ridges, adjacent to the gas tank, with resulting rupture or penetration of the tanks in crashes. As you have seen in our tests, these sorts of designed-in threats to the integrity of gas tanks are commonplace.

"Fuel lines, filler pipes and vent lines also can be designed and made in ways that make them far less likely to shear or to break under the impact forces.

"In the event of rupture, thoughtful design can substantially reduce the likelihood of spark – and, therefore, of ignition of the gasoline – by, for example, better insulation and placement of electrical lines.

"And there are design approaches to stop the exploding gasoline from erupting into the passenger compartment and searing its precious human cargo. Chief among these approaches is the provision of firewalls – protective shields that have been built into the fronts of cars for decades, but which are found on the rear ends of only a few vehicles now being sold in the United States. . . .

"This is a principle which, if applied, would stop one aspect of the present situation seen in our crashes and in tragedies all over, where the burning gasoline comes directly from a ruptured tank over or through the back seat upholstery and into the passenger compartment, as you already have seen in one of these crashes.

"Finally, if the design is a humane one, designs and structures are provided to insure that the car doors themselves do not jam under impact conditions, trapping the people inside and locking rescurers out. Nothing is so tragic – and yet they are seen over and over again – as situations in which people, screaming in terror and trying to get out of crashed cars as they are reached by the burning gasoline, find their escape routes cut off by designs that allow this kind of jamming."

Concluding, Haddon told the subcommittee:

"Nobody died in our test demonstrations. But out on the highway, thousands of people will die over the next several years in real-world demonstrations of the lethal design inadequacies represented by the gas tanks of even the newest cars. For years to come, tens of millions of cars with this frightening design deficiency will continue to be travelling – and crashing – on America's roads. No new federal standard can help their occupants. But a tough federal standard made effective promptly would help people in cars that will be made and sold in future years. For them, it's still not too late."

FURY FAILURE – The VW's front-end tank survived this crash undamaged, but the Plymouth's tank ruptured, allowing this blaze.

DOT ON NOTICE

Haddon pointed out that dozens of studies conducted or sponsored by DOT in recent years have given notice that automobile fuel tanks and fuel systems are failing, rather than resisting rupture, in even moderate-speed crashes – with fatal fire too often the consequence. He gave these examples:

1968

Car crash fire "constitutes a hazard that is responsible for more deaths than the lives lost in all types of commercial aircraft accidents annually." Fairchild Hiller, Experimental Safety Car Study, Phase I Final Report, Department of Transportation Contract No. FH 11-6820, Aug. 26, 1968.

1970

". . . about one in 24 collisions produced fuel leakage and/or collision fire." Vehicle Postcollision Considerations, a study of crashes in the Los Angeles area sponsored in part by the U.S. Department of Transportation and authored by A.W. Siegel and A.M. Nahum. Published in the International Automobile Safety Compendium, Society of Automotive Engineers, 1970.

1971

". . . the National Highway Traffic Safety Administration and the Automobile Manufacturers' Association (should) initiate programs leading to the development of automotive fuel-tank systems which will minimize the escape of fuel in collisions." Recommendation of the National Transportation Safety Board on the basis of multiple-vehicle collisions under fog conditions, followed by fire, on the New Jersey Turnpike on Nov. 29, 1969. Report NTSB-HAR-71-3, adopted Jan. 20, 1971.

"An unsafe condition is assumed to exist in the presence of any liquid or vaporized fuel spillage, as potential ignition may be assumed to exist at all times." In the event of fuel spillage in a crash, a fire may be sparked "not only from the electrical and exhaust sytstems of the vehicle, but also from outside sources . . . such items as a flame, for instance, from a burning tire, a lighted cigarette, sparks from a broken power line, and as a prime ignition source during collisions . . . sparks generated from scraping of metallic vehicle parts against the pavement." Johnson, N.B., An Assessment of Automotive Fuel System Fire Hazards, Final Report of Dynamic Sciences under Department of Transportation Contract No. DOT FH 11-7579, Dec. 1971.

1972

"The Board reaffirms that recommendation (for action by NHTSA and the Automobile Manufacturers' Association to make gas tanks more crashworthy) which is further supported by this report of a typical collision accident between two passenger automobiles." National Transportation Safety Board report on its investigation of a two-car collision on the Dulles Airport access road in 1971, in which fire occurred after one car struck another at an estimated speed of 15-20 miles per hour. Report NTSB-HAR-72-1, adopted March 15, 1972.

There has been "no significant reduction of vehicle fires in recent years . . . as highway speeds increase, the likelihood of fire becomes greater. . . .

"As long as motor vehicles use liquid combustible fuels, the hazard of fire will persist. But this hazard can be materially reduced by attacking both the spillage of fuel and the common ignition sources, since combustion requires both a readily ignitable fuel and ignition.

"Therefore, two main goals need to be considered in a systems approach to this problem:

"a. The limiting of fuel spillage . . . at present, fuel system components seem to be among the most vulnerable of all vehicle subsystems in terms of design, construction and placement, when vehicles are involved in crashes; and

"b. Better management of electrical ignition potentialties. . . .

"Both of these approaches are well within the state of the art and should require only direction and development to meet realistic performance parameters." National Transportation Safety Board conclusion on the basis of multiple vehicle collisions and fires in Ventura, Cal., on Aug. 18, 1971. Report NTSB-HAR-72-4, adopted July 6, 1972.

Fuel Tank Rule: A Study In Delay

Six years ago the U.S. Department of Transportation announced a plan to set a standard to protect fuel tanks and lines from fire-threatening rupture in rear-end and side crashes and rollovers. Somehow, the plan has never materialized. As Haddon said in his appearance before the House hearing: "The U.S. Department of Transportation, which has been on notice as to this problem for years, has been legally empowered for seven years under the National Traffic and Motor Vehicle Safety Act of 1966 to mandate a remedy by using a safety standard. Yet still, there has been no action.

- "In October of 1967, the department issued an Advance Notice of Proposed Rule Making (Docket 3-1, subsequently renumbered Docket 70-20) announcing its plan to set a standard to require 'lateral and rear end longitudinal collision tests, prevention of fuel spillage due to rollover, puncture-resistant fuel tanks, and protection of fuel lines and fittings.'

- "...In January of 1969, it issued a Notice or Proposed Rule Making – a more formal step that normally leads to a final standard – proposing that effective Jan. 1, 1970, every new passenger car be required to withstand a 20 mile per hour rear-into-barrier crash without hazardous fuel spillage.

- "...In subsequent 'Program Plans,' the agency revealed that it had no intention of putting the standard into effect before Sept. 1, 1976, if at all."

"...Had that first step been taken – had the standard been adopted – today's new automobiles would reflect designs that tended to protect the integrity of fuel tanks in crashes and, therefore, to protect those cars' occupants against crash fires. But it was not taken," Haddon said.

APPENDIX E

Vol. 9, No. 8 EXCERPT April 16, 1974

NHTSA Issues New Fuel System Rule

The National Highway Traffic Safety Administration has issued a long-awaited broadening of its motor vehicle fuel system integrity standard.

- The new standard requires that fuel systems of all motor vehicles under 10,000 pounds be able to withstand front-angular, lateral moving and rear moving-barrier crashes. After each crash, a rollover test is required. The new standard also specifies the amount and rate of allowable fuel spillage following both barrier crashes and rollovers.

The new standard, issued March 18, 1974, somewhat weakens a set of fuel system requirements and proposals issued last August by NHTSA. At that time Congressman John E. Moss (D-Cal.) called them "unacceptable" and "consumer abuse by government." (See *Status Report,* Vol. 8, No. 17, Sept. 10, 1973.)

NHTSA issued the August fuel system crash test requirements and proposals after Insurance Institute for Highway Safety crash test research showed design deficiencies existed that led to leaking gasoline and fires, even in moderate-speed rear-end crashes. (See *Status Report*, Vol. 8, No. 17, May 29, 1973.)

Because of NHTSA's delay in issuing an improved fuel system integrity standard, Moss and Sen. Joseph Montoya (D-N.M.) have both introduced bills to require the agency to issue an upgraded fuel system crashworthiness standard. (See *Status Report*, Vol. 9, No. 3, Feb. 6, 1974.)

The new standard does not contain requirements for rear corner impact tests. Rep. Moss had asked DOT Secretary Claude S. Brinegar why such requirements were not included when the proposed standards for front-angular, rear and lateral impacts were issued last August. (See *Status Report,* Vol. 8, No. 17, Sept. 10, 1973.) According to a 1972 insurance study, 30 per cent of property damage crash claims involved rear corners of cars.

Under the new standard's lateral impact provisions, the barrier will strike only the doors of a typical full-size four-door sedan, not the areas where the vehicle's filler pipes, filler system or fuel tank are installed.

Although NHTSA had proposed dynamic rollover tests last August, the agency deleted this test from the new standard, noting that "commenters pointed out the exceptional difficulty in measuring or even ascertaining a leakage when the vehicle is rolling over at 30 mph."

The standard does not include provisions for preventing doors from jamming closed during a crash. IIHS crash test research showed that design structure of some cars is "so inadequate that one or more doors jammed on impact." This could result in occupants being trapped in burning cars. A Senate Commerce Committee staff member told *Status Report* that Sen. Vance Hartke (D-Ind.) intends to urge NHTSA Administrator Dr. James B. Gregory to issue a standard to correct this hazard.

Vehicles that use propane or gaseous fuel are exempt from the new standard.

AUTO MAKER REQUESTS DENIED

In addressing other auto maker objections, NHTSA rejected a Ford Motor Co. analysis that, according to the auto maker, showed that the costs of rollover and fuel spillage requirements ($11.20 per car and $12.15 per light truck) would outweigh their benefits. Ford had asserted that – in rollover crashes alone – the consequences of fire "can be estimated as 180 deaths, 180 non-fatal injuries and 2100 other fire crashes." The auto maker said, "Similar analysis for other impact modes would be expected to yield comparable results, with the implementation costs far outweighing the expected benefits."

A Chrysler request that fuel which escapes from the carburetor bowl during a rollover not be included in the amount of fuel discharged from the vehicle was found "unacceptable." A Mercedes-Benz request that diesel-powered vehicles be excluded was also denied. The agency said that a Nissan request that the barrier test crash and the rollover test be conducted with two different vehicles was "without merit." NHTSA denied petitions asking that testing temperatures be between 60 degrees and 80 degrees and not be allowed to vary more than five degrees.

The agency also said that "no justification exists" for a Triumph Motors request that the weight of the 4,000-pound barrier be changed to the curb weight of the vehicle being tested. NHTSA notes that vehicles in use are often over 4,000 pounds in weight and a small vehicle is "as likely to collide with a vehicle of that size as one smaller."

APPENDIX F

Vol. 9, No. 2 EXCERPT January 28, 1974

Side Damage Up

Some 1974 Models More Damage-Prone

Taken on balance, 1974 model domestic cars are "no less damage-inviting than the 1973 versions – and, in some cases, considerably more vulnerable to damage," the Insurance Institute for Highway Safety has told the House Commerce and Finance Subcommittee.

Reporting on the filmed results of IIHS's low-speed crash tests of a range of 1974 model cars of American manufacture, the Institute's president, Dr. William Haddon, Jr., told a subcommittee hearing:

"The designs of many of these new car models resulted, in the very low speed crashes used in our tests, in promoting damage that could have been completely avoided had the vehicles been equipped with front- and rear-end energy absorbing capabilities such as those provided by numerous systems long available to the auto manufacturing industry and long known to Congress, DOT and the public."

Haddon stressed that not every 1974 model performed as expensively as the 1973 version – "proving," he said, "that it is most realistic to expect *all* new cars not only to meet the current, modest

federal standards for bumper performance, but also to be designed at long last to reduce, and certainly not to increase, the avoidable damage costs of low-speed crashes."

The nation is "increasingly squeezed by drains on our natural resources -- including the petroleum-consuming plastics and the metals used for making both the new-car parts damaged in such crashes *and* the countless dollars worth of replacement parts sold by their manufacturers each year to repair them – so it is very reasonable indeed to expect management decisions in auto manufacturing that will discourage, instead of continue and even expand, the use of our shrinking resources for such completely wasteful, completely avoidable ends."

The Institute test results included the following:

- For seven representative domestic sedan and small-car models, the average designed-in damage cost for the 1974 versions at the very low impact speed of five miles per hour front-into-barrier was nearly triple the average cost for the 1973 models – $43.18 as compared with $15.18 – even though nearly all the 1974 models met the prevailing federal bumper standard.

"The 1974 car that performed worst in the five mile per hour front-into-barrier test, the Plymouth Fury, produced nearly $120 worth of such damage," Haddon reported. "The best performers, the AMC Gremlin and Chevrolet Impala, proved the ease with which cars can be designed to permit not even a penny's worth of damage in such an impact."

- In all but one case – again the Plymouth Fury – the seven 1974 models produced substantially less damage on average than the 1973 models in five mile per hour rear-into-barrier crashes, presumably in reflection of the slightly toughened federal standard requiring that these new cars be designed to resist "safety-related" damage in such crashes.

- But in 10 and 15 mile per hour front-into-barrier tests, the designs of the 1974 models encouraged damage costing as much to repair, on average, as for the comparable 1973 models. "Variations in repair costs from model to model were very large, reflecting manufacturers' design choices granting a measure of low-speed crash protection to the buyers of some 1974 models, but withholding it from others," Haddon told the subcommittee. He noted that repair estimates are "based on the conservative labor cost of eight dollars per hour" – the same amount used to estimate damage in IIHS crash tests in the two previous years. He pointed out that the eight dollar hourly rate "is considerably lower than the auto repair rate prevailing in many parts of the country."

"Despite the stated intent of the current federal bumper standard to reduce bumper mismatch in intervehicular crashes, some 1974-model cars are so designed that in our tests their bumpers actually promoted underride and override in jogging speed, 10 mile per hour, front-into-corner crashes, thus aggravating, by design, the damage and repair cost resulting from such commonplace crashes," he said.

"Designed-in, inappropriate vulnerability of automobile sides to needless, expensive damage in very low speed crashes has reached a remarkable high," Haddon said, "an average of $452.66 for side damage in

INSURANCE INSTITUTE FOR HIGHWAY SAFETY
1974 MODEL LOW SPEED BARRIER CRASH TEST RESULTS
DOMESTIC SERIES

		5 MPH FRONT INTO BARRIER	5 MPH REAR INTO BARRIER	10 MPH FRONT INTO BARRIER	15 MPH FRONT INTO BARRIER
FULL SIZE	CHEVROLET IMPALA	0	0	289.02	709.23
	FORD GALAXIE	89.10	0	450.45	992.06
	PLYMOUTH FURY	119.75	95.12	1,061.99	1,783.96
	AMC AMBASSADOR	19.00	81.45	255.98	1,170.33
SUB-COMPACTS	CHEVROLET VEGA	70.40	67.80	441.20	1,071.33
	FORD PINTO	4.00	11.00	268.60	792.30
	AMC GREMLIN	0	0	205.79	675.80
	AVERAGES	43.18	36.48	424.72	1,027.86

front-to-side collisions at 10 miles per hour, compared with substantially lower amounts in the 1969 through 1973 models. Judging from the results of our front-into-barrier tests, this increase cannot be explained away by claims that the front bumpers of 1974 model cars are relevantly tougher than the 1973 versions.

"Across the board, the modest improvements found last year in the damageability of new-car sides have been more than negated by the design and manufacturing choices represented by the 1974 models."

Haddon noted in his testimony that DOT "has no authority to protect American consumers" with standards directed at minimizing the economic penalties of side impacts. "DOT is empowered under the Motor Vehicle Information and Cost Savings Act of 1972 to set such a standard for 'bumper' performance in crashes at low speeds – speeds that DOT has appropriately defined as 20 miles per hour and below. But so far, it has issued only a *proposed* standard, which is limited to five mile per hour impacts," he pointed out.

For detailed damage costs and comparisons, see charts on the following pages.

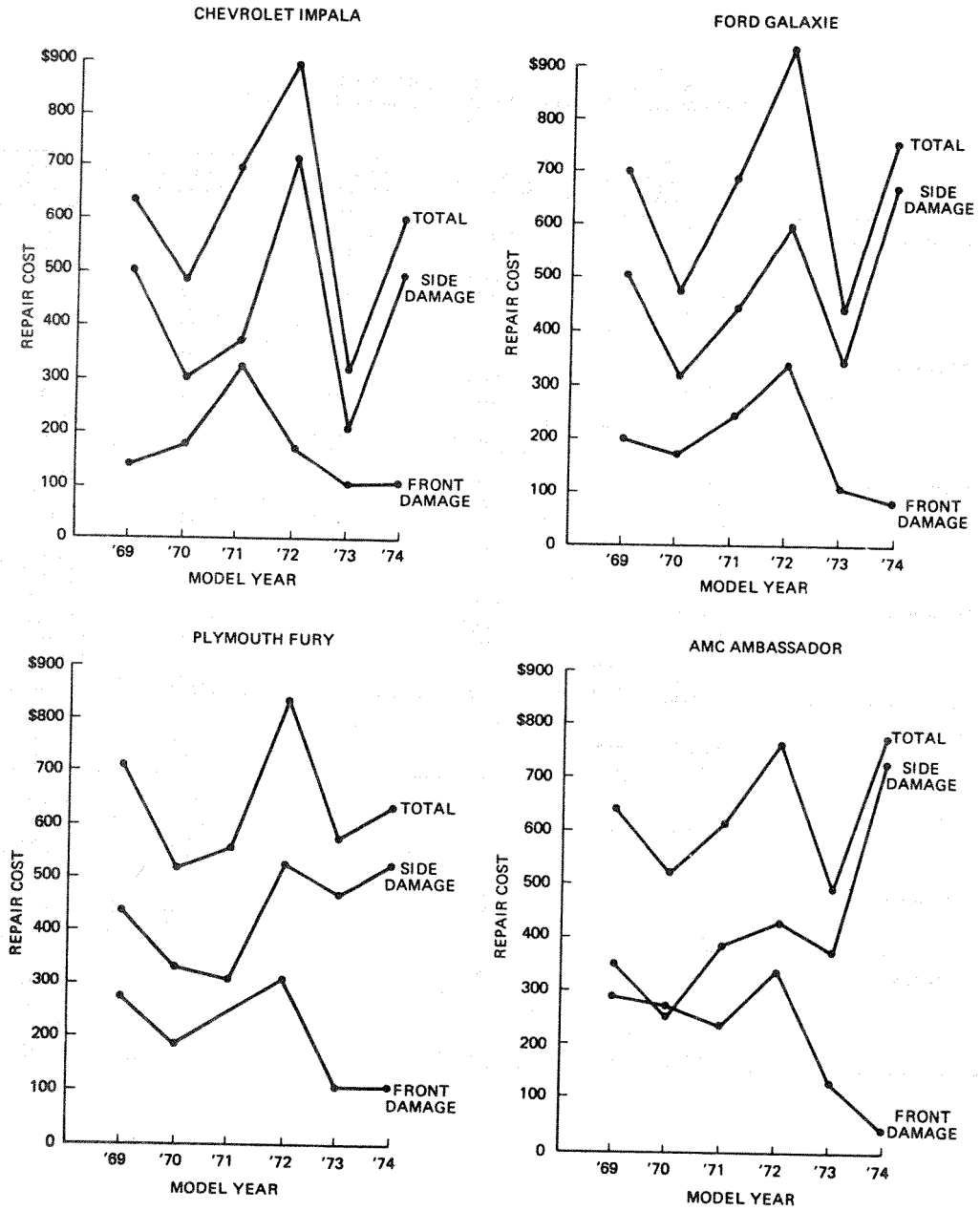
INSURANCE INSTITUTE FOR HIGHWAY SAFETY
1974 MODEL LOW SPEED CAR-TO-CAR CRASH TEST RESULTS
DOMESTIC SERIES

		10 MPH FRONT TO REAR			10 MPH FRONT TO SIDE		
		FRONT DAMAGE	REAR DAMAGE	TOTAL DAMAGE	FRONT DAMAGE	SIDE DAMAGE	TOTAL DAMAGE
FULL SIZE	CHEVROLET IMPALA	124.80	71.15	195.95	109.87	497.50	607.37
	FORD GALAXIE	150.25	83.85	234.10	85.10	676.15	761.25
	PLYMOUTH FURY	227.97	97.92	325.89	110.77	528.07	638.84
	AMC AMBASSADOR	19.00	102.70	121.70	42.00	736.77	778.77
SUB-COMPACTS	CHEVROLET VEGA	89.70	102.00	191.70	125.38	215.00	340.38
	FORD PINTO	72.75	73.10	145.85	152.19	240.80	392.99
	AMC GREMLIN	81.00	71.08	152.08	67.53	274.36	341.89
COMPACTS	CHEVROLET NOVA	–	–	–	–	–	–
	FORD MAVERICK	–	–	–	–	–	–
	PLYMOUTH VALIANT	–	–	–	–	–	–
	FORD MUSTANG II	–	–	–	–	–	–
	AVERAGE	109.35	85.97	195.32	98.98	452.66	551.64

INSURANCE INSTITUTE FOR HIGHWAY SAFETY
1974 MODEL LOW SPEED CAR-TO-CAR CRASH TEST RESULTS
DOMESTIC SERIES

		10 MPH FRONT TO FRONT CORNER			10 MPH FRONT TO REAR CORNER		
		FRONT DAMAGE	CORNER DAMAGE	TOTAL DAMAGE	FRONT DAMAGE	CORNER DAMAGE	TOTAL DAMAGE
FULL SIZE	CHEVROLET IMPALA	79.70	108.30	188.00	137.05	131.80	268.85
	FORD GALAXIE	112.40	379.75	492.15	132.05	160.85	292.90
	PLYMOUTH FURY	177.72	372.56	550.28	195.92	202.30	398.22
	AMC AMBASSADOR	200.18	305.30	505.48	94.53	244.33	338.86
SUB-COMPACTS	CHEVROLET VEGA	120.90	149.43	270.33	94.20	160.30	254.50
	FORD PINTO	79.10	294.05	373.15	110.70	162.25	272.95
	AMC GREMLIN	72.33	145.33	217.66	75.39	143.48	218.87
COMPACTS	CHEVROLET NOVA	92.56	160.41	252.97	167.46	104.30	271.76
	FORD MAVERICK	115.85	266.60	382.45	97.20	286.85	384.05
	PLYMOUTH VALIANT	54.83	263.94	318.77	54.83	224.30	279.13
	FORD MUSTANG II	53.35	379.40	432.75	493.25	230.55	723.80
	AVERAGE	105.36	256.82	362.18	150.23	186.48	336.72

SEDANS – 10 MPH FRONT-TO-SIDE
INSURANCE INSTITUTE FOR HIGHWAY SAFETY CRASH TEST RESULTS



Status Report

January 28, 1974

INSURANCE INSTITUTE FOR HIGHWAY SAFETY
1969-73 MODEL LOW SPEED BARRIER CRASH TEST RESULTS
DOMESTIC SERIES

			2.5 MPH REAR INTO BARRIER	5 MPH FRONT INTO BARRIER	5 MPH REAR INTO BARRIER	10 MPH FRONT INTO BARRIER	15 MPH FRONT INTO BARRIER
FULL SIZE	CHEVROLET IMPALA	1969	–	$187.15	$195.80	$665.80	$1,179.50
		1970	–	196.20	247.30	491.40	740.40
		1971	–	367.90	447.00	828.50	1,170.50
		1972	$112.60	153.75	197.05	576.65	1,133.52
		1973	12.00	0	98.80	281.80	1,035.18
	FORD GALAXIE	1969	–	$174.30	$173.70	$485.00	$854.18
		1970	–	185.80	325.25	459.05	703.10
		1971	–	341.20	318.55	781.50	1,207.45
		1972	$ 20.00	402.10	242.60	917.17	1,243.30
		1973	0	16.00	130.00	417.10	1,008.60
	PLYMOUTH FURY	1969	–	$134.35	$134.40	$644.15	$1,049.25
		1970	–	171.30	202.05	600.05	652.30
		1971	–	202.25	266.35	633.50	870.65
		1972	$ 96.50	331.15	224.50	722.15	1,035.10
		1973	9.97	0	54.37	856.73	1,252.55
	AMC AMBASSADOR	1969	–	$305.15	$352.55	$814.90	$823.27
		1970	–	309.25	100.05	615.75	819.50
		1971	–	415.40	285.20	699.25	1,206.98
		1972	$ 12.00	168.25	159.10	830.50	1,095.90
		1973	20.00	25.60	65.70	263.39	1,112.27
SUB COMPACT	CHEVROLET VEGA	1971	–	$181.30	$228.45	$439.05	$785.60
		1972	$ 71.70	190.90	274.45	619.20	777.41
		1973	36.38	50.27	152.18	420.35	681.69
	FORD PINTO	1971	–	$164.20	$210.00	$535.79	$816.34
		1972	$ 36.80	125.20	267.50	667.05	805.25
		1973	9.60	8.00	195.05	193.40	809.50
	AMC GREMLIN	1971	–	$121.30	$286.90	$576.92	$830.06
		1972	$ 44.80	334.65	235.65	653.35	861.15
		1973	67.65	6.40	124.58	290.32	999.00
	AVERAGES	1969	–	$200.24	$214.11	$652.46	$976.55
		1970	–	215.64	218.66	541.56	728.82
		1971	–	256.22	291.78	642.07	983.94
		1972	$ 56.34	243.71	228.69	712.30	993.09
		1973	22.23	15.18	117.24	389.01	985.54

INSURANCE INSTITUTE FOR HIGHWAY SAFETY
1969-73 MODEL LOW SPEED CAR-TO-CAR CRASH TEST RESULTS*

DOMESTIC SERIES

			10 MPH FRONT-TO-REAR			10 MPH FRONT-TO-SIDE			10 MPH FRONT-TO-FRONT CORNER		
			FRONT DAMAGE	REAR DAMAGE	TOTAL DAMAGE	FRONT DAMAGE	SIDE DAMAGE	TOTAL DAMAGE	FRONT DAMAGE	CORNER DAMAGE	TOTAL DAMAGE
FULL SIZE	CHEVROLET IMPALA	1969	162.30	347.95	510.25	137.10	502.75	639.85	–	–	–
		1970	216.25	205.05	421.30	178.55	303.25	481.80	–	–	–
		1971	280.50	221.05	501.55	328.85	375.30	704.15	–	–	–
		1972	163.35	261.70	425.05	173.90	719.25	893.15	–	–	–
		1973	8.00	94.35	102.35	106.50	211.20	317.70	131.14	272.41	403.55
	FORD GALAXIE	1969	218.70	266.75	485.45	196.95	505.85	702.80	–	–	–
		1970	240.25	273.20	513.45	166.30	312.05	478.35	–	–	–
		1971	248.15	469.60	717.75	241.00	439.35	680.35	–	–	–
		1972	265.65	423.81	689.46	341.70	598.65	940.35	–	–	–
		1973	35.10	213.85	248.95	103.50	339.50	443.00	189.20	225.10	414.30
	PLYMOUTH FURY	1969	310.80	138.20	449.00	273.80	437.00	710.80	–	–	–
		1970	227.40	256.20	483.60	183.60	332.15	515.75	–	–	–
		1971	201.85	246.80	448.65	247.10	306.55	553.65	–	–	–
		1972	269.35	151.25	420.60	310.20	524.75	834.95	–	–	–
		1973	94.97	148.41	243.38	105.46	466.59	572.05	238.91	280.69	519.60
	AMC AMBASSADOR	1969	281.25	314.60	595.85	290.15	351.05	641.20	–	–	–
		1970	446.40	367.40	813.80	270.65	250.90	521.55	–	–	–
		1971	256.30	141.35	397.65	233.25	379.65	612.90	–	–	–
		1972	220.00	280.40	500.40	338.65	421.65	760.30	–	–	–
		1973	130.80	223.30	354.10	126.06	366.49	492.55	268.88	304.32	573.20
SUB COMPACT	CHEVROLET VEGA	1971	276.55	244.60	521.15	191.05	195.90	386.95	–	–	–
		1972	250.53	273.75	524.28	128.35	218.30	346.65	–	–	–
		1973	224.12	233.88	458.00	70.66	181.75	252.41	172.61	209.34	381.95
	FORD PINTO	1971	183.35	196.10	379.45	151.90	244.15	396.05	–		–
		1972	176.55	263.10	439.65	145.92	197.50	343.42	–	–	–
		1973	59.30	114.50	173.80	47.00	158.30	205.30	58.70	229.55	288.25
	AMC GREMLIN	1971	253.95	137.65	391.60	172.00	329.65	501.65	–	–	–
		1972	243.74	108.85	352.59	233.70	314.95	548.65	–	–	–
		1973	68.05	116.38	184.43	104.40	291.79	396.19	366.13	327.60	693.73
	AVERAGES	1969	243.26	266.88	510.14	224.50	449.16	673.66	–	–	–
		1970	282.58	275.46	558.04	199.78	299.59	499.36	–	–	–
		1971	242.95	236.74	479.69	223.59	324.36	547.96	–	–	–
		1972	227.02	251.84	478.86	238.92	427.86	666.78	–	–	–
		1973	88.62	163.52	252.14	94.80	287.95	382.74	203.65	264.14	467.80

*REAR CORNER IMPACT TESTS NOT CONDUCTED FOR THESE MODEL YEARS

Copies of the complete testimony are available by writing to "1974 Model, Domestic," Insurance Institute for Highway Safety Watergate Six Hundred, Washington, D.C. 20037.

Crash Test Results On 1974 Imports Reported

Although imported 1974 cars are somewhat less damage-prone than last year's models, they still suffer "needless, costly damage" in low speed crashes, especially when sides and corners are involved, according to testimony given to the Senate Committee on Commerce today.

William Haddon, Jr., M.D., president of the Washington-based Insurance Institute for Highway Safety, told the committee during motor vehicle safety oversight hearings that low speed crash test results of representative 1974 imports "are encouraging in their indications that somewhat less damage-prone designs have been adopted by some foreign-car manufacturers, but not so encouraging in their indications that much needless, costly damage still is being designed into, in particular, the corners and sides of the tested models."

Haddon pointed out that "DOT has no authority to protect consumers with standards directed at minimizing the damageability of sides and the consequent economic penalties of side impacts – including the forced purchase of untold millions of replacement parts. DOT is empowered under the Motor Vehicle Information and Cost Savings Act of 1972 to set such a standard for 'bumper' performance in crashes at low speeds – speeds that DOT has appropriately defined as 20 miles per hour and below. But so far, it has issued only a proposed standard, which is limited to five mile per hour front and rear barrier impacts."

In its tests of the 1974 imports, IIHS crashed six models. The Volkswagen Superbeetle, Volvo 142, Toyota Corolla, Opel Manta and Datsun 610 were tested in front-to-rear, front-to-side and barrier crashes.

INSURANCE INSTITUTE FOR HIGHWAY SAFETY
1974 MODEL LOW SPEED BARRIER CRASH TEST RESULTS
IMPORTED SERIES

	5 MPH FRONT INTO BARRIER	5 MPH REAR INTO BARRIER	10 MPH FRONT INTO BARRIER	15 MPH FRONT INTO BARRIER
VW SUPERBEETLE	$ 5.30	$ 24.90	$391.45	$828.50
VOLVO 142	15.56	31.14	235.92	924.10
TOYOTA COROLLA	99.66	81.73	404.52	986.95
OPEL MANTA	13.20	0.00	221.65	816.30
DATSUN 610	0.00	0.00	232.40	949.65
AVERAGE	$ 26.74	$ 27.55	$297.19	$901.10

INSURANCE INSTITUTE FOR HIGHWAY SAFETY
1974 MODEL LOW SPEED CAR-TO-CAR CRASH TEST RESULTS
IMPORTED SERIES

	10 MPH FRONT-TO-REAR			10 MPH FRONT-TO-SIDE		
	FRONT DAMAGE	REAR DAMAGE	TOTAL DAMAGE	FRONT DAMAGE	SIDE DAMAGE	TOTAL DAMAGE
VW SUPERBEETLE	$208.15	$126.70	$334.85	$124.65	$386.70	$511.35
VOLVO 142	37.32	46.43	83.75	6.04	462.78	468.82
TOYOTA COROLLA	114.27	227.47	341.74	78.29	412.34	490.63
OPEL MANTA	0.00	0.00	0.00	47.90	545.50	593.40
DATSUN 610	280.63	19.00	299.63	207.51	294.27	501.78
AVERAGE	128.07	$ 83.92	$211.99	$ 92.88	$420.32	$513.20

INSURANCE INSTITUTE FOR HIGHWAY SAFETY
1974 MODEL LOW SPEED CAR-TO-CAR CRASH TEST RESULTS
IMPORTED SERIES

	10 MPH FRONT-TO-FRONT-CORNER			10 MPH FRONT-TO-REAR-CORNER		
	FRONT DAMAGE	CORNER DAMAGE	TOTAL DAMAGE	FRONT DAMAGE	CORNER DAMAGE	TOTAL DAMAGE
VW SUPERBEETLE	$ 42.55	$173.55	$216.10	$121.40	$343.93	$465.33
VOLVO 142	3.56	286.00	289.56	67.25	128.36	195.61
TOYOTA COROLLA	124.77	278.50	403.27	180.96	182.25	363.21
OPEL MANTA	0.00	182.15	182.15	91.55	372.70	464.25
DATSUN 610	296.54	365.34	661.88	442.83	153.22	596.05
MAZDA RX-2	31.90	276.11	308.01	41.56	352.38	393.94
AVERAGE	$ 83.22	$260.28	$343.50	$157.76	$255.47	$413.07

Those five models, and the Mazda RX-2, were corner-crash tested. In 1973 the Datsun 510 and Opel 1900 were tested. Since these were not offered in 1974, the Datsun 610 and the Opel Manta were tested.

Haddon told the committee that:

- "With one discouraging exception – the Toyota Corolla – the representative makes of imported cars that we tested both for the 1974 and the 1973 model years showed, for the current models, reduced vulnerability to damage in their impacts of 5 miles per hour front-into-barrier. Moreover, all five of the current models showed reduced damageability in their impacts of 5 miles per hour rear-into-barrier. The Datsun was so designed that it experienced no damage whatsoever in either test, and the Opel – whose 1973 model was a poor performer in similar tests – experienced no damage at all in the rear-into-barrier test and only a small amount in the front.

- "Results were mixed for the same five 1974-model foreign cars in their 10 mile per hour front-into-barrier tests. None of this year's models was as damageable as the 1973 Opel, which registered $590 in damage compared with $222 in damage with the 1974 version. But, none was as damage resistant as the 1973 Volvo, which experienced only $63 in damage as compared with the substantially higher figure of $236 for the 1974 model.

- "With only one exception – the Opel, which showed substantial improvement from last year – the 1974 models were more vulnerable to damage in the 15 mile per hour front-into-barrier crashes than the 1973 models.

- "The five tested 1974 imports showed, on balance, more damage resistance in the 10 mile per hour front-to-rear intervehicular tests. (GM's Opel experienced not one cent worth of damage in this test, a record that has not been provided by any model, domestic or imported, tested by us to date despite the fact that it has for years been technologically easy to accomplish.)

"But the five also showed increased damageability, virtually across the board, especially in their sides and front corners – routinely struck locations on the car, for which damage-resistant design should be the rule."

Copies of the testimony are available by writing to "1974 Imports," Insurance Institute for Highway Safety, Watergate Six Hundred, Washington, D.C. 20037.

APPENDIX G

Vol. 9, No. 7 **EXCERPT** April 9, 1974

Restraint Development May Outpace Rulemaking

The Department of Transportation's new passive restraint proposals, which would take effect no earlier than the 1977 model year, may already have been outpaced by the passenger protection performance levels of some existing passive air bag and belt systems.

Passive restraint developmental work currently being carried out in the public and private sectors indicates that, for example, manufacturers would have no technological difficulty at all in meeting the proposed standard's requirement that passive protection be provided to all front seat occupants in 30 mile per hour frontal and angular and 20 mile per hour lateral barrier crash tests. (See *Status Report*, Vol. 9, No. 6, March 26, 1974.) Nor, it appears, would they have much difficulty meeting tougher requirements. Simultaneously with publishing its proposal (FMVSS 208) last month, DOT issued an "advance notice" seeking reaction to the idea of a future standard, projected for no earlier than the 1981 model year, that may require 45 or 50 mile per hour passive protection for occupants. The thrust and output of passive systems development to date suggest that such a standard already is within the state of the art.

The leading indicators of progress in passive restraint systems development – indicators that suggest faster movement on the technological front than the standards-making front – fall into the two general categories of "small car" and "standard-sized car":

SMALL CAR OCCUPANT PROTECTION

Even though sub compact cars "present the most difficult case for satisfactory performance of a passive restraint system," a DOT contractor has demonstrated that a vehicle as small as the 1972 Ford Pinto, equipped with off-the-shelf air bags, can "consistently satisfy the requirements of FMVSS 208" as now proposed.

In a report submitted to DOT in December, 1973, (DOT-HS-801-003), Agbabian Associates described the results of a test series that it ran for DOT with 1972 Pintos equipped with air bag systems developed by Olin Corp. and Rocket Research Corp. The cars were crashed front-into-barrier at 30 and 35 miles per hour; ten crashes were run with air bag-equipped Pintos.

"Test data gathered in this project indicate that, given development of an inflatable occupant restraint system for a particular model of production vehicle and attendant modification of that vehicle, it should be possible to consistently satisfy the requirements of FMVSS 208 with a fully passive inflatable occupant restraint system," Agbabian told DOT.

It also said that compliance with DOT's proposed standard with passive restraint systems in current vehicle structures of sub compacts "is feasible."

"No significant structural modifications to passenger compartment framework, cowl or instrument panel are apparently required, even in the potentially difficult sub compact vehicle, for adequate passenger compartment integrity in 30 mile per hour flat barrier impacts."

Agbabian said its crash test results "show that the Pinto structure is very effective in controlling passenger compartment volume up to 30 miles per hour flat barrier impact severity." The "upper limit" of acceptable passenger compartment integrity for the 1972 Pinto, it added, was somewhere at or above 35 miles per hour; "It was determined that deformation of the passenger compartment was not a significant problem in impacts up to 35 miles per hour."

Another DOT contractor, Minicars, Inc., is midway through a federally-funded project (DOT-HS-113-3-742) whose goal is to develop passive passenger restraint systems enabling occupant survival, under FMVSS 208, at front-into-barrier speeds of 50 miles per hour and 30 degree oblique crash speeds up to 40 miles per hour. In this project also, the 1972 Pinto is being used as the test vehicle.

Meanwhile, in the words of a senior Japanese automotive engineer, that country's small car manufacturers for some time have been "pouring efforts" into studies aimed at providing "protection of the driver at the 80 kilometer per hour [50 mile per hour] head-on collision." Recently both Toyota Motor Co. and Nissan Motor Co. delivered experimental safety vehicle models to the U.S. government for extensive crash testing; the results, which probably will be announced at the International Research Safety Vehicle Conference to be held in London in June, will disclose whether the efforts have paid off.

At last year's conference, held in Tokyo, Nissan said that when its then-developed experimental small car was "caused to collide with a large-sized vehicle of 1,800 kg [4,000 pounds] at a relative speed of about 90 km/h [56 miles per hour], both cars sustained nearly the same degree of crush, and it was ensured that the occupant safety was also secured during collisions." It added that it also had developed an alternative – a belt system that "requires a pre-load device and a retractable steering column unit for occupant protection in collision at speeds as high as 80 km/h."

STANDARD-SIZE CAR OCCUPANT PROTECTION

Impressive evidence as to the proposed 1977-model standard's feasibility is being provided by air bag equipped vehicles already in operation across the United States, both in test fleets and individual ownership. (See *Status Report*, Vol. 9, No. 4, Feb. 21, 1974.)

In addition, work sponsored by DOT is providing additional support both for the proposed standard and for DOT's idea of a future standard geared to the somewhat higher – 45-50 mile per hour – crash range:

• Rocket Research has completed a study to "provide protection to normally seated occupants and minimize injury to out-of-position occupants," according to the National Highway Safety Administration's internal newsletter, *Weekly Bulletin*, dated March 15, 1974. "To date," it reported, "a six-year-old surrogate leaning flush against the bag (to simulate an out-of-position child) has been subjected to sled tests of up to 30 miles per hour; all readings satisfied the established injury criteria. Problems currently encountered with the 50th percentile surrogate" – an adult size test dummy – "in sled tests at 50 miles per hour are considered solvable by the contractor."

• DOT has tested, with human volunteers, a passive "advanced belt restraint system" developed by Takata-Kojyo Co., Ltd. The "dynamic human volunteer test runs" were at 22.5 and 25 miles per hour, DOT said; "the volunteer has sustained no injuries and will probably volunteer to ride again at 27.5 miles per hour. These runs are the highest injury-free tests ever attained by human volunteers restrained by automotive type lap and shoulder belts."

• Under still another DOT contract (DOT-HS-053-2-280), Calspan Corp. has developed a system that combines air bags and "collapsible" energy-managing dashboards. As described by Calspan in a "final report" published last month, the energy-managing "collapsible panel section of the restraint system provides adequate protection for all occupant sizes larger than 50 pounds at impact speeds up to at least 20 miles per hour without the necessity of deploying the air bag.

"The restraint system performance is generally satisfactory in restraining a 95th percentile male in a 45 mile per hour frontal impact. Somewhat better performance was achieved in protecting a 50 pound child (6 year old) and a 50th percentile male at speeds of 40 and 50 miles per hour, respectively."

***

FRONT ENERGY MANAGEMENT PARAMETRIC VARIATION STUDY

WILLIAM RUPP
Advanced Systems Laboratory
AMF Incorporated

ABSTRACT

Analytical investigations of nine front energy management systems were conducted to evaluate their performance in terms of a) the protection provided a restrained occupant, b) aggressiveness in frontal impacts with conventional cars, and c) the crush distance required by each system. Two classes of energy management were considered, 1) hybrid hydraulic/structural systems with velocity-sensitive force-deflection characteristics providing a near square acceleration pulse shape, and 2) fixed force ramped systems with force-deflection characteristics providing a relatively long ramp to a fixed force level.

Each system was designed for a 4,700 pound vehicle impacting a) frontally, a rigid pole at 40 mph, b) a rigid barrier, frontally, at various oblique angles and speeds up to 45 mph, and c) frontally, another car whose acceleration was limited to 40 g's.

The restraint system considered was a 10 cubic foot air bag for the right front occupant. Bag characteristics were selected to suit the pulse associated with each energy management system, and then occupant response was determined for each system in 30 and 45 mph frontal impacts. Bag deployment times were 30 milliseconds and 40 milliseconds.

The relative aggressiveness of the systems was analyzed by simulating impacts with conventional vehicles ranging in weight from 1,875 to 3,900 pounds.

A preferred hybrid system was selected after evaluating the analytical results. This system, having a 14 inch buffer stroke, an average acceleration of 27.7 g's and peak of 32.6 g's, was built and tested as a complete system. The test series, reflecting the range of impact conditions specified, verified, in most instances, the predicted behavior of the preferred system.

INTRODUCTION

The subject of this paper was one of a series of sixteen studies that comprised the ESV Tradeoff & Integration Systems Studies program, conducted by the Advanced Systems Laboratory, AMF Incorporated, under the sponsorship of the National Highway Traffic Safety Administration, U.S. Department of Transportation.

The objective of the ESV Tradeoff & Integration Systems Studies was to obtain the information and data, through analyses and testing, necessary to support the development and validation of performance goals for a full-sized five-passenger sedan with a maximum curb weight of 4,700 pounds. The performance goals were to be the optimum ones in terms of accident avoidance and crash injury reduction with a minimum of aggressive behavior in

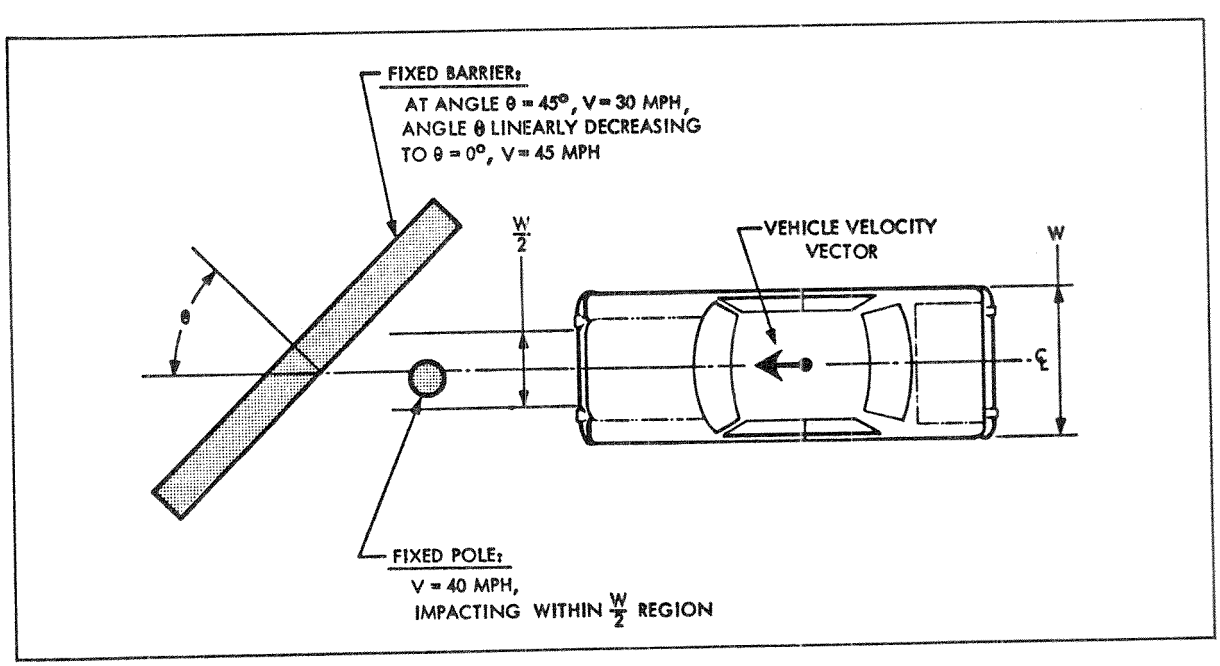


Figure 1. Frontal Impact Collision

vehicle-to-vehicle impacts. It was the intent of the program to map out various levels of performance over a range of design parameters rather than attain predetermined performance levels.

At the Fourth International ESV Conference held last year, AMF discussed the impact conditions, the parameters of interest, and the approach taken in the study of front energy management. A summary review of that discussion is appropriate before presenting the study findings.

Two classes of energy management, hybrid hydraulic/structural systems and fixed force systems, were investigated analytically to predict and compare their performance in frontal impacts. The baseline impact conditions are shown in Figure 1.

Performance comparisons were made in terms of occupant response and aggressiveness. Although it would have been desirable to complement these performance comparisons with comparisons of weight, cost, and packaging requirements, certain study constraints precluded this. The one factor, not directly related to crash performance, which was considered for both the hybrid and fixed force systems, was the required crush distance.

Upon evaluation of several versions of the two systems, a preferred hybrid configuration was selected for impact testing.

HYBRID HYDRAULIC/STRUCTURAL SYSTEMS

These systems utilize hydraulic buffers to obtain velocity-sensitive force-deflection characteristics. Crush distance in excess of that provided by the buffers is provided by a crushable structure supporting the buffers. To minimize the total required crush distance, all hybrid systems considered had a nearly square acceleration pulse in barrier impacts.

Six hybrid configurations were considered for energy management in a 45 mph on-axis frontal impact into a rigid flat barrier. Three configurations with buffer strokes of 9 inches, 12.5 inches, and 16 inches respectively were to limit the peak passenger compartment acceleration in the barrier impact to 40 g's and three configurations with the same range of buffer strokes were to limit the peak accelerations to 30 g's.

Table 1 lists the six configurations considered and the computed characteristics for each. The table also lists the computed values for a 30 mph barrier impact since this case was also considered in the occupant restraint investigations. In computing the stroke-average acceleration and hence, the total stroke, for the 30 and 40 g peak accelerations, a ratio of average to peak acceleration of 0.85 was assumed. This value was based on data obtained from static crush tests of the major structural energy elements in the system, and from prior experience with hydraulic energy absorbers of the type considered.

The acceleration pulse for the hybrid systems was represented by a trapezoid. A typical pulse shape is shown in Figure 2; the rise and decay times of 5 and 10 milliseconds (msec) respectively were based on response data taken from prior tests of similar systems. The assumption was made that the actual pulse can be approximated well enough by a trapezoid for the purposes of the study. For this to be valid, both the time-average and stroke-average accelerations have to be the same for the trapezoidal shape as for the actual pulse over periods of time significant in terms of restraint system response (e.g., over the deployment interval of 30 to 40 msec). Data from crush testing of system components indicated that the assumption made on pulse shaping was a reasonable one.

FIXED FORCE SYSTEMS

Fixed force systems are those whose force-deflection characteristics are not velocity-sensitive. Whereas the velocity-sensitive characteristics of hybrid systems serve to minimize acceleration in low velocity impacts and aggressiveness in vehicle-to-vehicle impacts, the only

Table 1. Characteristics of Hybrid/Hydraulic Structural Systems

No.	Buffer Stroke	Peak g's		Stroke - average g's		Crush Distance		Duration (msec)*	
		@ 45 mph	@ 30 mph	@ 45 mph	@ 30 mph	@ 45 mph	@ 30 mph	@ 45 mph	@ 30 mph
1	16.0"	40	26.5	34.0	22.5	23.8"	16.0"	60.3	60.7
2	12.5"	40	33.9	34.0	28.8	23.8"	12.5"	60.3	47.5
3	9.0"	40	40.0	34.0	34.0	23.8"	10.6"	60.3	40.2
4	16.0"	30	26.5	25.5	22.5	31.8"	16.0"	80.5	60.7
5	12.5"	30	30.0	25.5	25.5	31.8"	14.1"	80.5	53.6
6	9.0"	30	30.0	25.5	25.5	31.8"	14.1"	80.5	53.6

*Without rebound

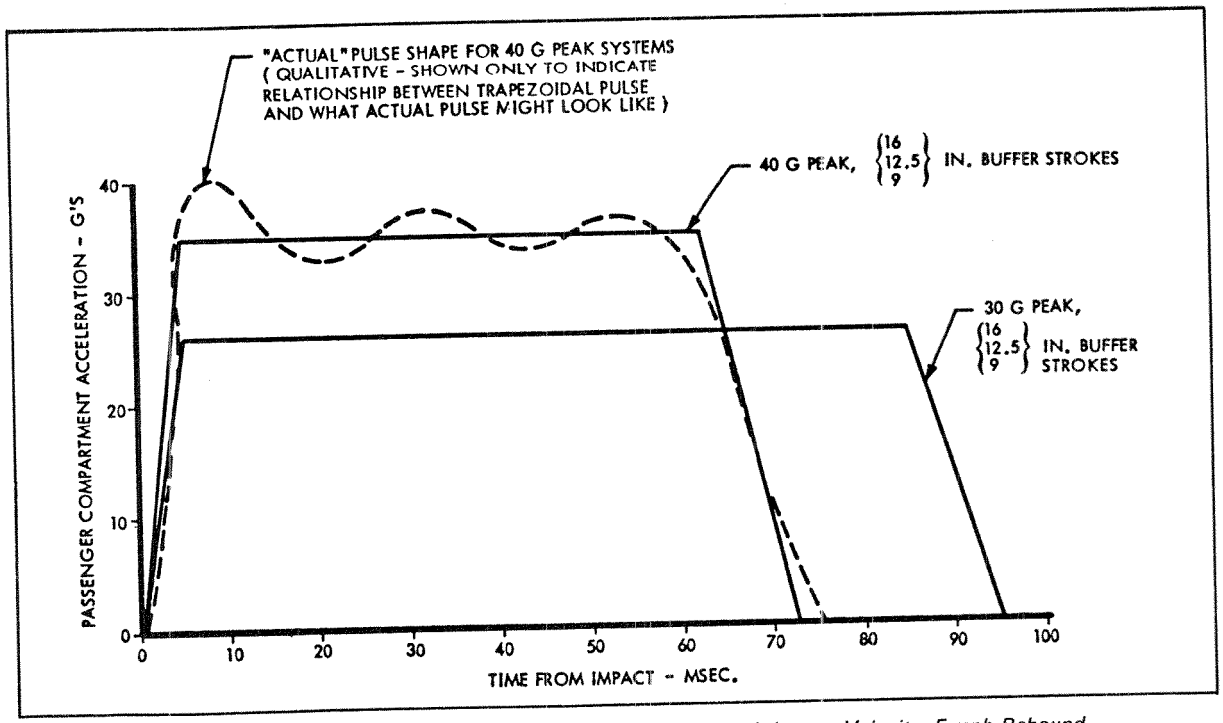


Figure 2. Hybrid System Pulse Shapes for Occupant Response Study — 45 mph Impact Velocity, 5 mph Rebound

way this can be accomplished in a fixed force system is to utilize a rising force-deflection characteristic. For this study a ramp characteristic was used in which the forces rises linearly up to a certain deflection and remain constant thereafter.

Table 2 lists the computed characteristics of the two fixed force systems considered in the study; Figure 3 presents their acceleration pulse shapes for the two impact speeds of interest.

Table 2. Characteristics of Fixed Force Systems

No.	Ramp to g Level	Peak g's		Stroke - average g's		Crush Distance		Duration (msec)*	
		@ 45 mph	@ 30 mph	@ 45 mph	@ 30 mph	@ 45 mph	@ 30 mph	@ 45 mph	@ 30 mph
7	32" to 40	40	30	22.7	15.0	36.3"	24.0"	73.3	71.5
8	16" to 40	40	40	28.7	21.2	28.3"	17.0"	61.8	51.1

*Without rebound

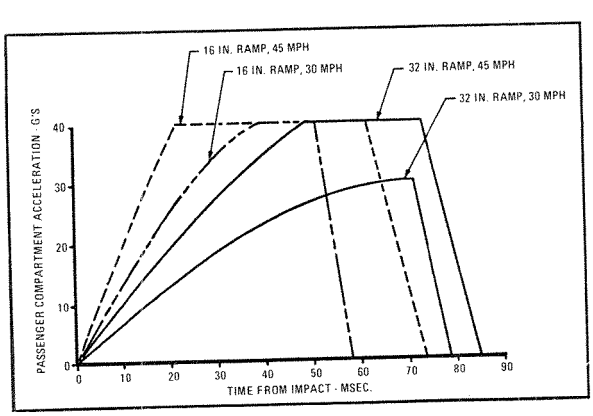


Figure 3. Fixed Force System Pulse Shapes for Occupant Response Study
45 mph Impact, 5 mph Rebound
30 mph Impact, 3 mph Rebound

OCCUPANT RESPONSE

A computer program for a one-dimensional mathematical model was used to determine the response of an occupant restrained by a 10 cubic foot (cu. ft.) air bag when subjected to each of the described crash pulses. Several runs were made for each 45 mph pulse to optimize the bag parameters (initial pressure, vent pressure, and vent area). In each case, the optimum bag was considered to be the one which required the lowest initial pressure to obtain peak occupant accelerations of less than 60 g's for each of the three occupant sizes considered (5th percentile female, 50th and 95th percentile males) at 30 msec of bag deployment time.

The resulting values for these three variables are listed in Table 3.

Table 3. Optimized Air Bag Parameters

System	Initial Pressure (psi)	Vent Pressure (psi)	Vent Area (sq.in.)
All 40 g peak, Hybrids	7	11	53
All 30 g peak, Hybrids	3	8	40
32" ramp to 40 g peak, Fixed Force	6	10	55
16" ramp to 40 g peak, Fixed Force	8	12	55

Occupant response was determined for each energy management system for both 45 mph and 30 mph impact speeds; the 45 mph bag parameters were used for both speeds. Table 4 presents the resulting occupant response data.

Based on this response data, the eight systems were evaluated by a weighted rating procedure. Typically, the weighting factors emphasized the 50th percentile male and the 45 mph impact speed, and favored the lower initial bag pressures (since lower initial bag pressure implies that the restraint system is closer to the state of the art and is more likely to have a lower noise level). The rating scheme utilized penalty scoring, scaled from 1 to 10, wherein the better performing systems receive a lower numerical value.

The scores are shown in Table 5, in order of their ranking, for the six hybrid systems and the two fixed force systems.

Table 4. Occupant Response (30 msec Deployment)

System		Peak Occupant g's @ 45 mph Impact			Peak Occupant g's @ 30 mph Impact		
No.	Description	5th%F	50th%M	95th%M	5th%F	50th%M	95th%M
1	16.0" stroke, 40 g, Hybrid	57.7	53.0	55.5	53.0	44.7	32.8
2	12.5" stroke, 40 g, Hybrid	57.7	53.0	55.5	55.3	45.4	38.3
3	9.0" stroke, 40 g, Hybrid	57.7	53.0	55.5	57.5	47.6	41.7
4	16.0" stroke, 30 g, Hybrid	49.0	49.1	50.6	49.2	42.8	39.2
5	12.5" stroke, 30 g, Hybrid	49.0	49.1	50.6	52.4	46.9	44.0
6	9.0" stroke, 30 g, Hybrid	49.0	49.1	50.6	52.4	46.9	44.0
7	32" ramp to 40 g, Fixed Force	48.8	48.5	56.3	50.3	40.2	52.1
8	16" ramp to 40 g, Fixed Force	57.2	54.3	57.3	56.4	47.4	40.7

Table 5. Occupant Response Scoring

System		Penalty Score	
No.	Description	30 msec	40 msec
4	16.0" stroke, 30 g, Hybrid	3.95	4.33
5	12.5" stroke, 30 g, Hybrid	4.32	4.92
6	9.0" stroke, 30 g, Hybrid	4.32	4.92
P	14.0" stroke, 32.6 g, Hybrid	5.60	6.35
7	32" ramp to 40 g, Fixed Force	6.52	6.35
1	16.0" stroke, 40 g, Hybrid	8.63	
2	12.5" stroke, 40 g, Hybrid	8.75	
3	9.0" stroke, 40 g, Hybrid	9.10	
8	16" ramp to 40 g, Fixed Force	10.00	

The higher ranking systems were also evaluated on their performance with a restraint deployment time of 40 msec; these scores are also indicated on Table 5. The computed occupant accelerations for these systems are shown in Table 6.

Table 5 also shows scores for a hybrid system labeled "P"; this is the preferred energy management system which was later built and impact tested. It has a 14 inch hydraulic buffer stroke, a peak acceleration of 32.6 g's for the 45 mph impact case, stroke average accelerations of 27.7 g's and 25.8 g's for the 45 mph and 30 mph impacts respectively, and crush distances of 29.3 inches and 14.0 inches at 45 mph and 30 mph respectively. These particular parameters resulted from a design effort to package a hybrid system approaching the performance of the top-rated 16 inch stroke, 30 g, system into a functional vehicle. They represent a compromise between vehicle functional requirements and crash performance. For the preferred system configuration, a 10 cu. ft. air bag with an initial pressure of 4 psi, vent pressure of 9 psi, and 47 square inches of vent area resulted in occupant response listed in Table 7.

As indicated in Table 5, the preferred system ranked higher than the 32 inch fixed force ramp system for 30 msec bag deployment time and had equal ranking for the 40 msec case. It should be noted, however, that the fixed force system requires an additional 7 inches of crush distance to achieve this performance. If comparisons were to be made of systems with equal crush distance – e.g., the 36.3 inches required by the fixed force system – the hybrid system would offer significantly higher performance. (A trapezoidal pulse for a hybrid system with this same crush would have a peak of

Table 6. Occupant Response (40 msec Deployment)

System		Peak Occupant g's @ 45 mph Impact			Peak Occupant g's @ 30 mph Impact		
No.	Description	5th%F	50th%M	95th%M	5th%F	50th%M	95th%M
4	16.0" stroke, 30 g, Hybrid	55.6	51.0	51.2	51.6	42.0	38.8
5	12.5" stroke, 30 g, Hybrid	55.6	51.0	51.2	55.6	49.0	44.7
6	9.0" stroke, 30 g, Hybrid	55.6	51.0	51.2	55.6	49.0	44.7
7	32" ramp to 40 g, Fixed Force	56.1	51.7	58.0	56.2	47.2	34.5

Table 7. Peak Occupant Acceleration (g's)

	Deployment time = 30 msec			Deployment time = 40 msec		
Pulse	5% F	50% M	95% M	5% F	50% M	95% M
45 mph	56.5	50.1	51.9	57.7	56.3	58.6
30 mph	56.1	46.2	38.7	52.6	47.0	42.7

26.2 g's and a stroke-average acceleration of 22.3 g's). In this comparison, the following should be noted:

1. At the 40 msec deployment time, the 30 g peak hybrid system scored noticeably better than the 32 inch ramp due to the much lower initial bag pressure required. Because of the lower acceleration, a 26.2 g peak hybrid system would result in even more favorable performance.
2. With a restraint system having a deployment time equal to the duration (prior to rebound) of the ramp system's pulse, the 26.2 g trapezoid would provide better performance because, when the bag deploys, the vehicle is still crushing (due to the longer pulse) with some associated ride-down benefit. The vehicle with ramp characteristics would have reached zero velocity by the time the bag deployed.
3. At restraint deployment times greater than the duration of the trapezoidal pulse, both systems would provide identical performance since the pulse shape no longer matters – both vehicles would have stopped by the time the bag deploys.

Based on the foregoing, the following conclusion is reasonable: For equal crush distance, a hybrid system (trapezoidal pulse shape) provides superior performance to a long ramp system for any restraint deployment times which still provide a reasonable chance of protecting the occupant.

AGGRESSIVENESS

The relative aggressiveness of each of the nine energy management systems considered in the occupant response study was analyzed by conducting simulated impacts with conventional vehicles. The measure used for the aggressiveness of a given system was the amount of crush computed for the conventional vehicle after it is impacted by a vehicle with that front energy management system. The conventional vehicles referred to in this paper as the "impacted vehicle" included a VW "Beetle" (1,875 pounds), a Volvo (2,530 pounds), and a 1969 Ford (3,900 pounds); a weight of 4,700 pounds was used for the impacting vehicle. The force-deflection characteristics of the impacted vehicles were determined from barrier impact acceleration traces available in published literature.

Because of modeling simplifications, and the assumption that a perfect structural interface existed between the two vehicles, the crush distances obtained for the impacted vehicles are not interpreted as realistic in terms of an actual collision. Their value is in terms of providing comparative aggressiveness data on the various systems under study.

The simulations were all run at the same relative impact speed, 54 mph. This speed was chosen to limit the amount of crush of the impacted vehicle to the crush suffered in the barrier impact (usually at 30 mph) from which the force-deflection data were derived.

A summary of the results of the simulated impacts is presented in Table 8. The maximum crush of the impacted vehicle is the variable of interest. In all the simulated impacts involving the hybrid systems, the crush of the impacting vehicle was limited to the stroke of the hydraulic buffer. The slight differences between the results obtained with the 30 g peak acceleration and with the 40 g peak acceleration

hybrid systems were caused by the slightly higher stiffness of the upper front end structure used in the 40 g system configurations. If the upper structure had been omitted, no differences would have been noted between the 30 g and 40 g cases.

Table 8. Aggressiveness of Front Energy Management Systems

Impacting Vehicle with System		Impacted Vehicle Crush (inches)		
No.	Description	VW	Volvo	Ford
1	16.0" stroke, 40 g, Hybrid	21.0	18.7	23.5
2	12.5" stroke, 40 g, Hybrid	23.2	20.7	24.5
3	9.0" stroke, 40 g, Hybrid	25.5	23.1	30.4
4	16.0" stroke, 30 g, Hybrid	20.8	18.5	23.5
5	12.5" stroke, 30 g, Hybrid	22.9	20.5	24.7
6	9.0" stroke, 30 g, Hybrid	25.5	23.3	32.2
P	14.0" stroke, 32.6 g, Hybrid	21.5		
7	32" ramp to 40 g, Fixed Force	22.3		
8	16" ramp to 40 g, Fixed Force	28.2		
-	1969 Ford with production S-frame	30.3		

It became evident, after running all of the original hybrid system simulations, that there was no advantage in considering all three impacted vehicles, and that a reasonable evaluation could be made based on the VW results only. Therefore, the preferred hybrid and the fixed force ramp systems were run only against the VW.

To complete the aggressiveness study, an impact between conventional cars, a VW and a 1969 Ford, was simulated. As shown in Table 8, this collision resulted in more crush of the VW than that computed for any of the simulated impacts with front energy management systems, either hybrid or fixed force ramp. The greater crush was attributed to the very high initial stiffness and collapse load of the S-frame front end.

The relative aggressiveness of each system was evaluated using the results of simulated impacts with the VW "Beetle". The results of this evaluation, using the penalty scoring approach, are shown in Table 9.

The best comparable systems are the hybrids with hydraulic buffer stroke in the 12.5 to 16.0 inch stroke range and the fixed force system with the 32 inch ramp.

PACKAGING EVALUATION

No design studies were made of the fixed force ramp systems, so the only way in which they could be compared with the hybrid systems in terms of packaging requirements was on the basis of required crush distance. The scoring was done by assessing a penalty for any crush distance required in excess of 20 inches. The results of this evaluation on a penalty scoring basis are summarized in Table 10.

Table 9. Aggressiveness Scoring

System		
No.	Description	Score
4	16.0" stroke, 30 g, Hybrid	0.78
1	16.0" stroke, 40 g, Hybrid	0.97
P	14.0" stroke, 32.6 g, Hybrid	1.46
7	32.0" ramp to 40 g, Fixed Force	2.23
5	12.5" stroke, 30 g, Hybrid	2.82
2	12.5" stroke, 40 g, Hybrid	3.40
6	9.0" stroke, 30 g, Hybrid	5.34
3	9.0" stroke, 40 g, Hybrid	5.34
8	16.0" ramp to 40 g, Fixed Force	7.96
-	1969 Ford with S-frame	10.00

Table 10. Packaging Scores

System		
No.	Description	Score
1	16.0" stroke, 40 g, Hybrid	2.33
2	12.5" stroke, 40 g, Hybrid	2.33
3	9.0" stroke, 40 g, Hybrid	2.33
4	16.0" stroke, 30 g, Hybrid	7.14
5	12.5" stroke, 30 g, Hybrid	7.14
6	9.0" stroke, 30 g, Hybrid	7.14
8	16" ramp to 40 g, Fixed Force	5.10
P	14.0" stroke, 32.6 g, Hybrid	5.70
7	32" ramp to 40 g, Fixed Force	10.00

OVERALL EVALUATION

An overall evaluation of the hybrid and fixed force systems was conducted in terms of the following factors: occupant response, aggressiveness, and packaging (crush distance) requirements.

To assure that the system evaluated was not unduly sensitive to any one factor, three scoring methods were used, each method weighting the evaluation factors differently. The first scoring method, designated T_1, weighted occupant response, aggressiveness, and packaging equally. The second, T_2, emphasized performance, weighted occupant response and aggressiveness equally and ignored packaging. The third, T_3, emphasized packaging and weighted that factor twice as much as either occupant response or aggressiveness.

The results of the overall evaluations for each scoring method are listed in Tables 11 and 12. Table 11 is based on occupant response performance with a 30 msec air bag deployment time, and Table 12, a 40 msec deployment time. The scores, penalty oriented, were obtained as an appropriately weighted sum of the previously computed scores for the three evaluation factors.

Table 11. Overall Evaluation Scoring (30 msec Air Bag Deployment)

System		Scoring Method		
No.	Description	T_1	T_2	T_3
1	16.0" stroke, 40 g, Hybrid	3.98	4.80	3.56
2	12.5" stroke, 40 g, Hybrid	4.83	6.07	4.20
3	9.0" stroke, 40 g, Hybrid	5.59	7.22	4.77
4	16.0" stroke, 30 g, Hybrid	3.96	2.36	4.75
5	12.5" stroke, 30 g, Hybrid	4.76	3.57	5.35
6	9.0" stroke, 30 g, Hybrid	5.60	4.83	5.97
P	14.0" stroke, 32.6 g, Hybrid	4.25	3.53	4.61
7	32" ramp to 40 g, Fixed Force	6.25	4.38	7.18
8	16" ramp to 40 g, Fixed Force	7.69	8.98	7.03

Table 12. Overall Evaluation Scoring (40 msec Air Bag Deployment)

System		Scoring Method		
No.	Description	T_1	T_2	T_3
4	16.0" stroke, 30 g, Hybrid	4.08	2.56	4.84
5	12.5" stroke, 30 g, Hybrid	4.96	3.87	5.50
6	9.0" stroke, 30 g, Hybrid	5.80	5.13	6.13
P	14.0" stroke, 32.6 g, Hybrid	4.50	3.90	4.80
7	32" ramp to 40 g, Fixed Force	6.19	4.29	7.14

With a restraint deployment time of 30 msec, the 16.0" stroke, 30 g hybrid system ranks marginally first in the balanced, T_1, scoring, first in the performance weighted, T_2, scoring, and fourth in the packaging weighted, T_3, scoring.

It was noted previously that the 40 g hybrid systems and the 16 inch ramp fixed force system were not considered in the occupant response simulations using the 40 msec deployment time because of their poor performance with 30 msec deployment. Of the surviving systems, Table 12 indicates that 16.0 inch stroke, 30 g hybrid system still ranks first in the T_1 and T_2 scoring methods, and is marginally second in the T_3 method.

The 14.0" stroke, 32.6 g hybrid system, which evolved during the course of the study as the preferred system from design feasibility considerations, closely approximates the performance of the 16.0" stroke, 30 g hybrid system. The preferred system ranks high in all the scoring methods and exhibits relatively small variations with changes in weighting, which is indicative of a well-balanced design. On this basis, this system was chosen as the one to be built and tested.

FRONT END DESIGN

A plan view of the test configuration of a front end incorporating the preferred 14.0" stroke, 32.6 peak acceleration hybrid system is shown schematically in Figure 4. The system consisted of two hydraulic buffers, each backed up by two stages of crushable tube providing a maximum available crush of 34 inches (including 14 inches of buffer

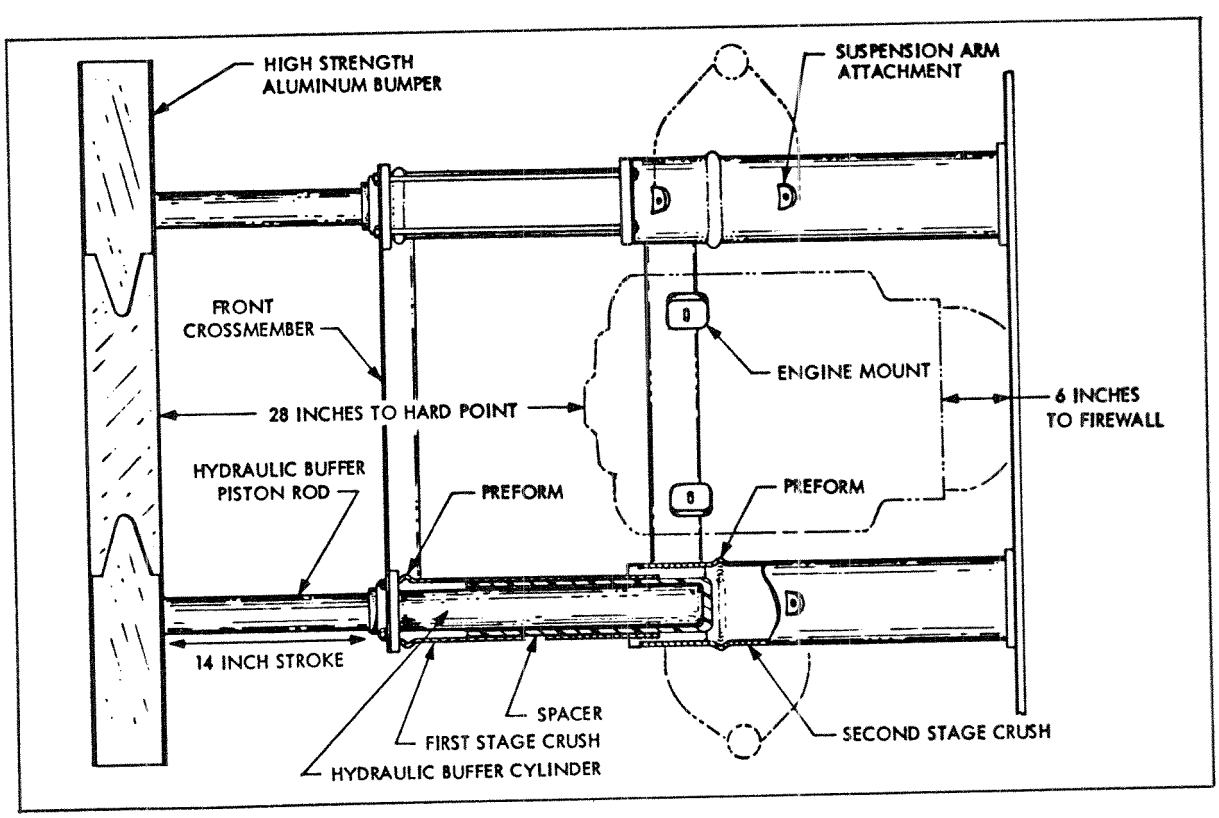


Figure 4. Front End Energy Management System

stroke) with the bumper in contact with the engine and the engine in contact with the firewall. Although only 30 inches of crush was expected in the 45 mph barrier impact, it was estimated that approximately 40 inches would be needed in the offset pole impact at 40 mph, resulting in local intrusion in the passenger compartment of six inches.

The variable stroke hydraulic buffers were scaled down, simplified versions of designs used previously by AMF. The variable stroke was obtained by aligning different orifice patterns, depending on pressure, in a pair of concentric sleeves. Two orifice patterns were used: one operative in impacts below 5 mph and permitting a six inch stroke, and the second operative above 14 mph and permitting the full 14 inch stroke. Between 5 and 14 mph, intermediate values of stroke were obtained, depending on the impact velocity. The buffers were completely self-contained by incorporating the accumulator in the piston rod of the buffer. Re-extension of the buffers and bumper following a low-speed impact was accomplished by a gas precharge of 200 psi in the accumulators.

The crushable tube stages in the front end structure were made up of mild steel round tube; the first stage had a 4.5 inch OD with a .095 inch wall, and the second, a 5 inch OD with a .083 inch wall. The tubes were locally preformed to ensure that crush would start at the desired locations.

The system was designed for structural crush to occur only at barrier impact velocities in excess of about 25-30 mph, with the crush occurring simultaneously with stroking of the hydraulic units. Crush of the first tube stage was intended to be completed before a significant amount of crush of the second stage would begin. This meant that the second stage would crush around the hydraulic cylinder and spacer, rather than having these components find a path through an already crushed tube, which would be the case if the first tube stage crushed later than the second stage.

The spacer was provided primarily to reduce the variation in crush characteristics of the first tube stage in oblique impacts by permitting transverse loads and moments to be carried out into an area of the tube not free to crush (due to the support of the spacer). The area of the tube free to crush was thus subjected primarily to longitudinal loads even in oblique impacts, so its crush characteristics would be nearly invariant.

The engine mounts, which were caged to restrain the engine in all directions, and the front suspension mounts were located between the first and second stage tubes. The resultant higher mass loading on the first stage tubes was designed to ensure their crushing before the second stage. In addition, the lateral inertia of the engine and suspension was expected to provide some beneficial stabilization of the second stage tubes in oblique impacts.

SYSTEM TESTS

A series of eight tests of the preferred front energy management system was conducted to determine the system's response under the impact conditions of interest (listed in Table 13). Two test vehicles were used in this test series: one vehicle, designated the "bogey" incorporated the front end system including functional vehicle components, and a rear energy management system, and the second, designated the "integrated test vehicle," incorporated a passenger compartment structure with the front energy management system and the required functional components.

Table 13. Front Energy Management Tests

Test No.	Impact Condition
317	Frontal at 0° with a rigid flat barrier at 22.75 mph
318	Frontal at 0° with a rigid flat barrier at 45.71 mph
319	Frontal oblique at 30° with a rigid flat barrier at 34.49 mph
321	Frontal with a rigid pole at vehicle centerline at 40.69 mph
427	Frontal at 0° with a rigid flat barrier at 44.84 mph
428	Frontal oblique at 45° with a rigid flat barrier at 29.85 mph
428A	Frontal oblique at 30° with a rigid flat barrier at 35.5 mph
429	Frontal with a rigid pole, off-center, at 41.07 mph

A summary of the findings of each front energy management system test follows.

Test 317, Frontal Impact at 0° with a Flat Barrier at 22.75 mph. This impact was conducted to test the performance of the hydraulic buffers; no structural deformation was anticipated.

The test indicated that both buffers traveled through their full stroke. An internal malfunction in both buffers caused some excess pressure to develop during part of the stroke, thereby increasing the resisting force of the buffers and causing slight structural deformation (about 0.5 inches). However, this failure did not interfere with the functioning of the units during impact. Figure 5 shows the post-impact condition of the front end with the buffer in its fully-compressed position.

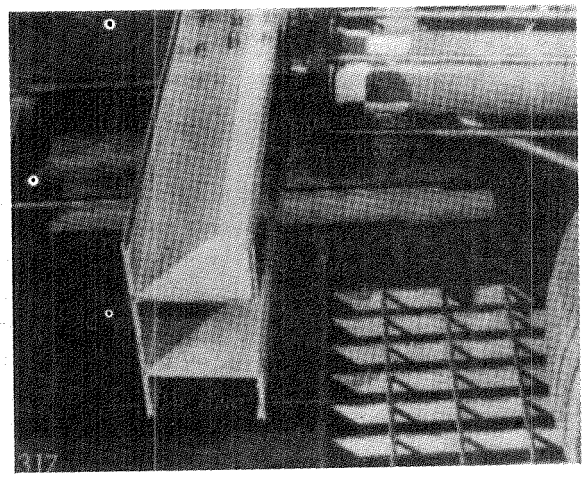


Figure 5. Post-crash View, Test 317

The maximum passenger compartment acceleration was 22.8 g's at 31 msec. The duration of the crash pulse was approximately 80 msec with the major part of the crash pulse below 20 g's.

Test 318, Frontal Impact at 0° with a Flat Barrier at 45.71 mph. This impact was conducted to test the behavior of the total front energy management system at the maximum design condition for frontal impact with a flat barrier.

The hydraulic buffers functioned properly, the malfunction which had been discovered in Test 317 having been corrected prior to the conduct of this test.

The engine broke free of its mounts at about 15 msec and impacted the rear of the bumper at about 40 msec after impact. The engine-bumper impact had a duration of approximately 15 msec. As a result of the engine breaking free, the front (first stage) tubes were not completely crushed; this allowed the second stage tubes on the right side to partially crush before the buffer cylinder entered it.

At approximately 45 msec, a hinge formed behind the buffer cylinders, just aft of the front suspension, due to the torque applied as a result of the offset of the vehicle CG above the line of resistance of the front end structure. Figure 6 shows the post-crash condition in this area. This offset caused the rear of the test vehicle to pitch upward to a maximum angle of 11° (with the ground). Post-crash inspection indicated the front end structure rotated 15° to 20° relative to the rest of the vehicle. During the pitching action, the frame contacted the ground at about 95 msec resulting in a 28.1 g vertical acceleration at that time.

The crash pulse recorded at the vehicle CG and in the rear seat area shows two peaks, ranging from 33.7 g's to 36.8 g's, one peak at 18 msec and the other at 76 msec after impact. Halfway between the peaks, the acceleration dropped to a minimum of approximately 20 g's. The sharp decrease in acceleration at 45 msec coincided approximately with the bumper-engine impact and the formation of the hinge. Ground friction appears to be the major cause of the second peak. At this point, compression of the front and rear tubes ceased and deflection about the hinge was predominant.

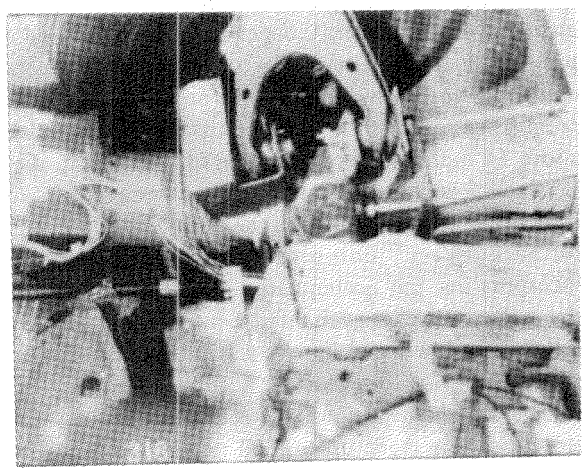


Figure 6. Post-crash View, Test 318

Average residual crush of test vehicle was 29 inches. The vehicle centerline remained normal to the barrier during the crash. The buffer cylinders and front tubes stroked symmetrically. On the left side, the rear tube collapse was in sequence providing the intended failure mode. As mentioned previously, on the right side, the rear tube collapse departed somewhat from the intended sequence.

Test 319, Frontal Oblique Impact at 30° with a Flat Barrier at 34.49 mph. The objective of this test was to determine the response of the front energy management system in an oblique impact.

In this test, it was expected that the vehicle would rotate into the barrier at impact; that is, the vehicle centerline would become more nearly perpendicular to the barrier face. (The pre-crash attitude of the test vehicle at the barrier is shown in Figure 7.) Instead, the vehicle rotated away from the barrier. A plywood facing on the barrier detached itself 47 msec after impact and apparently, low friction between the facing and the barrier allowed the bumper to slide across the barrier moving the facing with it.

The energy management system performed as intended in this test. The right hand system components were deflected less than one inch axially during impact. Some deformation in hinging and local collapse occurred in the rear tube at its attachment to the firewall.

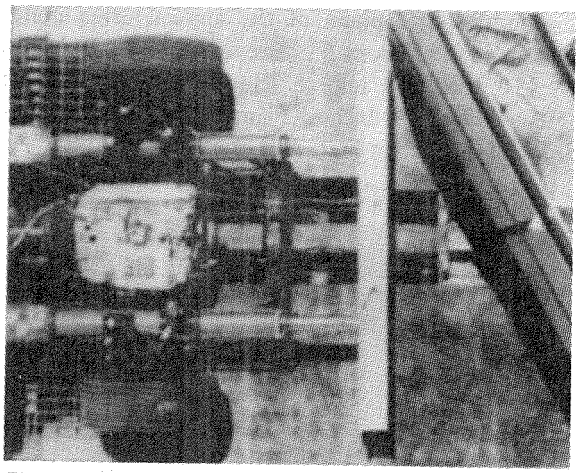


Figure 7. Pre-crash View, Test 319

Figure 8 depicts the post-crash condition of the left side, the impact side, of the front end. The buffer stroked 5½ inches in the first 20 msec after impact at which time the bumper end failed and the vehicle was free of the barrier for approximately 7 msec. Buffer stroking resumed at 27 msec and continued until 11 inches of stroke was reached at about 60 msec after impact. Rear stage tube collapse started at 27 msec and 5 inches of collapse was attained at 60 msec.

When the left side system components stopped deforming axially at around 65 msec, a definite transverse motion was occurring due to the sliding action across the barrier. The hinge at the base of the right side system formed at 80 msec and at 88 msec, the first stage tube on the left side was pushed into the engine. At 109 msec, the left front tire contacted the left rear tube and pushed it into a "Z" shape, and resulted in a peak transverse acceleration of 29.1 g's at the vehicle's CG 115 msec after impact. The maximum longitudinal acceleration at the vehicle CG was approximately 16 g's.

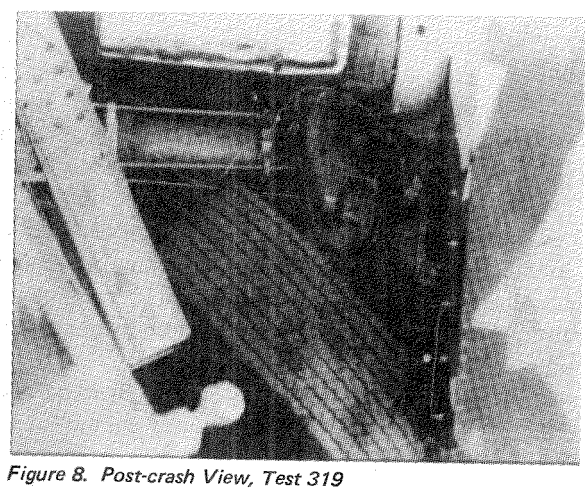


Figure 8. Post-crash View, Test 319

The vehicle was free of the barrier at 135 msec. The residual velocity of the vehicle was 16.0 feet per second (11 mph) at 163 msec after impact; approximately 90 percent of the vehicle's initial energy dissipated by this time.

Test 321, Frontal Impact with Rigid Pole at Vehicle ℄ at 40.69 mph. This test was conducted to determine the adequacy of the bumper design as well as the front energy management system in a centered pole impact.

The barrier force transducer indicated a peak of 142 kips and an average of 90 kips during impact. The bumper withstood these loads and transmitted them into the energy management system.

During this impact, the engine mounts failed and again caused a premature stop to the collapse of the first stage tubes. This condition initiated collapse of the rear stage tubes without the benefit of being supported by the buffer cylinder in the first stage tubes. Nevertheless, the rear tube stages collapsed uniformly, providing about 9 inches of the total crush of 26 inches.

Vehicle deformation was symmetrical. The buffers stroked completed and normally. Both front tubes started collapse at 8 msec. The engine broke free of its front mounts and was uncoupled from the first stage (front) tubes starting at 19 msec. The front tube collapse was thereby limited to about 3 inches. At 32 msec, the engine moved forward relative to the frame and at 58 msec, impacted the rear face of the bumper.

Collapse of the rear stage tubes started at 12 msec, was interrupted momentarily at 60 msec when the engine was impacting the bumper, and continued until 120 msec after impact.

The front of the vehicle frame started diving at about 40 msec after impact, and struck the ground at 100 msec. At 80 msec, after being impacted by the engine, the bumper began riding up the pole, carrying the front energy management components with it. Thus, the collapse of the rear stage tubes occurred in the presence of large bending moments, collapse being caused by a combination of compression and bending forces.

The peak acceleration in the passenger compartment (at the vehicle CG) reached 27.5 g's 16 msec after impact, the acceleration pulse approximated a square with a 90 msec duration.

Test 427, Frontal Impact at 0° with a Rigid Flat Barrier at 44.84 mph. This test had the same objective as Test 318 except that the integrated test vehicle, rather than the bogey, was used. As a result of the findings of the 300 series of tests, some modifications were made to the front energy management system; the rear stage tubes were

stiffened to prevent premature collapse in the area near the firewall and the engine mounts were reinforced.

During the impact, the engine mounts remained intact for a longer period than in prior tests, but failure began at 25 msec and the engine was moving freely by 35 msec. Both buffers performed normally, reaching their full 14 inch stroke at about 60 msec after impact. The collapse of the front stage tubes was normal, but the rear tube collapse was initiated prematurely because of the failure of the engine mounts and subsequent unloading of the front stage tubes. However, rear tube collapse continued with the buffer cylinders providing the tube support as the buffer moved rearward.

The crash pulse was approximately rectangular with a peak acceleration of 36 g's at 36 msec after impact, and a stroke-average acceleration of 28 g's. The peak accelerations were somewhat higher than predicted. The total front end crush was approximately 29 inches.

Test 428, Frontal Oblique Impact at 45° with a Rigid Flat Barrier at 29.85 mph. This impact was conducted to test the behavior of the front energy management system in a moderate speed impact at a high angle of incidence with the barrier.

The left side of the test vehicle was the impacted side. The left buffer performed normally and reached a maximum stroke of 13 inches at 50 msec after impact. The right side of the bumper never contacted the barrier and the right buffer was under tension throughout the impact. Neither of the front stage tubes was compressed.

The deflection of the rear stage tube on the impact side was approximately 4½ inches, including both longitudinal crush and parallelogram deformation of the energy management system which began about 70 msec after impact. (The post-crash condition of the front end is shown in Figure 9). A hinge formed at the forward and aft ends of the left side rear stage tube. The right side components deflected as a single column about a hinge point at the firewall.

The crash pulse was of low amplitude with a peak acceleration of 13.4 g's.

Test 428A, Frontal Oblique Impact at 30° with a Rigid Flat Barrier at 35.5 mph. This test had the same objectives as Test 319 except that the integrated test vehicle was used for the impact.

In contrast to Test 319, the test vehicle was not deflected away from the barrier during impact and the barrier facing remained intact. Both buffers reached full stroke sequentially. The left buffer (impacted side) stroked 13 inches by 40 msec after impact and completed its last inch of stroke more slowly thereafter. The right buffer, under tension during the first phase of the impact, came under compression at about 55 msec and stroked 13 inches by 110 msec after impact. The first stage tube on the left collapsed 2½ inches. Figure 10 shows the right front of the system in its post-crash state.

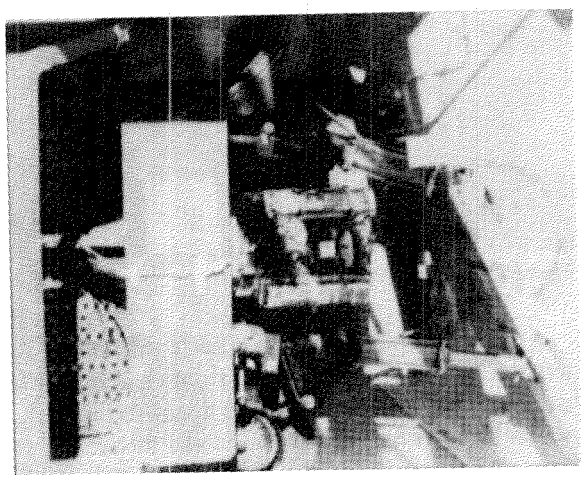


Figure 9. Post-crash View, Test 428

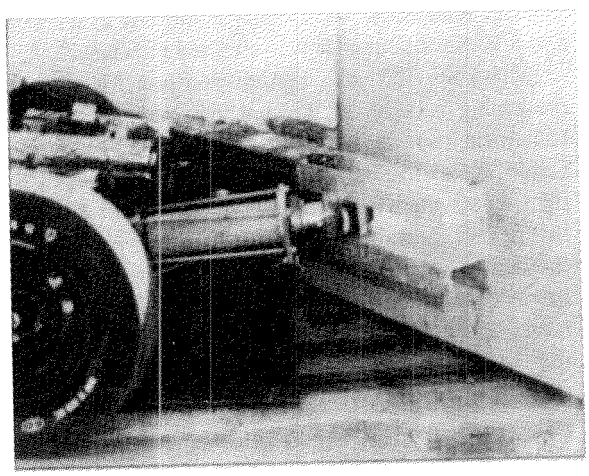


Figure 10. Post-crash View, Test 428A

The barrier drove the left front wheel against the left rear stage tube at 65 msec forming a plastic hinge that moved inward. The right tubes deflected away from the barrier as a single column, hinging about a point at the firewall. Deflection about this hinge was evident at about 90 msec.

The front energy management system behaved as intended in this test with most of the impact energy absorbed by the hydraulic buffers. The crash pulse had a maximum longitudinal peak acceleration of 20 g's.

Test 429, Frontal Impact Off-Center, with a Rigid Pole at 41.07 mph. This impact was conducted to test the performance of the system when the point of impact with a rigid pole is within the center

half-width of the bumper (16 inches maximum offset). In assessing the design of the front energy management system, it was anticipated that the maximum intrusion in this type of impact might exceed 3 inches. In the offset pole impact, if it is assumed that the average energy absorbed is 60% of that absorbed in a 0° flat barrier impact (for the same crush or penetration distance), then the speed at which the available 34 inches of crush is used up is about 37 mph. For the 40 mph offset pole impact, the passenger compartment residual speed, when the 34 inches is used up, would be about 15 mph which might result in exceeding the 3 inch intrusion limit.

In this test, the point of impact was offset 19-3/8 inches instead of 16 inches. The impact speed was 41.07 mph.

The bumper, hydraulic buffer and front collapsible tube performed properly over a short distance and then were pushed aside. The left side buffer stroke reached 9 inches and the left front tube collapsed 4.5 inches in this period. The engine accelerometer record indicated a sharp side impact at 50 msec resulting from contact by the front tube as it was pushed aside by the pole. The pole progressed unimpeded alongside the engine and energy management components and contacted the suspension arms, rotating the tire into a position between the pole and the frame. The forward motion of the vehicle stopped when the pole contacted the firewall.

A reasonable crash pulse started to develop in the early stages of the crash. By 17 msec, the vehicle CG acceleration had reached 15 g's; but, by 29 msec, the acceleration dropped below 10 g's as the energy management system was pushed aside. A low acceleration level was maintained until about 90 msec, when the pole encountered the front suspension. A level of 12 to 25 g's was recorded until 116 msec at which time the vehicle velocity was approximately 14 mph. A final pulse of about 15 msec duration and peak amplitude of about 30 g's (38 g's on the impacted side of the vehicle) occurred as the pole pushed the tire into the firewall and deformed the frame.

The vehicle centerline remained approximately in alignment with the original direction of travel during the impact. The vehicle yawed slightly to the right at about 100 msec as the front end was deflected by the pole. At about 150 msec, the vehicle began to pivot around the pole in the opposite direction. At the rear seat position, the transverse acceleration reached a peak of 14 to 17 g's with a pulse duration of approximately 10 msec.

The vehicle frame contacted the ground at 160 msec, resulting in reversing vertical accelerations of 10 to 18 g's peak amplitude and approximately 10 msec in duration. The post-crash condition of the front end is shown in Figure 11. Intrusion was on the order of 5 to 8 inches and was confined to the vicinity of the driver's feet.

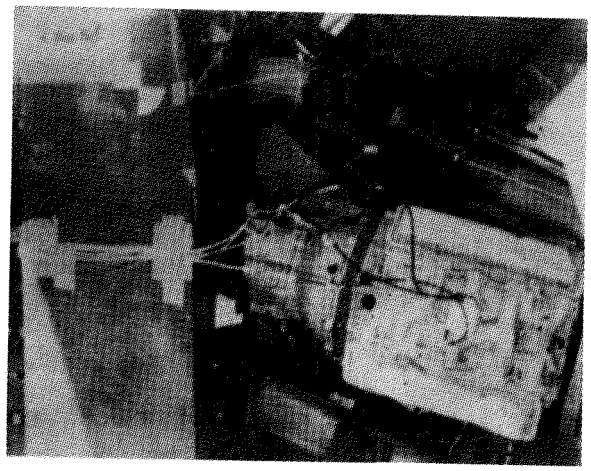


Figure 11. Post-crash View, Test 429

CONCLUSIONS

The test program demonstrated that the selected front energy management system is capable of protecting the passenger from excessive acceleration and compartment intrusion in all baseline impacts with the possible exception of the off-center pole impact. In this case, the structure was forced to one side by the pole and was not fully effective in retarding the vehicle.

Improvement of the system's performance in this pole impact, as well as in the oblique impacts, would result from increasing the lateral stiffness of the front end structure. This could be accomplished by tying in the basic front end structure with the fender and hood structures, none of which were included in the test vehicles. Inclusion of the fender and hood structures would also tend to reduce kick-up of the rear end. Performance in the off-center pole impact would also have been improved by increased toughness in the bumper-to-buffer connections; the brittleness of this connection was due, however, to faulty welding, not the basic design or material.

Another area requiring improvement involves the sequencing of the crushing of the front and rear tube stages. The basic problem here was the inability to get the engine inertia load into the front tube stage. This was due in most cases to failure of the engine mount bosses on the engine block, and in the remainder of the cases, to weakness in the cage structure surrounding the engine mounts. An improved engine restraint system is required here.

For a system incorporating the improvements mentioned above, some estimates were made of the maximum impact velocities which could be sustained without subjecting the passenger compartment to excessively high acceleration or intrusion in 0°, 30°, and 45° flat barrier impacts. These estimates were based on an analysis of the test results in terms of the energy absorbed be each component, the kinematic behavior of the vehicle, and the geometry of the entire front end assembly. The factors which limit the allowable impact velocity are: the total energy absorption capacity in the 0° case, the relative geometry of the engine, deformed structure, and passenger compartment in the 30° case, and the vehicle kinematics (eventual rotation of the side of the vehicle into the barrier with the resultant high lateral acceleration) in the 45° case. The estimated limiting impact velocities in these three cases were: for 0°, 48 mph; for 30°, 42 mph; and for 45°, 40 mph.

ANALYSIS OF TEST RESULTS FOR AMF/FIAT ESV HEAD-ON COLLISION

SOL DAVIS
Ultrasystems, Inc.

ABSTRACT

As part of its ESV evaluation effort, Dynamic Science has recently conducted vehicle-to-vehicle aggressive testing of the large AMF ESV with two small Fiat ESVs (Fiat 2,000-pound and Fiat 2,500-pound classes) at a closure speed of 75 mph. Preliminary analyses of crash test results are presented in this report. The vehicle-to-vehicle tests were highly successful in that structural integrity of the Fiat passenger compartments was adequately maintained. The AMF vehicle's hydraulic subsystem absorbed a significant portion of the crash energy in its 20-inch stroke, thus offering reduced aggressiveness to the smaller Fiat vehicles. However, human survival in terms of meeting current occupant injury criteria was not evaluated since dummies were not included in this preliminary test series. Such an evaluation is planned as part of future phases of this ESV evaluation program.

During the Fourth ESV Conference, Dynamic Science reported on the results of a head-on collision between a small Italian ESV (Fiat 1,500-pound class) and a large American ESV (AMF) at a closure speed of 60 mph. Dynamic Science has just completed similar vehicle-to-vehicle aggressiveness testing of the AMF ESV with two heavier Fiat ESVs, the Fiat 2000 and the Fiat 2500A. These structural tests were conducted at a closure speed of 75 mph. In addition, testing of the AMF into a Fiat 2500B is planned at a closure speed of 50 mph. The Fiat 2500B vehicle will include two front occupants restrained by experimental belt/seat subsystems. This paper presents preliminary analyses of the results for the two structural tests conducted at a 75 mph closure speed.

TEST RESULTS

Table 1 presents dimensional and weight data for the Fiat vehicle classes as reported by Fiat in Reference 1. The Fiats used in this program were structural test vehicles and did not have completely furnished interiors. The lower as-received weights, shown in Table 1, permitted the addition of the required on-board test instrumentation.

Figure 1 shows a pre-test view of the front bumper interface for the AMF/Fiat 2000 test. Similarly, Figure 2 shows the pre-test interface for the AMF/Fiat 2500A test. Note the excellent match in vehicle bumper heights. Film strips of the two crashes were presented by Mr. N. Stahler earlier in the conference. Views of the post-test vehicle interfaces are shown in Figures 3 and 4, respectively.

TABLE 1.
FIAT ESV DIMENSIONS AND WEIGHT DATA

PARAMETER	FIAT 1500	FIAT 2000	FIAT 2500
Length (in.)	132.7	165.9	172.2
Width (in.)	55.1	66.5	68.2
Height (in.)	52.8	55.3	56.2
Wheelbase (in.)	72.4	96.4	95.3
Tread Width, F/R (in.)	47.4/45.0	52.0/48.8	54.0/53.6
Curb Weight (lb)	1720	2570	2940
Occupant Capacity (No.)	4	4	4
GVW (lb)	2425	3275	3645
As-Received Weight (lb)	1563	2190	2480
Test Weight (lb)	1773	2436	2688

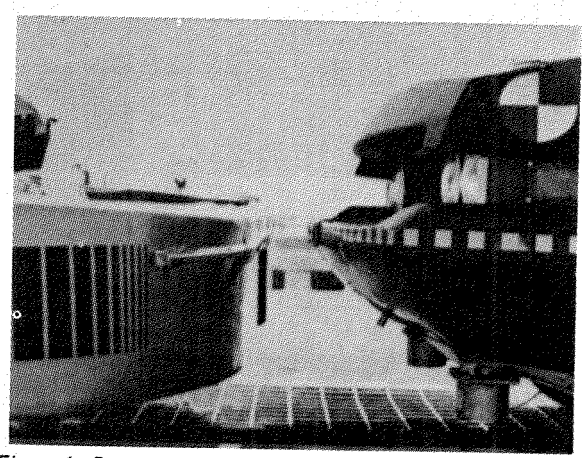


Figure 1. Pre-test View of AMF/Fiat 2000 Vehicle Interface.

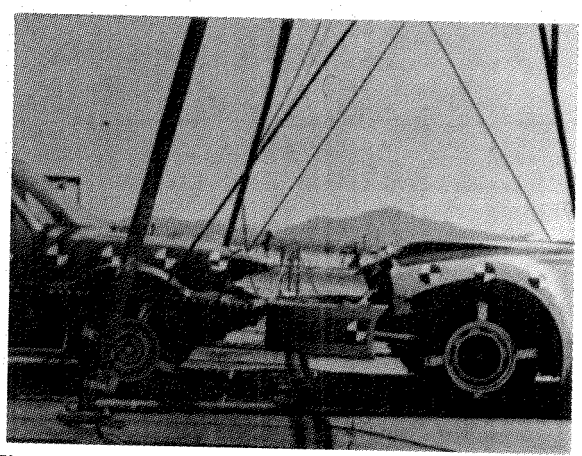


Figure 2. Pre-test View of AMF/Fiat 2500A Vehicle Interface.

A summary of the crash test results for the Fiat 2000 and Fiat 2500A tests is shown in Table 2. The AMF/Fiat weight ratios were 2.16 for the Fiat 2000 and 1.96 for the Fiat 2500A, or approximately 2:1. The closure speeds were nominally 75 mph in both tests. The Fiat velocity changes were about 56 mph. The ratio of Fiat/AMF velocity change was almost inversely proportional to the vehicle weight ratio, as expected.

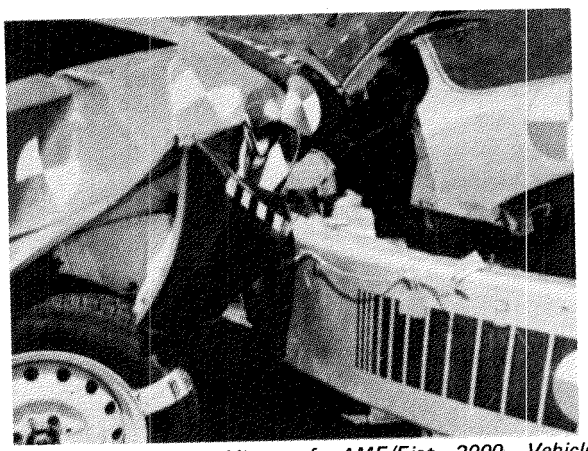


Figure 3. Post-test View of AMF/Fiat 2000 Vehicle Interface

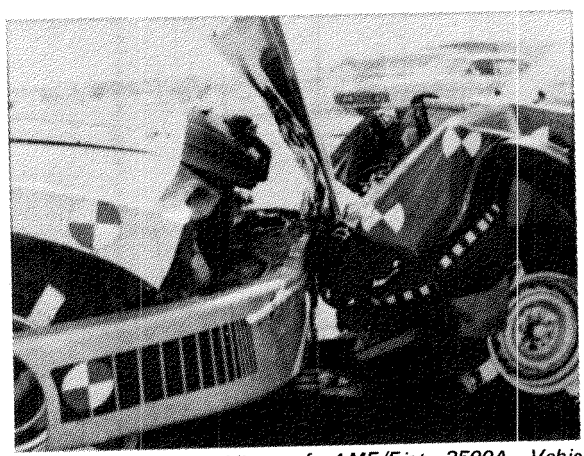


Figure 4. Post-test View of AMF/Fiat 2500A Vehicle Interface

TABLE 2
SUMMARY OF STRUCTURAL TEST DATA

PARAMETER	AMF/FIAT 2000 TEST		AMF/FIAT 2500A TEST	
	AMF	FIAT	AMF	FIAT
Test Weight (lb)	5273	2436	5273	2688
Vehicle Speed (mph)	37.3	-37.3	37.8	-37.8
Velocity Change (mph)	24.6	56.5	27.0	56.1
Dynamic Crush (in.)	21.8L/ 16.2R	24.4 @ 65 msec	19.8L/ 18.8R	23.9 @ 84 msec
Residual Intrusion (in.)	0	5.3L/ 5.4R	0	2.6L/ 4.5R
Peak Acceleration (G)	-26.2 @ 25 msec	-64.9 @ 46 msec	-28.7 @ 24 msec	-64.5 @ 31 msec

The advantage of a force-sensitive hydraulic system design in reducing the aggressiveness of larger vehicles toward smaller vehicles was demonstrated by the large total stopping strokes obtained. Total crush strokes of 43-46 inches were obtained (Figures 5 and 6), compared to perhaps 30 inches to be expected in similar Fiat impacts with conventional fixed-force front end structures.

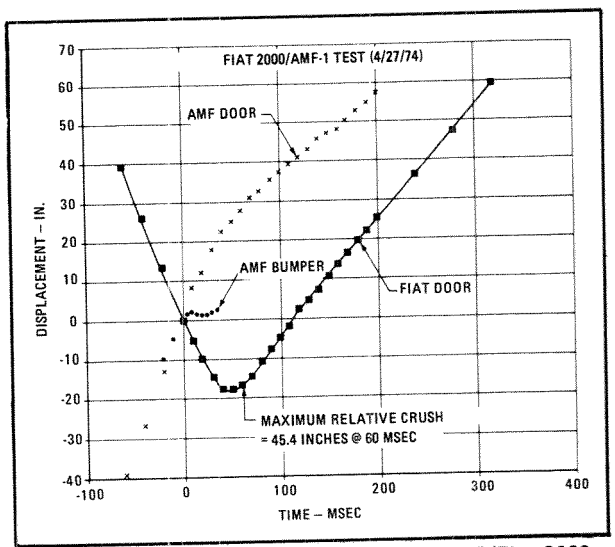


Figure 5. Displacement/Time Histories for AMF/Fiat 2000 Test

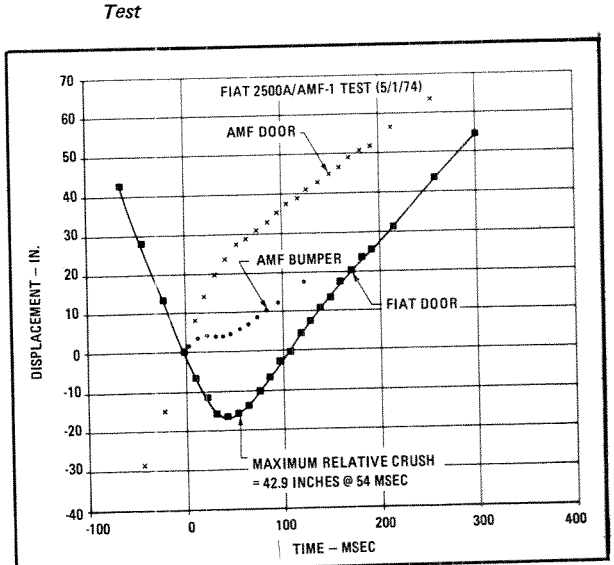


Figure 6. Displacement/Time Histories for AMF/Fiat 2500A Test

The hydraulic stroke of the AMF vehicle was approximately 20 inches compared to 24 inches of dynamic crush for the Fiats. Clearly, the larger vehicle absorbed a significant portion of the crash energy in these collisions. It should be noted that a fixed-stroke hydraulic design similar to the Fairchild ESV hydraulic subsystem would probably provide a longer hydraulic stroke (Figure 7). Such a design

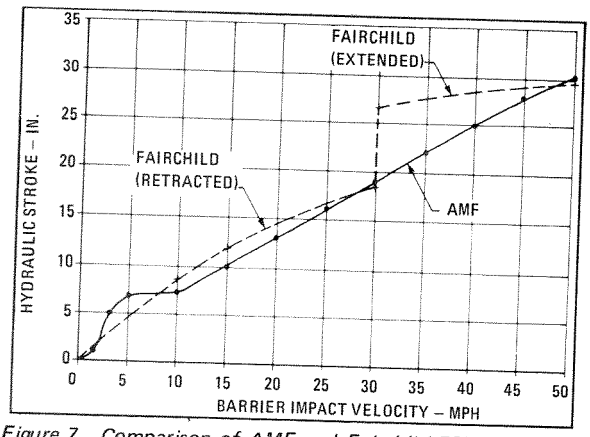


Figure 7. Comparison of AMF and Fairchild ESV Hydraulic Strokes in Flat Barrier Impacts

could therefore be even less aggressive than the AMF variable-stroke design at these closure speeds.

The maximum residual intrusions into the Fiat compartment (Table 2) were near the firewall centerline and did not present any real hazard to occupant safety. The intrusion into the AMF compartment was negligible since the AMF stroke was much less than its 30-inch allowable. As a corollary, a lighter-weight AMF-type cylinder design having a maximum stroke of only 22 inches would have been adequate for Fiat crashworthiness protection at this 75-mph closure speed.

The AMF and Fiat acceleration/time histories for the Fiat 2000 test are shown in Figures 8 and 9. These data are based on SAE J211 Class 60 filters. The corresponding velocity/time histories are shown in Figures 10 and 11, based on integration of the accelerometer data. The acceleration/time histories and velocity/time histories for the Fiat 2500A test are shown in Figures 12 through 15. These data are based on the average for the right and left frame accelerometers near the B pillar of the vehicles. The peak vehicle decelerations for these locations are included in Table 2. It should be noted that the peak G for the AMF and Fiat compartments did not occur at the same time, although there is a transient g crest for the AMF at the time of Fiat peak g (Figures 8, 9, 12, and 13). The bumper/engine dynamics at the

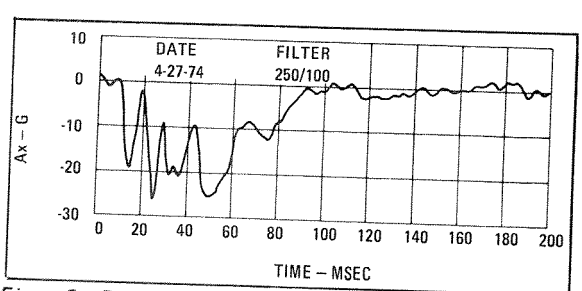


Figure 8. Deceleration of AMF Compartment During AMF/Fiat 2000 Collision

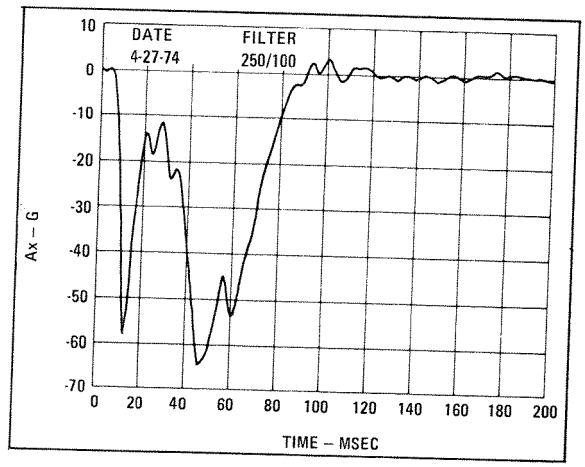


Figure 9. Deceleration of Fiat 2000 Compartment During AMF/Fiat 2000 Collision

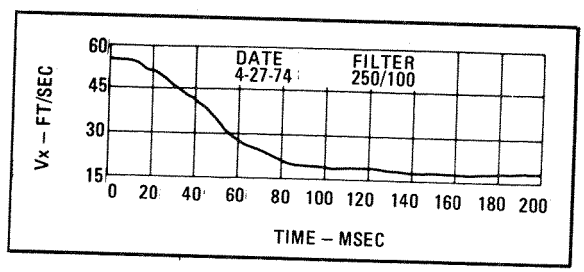


Figure 10. Velocity of AMF Compartment During AMF/Fiat 2000 Collision

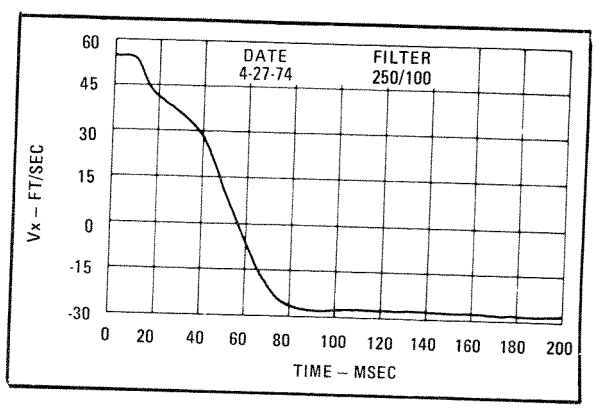


Figure 11. Velocity of Fiat 2000 Compartment During AMV/Fiat 2000 Collision

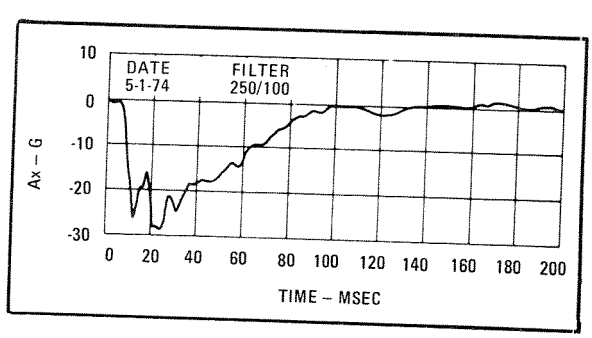


Figure 12. Deceleration of AMF Compartment During AMF/Fiat 2500A Collission

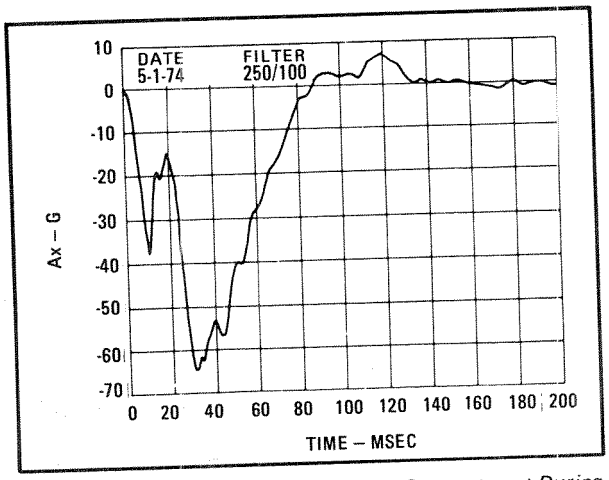


Figure 13. Deceleration of Fiat 2500A Compartment During AMF/Fiat 2500A Collision.

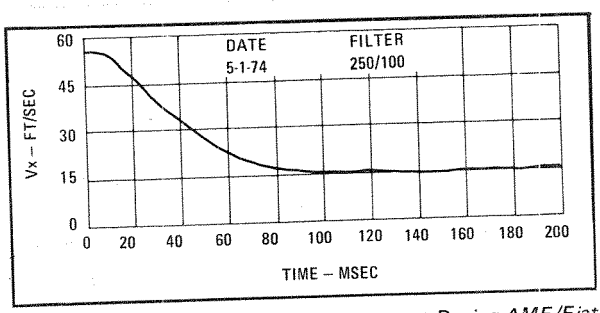


Figure 14. Velocity of AMF Compartment During AMF/Fiat 2500A Collision.

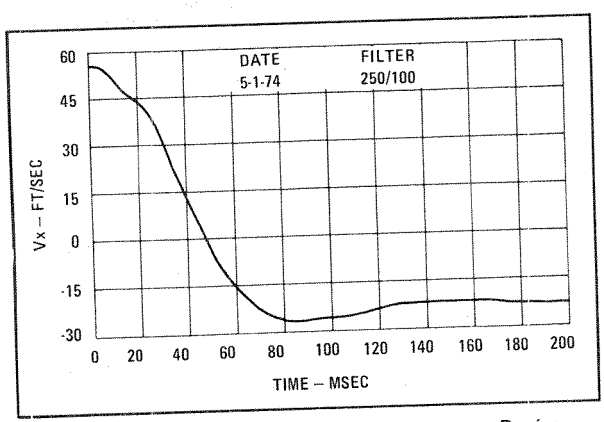


Figure 15. Velocity of Fiat 2500A Compartment During AMF/Fiat 2500A Collision.

vehicle interface apparently had a significant effect on AMF compartment response during the initial impact phase.

The unequal stroking of the AMF cylinders during the Fiat 2000 test (Figure 16) may have been due to the dynamics of the Fiat engine configuration (front wheel drive) which has the engine located on the right side of the vehicle. Time-correlated traces of vehicle frame and engine accelerations are presented in Figures 17 and 18. From Figure 17 we can conclude that the deceleration response of the Fiat was fairly symmetrical about its longitudinal centerline. However, the AMF compartment response (Figure 17) is radically different on the right and left sides.* There are several theories that are being investigated to explain this unexpected phenomenon, including the possibility that the off-center engine in the small car behaved aggressively toward the larger car.

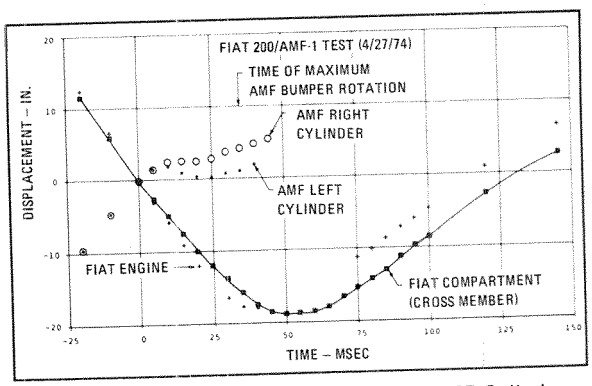


Figure 16. Displacement/Time Histories of AMF Cylinders and Fiat 2000 Components.

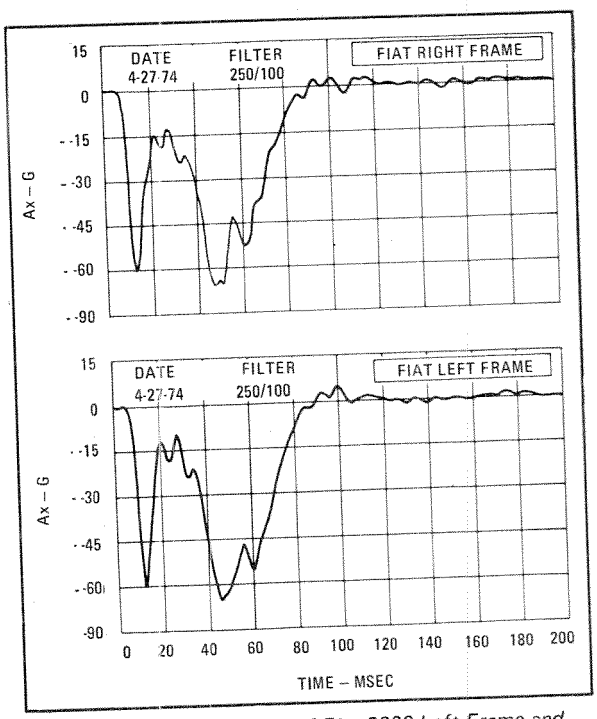


Figure 17. Time Correlation of Fiat 2000 Left Frame and Right Frame Acceleration Data.

*References to right and left sides are based on the driver position in each vehicle. Thus, the Fiat right side is opposite the AMF left side. Similarly, decelerations are referenced to initial velocities of each vehicle being in the positive X direction for that vehicle. The Fiat +X direction is opposite to the AMF +X direction.

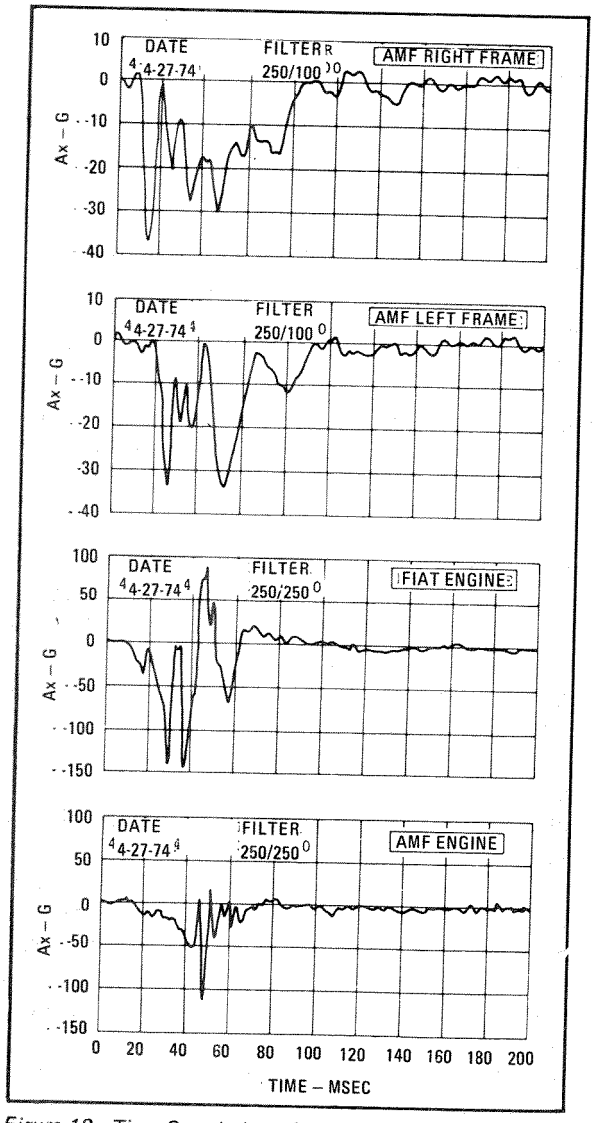


Figure 18. Time Correlation of AMF Engine, Fiat Engine, AMF Left Frame, and AMF Right Frame Acceleration Data.

CONCLUSIONS

1. The vehicle-to-vehicle tests were successful in that structural integrity of the Fiat passenger compartments were adequately maintained.

2. The AMF vehicle's hydraulic system stroked 20 inches compared to 24 inches of dynamic crush for the Fiats, indicating that the heavier AMF vehicle absorbed a significant portion of the crash energy.

3. A lighter-weight AMF-type cylinder design having a maximum stroke of only 22 inches would have been adequate for Fiat crashworthiness protection at this 75-mph closure speed.

4. Human survival in terms of meeting current occupant injury criteria was not evaluated since dummies were not included in this preliminary test series.

REFERENCES

1. G. Puleo, "Experimental Safety Vehicles in the 1,500 lb, 2,000 lb, and 2,500 lb Weight Classes – Body Design and Crashworthiness," SAE Paper 740207, March 1974.
2. "Experimental Safety Vehicle (Family Sedan)," AMF Incorporated, March 1972 (DOT-HS-800-689).
3. "The Fairchild ESV Family Sedan, Volume 2," Fairchild Industries, May 1972 (DOT-HS-800-694).

BIG AND LITTLE CAR COMPATIBILITY

JEROME M. KOSSAR
Office of Vehicle Safety Research
National Highway Traffic
Safety Administration

ABSTRACT

To reduce highway deaths and injuries, it is necessary to consider new design approaches for vehicles and their occupant restraint subsystems. We must give attention not only to the safety of the occupants of our vehicle but also to the safety of the human cargo aboard the vehicle with which we may collide. The higher rates of injury and fatality currently suffered by smaller car occupants and the increasing number of these smaller vehicles in our traffic mix demand that efforts be made to reduce the aggressiveness practiced by large cars in collisions with small cars. It is also essential that small cars themselves be better designed for their occupants' safety.

A theoretical evaluation of potentials for achievement of these goals through use of improved occupant restraints together with hybrid vehicle front end designs has been accomplished in the ESV Program Office of the Research Institute, NHTSA. The hybrid vehicle front end consists of short stroke, self force regulating hydraulic cylinder in tandem arrangement with crushable vehicle structure. The improved occupant restraint provides most of the best features of pretensioning, force limiting, early occupant engagement and good load distribution on the occupant with resulting restraint functioning efficiencies not otherwise possible.

The studies have demonstrated a design approach for vehicles which produces vehicle compatibilities in head-on closures of 70 mph between a 1,500 pound car and any other vehicle up to 5,000 pounds causing a 29½ inch crush of the small car. Another design approach is presented which indicates potential for reducing even this level of small car crush in collisions with large vehicles.

INTRODUCTION

This paper will present some of the work recently accomplished by the author in the Experimental Safety Vehicle Programs Office of the Research Institute at NHTSA. The research findings and opinions expressed herein do not necessarily reflect current or future Department of Transportation policy or direction. The paper does, however, reflect the ever-growing conviction among researchers and policy makers of the need to approach auto safety problems on the basis of total systems. A purpose of this paper is to advance general knowledge among the automotive community of the significant potentials for overall safety offered by hybrid vehicle designs and improved restraint systems. Another and equally important objective is to encourage other researchers to present their fundamental evaluations of other potential paths to overall automotive safety. Such written efforts are essential to insure the adequacy of final decisions.

Alleviation of significant injuries in automotive crashes involves a crucial consideration of both vehicle and occupant restraint design. The functions of each must be compatible for ideal protection to a vehicle's occupants. However, occupants of struck vehicles are not necessarily assured protection merely on the basis of the design of their struck vehicle and its restraints. Of equal importance to their survival is the crash performance characteristics of the other vehicle which strikes their car. This paper will endeavor to present a clear treatment of these considerations and will present analytical logic leading to good solutions. Most of the subsystems described herein have been the subject of extensive development over the past years, primarily under the aegis of the ESV program. There has been, however, an inadequate communication of some of the most fundamental and gross potential advantages of these system efforts. This paper may partially correct this fault.

For an idealized impact between two vehicles of any mass (assuming colinear crash responses, no roadway drag loading introduced by crush, and no restitution at the end of vehicle crush phase), the sum of energies absorbed in the two vehicles during crash are readily determined.

M_1, M_2	mass of vehicles 1 and 2, respectively
V_1, V_2	pre-impact velocities of vehicles 1 and 2
V_c	pre-impact closing velocity ($V_c = V_1 - V_2$)
V_f	post-crash final common velocity of vehicles 1 and 2
E_{at}	total energy absorbed during crash

To satisfy conservation of momentum requirements:

$$M_1V_1 + M_2V_2 = (M_1 + M_2)V_f \tag{1}$$

$$V_f = \frac{(M_1V_1 + M_2V_2)}{(M_1 + M_2)}$$

To satisfy conservation of energy requirements:

$$1/2\, M_1V_1^2 + 1/2\, M_2V_2^2 = 1/2(M_1 + M_2)V_f^2 + E_{at}$$

substituting for V_f from equation (1), defining V_c as above, and solving for E_{at} we find

$$E_{at} = 1/2\,(M_1M_2)V_c^2/(M_1 + M_2) \tag{2}$$

Equations (1) and (2) clearly demonstrate that the post-crash common velocity of the two vehicles and the sum of the energies absorbed in both vehicles is determined only by the masses and pre-crash velocities of the two vehicles and are totally independent of the individual crash characteristics of the vehicles. Although equation (1) shows that the post-crash common velocity is dependent on the pre-crash speed and mass of each vehicle, equation (2) indicates the total absorbed crash energy to be independent of the absolute velocities of each, but rather is dependent only on the difference in their pre-crash velocities (V_c). Equation (2) also demonstrates that a large mass stationary vehicle struck by a smaller mass vehicle moving at velocity V results in no less energy absorption in the two vehicles than if the smaller massed vehicle were initially stationary and struck by the heavier vehicle moving at the same velocity V.

In actual crashes it is usually observed that the two vehicles separate after the crash event. This separation is the result of some elastic compressive strain existing in the vehicles at the time that the E_{at} requirement of equation (2) is satisfied. The elastic compressive strains then return to zero, but in the process exert forces at the interface between the two vehicles. These forces generated during expansion of the crushed vehicles represent work which imparts a differential velocity to the two vehicles which had both been moving at the same velocity at the instant of maximum energy absorption, this instant being defined by equation (2).

Before proceeding further, it is advisable to reflect on the limitations of equations (1) and (2) relative to real-world accidents. There probably never has been a real-world vehicular accident that perfectly satisfies all the condition assumptions made. Even in perfectly aligned head-on crashes, vertical accelerations are invariably produced on the vehicles, usually in the form of pitch, resulting from the vehicle center of gravity being above the effective reacting line of crash force. The amount of energy absorbed during these vertical excursions is only a small percentage of the total crash energy and may, therefore, be rationally ignored in approximate work. The same rationale of relatively low energy excuses the overlooking of wheel lock-up caused by vehicle crush which is common in severe crashes.

Of all accidents involving two vehicles on real roadways, only a small number involve truly aligned head-on collisions. The vast majority involve some non-symmetry or angularity between the vehicles. These conditions allow to varying extents for relief from the velocity change and energy absorption requirements dictated by equations (1) and (2), since clearly any energies retained in the vehicles in the form of ricochet velocities need not be absorbed by the work of vehicle deformation. In fact, Langwieder, in his study of German real-world accidents, has identified as a factor in favor of small cars the ease with which they are redirected in angular and asymmetric impacts.[1]*

Although the derived equations are also applicable to side impacts, the incidence of catastrophic structural failures, vehicle wraparound and tire side drag forces associated with the most severe front-to-side impacts makes an accounting of absorbed energy very difficult.

Since an aligned head-on crash imposes the most severe velocity change and energy absorption requirement on the striking vehicles of all car-to-car crash modes, this will be the primary event addressed.

There is only one valid criteria for judging the degree of safety afforded a vehicle occupant and that is his physical condition after the collision. His safety partially depends on structure which is capable of absorbing required crash energies without impinging into his body as he travels through some volume within the passenger compartment during the crash. Another requirement is that his body suffer no loadings beyond what it can tolerate without exceeding moderate level injuries. These are two mutually related requirements, neither of which may be ignored.

Since the recent dramatic recognition of limited energy resources has settled on the American scene, the car buying public has further accelerated its prior

*Superscript numbers refer to references at end of this paper.

trend towards smaller vehicles. Figures 1 and 2, reproduced here from an *NHTSA Technical Note* by Mr. Donald F. Mela,[2] clearly demonstrates the hazards to which small car occupants are subjected in the current traffic mix based on a recent New York State study. The current upsurge of small car buying surprisingly may eventually result in an improvement in the safety of small car occupants in that fewer and fewer large (by current definition) cars will remain in the future traffic mix, thereby reducing the probability of big car collisions with small cars as they now are constituted. However, this eventuality would be far downstream and it is very unlikely that we will ever have only a single weight vehicle constituting our entire traffic scene. To put it simply, there is likely to be a big car-small car problem existing for the foreseeable future although our weight definitions of big and small may dramatically change.

Therefore, in addressing vehicle safety, it is obvious that some accommodation of small cars must be produced in the designs of large cars. We cannot change the fact that in a head-on between two vehicles having a weight ratio of two, the smaller vehicle will experience approximately twice the deceleration of the larger vehicle. What must be accomplished is an accommodation that limits forces and resulting decelerations in the small vehicle to levels that its occupants can survive when reasonable closing speeds are considered.

All cars must exhibit crush characteristics which allow absorption of all required energy in barrier impacts (its entire initial kinetic energy) without intrusion of elements into that portion of the passenger compartment volume through which an occupant travels during his deceleration. The total crush of the vehicle in barrier crash at design barrier speed represents its maximum allowable stroke capability. The burden of accommodating a small car imposes a restriction on the allowable force generated by the large car as its crush progresses through at least a portion of its maximum stroke capability. The key

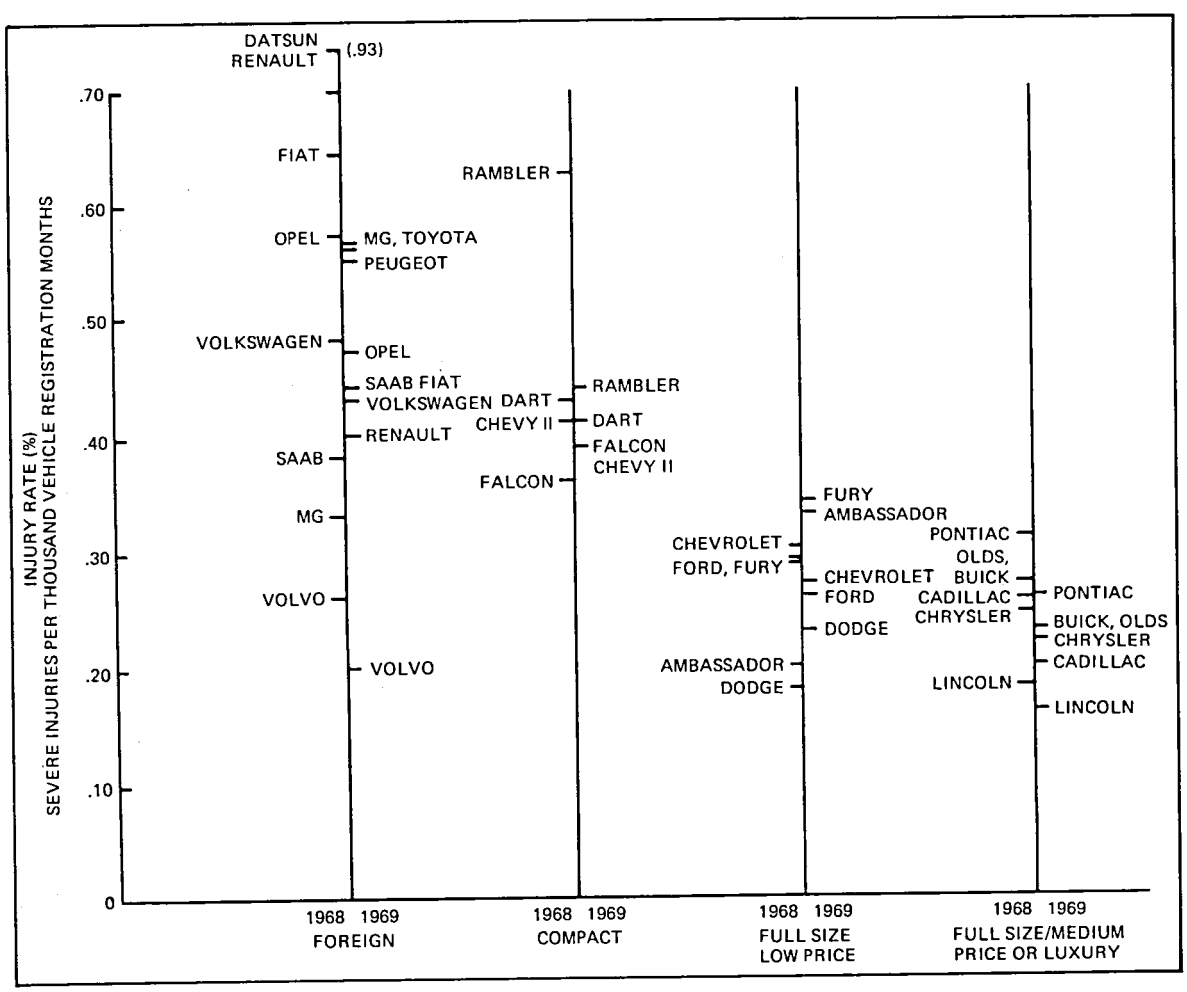


Figure 1. Injury Rates for Four Groups of Cars 1968 and 1969 Models

requirement is that each car, small and large, must be capable of performing adequately in barrier crash at design barrier speed without violating survival prerequisites of its occupants.

This paper will now address engineering approximation techniques that may be applied to the achievement of a specified minimum safe closing speed between two cars, this speed being independent of the relative mass of each. The selection of such a minimum safe closing speed is, of course, a difficult task and should be the result of extensive statistical research establishing benefits coupled with good cost-accounting to establish the levels of additional consumer costs as a function of increasing tolerable closing speeds.

The conditions which must be met by the two opposing cars at design closing speed are not overly complex. The smaller car may be allowed to suffer the same extent of crush as it experiences in its design barrier speed crash. However, since the engagement of forward structure in a barrier crash is more complete and uniform that generally can be anticipated in car-to-car collisions, it may be reasonable to assume that only 90 percent of the barrier test energy absorption can be achieved in the small vehicle at its maximum safe stroke capability demonstrated in barrier crash.

In order to achieve any reasonable high safe minimum closing speed, it is necessary to demand that the larger car in the crash absorb some portion of the total absorbed crash energy determined from equation (2). This is the area of accommodation required of larger cars.

A paper from NHTSA[3] had suggested an approach to this accommodation involving deflection-sensitive structures whose axial crush characteristics produce linearly increasing forces up to a specified force level with additional crush occurring at that constant plateau level of force. Some theoretical work has indicated possible disadvantages relative to current restraint systems function resulting from the slow onset loading aspect of this approach and the high maximum vehicle decelerations required of light cars.[4,6]

Work reported by Dr. H. Appel[7] during Volkswagen ESV efforts suggests the use of hybrid energy management systems. The hybrid system employs fixed stroke hydraulics in series with controlled crush structures to gain accommodation between different vehicle sizes. A schematic depicting a hybrid is seen in Figure 3.

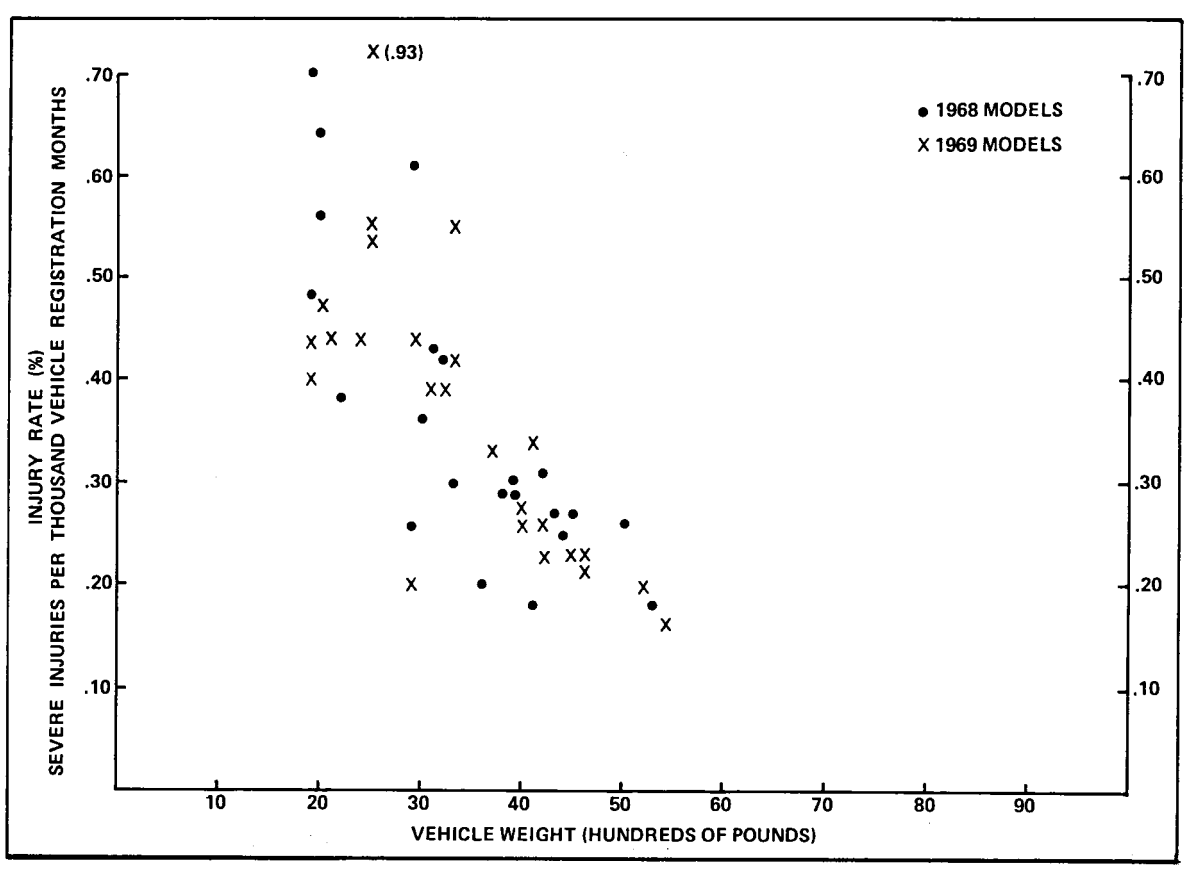


Figure 2. Injury Rate as a Function of Vehicle Weight

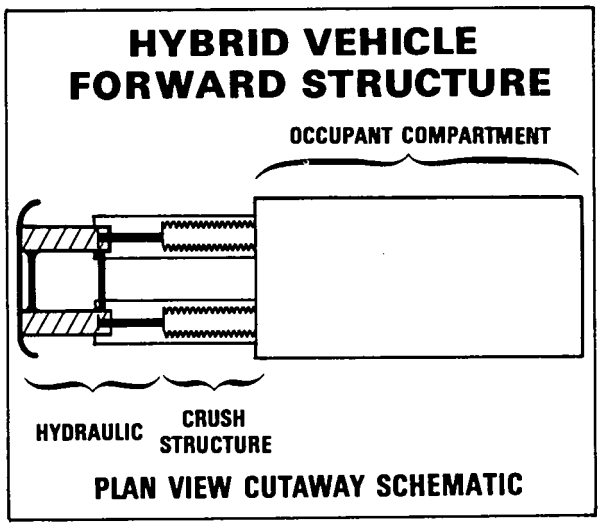


Figure 3. Hybrid Vehicle Forward Structure

In order to understand the accommodation mechanism in hybrid design, it is necessary to briefly review efficient hydraulic stroke characteristics. A fixed stroke hydraulic cylinder with piston can be designed to produce nominally unvarying resistance to piston motion within the cylinder throughout barrier crash stroke at all crash speeds. Such designs incorporate methods to diminish hydraulic fluid escape orifice areas from the cylinder as piston stroke progresses. Typically, orifice areas can be reduced as a function of piston position by means of a tapered metering pin fixed to one member (the piston or cylinder) which progresses through a fixed diameter orifice in the other member, thereby producing an annular oil escape orifice which diminishes in area as piston stroke progresses. This was done in the Fairchild Hiller ESV design.[8] Another technique was used by AMF in its ESV design. AMF employed orifices in the cylinder wall arrayed along the cylinder length. These orifices were progressively covered and passed by the piston face during its stroke, thereby diminishing total oil escape orifice area as the piston stroked.[9]

Regardless of the specific cylinder design approach employed to achieve a constant force response in barrier crash, the resistance to piston stroke is directly proportional to the square of instantaneous piston velocity within the cylinder. The basis for design is the achievement of a constant force throughout entire piston stroke. For the saké of simplicity in this paper, the normal reduction of decelerated vehicle mass and addition of effective mass by the occupants occurring during vehicle crush is overlooked. It now becomes a simple matter to approximate forces resisting piston motion in any crash knowing instantaneous piston position in the cylinder, instantaneous piston speed relative to the cylinder, and the system design barrier speed with associated design constant load. It is this feature of hydraulic designs which allows them to be self-regulated in their response to the crush resistance offered by a struck vehicle.

A simplified development of the implicit force, speed, and displacement relationships for an appropriately designed hydraulic cylinder will now be developed. Basically, stroke resistance of a cylinder is proportional to the square of piston relative velocity, therefore:

$$F_d = K_o V_{od}^2 \tag{3}$$

where

F_d = design stroke resistance

K_o = cylinder factor of proportionality at initial barrier contact

V_{od} = initial piston speed at design vehicle barrier impact velocity

Since the resistive cylinder force is designed to be constant through the entire barrier impact crush stroke, factor K must be made a variable factor by appropriate selection of cylinder orifice area change during piston stroke such that:

$$K_x = F_d / V_{xd}^2 \tag{4}$$

where

K_x = cylinder proportionality factor at each position (x) of the piston

V_{xd} = instantaneous design velocity of piston in cylinder at any position during its design barrier crash

The instantaneous design velocity of the piston at any position during its stroke within the cylinder may be expressed, consistent with the desired constant vehicle deceleration, as:

$$V_{xd}^2 = V_{od}^2 - 2A_d X \tag{5}$$

where

A_d = desired constant vehicle deceleration during barrier impact at design impact velocity

X = stroke of piston in cylinder when design instantaneous piston velocity is to exist

Substituting equation (5) into equation (4):

$$K_x = F_d/(V_{od}^2 - 2A_dX) \qquad (6)$$

The cylinder proportionality factor, K_x, is a relationship that must be satisfied by a variation of oil escape orifice area during piston stroke in order to maintain a constant force throughout the entire piston stroke during a barrier impact.

Generalizing equation (3), the resistive force created by a cylinder designed for constant force barrier impact as a function of piston position and velocity in the cylinder is:

$$F_x = K_xV_x^2 \qquad (7)$$

where

F_x = resistive cylinder force at any position of the piston within the cylinder

Substituting equation (6) into equation (7) we get:

$$F_x = F_dV_x^2 /(V_{od} - 2A_dX) \qquad (8)$$

A relationship establishing the maximum crush stroke under the desired barrier crash condition of constant deceleration is:

$$V_{od}^2 = 2A_dX_{mxd} \qquad (9)$$

where

X_{mxd} = the full crush stroke

Rearranging equation (8) we can get:

$$F_x/F_d = V_x^2/(V_{od}^2 - 2A_dX) \qquad (10)$$

Substitution of equation (9) in equation (10) leads to:

$$F_x/F_d = (V_xV_{od})^2/(1-X/X_{mxd}) \qquad (11)$$

Equation (11) provides direct means for determining cylinder force during any collision event if stroking velocity and position can be established for the piston.

A hybrid design incorporates a series array of hydraulic cylinder and back-up supporting structure, the back-up structure being designed to crush plastically at the design barrier impact load. With such a front energy absorber design, the piston stroking speed adjusts to obtain a piston resistance corresponding to the instantaneous crush resistance of the struck vehicle. This accommodation will be automatically accomplished allowing stroke to occur simultaneously in both vehicles during the full stroke of the hydraulic cylinder portion of the hybrid system. If the closing velocity of the vehicles required more energy absorption than is accounted for in the crushing of the small vehicle to its design barrier crush limit plus the energy absorbed in the hydraulic stroke of the larger vehicle, then a combination of two undesirable results occurs. The small vehicle is exposed to both accelerations and crush levels higher than its basic barrier impact design. These results are not acceptable since the small car occupants would be excessively endangered if restrained by currently existing restraint systems, or if resulting excessive crush impinged on the occupants.

The hybrid front end also offers obvious advantage in front-to-side vehicular crash interactions. In the vast majority of cases, the greatest imperiled are the occupants of the side-struck vehicle. The energy absorbing capabilities of vehicle side structures are limited by both relatively low allowable safe deformations and compartment internal design adequacy for safe transfer of high lateral impulse loadings to the occupants. Improvement of the crash function of the side-struck vehicle is, of course, highly beneficial. However, the design of the striking car can also provide appreciable relief to the occupants of the struck car. When the hybrid car is the striking vehicle, its self-regulation of force during its stroke, to match the deformation resistance of the weaker structure being struck, assures maximum collision energy absorption in the striking vehicle for a given stroke distance. Since the momentum transfer is pre-ordained by physics, the ability of the hydraulics to stroke under the relatively low crushing resistance of the side struck car tends to extend the time duration of the momentum transfer. This, of course, results from the relationship that momentum change is directly equal to the time integration of force. Therefore, the hybrid vehicle both reduces the net amount of energy to be absorbed by the struck vehicle and also reduces its maximum accelerations.

The establishment of a nominal vehicle g level which shall not be exceeded and is compatible with occupant restraint system function may be the first step towards arriving at compatible vehicle designs. Since the lowest level of vehicle deceleration provides the greatest assurance of restrained occupant survival and provides the least aggressive impact against other vehicles, the design vehicle g level should be minimized. The forward crush dimensions available in vehicles coupled with selected design barrier speed establish the lower bound of attainable vehicle g level. Table 1 presents design maximum g levels and associated vehicle crush for 35, 40, 45, 50 and 55 mph barrier design speeds. The assumption made in

preparation of this table is that with optimum design, a stroke averaged g level can be achieved which is 85 percent of the maximum g level maintained or exceeded continuously for an 8 millisecond interval. By defining maximum g as a level of 8 ms. duration, we essentially are attempting to direct concern only on maximum accelerations which would exert influence on loading of a properly restrained occupant within the vehicle. Dynamic forcing functions of substantially higher frequency than the natural frequency of the restraint/occupant dynamic system, superimposed on the main crash pulse shape, can be anticipated to have an insignificant effect on the occupant response. Therefore, amplitudes of higher frequencies need not be considered in designs for occupant survival. Davis had derived natural frequencies of conventional belted restraint/occupant systems to be in the 8 to 12 Hz range in his excellent paper advocating application of shock response spectrum techniques.[10]

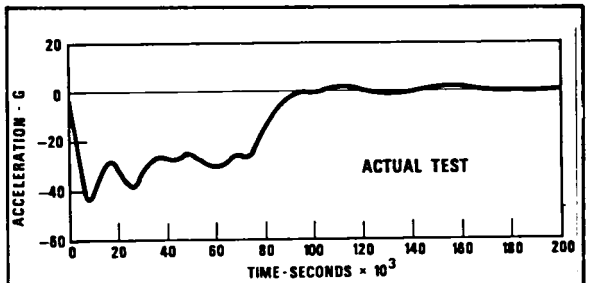
TABLE 1
MINIMUM VEHICLE CRUSH*
(INCHES)

Maximum Vehicle Design Acceleration	Design Rigid Barrier Crash Speed				
	35 MPH	40 MPH	45 MPH	50 MPH	55 MPH
25g	23.1"	30.2"	38.2"	47.2"	57.1"
30g	19.3"	25.2"	31.9"	39.3"	47.6"
35g	16.5"	21.6"	27.3"	33.7"	40.8"
40g	14.5"	18.9"	23.9"	29.5"	35.7"

*Assumes stroke averaged acceleration equals 85 percent of maximum vehicle design G.

Figure 4 depicts an actual deceleration history obtained during 50 mph barrier crash of the AMF Experimental Safety Vehicle. The figure demonstrates the validity of the assumption that the stroke averaged nominal acceleration is 85 percent of the maximum acceleration maintained or exceeded for an 8 millisecond interval. The chart on Figure 4 demonstrates another highly significant fact. Double integration of the actual test deceleration trace and the stroke averaged rectangular wave yield extremely good crush versus time agreement. This excellent agreement provides justification for the vehicle and occupant restraint analyst to simplify his initial efforts by paying heed to only the low frequency primary pulse shape of the crashing vehicle deceleration history. This provides easy computer modeling of the occupant in a crash and encourages more thorough and economical parametric evaluations.

Table 1 clearly demonstrates the significant increases in controlled frontal crush required to achieve lower g loadings or higher barrier design impact speeds. In order to evaluate the practical significance of the matched dependent variables shown in Table 1, several factors affecting occupant safety must be considered. The highest barrier design velocity at lowest vehicle deceleration level provides maximum occupant safety, but associated crush dimensions would involve severe design hardships in most vehicles of current configurations. A maximum vehicle frontal crush of 32 inches or less is thought to be an achievable goal without significantly altering current nominal vehicle appearances.[11] In some vehicles, a 32-inch crush may entail controlled intrusion of front-mounted engines into the passenger compartment volume. This engine intrusion must be limited by specific and reliable design in its location and magnitude so as to not be the agent of occupant injury or entrapment. Engine deflection techniques may also be considered alone or in conjunction with controlled intrusion to gain the required frontal crush. Rear and mid-engine locations provide still other potential for attainment of required frontal crush.



RESULTS OF DOUBLE INTEGRATION OF ACCELERATION

TIME (ms)	DISPLACEMENT (INCHES)		TIME (ms)	DISPLACEMENT (INCHES)	
	TEST	RECTANGULAR WAVE		TEST	RECTANGULAR WAVE
0	0	0	40	25.32	25.54
5	4.32	4.22	45	27.12	27.41
10	8.28	8.16	50	28.68	28.98
15	11.94	11.79	55	29.88	30.25
20	15.24	15.13	60	30.96	31.23
25	18.24	18.18	65	31.68	31.91
30	20.88	20.93	70	32.16	32.30
35	23.22	23.38	75	32.40	32.40

Figure 4. AMF ESV Barrier Test

Referring to Table 1, it is seen that none of the crush strokes are reasonable for the 55 mph design rigid barrier crash speed. Obviously, an acceleration level above 40 g could be shown to require only a

32-inch crush in a 55 mph barrier crash. However, a 40 g level appears to be a practical maximum for the case of near constant vehicle deceleration when considering current occupant restraint performance. Rather than obtaining an attenuation of vehicle decelerations with current systems, the occupant normally experiences a 1.5-2.0 amplification. Since it is generally accepted that limit chest accelerations should be no greater than 60 g, a 40 g limit on vehicle acceleration during critical time of occupant restraint loading is a maximum which should be considered at this juncture in restraint development.

Accepting this limitation in restraint performance, Table 1 shows the 50 mph barrier impact design at 40 g, 45 mph barrier impact design at 30 g, 40 mph barrier impact design at 25 g, and 35 mph barrier impact design at less than 25 g to each offer reasonable potential for occupant survival. These represent the lowest compatible vehicle decelerations at each barrier crash design speed that satisfy both current occupant restraint capabilities and reasonably achievable vehicle front end crush magnitudes.

HEAD-ON IMPACTS AND OCCUPANT/SURVIVAL SPACE

To consider the question of compatibility in car-to-car head-on impacts relative to assuring adequate occupant survival space, we must address equation 2.

$$E_{at} = 1/2\,(M_1 M_2/(M_1 + M_2))\,V_c^{\,2}$$

E_{at} is the total energy to be absorbed during the crash event. Since the interface forces between the two vehicles cause equal loading on each vehicle, it is clear that higher accelerations are induced in the smaller vehicle. In order to assure occupant survival in the compartment of the smaller vehicle, it is necessary that its crush and acceleration not exceed that obtained in its barrier design crash speed case. Obviously, this crush and acceleration limitation is possible only by limitation of interface loads to a value not exceeding the small car's design loading in barrier crash. This, of course, also implies that the large car should experience acceleration levels during small car impact which are below its design barrier crash condition.

The problem of accommodation relative to limitation of small vehicle loading during head-on impact with large vehicles reduces to a simple accounting of absorbed crash energy (E_{at}). The small vehicle itself should not be anticipated to be capable of absorbing the full amount of energy that it does in its design barrier crash condition, even though it suffers the same extent of crush. This reduced energy absorption results from the physical difference between its rigid barrier interface during barrier test and its opposing vehicle interface in car-to-car crash. The former is an unyielding fixed vertical plane engaging all of the vehicle structural elements reaching its structural surface. However, in a car-to-car crash, only those structural elements whose frontal loading paths contact stiff areas on the opposing vehicle will be fully engaged in absorbing energy during element crush. Other elements not directly loaded by the opposing vehicle will also distort, but their associated load deformation characteristics will usually be significantly reduced from those associated with barrier crash due to the introduction of lower energy failure modes. It is, therefore, recommended by the author that approximately 90 percent of a vehicle's design barrier energy absorption be considered as potential for energy absorption in aligned head-on impacts with larger vehicles.

To approximate tolerable closing speeds between unequal weight vehicles then involves adding 90 percent of the smaller vehicle's kinetic energy at design barrier speed to the energy absorption capability of the larger car under loads not exceeding 90 percent of the smaller vehicle's design maximum barrier load. If the larger vehicle were to employ full crush stroke hydraulics for its barrier design energy absorption, it could theoretically contribute the same level of energy absorption as the smaller vehicle at the critical closing speed and still not exceed the smaller vehicle's load and deformation tolerance. Full crush stroke hydraulic energy absorbers, therefore, constitute the most accommodating of all current energy absorbing systems. However, based on developmental, nonproduction engineered designs of full stroke hydraulic energy absorbers, significant criticism as to their producibility, cost and weight has been made. At this stage, it is, therefore, more reasonable to employ shorter stroke hydraulics so as to keep these debatable factors to a minimum. Engineering efforts accomplished by Volkswagen in their ESV work[12] has resulted in much more practical hydraulic units designed for function in hybrid front ends than the full crush stroke hydraulic systems of Fairchild[8] and AMF.[9]

The length of hydraulic stroke required in a hybrid system for a particular vehicle is a function of the vehicle effective mass, the design barrier crash speed established, the lightest vehicle weight to be accommodated in the traffic mix, and the selected maximum closure speed at which the smallest vehicle is not to exceed its occupant survival requirements regardless of which heavier weight vehicle it encounters.

HYBRID REQUIREMENTS

Considering the lightest protected vehicle to be 1,500 pounds and the desired maximum safe closing speed to be a parameter to be investigated, Tables 2 thru 4 demonstrate the influence of barrier design velocity on required hydraulic stroke as part of a hybrid system. The design speeds and accelerations in barrier impacts listed are those which have been derived from Table 1 as requiring less than 32 inches of crush.

An example of the calculations involved in deriving Table 2 is shown below. The case considers that all vehicles are designed to accommodate a 45 mph barrier impact with a maximum acceleration of 30 g, requiring a total crash stroke of 31.9 inches. The subject car has an effective weight of 3,000 pounds.

M_1 = mass of lightest car to be accommodated

M_1 = 1500 lb./g = 46.58 slugs

M_2 = mass of subject car

M_2 = 3000 lb./g = 93.17 slugs

Total energy to be absorbed in crash from equation 2 is

$E_{at} = 1/2 M_1 M_2 (M_1 + M_2) V_c^2$ where V_c = 88 ft./sec.(60 mph)

E_{at} = 120,248.6 ft.-lb.

The maximum allowable energy absorbed by 1,500 pound car in each car-to-car impact is 90 percent of its barrier design energy, therefore,

$E_{al} = 1/2\, M_1 V_b^2\, (.90)$ where V_b = 66 ft./sec (45 mph)

E_{al} = 91,313.7 ft.-lb.

Since

$E_{al} + E_{a2} = E_{at}$, $E_{a2} = E_{at} - E_{al}$, from which E_{a2} = 28,934.9 ft.-lb., which is the crash energy that must be absorbed by the 3,000 pound car at a load level which will not jeopardize the small car occupant. Because the smaller car nominal load acceleration in barrier impact is 25.5 g (30 g x .85), and it is assumed that only 90 percent of this value can be safely generated in car-to-car impact without exceeding small car design barrier deformation, the tolerable car-to-car load is:

Force = (1,500 lb.) (25.5 g) (.90) = 34,425 pounds

Since the 3,000 pound car must absorb 28,934.9 ft. pounds at a load level of 34,425 pounds, it requires an effective constant force stroke length of 0.84 ft. or 10.09 inches.

Therefore, the hydraulic stroke length to be included in the 31.9 inch potential crush length of the 3,000 pound vehicle is 10.1 inches. This hydraulic stroke length will be the only crush of the 3,000 pound car during the head-on 60 mph closure speed collision with the 1,500 pound vehicle and will occur at a load level compatible with the 1,500 pound vehicle designed for barrier impact.

Table 2 demonstrates the influence of a selected design barrier impact speed which is common to all vehicles regardless of vehicle weight. The lower the design barrier speed, the more accommodation must be provided by heavier vehicles. This is clearly the result of low barrier design speed limiting the amount of energy that may be safely absorbed by the lighter car but not influencing the total energy which must be absorbed in a car-to-car collision.

TABLE 2
COMPATIBLE HYDRAULIC STROKES FOR 60 MPH CLOSURE SPEEDS AGAINST 1500# CAR

Barrier Impact Design Speed	Total Barrier Impact Stroke	Maximum Barrier Impact Acceleration	Hydraulic Stroke Requirement (inches)						
			Effective Vehicle Weight – Pounds						
			2000	2500	3000	3500	4000	4500	5000
40 MPH	30.2 In.	25g	13.0 in.	17.0 in.	20.2 in.	22.7 in.	24.7 in.	26.5 in.	27.9 in.
45 MPH	31.9 In.	30g	4. I in.	7.5 in.	10. I in.	12.2 in.	13.9 in.	15.4 in.	16.6 in.
50 MPH	29.5 In.	40g	0.0**	0.0**	2.0 in.	3.6 in.	4.9 in.	5.9 in.	6.9 in.

**Adequate energy absorption capability in 1500# vehicle to accommodate car-to-car crash energy without supplement from large car.

The effect of higher barrier impact design speed is apparent in the 50 mph design barrier speed cases where it is seen that 60 mph closure speed can be accommodated with no crash energy absorption in the 2,000 pound car, all the required absorption safely occurring in the 1,500 pound car without exceeding its barrier design condition.

The influence of design barrier impact decelerations may also be deduced from Table 2. Comparing the required increments of large vehicle stroke for accommodation of the 1,500 pound car, it is seen that the increments are reduced as the basic barrier impact design deceleration is increased. This results directly from the fact that accommodation stroke of the large vehicle can occur at higher loads without violating the design limitations of the 1,500 pound vehicle.

Table 3 considers the effect of increasing desired safe closing speed between the 1,500 pound vehicle and vehicles up to 5,000 pounds effective weight. The table addresses 65 mph closing speeds. Each case of Table 3 (V_c = 65 mph) represents a demand for 17.4 percent more energy to be absorbed in the collision than had been required at 60 mph. We note from Table 3 that required accommodation strokes of the vehicles above 1,500 pounds have increased beyond those of Table 2 reflecting the need for additional energy absorption. Also, note that although total crash energy absorption requirements rose by only 17.4 percent in going from 60 to 65 mph closure speeds, the accommodation strokes increased by much larger percentages. It is also shown that the 40 mph, 25 g barrier design class of vehicles cannot be made compatible beyond the 1,500 to 3,500 pound range for 65 mph safe closure speeds. Compatibility can be obtained only when required accommodation stroke in the larger vehicle is less than or equal to its stroke at design barrier speed.

Table 4 representing required accommodation strokes for a 70 mph head-on closure speed, demonstrates that 45 mph design barrier speed case cannot provide accommodation in cars of 4,500 pounds since accommodation strokes exceed barrier design strokes in cars of that weight.

Tables 2, 3 and 4 have shown the approximate values of accommodation stroke required in larger

TABLE 3
COMPATIBLE HYDRAULIC STROKES FOR 65 MPH CLOSURE SPEEDS AGAINST 1500# CAR

Barrier Impact Design Speed	Total Barrier Impact Stroke	Maximum Barrier Impact Acceleration	Hydraulic Stroke Requirement (inches) — Effective Vehicle Weight – Pounds						
			2000	2500	3000	3500	4000	4500	5000
40 MPH	30.2 In.	25g	20.5 in.	25.2 in.	28.9 in.	31.8 in.	–*	–*	–*
45 MPH	31.9 In.	30g	10.4 in.	14.3 in.	17.4 in.	19.9 in.	21.9 in.	23.6 in.	25.0 in.
50 MPH	29.5 In.	40g	2.2 in.	5.2 in.	7.5 in.	9.3 in.	10.8 in.	12.1 in.	13.1 in.

*Incompatible – required stroke exceeds barrier design stroke.

TABLE 4
COMPATIBLE HYDRAULIC STROKES FOR 70 MPH CLOSURE SPEEDS AGAINST 1500# CAR

Barrier Impact Design Speed	Total Barrier Impact Stroke	Maximum Barrier Impact Acceleration	Hydraulic Stroke Requirement (inches) — Effective Vehicle Weight – Pounds						
			2000	2500	3000	3500	4000	4500	5000
40 MPH	30.2 In.	25g	28.5 in.	34.0 in.*	–*	–*	–*	–*	–*
45 MPH	31.9 In.	30g	17.1 in.	21.7 in.	25.2 in.	28.1 in.	30.4 in.	32.4 in.*	–*
50 MPH	29.5 In.	40g	7.2 in.	10.7 in.	13.3 in.	15.5 in.	17.2 in.	18.7 in.	19.9 in.

*Incompatible – required hydraulic stroke exceeds barrier design stroke.

vehicles as the value of desired safe closing speed increased. The results demonstrate that within the confines of a specified maximum stroke, higher compatible closure speeds require higher design rigid barrier impact speeds with associated higher design accelerations for all vehicles. The dollar cost of providing structure to accommodate these higher design loads must be carefully weighed against potential value to society before selecting a specific design safe closing speed requirement.

Up to this point we have addressed: first, the problem of assuring survival space for the occupants of small vehicles; second, providing vehicle acceleration versus time histories which show promise for the adequate function of restraints; and third, providing vehicle deceleration levels as low as possible to provide minimum threat relative to aggressiveness in all inter-vehicular crash modes. It is necessary now to address requirements of occupant restraints and internal occupant compartment free space necessary to assure survivable occupant loadings.

SURVIVAL VIA OCCUPANT RESTRAINTS

First, it is necessary that occupants survive under rigid barrier crash conditions for which their vehicle is designed. Second, and probably more productive relative to saving lives, the occupants must be spared lethal loading during head-on impacts with vehicles of different weight.

Restraint systems of current vintage in general are comprised of many compromises from optimum performance for many and varied reasons. Despite their relatively inefficient performance, occupant restraints of both moderate and high costs have been consistently shown to be cost-effective in that significant net overall societal savings are appreciated in their implementation. Perhaps it is appropriate to reconsider techniques of restraining occupants with an aim of incorporating in one system the better features of currently competitive systems.

Advocates of air cushion technology point out, in addition to the obvious potential for passive applications, that air bags distribute loading over large areas of the body. Intuitively we know that higher safe body decelerations may be imposed on the occupants so restrained than can be tolerated by devices imposing more local loadings on the occupant. With suitable design, the air cushion may be relied on to prevent severe head whip and associated threats to the occupants' safety.

Three point belt advocates have in recent years made significant improvement of their technology. Techniques of relocation and/or adaption of belt anchorages have provided systems which are essentially passive. Inflating belt techniques have the apparent dual advantages of increasing load distribution and causing earlier engagement of the occupant. Pretensioning systems have also been successfully employed to provide earlier occupant engagement.

In addition, researchers have experimented with several mechanical techniques to enhance the efficiencies of restraint systems. Use of collapsible bolsters and specially designed energy absorbing steering columns to reduce the distance between inflatables and occupant are promising techniques which reduce the time to occupant engagement and may improve stroking efficiencies. Such approaches, however, share some of the drawbacks of other restraint systems. Occupant size range accommodation requirements lead to reduced stroke efficiency for each individual occupant. This comment, of course, applies to all current restraints known to the author. Bolsters also leave some question as to customer acceptance resulting from their protuberance into what has customarily been free compartment volume. In any event, further development work would be required before their primary restraint function could be adequately achieved over the desired range of accommodated occupant sizes.

Other research techniques have been exploited to improve occupant stroke efficiencies by providing improved control of occupant loading over a wide range of collision speeds. These techniques employ load limiting devices which can limit maximum occupant loading and concurrently provide absorption of a portion of occupant kinetic energy during the load limiter distortion. Improved energy absorption functions have also been achieved by energy absorbing webbing which can be designed to provide appreciable elongations at prescribed load levels.

A concise listing of the attributes of an idealized restraint system will be beneficial at this point. It should:

- be applicable to a wide range of occupant sizes.
- provide protection to occupants over a wide range of accident modes.
- provide restraint as early in the collision event as possible to best utilize vehicle external crush for absorption of occupant kinetic energy.
- distribute load on the occupant and control articulation of the head.
- provide high occupant stroke efficiency to best utilize available stroking dimensions within the passenger compartment for absorption of occupant kinetic energy.
- provide safe restraint function over wide ranges of occupant pre-crash orientation.

Ideally, the restraint system should also be non-intrusive, non-confining in non-crash storage, non-confining in post-crash function, rapidly self-restoring to a renewed ready condition after impact, and economical in both original cost and maintenance.

The foregoing comprises a listing, perhaps incomplete by some judgments, of what features an occupant restraint system should possess. I cannot currently describe even a hypothetical system that contains all of the ideal attributes. I would, however, like to describe a concept that includes most of these features and which could be developed into hardware without significant new development requirements. The technology requirements for the components have, for the most part, already been developed. The system development, however, does require sound engineering effort to array components in an efficient and practical manner. The resulting system would better supply the majority of stated ideal attributes than any other system of which I am aware.

Visualize a deep sea fishing reel having an adjustable drag fastened to the deck of a moving boat. As a fish, while frantically running, pulls on the line with sufficient force, he unreels it at the drag force which you had set. He is doing work. The work rate is equal to your drag setting in pounds multiplied by the sum, in foot per second, of the rate of unreeling of line and boat speed. As the fish tires, the rate of line unreeling diminishes until the fish can no longer muster the force required to overcome your drag setting. He is still pulling on your line, he is still doing work. The work rate is now that diminished force he is exerting multiplied by the boat speed. During this phase, the fish is no longer able to increase the distance between himself and the boat and is, therefore, constrained to the boat velocity, but yet he is still working. Upon exhaustion, the fish can no longer exert load on the line and no additional work is accomplished.

The above analogy serves to describe the function of the load limiter assembly which is a major component of the system I am proposing. During the crash event, the load limiter components, attached to car structure behind the occupant, allows the withdrawal of steel ribbons at self-adjusting load levels suited to the occupant's weight. The occupant will move forward in his seat during vehicle deceleration, transferring some of his kinetic energy into the work of drawing out the load limiter ribbon. If the occupant is decelerated to vehicle velocity before the completion of vehicle crush, he will no longer draw out force limiter ribbon, but remain fixed in his relative position to the seat while decelerating at car deceleration levels until completion of the crash event.

The steel ribbons are not in contact with the occupant but instead provide restraining forces to the three corners of a low volume air bag which is in front of the occupant. The bag in its undeployed state is folded into a triangular shape lying with appreciable slack over the chest of even the largest intended occupant. One edge of the triangular planform runs across the lap, another edge droops loosely approximately along the course of current torso restraint belts, while the third edge hangs nearly vertical beyond the outboard extremity of the occupant. During deployment, the air bag must grossly expand so as to extend in front of the occupant's face while initiating onset of occupant deceleration. The system should not require retractors, as snug fit is actually undersirable from the standpoint of user acceptance. Sufficient slack must be present to allow unhindered whole body and torso movement whereby most normal occupant activities can be accommodated. If the undeployed bag proves to be an annoyance due to its insulation properties while draped across the chest, accordion style folded tubes might be incorporated passing through the folded bag for occupant ventilation while preserving the bag's closed volume characteristics. Should the weight of the bag prove oppressive lying on the occupant the bag can be designed to stand-off the chest and lap by incorporating wire tendons within the bag construction, much the same as corset stays, the wire tendons being flexibly supported form the extremities of the steel load limiter ribbons.

Now let us review the potentials of such a system:

- It can be applicable to a wide range of occupant sizes since load is distributed on the occupant by a well-located air bag, the force level being essentially programmed by a force limiter which automatically adjusts to the occupant's weight. The details of this unique force limiter will be addressed in a subsequent section.
- Occupant protection can be provided over a wide range of accident modes. The proposed system can conceivably even augment near side, side impact occupant protection with suitable air bag design providing initial outboard occupant wrap-around and provision of adequate lateral impact sensor switches. Such side impact sensors might well be of the electrical contact type located within side structure where electrical contact is initiated by early plastic deformation of the vehicle outer shell components.
- Occupant restraint can be provided very early in the crash event since a relatively low volume inflatable, which is force limited by its anchorages,

must expand its fore-aft dimension only a small amount before exerting loads on the occupant. The effective moving mass of this bag (only the aft half) can be very small compared to conventional air bags, thereby diminishing bag slap effects. Since current impact sensors have been demonstrated which have sensing times in the three to five millisecond range, it is reasonable to consider lowest occupant onset loading to commence at ten milliseconds. Such frontal crash sensors include hydraulic cylinder pressure switches and bumper mounted impulse switches as used in the ESV program.

- Adequate load distribution and control of head articulation can readily be provided by the air bag.
- High occupant stroke efficiency within the passenger compartment is provided by the occupant force limiters structurally attached to the compartment. These limiters will allow occupant stroke to progress only when the occupant load matches the concurrent force limiter value.
- Occupants can receive adequate protection in crashes over a wide range of pre-crash occupant orientations. This is achievable as a result of the proximity and wrap-around nature of the inflatable and the controlled deceleration levels provided by the load limiter mechanisms.

Unlike the ideal restraint, this system cannot be claimed to be non-intrusive. It is there, in front of the occupant's chest, where it intrudes into areas which car occupants prefer to have free. It is relatively non-confining in non-crash configuration, but only to the degree that efficient layout can provide. It is not self-restoring to full ready condition after severe crash, but it can be anticipated to provide significant multiple impact protection during severe crash events by providing adequate, reasonable duration of influx gas to the air bag. In non-critical (non air bag deployment) crashes, the system can easily restrain occupants without degradation of a subsequent critical performance. This feature could allow elimination of the current energy absorbing function of steering columns. As to the quality of being non-confining in post-crash environment, this is primarily a function of the ingenuity of our designers. Conceivably, crash initiated timing delays can be designed to provide automatic disconnects after crash, should the deflated bag slack space prove to be insufficient for ease of egress.

The last, and perhaps most critical, consideration of economy is difficult to assess. The components are relatively simple. Air bag volumes are less than most current designs and gas sources need supply appreciably less volume. The force limiter design and fabrication is not complex, hopefully costing little more than current belt inertia reel retractors. Energy absorbing steering columns could be simplified with, hopefully, a cost savings to provide telescoping or low crush load function only. An honest, accurate system cost assessment cannot legitimately be made at this time. Only if development is implemented can vital cost factors be determined.

DESIRABLE OCCUPANT LOAD CHARACTERISTICS

Although researchers and designers have always recognized an advantage in spreading load distribution on the occupant, it was not until restraint systems for high speed protection came under development that considerations of rate of occupant onset load became a significant concern. Not a great deal of research has been accomplished to establish tolerable onset rates, or for that matter, to determine that onset rates are in fact either a dependent or independent parameter affecting occupant survival.

To emphasize the importance of this possible parameter and thereby induce accelerated research of this critical question, modeling work which is presented in this paper probes the influence of controlled onset loading effects on efficient utilization of available forward stroking distance within the occupant compartment.

The influence of maximum occupant acceleration and restraint delay is also evaluated. Since a single lumped mass (non-articulating) is the simulated occupant, it must be clear that the air bag design was not attempted. Resulting occupant acceleration was, however, taken as representative of upper thorax acceleration and, therefore, chest severity indices were computed.

The parametric variations of restraint function were performed during computer model simulations of hybrid vehicle/stationary rigid barrier crashes. Figure 5 shows the form of a restraint function versus time after initial barrier contact. The figure depicts a ten millisecond delay before occupant onset loading commences. The initial ramp portion reflects a 1500 g/sec. onset rate for occupant deceleration ending at 37 milliseconds when the 40 g level is attained. Beyond 37 milliseconds, the restraint function is capable of exerting 40 g decelerations on the occupant. The parameters of restraint delay, onset rate of loading and plateau g level were varied in the study.

As previously presented, based on the objective of employing minimum vehicle deceleration levels in vehicle crash design and not exceeding 32 inches of front end crush barrier impact, a special vehicle

design performance selection was made for each 50, 45, 40, and 35 mph barrier crash design condition. Since the 35 mph barrier crash design offers little promise for accommodation of high speed head-on vehicle to vehicle crashes, it was not carried forward for additional evaluation. The remaining three design characteristics were employed in computer simulations of barrier crashes to evaluate restraint characteristics. Figures 6 thru 9 depict typical computer run results from this study of restraint performance in barrier impact using the restraint characteristics shown in Figure 5. Figure 6 shows the motions incurred during barrier crash at 45 mph of the 30 g design vehicle. It demonstrates that although occupant travel over the ground was almost 39 inches, the restrained occupant moved forward with respect to his seat only seven inches. A tabulation of these motions is given in Table 5. Figure 7 shows the pertinent velocity and deceleration histories for this same 45 mph crash condition.

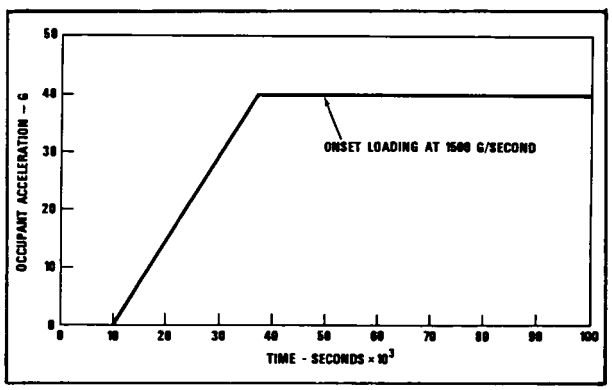


Figure 5. Occupant Restraint Time Function

Figure 7 shows that the occupant continued an undisturbed forward motion for 10 milliseconds after commencement of vehicle crash, after which he was decelerated while drawing out his force limiter. At 64 milliseconds he reached the vehicle's instantaneous velocity. After reaching vehicle velocity, he no longer drew-out his force limiter and was decelerated at the vehicle deceleration level until both he and the vehicle stopped at 81 milliseconds. As can be seen, the desired onset rate of 1500 g/sec. was achieved and a 40 g level then maintained until the occupant's motion relative to the vehicle ceased, after which he then underwent the lower deceleration of the vehicle. These results for the design condition impact of 45 mph with the restraint function chosen indicates even greater crash speed occupant protection potential if the cage structure of the vehicle passenger compartment were made capable of sustaining more than 30 g design level loads and allowed some additional foreshortening of the compartment space. This potential exists because the forward motion of the occupant within the compartment was only seven inches while most current vehicles can be counted on to supply 14 to 18 inches of free travel space, assuming for the driver the steering column to be of low load collapsible or telescoping design.

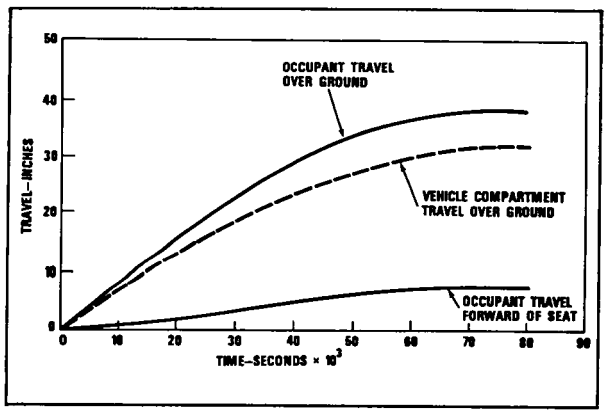


Figure 6. 45 mph Barrier Crash Travel Distances

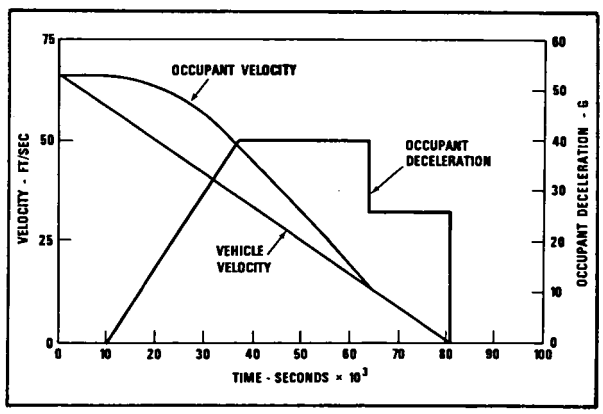


Figure 7. 45 mph Barrier Crash Dynamics

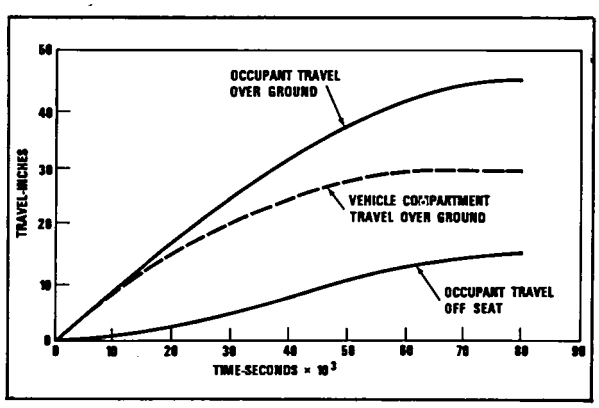


Figure 8. 50 mph Barrier Crash Travel Distances

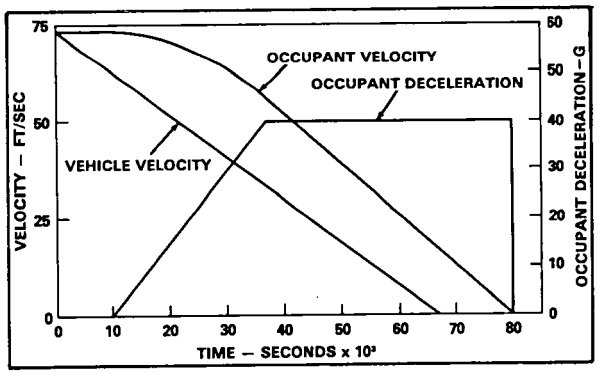


Figure 9. 50 mph Barrier Crash Dynamics

Figure 8 depicts the significant motions occurring during 50 mph barrier crash of a 40 g design vehicle. In this case, it can be seen that the occupant consumed over 15.5 inches of internal compartment space in his forward travel. This restrained motion accounts for a much more significant portion of his initial kinetic energy than was the case for the 45 mph design.

Figure 9 demonstrates the reason for the increased energy absorption demand on the restraint system in this 50 mph barrier crash. It results from the higher acceleration design of the vehicle which was required to accommodate the higher design crash speed within the bounds of allowable vehicle crush dimension. The occupant under the same restraint deceleration levels as in the 45 mph case is no longer decelerated to instantaneous vehicle velocity before the vehicle itself is stopped. He, therefore, draws out the force limiter ribbon throughout his entire stopping event and, unlike the 45 mph case, is subjected to a long period of 40 g deceleration. These two crash cases typify results of the parametric variations of restraint function variables.

It must be emphasized that at this point it was envisioned that all vehicles produced, regardless of weight, would be designed to provide survival in a specified speed, stationary rigid barrier crash. It was further assumed that each vehicle would be designed to produce the same vehicle deceleration levels at the specified crash speeds. These assumptions would allow all vehicle restraints to be designed to the same functional requirements and impose the same occupant load levels thereby standardizing minimum performance requirements on all restraints.

Since the vehicle design performance candidates had been reduced to three, there were three separate but complementary parametric restraint studies conducted, one for each design barrier crash speed.

Table 7 addresses the results obtained for the 40 mph 25 g vehicle candidate. As has been previously

TABLE 5
45 MPH BARRIER CRASH TRAVEL DISTANCES

Time (MS.)	Occupant Travel Over Ground (Inches)	Occupant Travel Forward of Seat (Inches)	Vehicle Compartment Travel Over Ground (Inches)
0	0	0	0
5	3.96	0.12	3.84
10	7.92	0.49	7.43
15	11.87	1.10	10.77
20	15.74	1.87	13.87
25	19.47	2.75	16.72
30	22.99	3.66	19.33
35	26.21	4.53	21.69
40	29.08	5.28	23.80
45	31.55	5.89	25.66
50	33.65	6.36	27.28
55	35.35	6.70	28.66
60	36.67	6.89	29.78
65	37.61	6.94	30.67
70	38.24	6.94	31.30
75	38.63	6.94	31.69
80	38.77	6.94	31.83

TABLE 6
50 MPH BARRIER CRASH TRAVEL DISTANCES

Time (MS.)	Occupant Travel Over Ground (Inches)	Occupant Travel Forward of Seat (Inches)	Vehicle Travel Over Ground (Inches)
0	0	0	0
5	4.4	0.16	4.24
10	8.8	0.66	8.14
15	13.2	1.47	11.72
20	17.5	2.53	14.97
25	21.7	3.78	17.89
30	25.6	5.14	20.49
35	29.3	6.54	22.75
40	32.6	7.91	24.33
45	35.5	9.22	26.30
50	38.0	10.47	27.58
55	40.2	11.66	28.53
60	42.0	12.80	29.15
65	43.3	13.88	29.45
70	44.3	14.84	29.47
75	44.9	15.44	29.47
80	45.1	15.65	29.47

explained, the occupant survival considerations are adequately satisfied by considering only the nominal vehicle deceleration pulse. For the hybrid vehicle designs proposed, the nominal pulse is well approximated by a rectangular wave whose plateau level is 85 percent of the effective maximum vehicle deceleration. Figure 4 has been employed to substantiate this procedure.

Cases A through D of Table 7 probe the influence of delay in the onset of restraint for this 40 mph barrier crash. In these cases, all the programmed onset rates and plateau maximum occupant decelerations of the restraint function were kept the same. The most significant column is seen to be that of occupant forward motion relative to his vehicle. As delay time is increased from ten to forty milliseconds a monotonic increasing function is indicated for the required frontal clear space in the compartment. It should be also noted that the thorax severity index is approximately doubled over the range of restraint delays.

Cases A and E through G investigate the influence of the maximum programmed occupant G when the onset rate and delay times are invariants. In these cases it is seen that the influences on occupant motion in the compartment and severity index are not strong but are consistent. Higher maximum G produced lower forward motions relative to the compartment but higher chest severity indices. The last three cases and Case B, representing a longer restraint delay, support these conclusions relative to occupant G level influence.

Table 8 concerns the 45 mph barrier crash restraint. Again, Cases A through D address the influence of increased time after initial vehicle to barrier contact before commencement of restraint function. The 45 mph case shows even greater increases of occupant travel within the compartment than did the previous 40 mph case. This is obviously the result of the increase in free travel speed of the occupant prior to restraint function. Examining the chest severity indices, one sees the anticipated increase for the ten through thirty millisecond restraint delays but the 40 millisecond delay does not cause additional increase. Both the thirty and forty millisecond delays produced occupant trajectories similar to that shown in Figures 8 and 9. Notice that this vehicle stops at eighty milliseconds but the end of restraint extension for these two cases occurred after this time. This means that the occupant was still moving forward when the vehicle stopped, therefore, at no time was the occupant allowed to enjoy the lower acceleration levels of the vehicle. Since the force limiters were programmed to provide the same

TABLE 7
PARAMETRIC RESTRAINT STUDIES IN BARRIER CRASH

Vehicle Performance Characteristics: 25g maximum in 40 MPH fixed rigid barrier crash with maximum crush of 30.18 inches occurring at 0.086 seconds.

Case Identification	Maximum Occupant Deceleration (G's)	Rate of Occupant Onset Loading (G/Sec.)	Restraint Delay (Seconds)	End of Restraint Extension (Seconds)	Occupant Forward Motion Relative to Vehicle (Inches)	Chest Severity Index ($G^{2.5}dt$)
A	54[a]	1500	.010	.047	4.18	307
B	60	1500	.020	.062	8.63	423
C	60	1500	.030	.078	14.35	543
D	60	1500	.040	.090	21.20	608
E	50	1500	.010	.047	4.18	305
F	40	1500	.010	.050	4.31	285
G	30	1500	.010	.069	5.44	255
H	50	1500	.020	.064	8.70	399
I	40	1500	.020	.072	9.27	356
J	30	1500	.020	.091	12.12	278

(a) Occupant attains instantaneous vehicle velocity before drawing out sufficient restraint force limiter to achieve force limiter maximum design load.

onset and plateau g levels, albeit commencing at different times, the amount of work done by the restraints on the occupants had to be identical and in the same relative time phasing from the start of restraint, since both occupants contained the same initial kinetic energy and were brought to rest. Notice that they were brought to rest ten milliseconds apart, the same incremental difference existing in restraint delay.

Cases A and E through G reflect incremental reductions in the plateau g level for the occupant. The results here are more pronounced than at 40 mph but were similar. Notice, though, that the 30 g case resulted in the occupant no longer being brought to rest with the vehicle and, therefore, causing disproportionate increase in occupant travel within the compartment. Cases F, H, I and J investigate influence of delay in restraint action when plateau occupant restraint function provides 40 g. It is interesting to note that only in Case F, where restraint delay is limited to ten milliseconds, is there appreciable time during which the lower vehicle decelerations act on the occupant. This results from the occupant being decelerated to instantaneous vehicle velocity well prior to vehicle stopping. The severity index for Case F is, therefore, lower than the other 40 g plateau cases.

Table 9 presents the parametric restaint study conducted on the 50 mph barrier crash designed vehicle which decelerates with a maximum effective level of 40 g.

Cases A through D study the influence of onset rate increases with the other parameters of restraint function held constant. The greater the onset rate is, the less time the occupant restraint initially functions below the plateau level. It is noted that both chest severity index and occupant forward travel within the compartment is lessened by higher onset rate. Cases E through H vary onset at the reduced plateau level of 50 g. As in the 60 g series the influence on passenger travel in the compartment is similar, that is, reduced occupant travel relative to the compartment occurs as increases in onset occupant g rate are made. Interesting phenomena are revealed in the severity index transition Case E (1500 g/second) resulted in a lower severity index than the 2,000, 2,500 and 3,000 g/second cases. This apparently is the result of an increased over the ground stopping distance for the occupant in Case E where the occupant was not at rest when the vehicle stopped. The durations of occupant exposure to the peak 50 g plateau were 28, 33, 32 and 30 milliseconds for Cases E through H respectively, which explains the inversion of S.I. trend.

Cases I through L vary onset while holding the plateau at 40 g. In all of these cases the vehicle stopping time of 67 milliseconds is exceeded by the occupant. As onset rate was increased, occupant

TABLE 8
PARAMETRIC RESTRAINT STUDIES IN BARRIER CRASH

Vehicle Performance Characteristics: 30g maximum in 45 MPH fixed rigid barrier crash with maximum crush of 31.83 inches occurring at 0.080 seconds.

Case Identification	Maximum Occupant Deceleration (G's)	Rate of Occupant Onset Loading (G/Sec.)	Restraint Delay (Seconds)	End of Restraint Extension (Seconds)	Occupant Forward Motion Relative to Vehicle (Inches)	Chest Severity Index ($G^{2.5}dt$)
A	60	1500	.010	.053	6.17	474
B	60	1500	.020	.070	12.16	623
C	60	1500	.030	.084	19.75	714
D	60	1500	.040	.094	27.67	714
E	50	1500	.010	.055	6.26	452
F	40	1500	.010	.065	6.94	412
G	30	1500	.010	.088	10.87	316
H	40	1500	.015	.079	10.45	461
I	40	1500	.020	.085	14.40	461
J	40	1500	.025	.090	18.36	461

forward motion in the compartment decreased resulting in reduced over the ground motions and increases in chest severity index. Occupant exposure times to plateau level decelerations (40 g) were 43, 47, 49 and 51 millisceonds for Cases I through L, respectively.

Cases M and N, when compared to Case A, show the influence of increasing restraint delay. Occupant motions within the compartment were seen to significantly increase due to the increased initial free flight time of the occupant, with no influence on severity index since all occupants stopped after the vehicle.

Cases O and P, when compared to Case C, also show the same influence for delayed onset to exist at the 2,500 g/sec. onset rate as existed at the 1,500 g/sec. onsets with one exception. The severity index substantially increased when delay was increased from ten to twenty milliseconds. The reason is clearly that the occupant with ten millisecond restraint delay decelerated to vehicle instantaneous velocity before the vehicle stopped and was, therefore, subjected to the reduced level of the vehicle deceleration for the remainder of his stopping time.

The parametric study of restraint function for each of the three barrier crash speeds and associated vehicle design functions has yielded the following general results:

1. Shortest delay between start of crash and start of restraint function is desirable.
2. Higher onset rates of occupant loading produce lower requirements for occupant forward travel in his compartment.
3. Higher onset rates of occupant loading can cause either increases or decreases in thorax severity index depending on selected values of other parameters.
4. Higher plateau levels of occupant restraint decelerations produce higher values of thorax severity index.
5. Higher plateau levels of occupant restraint decelerations produce small reductions of the requirement for occupant forward travel relative to his compartment if restraint delay time is low and his vehicle is undergoing relatively low deceleration.

Another generalization is of importance because of its fundamental nature and overall significance in

TABLE 9
PARAMETRIC RESTRAINT STUDIES IN BARRIER CRASH

Vehicle Performance Characteristics: 40g maximum in 50 MPH fixed rigid barrier crash with maximum crush of 29.47 inches occurring at 0.067 seconds.

Case Identi-fication	Maximum Occupant Decel-eration (G's)	Rate of Occupant Onset Loading (G/Sec.)	Restraint Delay (Seconds)	End of Restraint Extension (Seconds)	Occupant Forward Motion Relative to Vehicle (Inches)	Chest Severity Index ($G^{2.5}dt$)
A	60	1500	.010	.068	12.08	820
B	60	2000	.010	.058	8.61	795
C	60	2500	.010	.051	6.78	768
D	60	3000	.010	.047	5.68	750
E	50	1500	.010	.072	13.14	679
F	50	2000	.010	.068	9.87	711
G	50	2500	.010	.063	7.89	706
H	50	3000	.010	.058	6.68	691
I	40	1500	.010	.080	15.65	519
J	40	2000	.010	.077	12.92	533
K	40	2500	.010	.075	11.25	541
L	40	3000	.010	.974	10.13	547
M	60	1500	.020	.078	20.88	820
N	60	1500	.030	.088	29.68	820
O	60	2500	.030	.080	23.63	915
P	60	2500	.040	.090	32.43	915

restraint evaluations. The time integral of the product of occupant deceleration and occupant velocity must balance the initial kinetic energy of a one slug mass traveling at initial occupant velocity. It is not unreasonable to assume that from a given initial occupant velocity the least threatening deceleration history would be that offering least time exposure at a given maximum deceleration for a plateau type of restraint performance. Employing this criteria, it is seen that in cases where the occupant is stopped in his relative motion to the vehicle before the vehicle itself is stopped relative to the ground, high onset rates of occupant loading may be beneficial since the occupant deceleration is reduced to vehicle levels more quickly (see Figure 7 for such a case). If onset rate is determined to be an occupant damaging criteria, such increase of onset rates must be limited to safe levels. The conclusion that higher occupant onset ramps may be beneficial under the stated conditions, creates an anomaly relative to conventional logic. Higher ramps with fixed plateaus result in lower occupant travel within the compartment. The total occupant travel over the ground is the sum of his travel in the compartment, and the compartment travel over the ground. The compartment travel over the ground is not significantly affected by the occupant's onset rate of loading. Therefore, higher occupant onset rates of loading reduce occupant over the ground travel and yet indicate a reduced severity of occupant loading. This conclusion results from the unique performance characteristics of the proposed restraint design which permit the occupant to benefit from his vehicle's lower deceleration after the occupant is decelerated to vehicle velocity.

In the cases where, due to low chosen plateau g levels, the occupant's velocity is not reduced to that of his vehicle before the vehicle stops (see Figure 9) it is found that low occupant onset rates of loading are the more beneficial. This results from the fact that the occupant's velocity will be greater at any given time during his deceleration for the low onset ramp case than for a case employing a steeper onset ramp to the same plateau level. This higher occupant velocity provides a more rapid accrual of the time integral of the deceleration time produce, thereby, foreshortening the time requirement at the high g level of the plateau.

Before proceeding to evaluation of head-on collisions between vehicles, it was deemed most practical to select only one set of restraint function parameters for representation in the vehicles. Since the proposed concept offers a unique proximity and small volume for the inflatable restraint a low delay time was assumed. Prior programs have amply demonstrated the feasibility of three to five millisecond crash detection times. Rapid inflation of low volume bags has also been shown feasible. The occupant onset loading commences before full bag volume inflation is achieved. For these reasons, the time between car contact with barrier and start of occupant onset loading was assumed to be ten milliseconds. The occupant onset rate of loading in barrier crash was established at 1,500 g/second. The plateau level of restraint function was selected at 40 g. These restraint parameter selections correspond to Case F of Table 7 for the 40 mph fixed rigid barrier vehicle design; case F of Table 8 for the 45 mph barrier design vehicle; and Case I of Table 9 for the 50 mph barrier crash design vehicle. The parameter selections are shown graphically in Figure 5. These same restraint parameters were assumed in the computer modeling which produced occupant and vehicle time histories for the 45 and 50 mph barrier crashes depicted in Figures 6 through 9.

Remembering that restraint function timing was initially set to satisfy fixed rigid barrier crash conditions, it is apparent that the time versus g levels attained under different vehicle decelerations histories cannot be assumed to be the same as in the design barrier impact speed case. The novel proposed restraint system provides occupant loading as a function of the occupant's position relative to the compartment, in so far as this position relates to the amount of load limiter ribbon extended in his restraint system. Figure 10 shows the relationships of the restraint system force limiter extentions to the occupant deceleration for the 45 and 50 mph barrier impact designs. Table 10 tabulates these relationships. It is therefore possible, by utilizing the rigid fixed barrier computer runs for each different barrier impact speed vehicle design to relate restraint function to the deceleration history of the vehicles. The ten millisecond restraint delay and the 1,500 g/second onset ramp were established based on the time versus occupant loading in the design barrier case. By the double integral of vehicle deceleration during design barrier crash, we obtain a quantity representing the distance between a totally unrestrained mass originally traveling with the vehicle and the actual position of the vehicle undergoing barrier crash decelerations. This double integral of vehicle deceleration, when cross-plotted with the actual occupant deceleration existing during the design barrier crash, provides the basis for establishing occupant decelerations during any other vehicle deceleration history. This is accomplished by a double integration of the vehicle deceleration during a non-barrier crash condition and direct correlation of the values to occupant deceleration levels which

existed at the corresponding values of this double integral during design barrier crash.

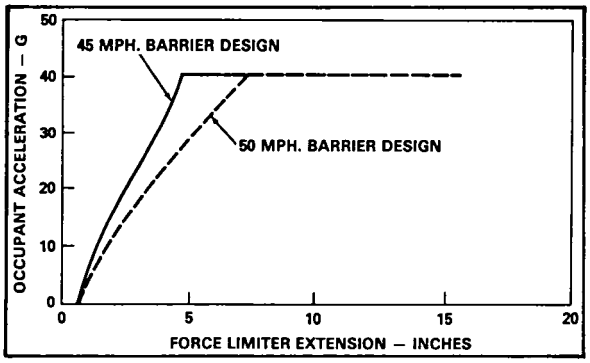


Figure 10. Occupant Restraint Displacement Functions

TABLE 10
OCCUPANT RESTRAINT DISPLACEMENT FUNCTION

50 MPH Design		45 MPH Design	
Force Limiter Ribbon Extension (Inches)	Occupant G	Force Limiter Ribbon Extension (Inches)	Occupant G
0	0.0	0	0.0
0.657	0.0	0.493	0.0
0.945	3.0	0.709	3.0
1.281	6.0	0.959	6.0
1.661	9.0	1.240	9.0
2.079	12.0	1.547	12.0
2.531	15.0	1.874	15.0
3.012	18.0	2.218	18.0
3.519	21.0	2.573	21.0
4.045	24.0	2.935	24.0
4.587	27.0	3.299	27.0
5.139	30.0	3.661	30.0
5.698	33.0	4.016	33.0
6.258	36.0	4.360	36.0
6.815	39.0	4.687	39.0
7.091	40.0	4.843	40.0
18.000	40.0	18.000	40.0

CAR-TO-CAR HEAD-ON ALIGNED CRASHES

As earlier stated, it appears that in order to protect the occupants of small vehicles, a crash performance requirement must be imposed on not only the small car but also those vehicles with which the small vehicle may collide. The highest energy absorption demands are imposed in aligned head-on crashes.

The assignment of design values of tolerable closing speeds must be based on real world crash statistics relating losses imposed on society as a function of closing speeds and car weights. These relationships must then be weighed against the cost to society of alleviating these losses for crashes occurring below prescribed design safe speeds for a range of vehicle weights. Research aimed at arriving at vehicle design performance conditions offering best investment return to American society, as it will exist in the mid-1980s, is currently underway. It is being accomplished as part of Phase I efforts of the Research Safety Vehicle Program (RSV) in which five competitive contracts are well underway and scheduled for completion by mid-April 1975.

The computer modeled design closing speed case studies and the range of vehicle weights employed in this paper are not intended to indicate a pre-conceived notion of the results of the complex work now underway for the RSV. The selections for these parameters were made only for purposes of illustration.

The range of vehicle weights for which compatible safe closing speeds are assigned, go from a low weight of 1,500 pounds to a maximum weight of 5,000 pounds. Within this range of vehicle weights, it will be necessary that aligned head-on crashes be tolerable to occupants at speeds equal to and less than the minimum design closing speeds assumed for these studies. That is to say, the occupants of both vehicles involved must suffer no more than minimum injuries regardless of the weight differences of the two involved vehicles. It is fundamental that the most threatened vehicle occupants are generally those in the lighter of the cars.

The cases simulated by computer modeling involve the vehicles and occupants of both vehicles in two car head-on collisions occurring at closing speeds of 70, 65 and 60 mph. Although the technical problem is not affected by the selected speed of each vehicle, but rather the resulting closing speed between them, for uniformity and dramatization of dynamic responses, the vehicles were assumed to be initially moving in opposite directions, each moving at half the desired closing speed.

For each of the three closing speeds, the cases studied always established the lighter vehicle to be the 1,500 pound car. The heavier vehicle weight was changed from case to case to provide data on car weights from 2,000 to 5,000 pounds in 500 pound increments. The heavier vehicle was assumed to contain the required hydraulic design accommodations developed earlier in the paper

(shown in Tables 2, 3 and 4), which, of course, varied according to the heavier vehicle weight and the design barrier impact speed vehicle class being used. The restraint characteristics employed in each of the two opposing vehicles were the same, being a function of the barrier impact speed used for those vehicles designs. The restraint functions are shown in Figure 10 and Table 10 for the 45 and 50 mph vehicle barrier impact speed designs employed.

Figure 11 graphically displays some of the dynamic effects encountered when the two colliding vehicles are only 500 pounds different in weight. It represents the case of a 70 mph aligned head-on crash between a 1,500 pound and 2,000 pound car. Both vehicles and their occupant restraints were designed to provide occupant safety in fixed rigid barrier 50 mph crash. The 2,000 pound car incorporated a 7.2 inch stroke hydraulic cylinder in tandem with a 22.3 inch constant force crush structure. Both vehicles were designed to produce a 40 g maximum effective vehicle deceleration (34 g nominal) and a 29.5 inch crush in their design barrier crash at 50 mph.

Study of Figure 11 and Table 11 reveals that the two cars initially going in opposite directions, each at 51.3 feet per second reach a common velocity of 7.3 feet per second, in the direction of initial motion of the larger car, sixty milliseconds after start of crash, and maintain that speed thereafter. The occupants of the 2,000 pound car attained final vehicle velocity three milliseconds later than the cars. The 1,500 pound car occupants attain final vehicle velocity eleven milliseconds after the vehicles. Note that the 40 g plateau occupant deceleration is of shorter duration for the occupants of the 2,000 pound vehicle and that the occupant deceleration drop to zero when vehicle final speed is matched by the occupants.

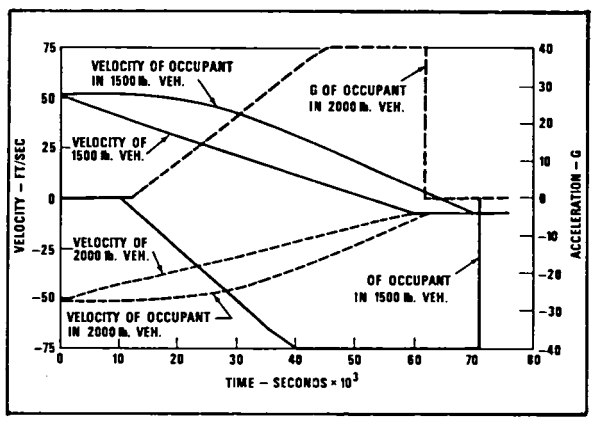


Figure 11. 2000 Pound Versus 1500 Pound Vehicle in 70 mph Head-On

TABLE 11
2000 POUND VERSUS 1500 POUND VEHICLE IN 70 MPH HEAD-ON

Time	1500 #VEHICLE			2000 #VEHICLE		
	Vehicle Velocity (Ft/Sec)	Occupant Velocity (Ft/Sec)	Occupant Acceleration (G)	Vehicle Velocity (Ft/Sec)	Occupant Velocity (Ft/Sec)	Occupant Acceleration (G)
0	51.33	51.33	0.00	-53.33	-51.33	0.00
5	46.41	51.33	0.00	-47.64	-51.33	0.00
10	41.48	51.33	0.00	-43.94	-51.33	0.00
12	39.51	51.29	-2.08	-42.47	-51.33	0.00
15	36.55	50.88	-6.35	-40.25	-51.18	+3.49
20	31.63	49.29	-13.46	-36.55	-50.12	+9.65
25	26.70	46.55	-20.58	-32.86	-48.07	+45.81
30	21.77	42.66	-27.70	-29.16	-45.03	+21.97
35	16.85	37.63	-34.80	-25.47	-40.99	+28.14
40	11.92	31.50	-40.00	-21.77	-35.97	+34.29
45	7.00	25.06	-40.00	-18.08	-29.96	+39.97
46	6.01	23.77	-40.00	-17.34	-28.67	+40.00
50	2.07	18.62	-40.00	-14.38	-23.52	+40.00
55	-2.86	12.18	-40.00	-10.69	-17.08	+40.00
60	-7.33	5.74	-40.00	-7.34	-10.64	+40.00
62	-7.34	3.16	-40.00	-7.34	-8.06	+40.00
63	-7.34	1.87	-40.00	-7.34	-7.34	0.00
70	-7.34	-7.14	-40.00	-7.34	-7.34	0.00
71	-7.34	-7.34	0.0	-7.34	-7.34	0.00

Figure 12 represents the same closing speed (70 mph) and same vehicle fixed rigid barrier crash design conditions (50 mph, 40 g maximum vehicle deceleration, 29.5 inch crush) as those of Figure 11. However, the heavier vehicle is now 5,000 pounds, containing a 19.9 inch hydraulic stroke front end accommodation. The weight ratio between the two vehicles is, therefore, 3.3.

As seen in Figure 12 and Table 12, the 1,500 pound vehicle again reverses direction during the crash, but now goes through a velocity change of almost 54 miles per hour ending up at a common velocity with the large car of 27.7 feet per second after 81 milliseconds. The 5,000 pound car undergoes slightly more than a 16 mph change in speed during the same 81 milliseconds. It is interesting to note that the large car occupant only reached a maximum deceleration of approximately 28 g, showing that he had not even required extension of his restraint force limiter up to the plateau region of 40 g. The large car occupant matched the instantaneous velocity of his vehicle at 56 milliseconds and for the next 25 milliseconds he experienced only that lower level of deceleration under which his car was approaching its final velocity.

Comparing the times at which occupant onset loading commenced in the two cars, it is seen that restraint for the large car occupant was delayed 19 milliseconds while the small car occupant onset started at 10 milliseconds. This difference in delay is related directly to the restraint action being a function of occupant position during his forward

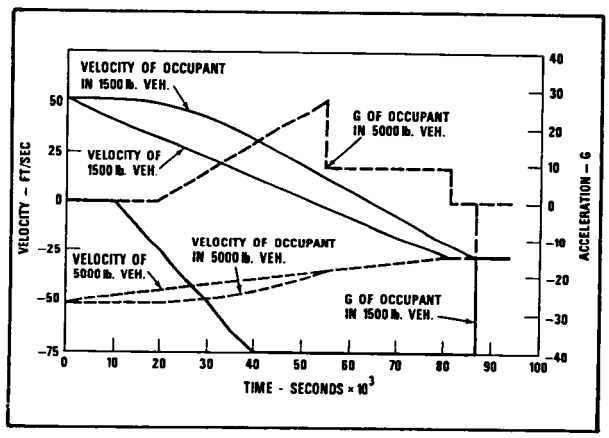


Figure 12. 5000 Pound Versus 1500 Pound Vehicle in 70 mph Head-On

TABLE 12
5000 POUND VERSUS 1500 POUND VEHICLE IN 70 MPH HEAD-ON

Time	1500 = VEHICLE			5000 = VEHICLE		
	Vehicle Velocity (Ft/Sec)	Occupant Velocity (Ft/Sec)	Occupant Acceleration (G)	Vehicle Velocity (Ft/Sec)	Occupant Velocity (Ft/Sec)	Occupant Acceleration (G)
0	51.33	51.33	0.00	-51.33	-51.33	0.00
5	46.41	51.33	0.00	-49.86	-51.33	0.00
10	41.48	51.33	0.00	-48.38	-51.33	0.00
15	36.55	50.88	-6.35	-46.90	-51.33	0.00
19	32.61	49.70	-12.04	045.72	-51.33	0.00
20	31.63	49.29	-13.46	-45.42	-51.33	0.51
25	26.70	46.55	-20.58	-43.94	-50.92	4.49
30	21.77	42.66	-27.20	-42.47	-49.89	8.38
35	16.85	37.63	-34.80	-40.99	-48.22	12.28
40	11.92	31.50	-40.00	-39.51	-45.93	16.18
45	6.99	25.06	-40.00	-38.03	-43.01	20.08
50	2.07	18.62	-40.00	-36.55	-39.47	23.98
55	-2.86	12.18	-40.00	-35.08	-35.29	27.88
56	-3.84	10.89	-40.00	-34.78	-34.78	9.18
60	-7.78	5.74	-40.00	-33.60	-33.60	9.18
65	-12.71	-9.70	-40.00	-32.12	-32.12	9.18
70	-17.64	-7.14	-40.00	-30.64	-30.64	9.18
75	-22.56	-13.58	-40.00	-29.16	-29.16	9.18
80	-27.49	-20.02	-40.00	-27.69	-27.69	9.18
81	-27.64	-21.31	-40.00	-27.64	-27.64	0.00
85	-27.64	-26.46	-40.00	-27.64	-27.64	0.00
86	-27.64	-27.64	0.00	-27.64	-27.64	0.00

travel relative to the vehicle. Since the smaller vehicle was subjected to 30.6 g deceleration and the larger vehicle 9.2 g, the occupant of the small vehicle reaches the displacement position at which restaint commences sooner than does the large car occupant. It is also significant that occupants of both vehicles were subjected to onset rates of restraint loading which were below the design value occurring in barrier crash. This is not surprising in the case of the big vehicle in that it underwent only a 16.15 mph velocity change in this collision while its restraint system was designed for adequate function in a 50 mph barrier crash. The 1,500 pound car underwent a 53.85 mph velocity change in this collision with the 5,000 pound vehicle and yet the occupant onset rate of loading did not exceed its 50 mph barrier design value. The reason for this is found in the comparison of vehicle deceleration levels in barrier and car-to-car crashes. Because of the structural differences in interface loading conditions imposed on a vehicle by another vehicle as compared to the fixed rigid plane of a barrier, the maximum load attained in car-to-car collision was taken as 90 percent of the level attained in barrier impact. These lower vehicle decelerations result in slower occupant displacements relative to his vehicle and, therefore, lower onset rates of loading for this unique restraint system.

Tables 13a, b, and c summarize results of computer simulations of 70 mph closing speed head-on collisions between a 1,500 pound and each of seven heavier vehicles. The heavier vehicle weights were 2,000, 2,500, 3,000, 3,500, 4,000, 4,500 and 5,000 pounds, Each vehicle and its occupant restraints were those designed for occupant safety in 50 mph fixed rigid barrier crash. The crush length of each vehicle in the 50 mph barrier crash was 29.5 inches occurring with a maximum vehicle deceleration of 40 g. The occupant restraint system in each vehicle was the same as previously described with the 40 g plateau level of maximum occupant deceleration. Accommodation hydraulic stroke lengths as previously derived were placed in each of the vehicles as determined from its weight.

As seen in the tabular results shown in Table 13a, the deceleration imposed on the 1,500 pound vehicle was the same regardless of the weight of the heavier vehicle. This results from the fact that in all cases, the small vehicle's barrier design crush is matched and not exceeded because of the ability of the hydraulic portion of the heavier car hybrid design to stroke fully even at loads below its maximum design level. It is also seen that the larger vehicle deceleration level decreases inversely with its weight. This results from the fact that interface loads between the two vehicles always match the crush load of the 1,500 pound vehicle and this crush load is independent of the heavier vehicle weight.

The velocity change experienced by each vehicle is seen to become more and more unequal as the heavier vehicle weight is increased. Heavier vehicle crush is seen to match the accommodation hydraulic stroke designed into each of the heavier vehicles and these accommodation strokes increase with increasing vehicle weight. It is also evident that the crash energy

TABLE 13a
ALIGNED HEAD-ON COLLISIONS BETWEEN 1500 POUND AND HEAVIER VEHICLES

Closing Speeds = 70 MPH, All Vehicles and Occupant Restraints Designed to Satisfy 50 MPH Fixed Rigid Barrier Requirements

Heavier Vehicle		Vehicle Deceleration	Vehicle Velocity Change	Vehicle External Crush	Absorbed Energy
Effective Weight (Pounds)	Design Hydraulic Accomodation Stroke (Inches)	Heavier/1500# (G's)	Heavier/1500# (MPH)	Heavier/1500# (Inches)	Heavier/1500# (K.FT.-LB.)
2000	7.2	23.0/30.6	30.00/40.00	7.20/29.47	27.56/112.71
2500	10.7	18.4/30.6	26.25/45.75	10.65/29.47	40.71/112.72
3000	13.3	15.3/30.6	23.33/46.66	13.32/29.47	50.94/112.71
3500	15.5	13.1/30.6	21.00/49.00	15.46/29.47	59.12/112.69
4000	17.2	11.5/30.6	19.09/50.91	17.21/29.47	65.82/112.71
4500	18.7	10.2/30.6	17.50/52.50	18.67/29.47	71.40/112.72
5000	19.9	9.2/30.6	16.15/53.85	19.90/29.47	76.12/112.73

absorbed by the heavier vehicle increases as the weight of heavier vehicle increases while the absorbed energy in the 1,500 pound car remains constant at 90 percent of its design barrier impact speed requirement.

From Table 13b, it is clear that none of the occupants was exposed to hazardous decelerations in this mixture of vehicle weights impacting with closing speeds of 70 mph, all vehicles and restraints having been designed to satisfy occupant survival in 50 mph barrier crash. It is interesting to note that impacts with 4,000 through 5,000 pound vehicles produced velocity changes of the small car which were in excess of 50 mph, yet occupant response was within allowable limits.

TABLE 13b
ALIGNED HEAD-ON COLLISIONS BETWEEN 1500 POUND AND HEAVIER VEHICLES

Closing Speeds = 70 MPH, All Vehicles and Occupant Restraints Designed to Satisfy 50 MPH Fixed Rigid Barrier Requirements

Heavier Vehicle	Maximum Occupant Deceleration	Duration of 40G Occupant Deceleration	Occupant Chest Severity Index
Effective Weight (Pounds)	Heavier/1500# (G's)	Heavier/1500# (Seconds)	Heavier/1500= (Inches)
2000	40.0/40.0	.016/.030	275/400
2500	40.0/40.0	.007/.034	203/443
3000	40.0/40.0	.001/.037	144/477
3500	(a) 36.2/40.0	(a)/.040	104/504
4000	(a) 33.0/40.0	(a)/.043	79/526
4500	(a) 30.2/40.0	(a)/.045	62/544
5000	(a) 27.9/40.0	(a)/.046	50/560

(a) Occupant attained vehicle velocity before extending restraint force limiter ribbon beyond onset ramp.

TABLE 13c
ALIGNED HEAD-ON COLLISIONS BETWEEN 1500 POUND AND HEAVIER VEHICLES

Closing Speeds = 70 MPH, All Vehicles and Occupant Restraints Designed to Satisfy 50 MPH Fixed Rigid Barrier Requirements

Heavier Vehicle	Vehicle Collision Duration	Occupant Loading Duration	Occupant Forward Motion Relative to Vehicle
Effective Weight (Pounds)	(Seconds)	Heavier/1500# (Seconds)	Heavier/1500# (Inches)
2000	.060	.051/.061	7.6/11.9
2500	.066	.053/.065	5.8/12.6
3000	.070	.056/.068	4.8/13.0
3500	.073	.057/.070	4.2/13.3
4000	.076	.059/.073	3.7/13.6
4500	.079	.061/.074	3.4/13.8
5000	.081	.062/.076	3.1/13.9

Table 13c demonstrates that occupant forward motions relative to his compartment do not reflect a problem for passenger compartment interior designers. Note that for the cases where the small car underwent velocity changes in excess of 50 mph, the occupant forward motions relative to his seat were less than the 15.65 inches forward motion experienced in the design case for the restraints and vehicles, the design case being a 50 mph velocity change as produced by fixed rigid barrier impact. Shorter occupant restraint extensions reflect less work required of the restraint subsystem during occupant deceleration. Similarly, by reference to Table 9 (the vehicle and restraint barrier collision design case) it is noted that duration of vehicle collision for the car-ro-car collision producing the 53.85 mph velocity change is increased by 21 percent over the barrier collision duration. The extended durations of collision are also observed in all collisions which produced over a 46 mph velocity change in the 1,500 pound car. This time increase results from the reduced interface loads in car-to-car as opposed to barrier collisions and the need to satisfy the momentum laws.

In light of the longer vehicle deceleration times and reduced energy absorption required of the restraints for the same velocity change in car-to-car events as compared to barrier impacts, it is clear that benefits are appreciated by the occupants of the small car in car-to-car collision that are not present in barrier collision. These benefits result from the reduction in car crushing loads and, therefore, vehicle decelerations, incurred in car-to-car collisions as compared to barrier impacts producing the same velocity change. Therefore, as in those cases studied, if the larger vehicle is designed to provide the required supplement to the energy absorption of the small car so as to relieve requirements for additional crush of the small car in a car-to-car collision, then the small car occupant survival problem in these car-to-car collision cases is actually less severe than in a barrier impact of the small car producing the same small car velocity change.

Addressing thorax severity index (S.I.) as shown in Table 13b and vehicle velocity change in Table 13a, it is observed that S.I. increases with increasing velocity change. In graphing this relationship, it is found that S.I. is a near linear function of vehicle velocity change. By interpolation, it is also found that the S.I. value (519) obtained in barrier crash at 50 mph (the vehicle design condition) occurs at a velocity change of 50.5 mph in the big car-to-small car collision cases addressed.

Tables 14a, b and c address the case of a reduction in desired tolerable closing speeds, 65 mph. All vehicles addressed in these tables were designed to provide survival in 50 mph fixed rigid barrier impact. These vehicles, therefore, are the same vehicles as were employed for the choice of 70 mph closure compatibility except that the length of hydraulics is substantially reduced. As in the 70 mph closure designs (Table Set 13), no problems are observed in the 65 mph closure designs (Table Set 14).

TABLE 14a
ALIGNED HEAD-ON COLLISIONS BETWEEN 1500 POUND AND HEAVIER VEHICLES

Closing Speeds = 65 MPH, All Vehicles and Occupant Restraints Designed to Satisfy, 50 MPH Fixed Rigid Barrier Requirements

Heavier Vehicle		Vehicle Deceleration	Vehicle Velocity Change	Vehicle External Crush	Absorbed Energy
Effective Weight (Pounds)	Design Hydraulic Accomodation Stroke (Inches)	Heavier/1500# (G's)	Heavier/1500# (MPH)	Heavier/1500# (Inches)	Heavier/1500# (K.FT.-LB.)
2000	2.2	23.0/30.6	27.86/37.14	2.15/29.47	8.23/112.73
2500	5.2	18.4/30.6	24.37/40.63	5.12/29.47	19.57/112.72
3000	7.5	15.3/30.6	21.67/43.33	7.42/29.47	28.39/112.72
3500	9.3	13.1/30.6	19.50/45.50	9.27/29.47	35.45/112.70
4000	10.8	11.5/30.6	17.78/47.27	10.78/29.47	41.22/112.72
4500	12.1	10.2/30.6	16.25/48.75	12.04/29.47	46.03/112.71
5000	13.1	9.2/30.6	15.00/50.00	13.10/29.47	50.10/112.71

TABLE 14b
ALIGNED HEAD-ON COLLISIONS BETWEEN 1500 POUND AND HEAVIER VEHICLES

Closing Speeds = 65 MPH, All Vehicles and Occupant Restraints Designed to Satisfy 50 MPH Fixed Rigid Barrier Requirements

Heavier Vehicle Effective Weight (Pounds)	Maximum Occupant Deceleration Heavier/1500# (G's)	Duration of 40G Occupant Deceleration Heavier/1500# (Seconds)	Occupant Chest Severity Index Heavier/1500# (Inches)
2000	40.0/40.0	.014/.026	251/367
2500	40.0/40.0	.007/.030	197/407
3000	40.0/40.0	.001/.033	139/438
3500	(a) 36.3/40.0	(a) /.036	101/464
4000	(a) 33.0/40.0	(a) /.038	77/484
4500	(a) 30.2/40.0	(a) /.040	60/501
5000	(a) 27.9/40.0	(a) /.042	49/516

(a) Occupant attained vehicle instantaneous velocity before extending restraint force limiter ribbon beyond onset ramp.

TABLE 14c
ALIGNED HEAD-ON COLLISIONS BETWEEN 1500 POUND AND HEAVIER VEHICLES

Closing Speeds = 65 MPH, All Vehicles and Occupant Restraints Designed to Satisfy 50 MPH Fixed Rigid Barrier Requirements

Heavier Vehicle Effective Weight (Pounds)	Vehicle Collision Duration (Seconds)	Occupant Loading Duration Heavier/1500= (Seconds)	Occupant Forward Motion Relative to Vehicle Heavier/1500= (Inches)
2000	.056	(b) (b) .049/.057	7.5/11.4
2500	.061	(a) (b) .048/.061	5.8/12.0
3000	.065	(a) (b) .051/.064	4.8/12.5
3500	.068	(a) (b) .052/.067	4.2/12.9
4000	.071	(a) (b) .054/.069	3.7/13.1
4500	.073	(a) (b) .055/.070	3.4/13.3
5000	(c) .075	(a) (b) .056/.072	3.1/13.5

(a) Occupant reaches vehicle instantaneous velocity before vehicle reaches its final velocity.
(b) Occupant reaches final velocity after the vehicle.
(c) Duration of 50 MPH barrier crash is only 0.067 seconds.

Comparisons between Table Sets 13 and 14 reveal deceleration levels in each car are not altered by the reduced closure design speed but velocity changes of the 1,500 pound car are reduced. The crushes of the 1,500 pound car are the same for both closing speeds, but the crush and absorbed energy of the bigger cars are reduced in the reduced closing speed case.

Maximum occupant decelerations are not influenced by the 5 mph reduction in design closing speed but severity index is reduced for the occupants of both the large and small vehicle. The duration of the collision event is reduced by the reduced closing speed and small decreases in occupant forward motion relative to his seat are observed in the 65 mph design closing speed cases.

Tables 15a, b and c again address a 65 mph allowable closure speed goal but this time all the vehicles are considered to have been designed for 45 mph fixed rigid barrier crash. Comparisons between Table Set 14 and 15 show the influence of employing a lower barrier design impact speed when given a specific desired tolerable car-to-car closing speed.

The small reduction in vehicle design barrier speed results in significant increases in crush requirements on the larger cars but produces impressive vehicle deceleration reductions. Peak vehicle velocity changes are not influenced by changes in the car structure but only by the closing speed and car weights. Substantial increases in absorbed energy is seen for the larger cars with corresponding reductions of crash energy absorptions in the 1,500 pound vehicle when lower car fixed barrier design speed is used. Maximum occupant deceleration is not reduced in the 1,500 pound car but some reductions are appreciated by the larger car occupants. Occupants of both vehicles achieve significant reductions of thorax severity index with this reduced basic barrier design velocity for the cars which must still provide tolerable conditions in 65 mph closure collisions. Dramatic increases of the time of vehicle engagement are seen to occur. Also, the occupant loading durations are seen to increase, but occupant forward motions in his compartment are substantially reduced. The reduced work done in the occupant restraints subsystem is the result of occupants in all cases having been decelerated to their vehicle velocities before their vehicles had been brought to their final velocities. This phenomena was illustrated in Figure 12.

The results of reducing the desired safe closing speed between cars in aligned head-on collisions to 60 mph are shown in Tables 16a, b and c. All vehicles in this group were designed for safe barrier crash at 45 mph.

The results followed a similar pattern to those obtained from lowering desired safe closing speeds from 70 to 65 mph without changing barrier design crash speed. Large vehicle accommodation stroke requirements were reduced, vehicle g levels remained the same, vehicle velocity changes were reduced, maximum occupant accelerations remained unchanged, but S.I. was slightly reduced. Vehicle collision duration was diminished but occupant forward motion within his compartment remained the same. In lowering desired safe closing speeds from 70 to 65 mph, there were small reductions of small car occupant motion relative to his vehicle. No such occupant motion reductions occurred when closing speed was reduced from 65 to 60 mph.

TABLE 15a
ALIGNED HEAD-ON COLLISIONS BETWEEN 1500 POUND AND HEAVIER VEHICLES

Closing Speeds = 65 MPH, All Vehicles and Occupant Restraints Designed to Satisfy, 45 MPH Fixed Rigid Barrier Requirements

Heavier Vehicle		Vehicle Deceleration	Vehicle Velocity Change	Vehicle External Crush	Absorbed Energy
Effective Weight (Pounds)	Design Hydraulic Accomodation Stroke (Inches)	Heavier/1500= (G's)	Heavier/1500= (MPH)	Heavier/1500= (Inches)	Heavier/1500= (K.FT.-LB.)
2000	10.4	17.2/23.0	27.86/37.14	10.34/31.83	29.65/91.30
2500	14.3	13.8/23.0	24.37/40.63	14.29/31.83	40.99/91.30
3000	17.4	11.5/23.0	21.67/43.33	17.36/31.83	49.81/91.30
3500	19.9	9.8/23.0	19.50/31.82	19.82/31.82	56.87/91.29
4000	21.9	8.6/23.0	17.73/47.27	21.84/31.83	67.45/91.30
4500	23.6	7.7/23.0	16.25/48.75	23.51/31.83	67.45/91.30
5000	25.0	6.9/23.0	15.00/50.00	24.93/31.83	71.52/91.30

TABLE 15b
ALIGNED HEAD-ON COLLISIONS BETWEEN 1500 POUND AND HEAVIER VEHICLES

Closing Speeds = 65 MPH, All Vehicles and Occupant Restraints Designed to Satisfy 45 MPH Fixed Rigid Barrier Requirements

Heavier Vehicle	Maximum Occupant Deceleration	Duration of 40G Occupant Deceleration	Occupant Chest Severity Index
Effective Weight (Pounds)	Heavier/1500# (G's)	Heavier/1500# (Seconds)	Heavier/1500# (Inches)
2000	40.0/40.0	.003/.018	176/315
2500	(a) 38.0/40.0	(a) /.018	113/332
3000	(a) 32.3/40.0	(a) /.017	78/346
3500	(a) 28.8/40.0	(a) /.017	57/357
4000	(a) 26.9/40.0	(a) /.017	43/366
4500	(a) 24.5/40.0	(a) /.017	34/373
5000	(a) 22.5/40.0	(a) /.017	28/379

(a) Occupant attained vehicle instantaneous velocity before extending restraint force limiter ribbon beyond onset ramp.

TABLE 15c
ALIGNED HEAD-ON COLLISIONS BETWEEN 1500 POUND AND HEAVIER VEHICLES

Closing Speeds = 65 MPH, All Vehicles and Occupant Restraints Designed to Satisfy 45 MPH Fixed Rigid Barrier Requirements

Heavier Vehicle	Vehicle Collision Duration	Occupant (a) Loading Duration	Occupant Forward Motion Relative to Vehicle
Effective Weight (Pounds)	(Seconds)	Heavier/1500# (Seconds)	Heavier/1500# (Inches)
2000	.074	.062/.064	4.1/5.8
2500	.081	.068/.071	3.3/5.8
3000	.086	.072/.076	2.8/5.8
3500	.090	.074/.080	2.5/5.8
4000	.094	.077/.084	2.2/5.8
4500	.097	.071/.087	2.1/5.8
5000	.099	.080/.089	1.9/5.8

(a) All occupants reach vehicle instantaneous velocity before vehicle reaches its final velocity.

TABLE 16a
ALIGNED HEAD-ON COLLISIONS BETWEEN 1500 POUND AND HEAVIER VEHICLES

Closing Speeds = 60 MPH, All Vehicles and Occupant Restraints Designed to Satisfy, 45 MPH Fixed Rigid Barrier Requirements

Heavier Vehicle		Vehicle Deceleration	Vehicle Velocity Change	Vehicle External Crush	Absorbed Energy
Effective Weight (Pounds)	Design Hydraulic Accomoda-tion Stroke (Inches)	Heavier/1500# (G's)	Heavier/1500# (MPH)	Heavier/1500# (Inches)	Heavier/1500# (K.FT.-LB.)
2000	4.1	17.2/23.0	25.71/34.29	4.10/31.83	11.76/19.31
2500	7.5	13.8/23.0	22.50/37.50	7.47/31.83	21.42/91.30
3000	10.1	11.5/23.0	20.00/40.00	10.09/31.83	28.94/91.30
3500	12.2	9.8/23.0	18.00/42.00	12.18/31.83	34.95/91.30
4000	13.9	8.6/23.0	16.36/43.64	13.90/31.83	39.87/91.30
4500	15.4	7.7/23.0	15.00/45.00	15.33/31.83	43.97/91.31
5000	16.6	6.9/23.0	13.85/46.15	16.54/31.83	47.44/91.30

TABLE 16b
ALIGNED HEAD-ON COLLISIONS BETWEEN 1500 POUND AND HEAVIER VEHICLES

Closing Speeds = 60 MPH, All Vehicles and Occupant Restraints Designed to Satisfy 45 MPH Fixed Rigid Barrier Requirements

Heavier Vehicle	Maximum Occupant Deceleration	Duration of 40G Occupant Deceleration	Occupant Chest Severity Index
Effective Weight (Pounds)	Heavier/1500# (G's)	Heavier/1500# (Seconds)	Heavier/1500# (Inches)
2000	40.0/40.0	.003/.017	170/301
2500	(c) 37.9/40.0	(c) /.017	109/317
3000	(c) 32.3/40.0	(c) /.017	75/329
3500	(c) 28.8/40.0	(c) /.017	55/339
4000	(c) 26.9/40.0	(c) /.018	42/347
4500	(c) 24.5/40.0	(c) /.017	33/354
5000	(c) 22.4/40.0	(c) /.017	27/360

(c) Occupant attained vehicle instantaneous velocity before extending restraint force limiter ribbon beyond onset ramp.

TABLE 16c ALIGNED HEAD-ON COLLISION BETWEEN 1500 POUND AND HEAVIER VEHICLES			
Closing Speeds = 60 MPH, All Vehicles and Occupant Restraints Designed to Satisfy 45 MPH Fixed Rigid Barrier Requirements			
Heavier Vehicle	Vehicle Collision Duration	Occupant Loading Duration	Occupant Forward Motion Relative to Vehicle
Effective Weight (Pounds)	(Seconds)	Heavier/1500# (Seconds)	Heavier/1500# (Inches)
2000	.068	(a) (a) .056/.058	4.1/5.8
2500	.074	(a) (a) .061/.064	3.3/5.8
3000	.080	(a) (a) .066/.070	2.8/5.8
3500	.084	(a) (a) .068/.074	2.5/5.8
4000	.087	(a) (a) .070/.077	2.2/5.8
4500	(b) .089	(a) (a) .071/.079	2.0/5.8
5000	.092	(a) (a) .073/.082	1.9/5.8

(a) Occupant reaches vehicle instantaneous velocity before vehicle reaches its final velocity.
(b) Duration of 45 MPH barrier crash is only 0.081.

The failure to achieve reduced occupant motion with this reduction of design safe closing speed from 65 to 60 mph, is related to the fact that the occupant reached vehicle velocity before vehicle velocity had been reduced to final value in the cases of both design safe closing speeds. Under these conditions, the benefits of the initial 10 milliseconds of vehicle crush are denied to the occupant as "ride down benefit" because of restraint delay in both the 65 and 60 mph closures. Since the vehicle deceleration level is the same in both 65 and 60 mph closures employing the same vehicle barrier design impact velocity, the occupant restraints must supply the same supplement to the absorption of occupant kinetic energy as long as the occupants and vehicles reach their final velocities concurrently.

The four families of car-to-car crash compatibility reported above, have probed at the design influence that desired safe closing speed imposes on vehicles designed to meet specific rigid fixed barrier impact speed goals. In all cases, the same conservative occupant restraint pulse shape was employed, a 1,500 g/second rise to a plateau of only 40 g. Even with these conservative assumptions, it was found that in no case was the occupant anticipated to collide with dashboard structure, and in most cases, there was large clearance. This indicates that appreciable deviation from the target occupant deceleration history could be tolerated during restraint development.

Many researchers believe that higher onset rates and g levels are tolerable in the case of a well-distributed load on the occupant. The restraint system proposed should provide desired load distribution and, therefore, less conservatism in occupant restraint pulse may be justified.

Should one consider raising the restraint pulse, it is obvious that higher closing speeds could be well-tolerated in aligned head-on crashes than has yet been presented. Still another option remains which also has significant potential. By raising vehicle design g level and benefiting from the improved features of the proposed restraint higher closing speeds could readily be attained. This would be accomplished without necessarily increasing threat to side struck or rear struck vehicles. The abilityof the hydraulic portion of the front-end system of a hybrid to stroke and absorb appreciable energies under impact with both high and

low load resistant structures provides the desired accommodation. The fact that the hydraulic stroke must be larger and larger with increasing vehicle weight in order to accommodate head-on collisions is consistent with the goal of the compatibility in other crash orientations. The heavier the vehicle, the more compatibility accommodation stroke we desire in the striking vehicle regardless of the orientation of the struck vehicle.

Up till now, this paper has addressed hybrid vehicle designs for 1,500 through 5,000 pound vehicles which provide occupant safety at various velocities of impact. They have used specified vehicle decelerations in their fixed rigid barrier crashes. Using the resulting crush dimensions, the paper has shown how to designate hydraulic accommodation stroke lengths in the larger vehicles to provide safety to the 1,500 pound car occupants in impacts with these larger vehicles. Several families of hybrid vehicles were derived. Each family has its own unique rigid barrier design speed, rigid barrier design vehicle deceleration level, and tolerable maximum closing speed limit for aligned head-on collisions with the lightest family member, the 1,500 pound car.

I would now like to analytically demonstrate that the safe closing speed allowable between the 5,000 pound car and other vehicles in the same design family actually increases as a function of struck vehicle weight above the minimum value of the 1,500 pound vehicle.

Let us investigate the vehicle family designed to accommodate the 70 mph closure against the 1,500 pound member. All of these vehicles were designed for fixed barrier impact speeds of 50 mph producing maximum vehicle decelerations of 40 g. Since nominal deceleration has been shown to be about 85 percent of maximum for hydraulic front ends, the equivalent rectangular pulse level is 34 g with a crush length of 29.5 inches.

Table 17 shows some calculated values leading to the determination of tolerable closing speeds with which the 5,000 pound vehicle can impact other vehicles in its design family. The column showing force levels in car-to-car collisions is 90 percent of the force levels achieved in the car's design rigid barrier crash condition. The 90 percent factor approximates the effects of geometrical differences between barrier and car-to-car impacts. Maximum energy absorption in the 5,000 pound car is the product of force level capability of the lighter car and the accommodation hydraulic stroke length in the 5,000 pound striking vehicle. This accommodation stroke in the 5,000 pound vehicle (1.658 feet) had been determined from considerations of the safety of 1,500 pound vehicle occupants in 70 mph closures as previously shown.

The total energy absorption available in the 5,000 pound vehicle impact against lighter vehicles is the sum of 90 percent of the design barrier crash energy absorption in the lighter car (to account for geometric effects in car-to-car crashes as compared to barrier crashes) and the associated energy absorption in the 5,000 pound car.

TABLE 17
CALCULATIONS OF TOLERABLE CLOSING SPEEDS INVOLVING 5000 POUND CAR AND ANOTHER VEHICLE IN ITS DESIGN FAMILY(1)

Struck Vehicle Weight (Pounds)	Energy Absorption in 50 mph Barrier Crash (Foot Pounds)	Force Level in Barrier Crash (Pounds)	Force Level Capability in Car-to-Car Crash (Pounds)	Maximum Energy Absorption of 5000 LB. Car in Head-on Car-to-Car = 1.658 X Force(2) (Foot Pounds)	Total Energy Absorption Available (Foot Pounds)
2000	167011.7	68000	61200	101490.0	251800.5
3000	250517.5	102000	91800	152235.1	377700.9
4000	334023.3	136000	122400	202980.1	503601.1
5000	417529.1	170000	153000	253725.1	629501.3

(1) All vehicles in this family accommodate the 1500 pound vehicle in aligned head-ons at 70 mph.
(2) Hydraulic stroke of 5000 pound car to accommodate 1500 pound car in 70 mph closures is 19.9 inches or 1.658 feet.

Table 18 presents the effective mass of the lighter cars in impact with the 5,000 pound car. Using the values of total energy absorption and effective mass in equation 2 ($E_{at} = \frac{1}{2}(M_1M_2)V_c^2/(M_1 + M_2)$), the tolerable closing speed (V_c) is found.

Note that each vehicle in this family of designs can impact the 1,500 pound vehicle at 70 mph without endangering the small car occupants as shown in Tables 13a, b and c. Table 13a shows that the 1,500 pound vehicle had undergone a 53.85 mph velocity change in its 70 mph closure speed impact with the 5,000 pound vehicle. Tables 13b and c demonstrate that the occupants of the 1,500 pound vehicle were not endangered even though the velocity change suffered had been greater that at its design barrier impact speed of 50 mph. It is, therefore, probable that a computer simulation of the 2,000 pound vehicle impacting the 5,000 pound vehicle with a closing speed of 72.64 mph would predict survival of all occupants despite the fact that the 2,000 pound vehicle undergoes a velocity change 1.88 mph greater than its design barrier impact speed. Such computer simulation has not actually been done, but it appears probable that occupants of the smaller vehicles would not be endangered due to restraint function at the closing speeds shown in Table 18.

It has then been demonstrated that allowable closing speeds increase as the weight ratio between the 5,000 pound and lighter vehicles reduces for a given family of hybrid vehicle designs.

Only one more shallow investigation will be presented. To this point, maximum frontal crush had been set to not exceed 32 inches, regardless of barrier design impact speed or the desired safe head-on closing speed. Although it is felt that this crush magnitude is attainable, it is clear that designers of lighter cars would have more difficulty in providing this amount of crush than would producers of heavier cars. An attempt was made to probe the effect of reducing required crush as an inverse function of car weight. For this evaluation, the 1,500 pound car and its restraint was designed to provide occupant safety in a 40 g maximum, 40 mph fixed rigid barrier impact. Restraint function was again limited to the 1,500 g/second onset to a 40 g plateau. The total crush of the 1,500 pound car in its 40 mph fixed barrier design case was 23.9 inches, rather than the 30.2 inch crush which had been previously postulated for a maximum 25 g deceleration.

The 5,000 pound vehicle was designed to accommodate the lighter vehicle in a 60 mph aligned head-on collision. In addition, it was designed to provide occupant protection in barrier impact at 45 mph with a maximum vehicle deceleration of 30 g. In order to accommodate the 1,500 pound vehicle, a hydraulic buffer stroke capability of 12.4 inches was provided within the 5,000 pound vehicle total safe crush dimension of 31.9 inches.

Computer simulation of this 60 mph closure speed showed the 1,500 pound car to have crushed up to its

TABLE 18
TOLERABLE CLOSING SPEEDS BETWEEN THE 5000 POUND CAR AND OTHER CARS IN ITS DESIGN FAMILY(1)

Struck Vehicle Weight (Pounds)	Total Energy Absorption Available (Foot Pounds)	Effective Mass = M_1M_2/M_1+M_2 (Slugs)	Maximum Safe Closing Speed (MPH)	Velocity Change of Lighter Vehicle (MPH)
2000	251800.5	44.366	72.64	51.88
3000	377700.9	58.230	77.65	48.53
4000	503601.1	69.013	82.36	45.75
5000	629501.3	77.640	86.82(2)	43.41

(1) All cars in the design family can accommodate the 1500 pound car in aligned head-ons with 70 mph closures.

(2) This speed would not crush both 5000 pound cars to the full barrier crush level but theoretically can be thought of as crushing one fully and the other only to the length of its accommodation stroke. If both vehicles are allowed to crush to their design barrier values (29.5 inches) a speed of 94.86 mph could be tolerated.

safe designed crush distance of 23.9 inches and the 5,000 pound car to have crushed only up to its hydraulic accommodation stroke distance of 12.4 inches. The acceleration level of the small car was 30.6 g and the big car was decelerated at 9.2 g.

The occupants of the small car underwent a deceleration plateau of 40 g for 37 milliseconds and did not reach vehicle final velocity until 8 milliseconds after the vehicle. The occupants had moved forward in their car 12.92 inches.

The occupants of the 5,000 pound vehicle reached a maximum deceleration of 27.3 g and had moved forward in their compartment only 2.3 inches.

It is significant to note that the previous hydraulic stroke accommodation requirement for 60 mph closure (Table 2) employing the same barrier design requirement that the 5,000 pound vehicle employs in this case is 16.6 inches, 4.2 inches more than is required for this case. Therefore, it would appear that deviations toward stiffer, lower barrier speed designs for smaller cars may not jeopardize their compatibility in car-to-car collisions. It is probable that similar results may be obtained without decreasing the barrier speed design of the smaller cars; simply raise the small car deceleration design level. The range of those variations that would not cause jeopardy in head-on collisions cannot be conjectured without additional analytics. However, design frontal crush relief for small cars appears to be a potential using a hybrid vehicle design approach.

OCCUPANT RESTRAINT LOAD LIMITER

One of the important components of the proposed occupant restraint subsystem is the load limiter. Its function is to offer a programmed resistance to occupant forward motion as long as the occupant's speed is greater than that of the vehicle during a crash. If the occupant forward speed is reduced to that of his crushing vehicle before the completion of the vehicle collision event, the load limiter is to cease extending. From that point in the crash, and until crash completion, the load limiter is to impose decelerations on the occupant at a rate which is the same as the deceleration of the vehicle. In this manner, the same final velocity would be achieved by occupant and car at the same time. Such a case is seen in Figure 7. On the other hand, if the vehicle reaches its final speed condition before the occupant, then the force limiter is to continue extending, imposing the programmed deceleration loads on the occupant until he attains the final speed of the vehicle. Figure 9 depicts this latter alternative function.

The force limiter is envisioned to employ an energy absorbing mechanism working with a principle similar to that employed in a special application on the General Motors Corporation ESV. The GM device was used to absorb energy during the stroking of the rear passenger's knee reaction panels in a frontal crash. More recently, Minicars, Inc. has devised a system of absorbing occupant kinetic energy using a more sophisticated application of the same principle.

Essentially, the principle involves bending of a metallic strap (ribbon) during its extension over a shaping mandril, the work being accomplished in the bending of the ribbon is the source of energy absorption for such a mechanism.

Figure 13 is a schematic of the force limiter component envisioned for use in the occupant restraint subsystem proposed. Three rollers replace a single mandril in this proposed application. The flat metallic load limiter ribbon is drawn through the three roller assembly from loads applied by the occupant on his airbag during his deceleration in a vehicle crash. The loads applied to the airbag by the occupant are reacted by load limiter ribbons. Resistance to ribbon withdrawal through the roller assembly is established by the following controllable parameters of the load limiter design.

- The metal selected for the ribbon.
- The thickness of the ribbon.
- The width of the ribbon.
- The diameters of the rollers.
- The longitudinal spacing of the rollers along the ribbon.
- The vertical spacing of the rollers relative to each other.

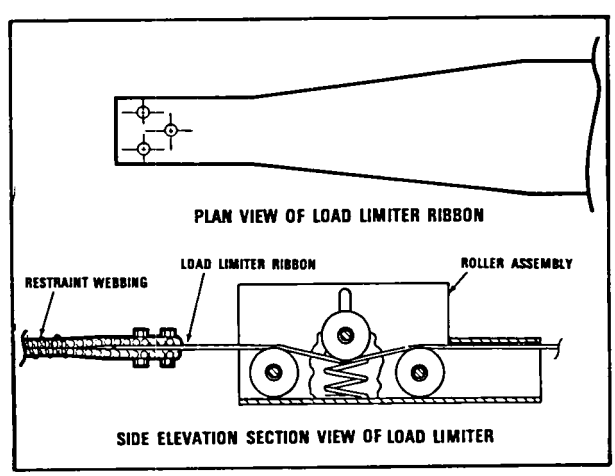


Figure 13. Restraint Load Limiter

To achieve a desired onset rate of loading in the critical design case of fixed rigid barrier impact at a specified speed, one calculates ribbon withdrawal positions during the barrier crash. These ribbon withdrawal positions are related to the occupant forward motion in the vehicle by the geometry of

force limiter locations relative to the occupant position. The information on occupant position relative to the seatback during the crash event is determined by direct analysis using the vehicle barrier crash velocity, the vehicle deceleration characteristics, delay in application of occupant restraint, desired occupant onset rate of loading (g/second) and the desired plateau of maximum occupant deceleration. This analysis, therefore, uses the desired time related restraint force functions under the design barrier impact speed case for the vehicle. The results yield a direct relationship between load limiter withdrawal position and those loads which the load limiter must supply at each position. Figure 10 typifies such a solution for two different barrier impact speeds of vehicles having different vehicle crash deceleration characteristics. Both the 50 mph and 45 mph restraint withdrawal functions shown will provide the same time related restraint delay, onset loading rate, and plateau shown in Figure 5 during fixed rigid barrier impacts at their respective design speeds. These restraint load versus position relationships were employed in the head-on collisions studied.

To provide the desired change in load resistance as the load limiter ribbon is withdrawn through the roller assembly required only that the load limiter be appropriately tapered in its width, becoming wider as a function of its length. To maintain the desired plateau level of force, the taper is discontinued and the ribbon width is constant for the remainder of its length.

In order to achieve best efficiency in occupant restraint relative to the amount of internal stroking distance consumed by the occupant, one more very important factor must be addressed. It is desired that an occupant restraint system adequately function over a wide range of occupant weights. We would prefer not to compromise the efficiency of restraint for any group of occupants in order to best accommodate the median weight group, or stated more firmly, do not jeopardize any group in favor of another group.

The load limiter system can provide good accommodation for all occupant weights by appropriate vertical adjustment of the center roller shown in Figure 13. This adjustment position determines the actual force levels at which the force limiter ribbon extend.

There are undoubtedly many ways that a designer may approach the problem of a passive technique to accomplish the roller adjusted position according to the weight of the unique occupant of the seat. I will briefly discuss only two potential approaches which have occurred to me.

Each restraint seat position could contain a closed fluid filled bladder under the seat fabric. The weight of the occupant would then be reflected by a unique pressure in the bladder fluid. This bladder fluid pressure can be readily amplified in lines outside the bladder by use of a small pressurized external fluid ballast bottle appropriately valved to external roller activating tubes. The pressure within these activating tubes must be appropriately amplified and controlled by the seat bladder pressure. The hydraulic pressure within the roller activating tubes can then be employed to move the center roller axis against the resistance of a spring designed to produce the ideal static position of the roller according to the occupant weight. Dashpot damping could readily be provided to prevent roller relocation during short intervals of apparent occupant weight fluctuations such as can be anticipated when the vehicle runs over bumps, the occupant shifts his seating posture, or during actual crash.

A second approach is more mechanical than the first described. It would involve hinging of the forward seat supports and would be more suitable for bucket seats than bench types, although bench types could also be accommodated by providing appropriate seat spring hinging. The hinging axes would be transverse to the vehicle longitudinal axis. The rear of the seat support would then require leverage links to the center roller of Figure 13. Dashpot damping of roller vertical motion would again be advisable.

Specific schemes for implementing both of the described systems for roller relocation are obviously envisioned by the author. Their presentation would actually be a disservice and might actually stifle the ingenuity of more competent designers than myself.

CONCLUSIONS

Hybrid vehicle designs in conjunction with occupant restraints are capable of providing protection to their own occupants as well as the occupants of other vehicles with which severe collisions may occur. The degree of protection offered associated with the least perturbation to existing vehicle dimensions and weights make hybrid vehicles and improved restraints extremely attractive for hardware development efforts.

It was analytically shown that the length of hydraulic accommodation stroke required was as low as two inches in 3,000 pound cars to provide safety to the occupants of 1,500 pound vehicles in head-on 60 mph closure speed collisions. For 65 mph closure speeds, the hydraulic accommodation stroke of the 3,000 pound car was shown to be as low as seven and

one-half inches to assure the safety of the occupants of the 1,500 pound vehicle. At 70 mph closure speed, the accommodation stroke was found to be 13.3 inches. It is obvious that long stroking hydraulic systems are not essential in hybrid vehicle designs.

It has been shown that the major determining factors in hybrid vehicle design are:

- Desired maximum survivable fixed rigid barrier impact speed.
- Weight and structuring of the lightest vehicle which must be accommodated by other vehicles in car-to-car collisions.
- Desired maximum deceleration level which is designated for the vehicles.
- Selected value of the maximum safe closing speed in collisions between vehicles.
- Absolute values of vehicle crushes selected as being consistent with providing internal survival space for occupant deceleration.

It has also been shown the factors most influencing occupant restraint design are:

- The time interval between initiation of vehicle crash and initiation of occupant restraint.
- Restrictions on the rate at which the occupant loads can safely be increased during the initial stage of his restraint function.
- The maximum deceleration level that the occupant is provided by the restraint subsystem.
- The available unobstructed distance within the vehicle through which the occupant may be decelerated by his restraint.
- The crash deceleration characteristics of the vehicle in which the restraint subsystem must function.
- Maximum barrier impact speed at which occupant is protected.

There are then at least eleven variables for car and restraint design to achieve crash survival, the majority of which are not independent of each other. This paper has demonstrated the influences of each and in the process demonstrated the interdependence of all for crashworthy non-aggressive vehicle and restraint design.

Some generalities affecting the hybrid car design are:

- Higher design fixed rigid barrier speeds lead to less demand for energy absorption in the larger vehicle in an aligned head-on collision at any given speed.
- Higher design allowable vehicle g in the smaller car reduces accommodation stroke dimension requirement in the larger car for any particular closing speed.
- Designing all vehicles, regardless of weight, to the same fixed rigid barrier design speed and the same value of nominal deceleration level imposes the same frontal crush dimension requirement on all of the vehicles. One by-product of this approach is to allow the same occupant restraint design function to work equally well in all cars.
- Within constraints, by establishing higher allowable deceleration levels for small cars than for large, it is possible to shorten their required frontal crush and still assure occupant safety within the small car in collisions with larger vehicles.
- Within constraints, by establishing lower design fixed rigid barrier speed requirements on smaller vehicles than on larger vehicles, it is possible to shorten small vehicle required frontal crush and still assure occupant safety within the small car in collisions with larger vehicles.
- As car weight increases, the required accommodation stroke increases for that car to provide a given safe closing speed with the lightest vehicle to be protected.
- Protection of the lightest car by hydraulic accommodation stroke of larger cars leads to increasing values of tolerable closing speeds within that family of cars as the smaller of the two cars involved in a head-on collision approaches the weight of the heaviest car in the protected family.

Some generalities affecting occupant restraint design are:

- Earliest possible commencement of occupant restaints is essential.
- Deceleration onset rate for the occupant should be as great as safely tolerable to the occupant.
- Highest deceleration level consistent with toleration levels reduces occupant stroke requirements within the vehicle.

The unique restraint subsystem proposed shows great advantage in its projected performance in severe collisions, well over the performance potential of most other restraint subsystems under current development.

The hybrid vehicle design approach offers significant improvement in tolerable closing speeds for all orientations of car-to-car crashes.

REFERENCES

1. Klaus Langwieder. Passenger Injuries in Collisions and Their Relation to General Speed Scale, Proceedings of Seventeenth Stapp Car Crash Conference, 1973: 26-33
2. Donald F. Mela. Another Look at Car Size and Safety, NHTSA Technical Note, October 1973: 4-5
3. Robert L. Carter. Passive Protection at 50 Miles Per Hour, DOT HS 810 197, June 1972: 14-21

4. Minicars, Incorporated. Development of Advanced Passive Restraint System for Subcompact Car Driver – Progress Report, Contract DOT-HS-113-3-742, November 1973: 4-5 and; Minicars, Incorporated. Subcompact Crashworthiness Program – Progress Report, Contract DOT-HS-113-3-746, October 1973: Attachment 4

5. Dynamic Science Division, Ultrasystems, Incorporated. Final Report – Vehicle Front End Structure Crash Evaluation Program, Contract DOT-HS-046-2-486, December 1973: v1, 28-29; v2, 203-210

6. Advanced Systems Laboratory, AMF Incorporated. Final Report – Experimental Safety Vehicle Tradeoff and Integration Systems Studies, Contract DOT-HS-257-2-514, September 1973: s3, 7-21

7. Herman Appel. Optimal Deformation Characteristics of Front, Rear and Side Structure of Motor Vehicles in Mixed Traffic, Report on the Second International Technical Conference on Experimental Safety Vehicles, 1971: 3-15

8. Fairchild Industries, The Fairchild ESV Family Sedan – Final Report, DOT HS-800 693, 1972: s5, 15-19

9. AMF Incorporated. Experimental Safety Vehicle – Final Report, DOT HS-800 689, 1972: s4, 4-9

10. Sol Davis. Application of the Shock Response Spectrum to Some Automotive Crashworthiness Problems, SAE 720071, 1972: 3-4

11. Herman Appel and J. Tomas. The Energy Management Structure for the Volkswagen ESV, SAE 730078, 1973: 2

12. Herman Appel and J. Tomas. The Energy Management Structure for the Volkswagen ESV, SAE 730078, 1973: 4-10

FLEXIBLE EXTERIOR AUTOMOTIVE TRIM

JONATHAN T. LANGE
E. I. duPont de Nemours & Co.
Elastomer Chemicals Department

ABSTRACT

Exterior automotive trim, a rapidly expanding new market for painted, flexible materials, is described. End use requirements, new processes, and functional tests are discussed. The properties of thermoset and thermoplastic elastomers now used or being considered for painted sight shields or filler panels in energy absorbing bumper systems, and for painted, full front and rear flexible skin systems, are presented.

INTRODUCTION

When the 1968 Pontiac was released in the fall of 1967, a new era in automotive use of rubber was born. The car featured a non-functional, front bumper piece made of cast microcellular urethane, painted with a flexible paint to match the exterior finish. Customer acceptance of this concept was good, and the technology has now been extended to other painted elastomers. Thus, a multi-million dollar market for rubber has been created in flexible, painted exterior automobile parts which must be both decorative and, to varying degrees, functional.

While the 1968 Pontiac bumper did not dent under minor impact, it was essentially a styling feature only. Therefore, government and insurance company pressure continued for cars that really would be safer and cheaper to repair after minor collision. Finally, in late 1970, legislation was first passed for the now familiar 5-mile-per-hour impact test. This intensified the search for bumper systems with: (1) impact resistance over a specified temperature range (2) styling (3) reproducible mass production capability (4) low cost and (5) long life.

Two years ago this month at the 1972 SAE Automotive Engineering Congress in Detroit, R. E. Knox of Du Pont pointed out that, "The application of an elastomeric facing material to an energy-absorbing system appears to be an economical and effective means of vehicle protection obtained without loss of styling." This turned out to be more prophetic than even he expected. Today, considerable long range development activity is under way on impact-resistant systems using painted, flexible skins on the full front and rear of the car to cover an internal energy manager.

SIGHT SHIELDS OR FILLER PANELS

Initially, a number of short range designs for impact-resistant bumper systems were tried in an attempt to meet the 5 mph legislation. The ones

most universally used at first consisted of protruding chrome bumpers with shock absorbers. These systems are still in use today. They have flexible painted sight shields or filler panels between the bumper and grille to make the whole design better looking. This application produced a total market for painted rubber parts of over 100 million dollars in calendar year 1973. Only the front of 1973 autos needed 5 mph protection; most 1974 models must pass impact requirements for both front and rear.

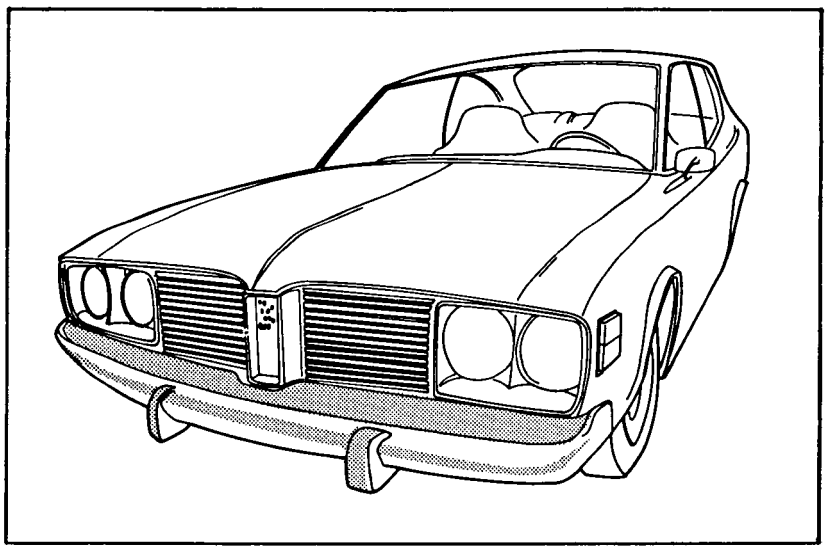


Figure 1. Sight Shield/Filler Panel

As illustrated in Figure 1, the sight shield or filler panel is a relatively small part weighing one-half to five pounds depending on design. It is usually positioned in a cantilevered manner from the car and must, therefore, be rigid enough to support its own weight yet flex upon impact. Most parts are painted to match the car's body color. Compatibility of the paint and substrate is required for good weathering and functionality.

Table I shows the physical properties required for filler panels. Stiffness of the part depends on the width of the gap spanned, type of support used and thickness of the part, but 3000 psi is about minimum as measured by flexural modulus. This test (ASTM-D-790B) is commonly used in the plastics industry. If the material stiffens too much at low temperature, breakage of the grille and die-cast light housings behind it can occur. Since the parts must cantilever yet appear flat, resistance to distortion and sag is important right through the manufacturing process and on into service on the car. A simple droop test on a 1" × 6" sample after one hour at 250° F gives a measure of resistance to sag, particularly for the heat of the paint bake.

Painting requirements are listed in Table II. The painted part must withstand rigorous exposure to the elements which challenge the paint and substrate in combination. Adhesion demands are severe. For example, the thermal shock test requires a sample at 0° F to be blasted with steam to

TABLE I

Physical Property Requirements Sight Shields/Filler Panels	
Flex Modulus: R. T.	3000 psi, min.
-20°F	150,000 psi, max.
160°F	2500 psi, min.
Distortion Resistance:	Out of Mold
	During Painting of Part
	During Vehicle Paint Repair
	In Service

TABLE II

Painting Requirements Sight Shields/Filler Panels		
Adhesion:	Thermal Shock	No Failure
	Water Immersion	No Failure
	Humidity	No Failure
Weatherability:	Florida Exposure	Gloss and Flexi-
	Accelerated Exposure	bility Retention
Cold:	Gravel-O-Meter	No Failure
	Mandrel Bend	No Failure
	Panel Impact	No Failure

simulate car wash situations. Samples must not show blistering during immersion in warm water or in a humid atmosphere for ten days. Weathering exposure data are musts, the critical test being 12 months at 5°F in Florida. Loss of paint gloss and flexibility should be as small as possible. Sight shields should have a minimum one year 20° gloss retention of 20-30%, depending on the color and original gloss. Accelerated weathering devices, such as the Atlas Weather-O-Meter and Fade-O-Meter, the Q-UV and others are sometimes useful for screening new candidates. They must be used very carefully though, and only within a given generic family of substrate or paint.

Survival of painted parts at low temperature without breaking is one of the toughest requirements. Many materials pass this easily when unpainted, but will shatter when tested after painting. Today's flexible paints crack at -20°F when flexed. When they do, the stressed substrate surface beneath is hit with an instantaneous, localized shock load. As shown in Figure 2, notching, tearing or complete breaking of the part can result, depending on the rate of bending, the breaking strength properties of the paint and the properties of the substrate at low temperatures.

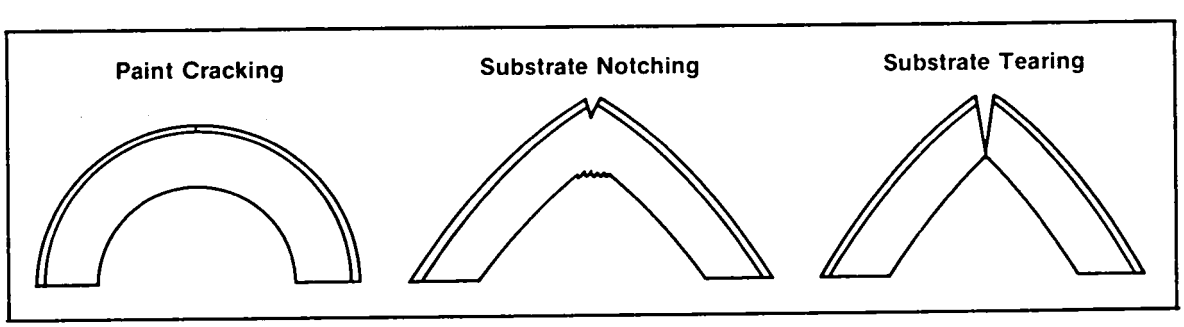


Figure 2. Flexible Painted Parts Cold Bend Failure

Several tests are used to predict this phenomenon. The Gravel-O-Meter blasts panels at 0°F with stones to measure paint chipping. Painted samples must also take a 180° bend at -20°F over a ½" mandrel without splitting the substrate. Most severe is actual impact testing with a ball dropped at 5 mph on a bent panel at -20°F as illustrated in Figure 3. Of course, the final test must be the performance of painted parts actually mounted on cars and impacted at 5 mph.

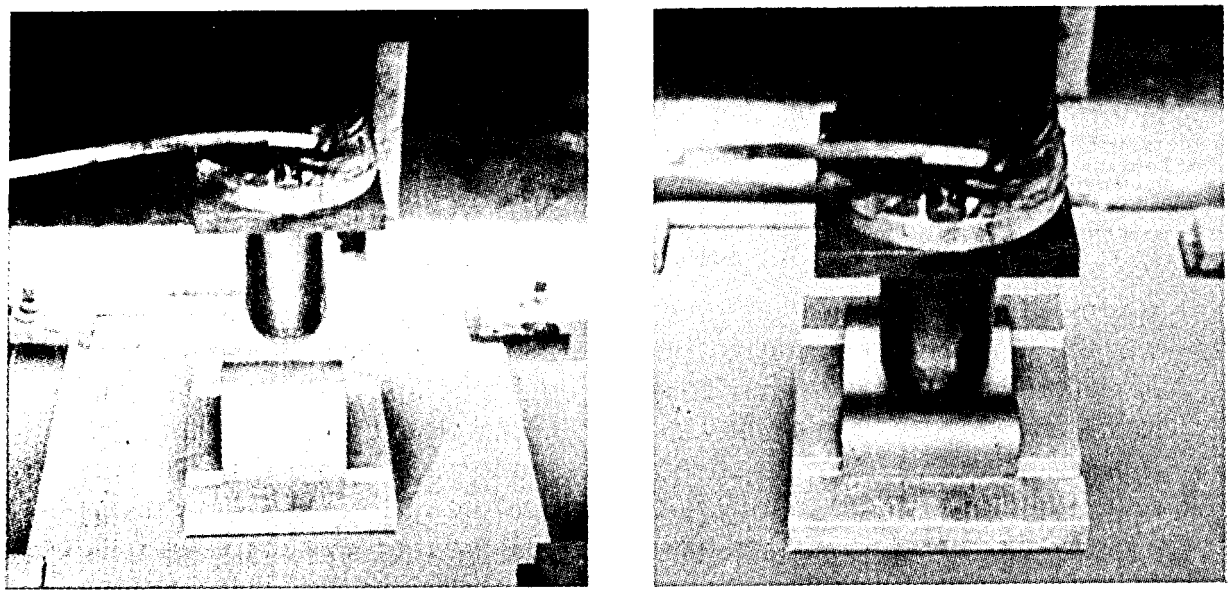


Figure 3. Bent Panel Impact Test

Many materials are being used, but three share the bulk of the sight shield/filler panel market today: thermoplastic urethane, vinyl and NORDEL® 2903, a thermoset EPDM compound developed by DuPont. A comparison of these materials is shown in Table III.

TABLE III

Sight Shield/Filler Panel Substrate Materials

	NORDEL® 2903	Thermoplastic Urethane	PVC
Cost, $/lb.	.25	.90-1.20	.30-.35
Specific Gravity	1.14	1.1-1.25	1.3-1.7
Flex Modulus, 10^3 psi			
R. T.	3.6	13-25	3.0
-20°F	8.2	200-300	175
160°F	3.0	4.8-6.4	1.0
Distortion Resistance	E	G	P
Paintability	G	E	F
Painted Cold Flex	E	P	F

E = Excellent, G = Good, F = Fair, P = Poor

Thermoplastic urethane is considered the standard because of its physical properties and paint history. However, its high cost and the tendency of painted TPU parts to shatter at low temperatures will likely limit its use in the future. Vinyl is potentially much less expensive, of course, but it distorts at high temperatures and must be carefully com-

pounded with special plasticizers to pass painted low temperature impact tests. The weatherability of some painted PVC compositions needs improvement as well. NORDEL® 2903 has a good balance of properties for sight shields or filler panels. Most important is its insensitivity to the effects of high or low temperatures.

FLEXIBLE FRONT AND REAR ENDS

Almost simultaneous with the emergence of protruding metal bumpers was the start of engineering work on making the whole front and rear of the car with flexible skin systems. This concept, sometimes called "facia," is illustrated in Figure 4. It offers exciting styling, weight and cost advantages to the automotive designer and is expected to become more than a two-billion-dollar business in four to five years. A few prototype systems are already on cars today. The front and rear of the 1974 Corvette have painted, flexible thermoplastic urethane skins as does the full front of the Pontiac Grand Am. The Chevrolet Laguna has a microcellular urethane full front end surrounding the headlights and grille.

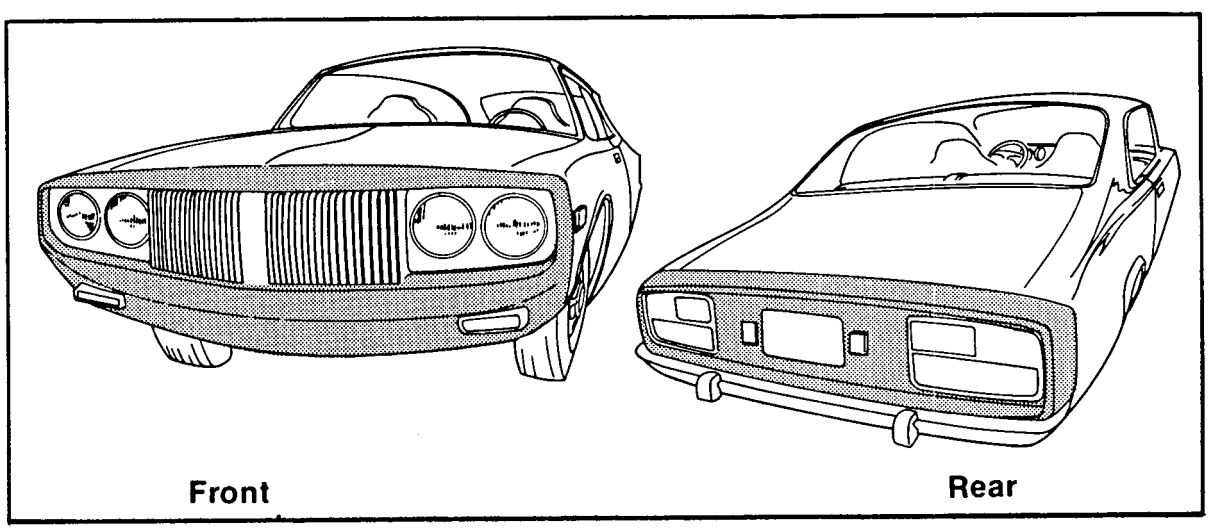


Figure 4. Flexible Skin Front and Rear

Requirements for materials to perform in such an application are very demanding. The flexible skin is painted to match the body color and it must deform or collapse under impact, then recover completely. The energy manager behind it may be mechanical, hydraulic, elastomeric buckling columns or foam blocks. The leading condidates for facia are thermoplastic urethane, liquid reaction urethanes and NORDEL® hydrocarbon rubber.

Most likely, no one material will win all the business but painted NORDEL® offers many advantages when the factors of cost, properties and proven performance are considered. NORDEL® has a wide range of compounding latitude to meet different end-use requirements and new tech-

niques for reinforcing NORDEL® with glass fibers have been developed for flexible skins. This gives the desired stiffness without loss of resistance to distortion at high temperatures or to breakage at low temperatures as shown in Figure 5. The thermoplastic urethanes are expensive and they become so stiff at low temperature that, when painted, they have little resistance to shattering on impact. The liquid urethane systems are potentially attractive but currently, have many unsolved problems.

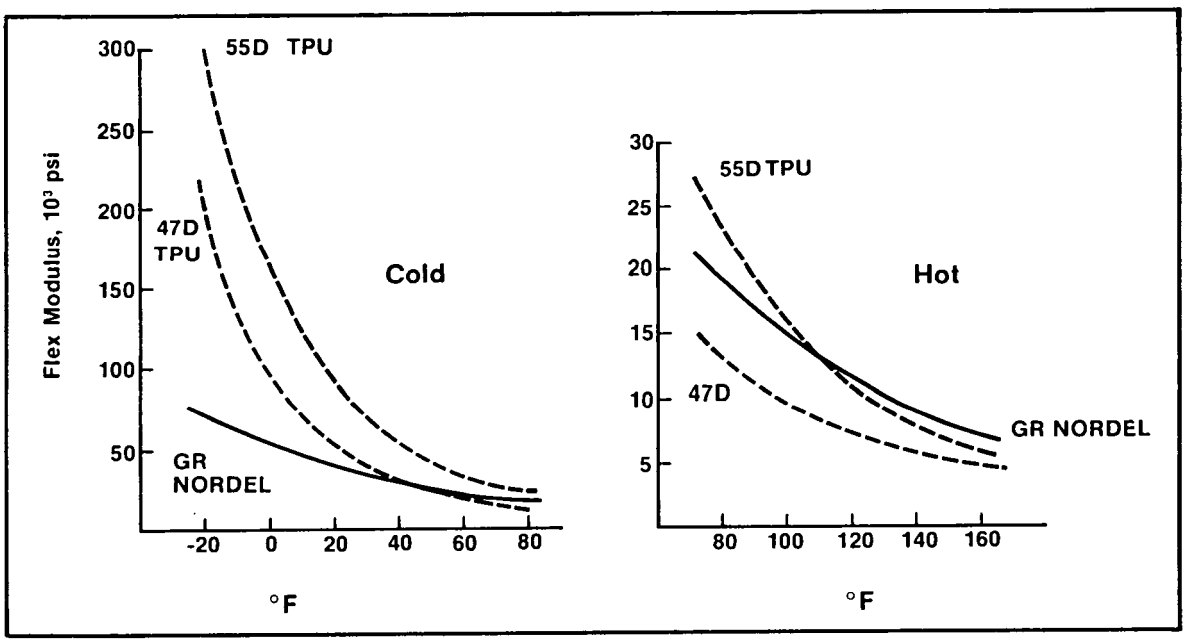


Figure 5. Stiffness vs. Temperature

THE SYSTEMS APPROACH

Most of the Ford and Chrysler automobiles in production today use sight shields made from painted NORDEL® 2903. The story of this development bears telling as an example of the challenge and opportunity this market presents and of the unique approach and commitment required.

The traditional role of a raw materials supplier has been to recommend various compounds as starting points and then to sell only the raw polymer. This didn't work for the development of painted NORDEL® 2903 sight shields. A total systems approach was needed and will undoubtedly be the key to success of future automotive trim applications. As shown in Figure 6, the important factors were: polymer manufacture; selection of compounding ingredients; compound manufacture; processing, which included injection or compression molding; extrusion and painting; functional testing; paint interaction; design of parts. All were critical to acceptance. Unfortunately, there were no short cuts. Each piece of the puzzle had to be optimized yet meshed with other factors to be successful.

The primary goal was to develop specially designed polymers for easy processing, particularly for injection molding. At the same time, these polymers had to be used in compounds that would be easily processed,

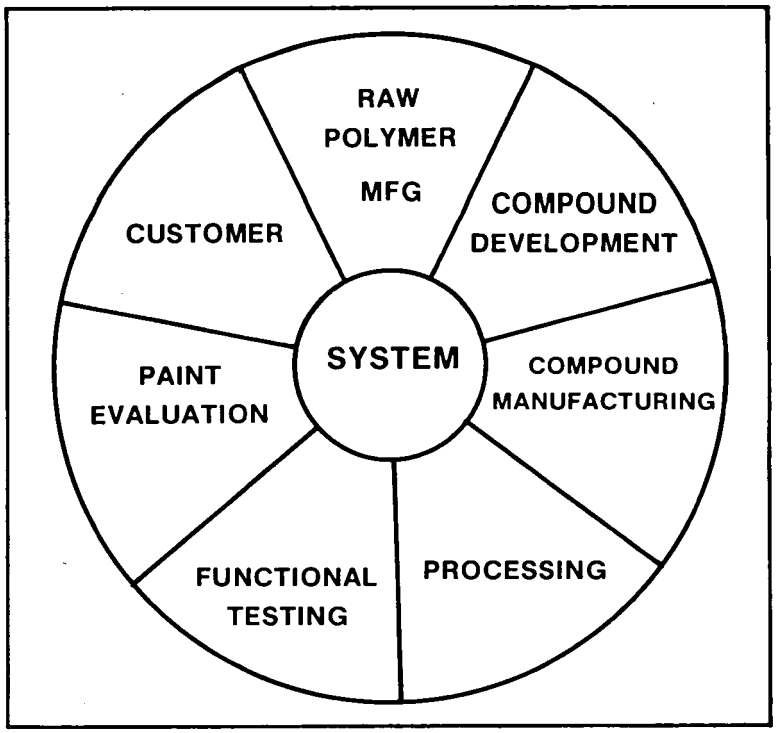


Figure 6. The Systems Approach

paintable, functional and reasonable in cost. Candidate formulations took a long time to develop since weathering in Florida and extensive field trials were necessary for acceptance. In a compound, many of the ingredients (such as fillers, oils and curatives) can affect weathering of the paint very drastically. A simple change in cure system can cause one year 20° gloss retention of a painted NORDEL® compound to drop from 40% to 2% for a given color and type of paint. Fortunately, the NORDEL® polymer itself does not require special additives to protect it from oxygen ozone or ultraviolet light. Specific ingredients also had to be carefully selected from various suppliers to get a reproducible formulation. All new compounds had to be evaluated in scaled-up mixing and injection molding equipment.

Custom mixing became a necessity to assure uniform material quality to Ford, to allow those sight shield suppliers without mixing capability to participate, and to protect hard earned information from general disclosure. An arrangement was made for a new mixing plant in Kingstree, South Carolina, which just opened this past August.

Of course, fast, automated processing is the key to the automotive industry. Fabrication and painting procedures had to be developed and put into production. About five years ago, DuPont started active research which led to the optimization of techniques and equipment designed for injection molding and curing large parts of thermoset NORDEL®. In less than two minutes, five-pound parts could be formed at 350-360° F and removed from the mold. This made processing costs for NORDEL® more competitive with thermoplastics. Compression molding and extrusion techniques also had to be refined and demonstrated.

The finishing operation called for innovation too. NORDEL® is not an easy surface to which to adhere paint. After much work on painting techniques, a DuPont task force found a unique feasible commercial process. The parts needed only to be cleaned, sensitized, irradiated with ultra-violet light and primed before topcoating. Finally, three fabricators for 1973 production and another two for 1974 were helped through plant construction and start-up in seven months time: total floor space — over 750,000 sq. ft. Each plant typically used over ten molding machines from 300-2500 tons in size plus compression molding and extrusion equipment. Each had the capacity to produce thousands of sight shield sets a day in 36 colors.

The end-use performance of these parts was quite unusual and many new functional tests had to be developed to characterize them, which have already been discussed. They led to the realization of the important role that the flexible finish played in part performance. Paint recipes, like rubber compounds, were very different and had to be evaluated over candidate substrates. The use of the wrong primer can cause one year 20° gloss retention of painted NORDEL® 2903 to drop from 40% to 10% for a given color and type of paint. Table IV lists the various flexible paints and their suppliers. A new flexible finish, FAE (flexible acrylic enamel), was developed by DuPont which had improved weatherability and excellent compatibility with NORDEL®.

TABLE IV

Flexible Finishes for Sight Shields/Filler Panels	
• Durethane 100	PPG, Polyester Urethane Enamel
• Autoflex	Midland, Polyester Urethane Enamel
• FAE	DuPont, Flexible Acrylic Enamel
• VC-101/VT-102	Bee Chemical, Urethane/Acrylic Lacquer
• Durethane 200	PPG, Polyester Urethane Enamel

Throughout this development, close contact was maintained with Ford and their fabricators. Awareness of the constant changes in materials, engineering, purchasing and processing activities would have been impossible without it. This, then, was the type of commitment that had to be made to develop a system to sell the flexible exterior part concept.

THERMOPLASTIC EPDM

The sight shield/filler panel market is now in full operation and a variety of new materials are being considered. The shortage of vinyl and high price of TPU is creating interest in many thermoplastic polymers. ABS, HYTREL® polyester elastomer from DuPont and ethylene-ethyl acrylate have seen use for some specific applications. Styrene block copolymers

are also being tried. However, the materials with the most to offer and receiving the most interest are thermoplastic EPDM polymers as characterized in Table V.

TABLE V

Thermoplastic EPDM for Sight Shields/Filler Panels	
Cost, $/lb.	.35-.65
Specific Gravity	.9-1.02
Flex Modulus, 10^3 psi	
R. T.	20-35
-20°F	90-140
160°F	8-12
Distortion Resistance	G
Paintability	E
Painted Cold Flex	E
E = Excellent; G = Good	

Lower cost is the primary driving force for interest in these materials for sight shields or filler panels. Their high stiffness suggests the use of thinner, lighter parts. Injection molding is fast and the scrap is reprocessable. Cycles of 30-45 seconds are possible. Painting requires a surface treatment such as UV or a flash primer, but the enamel baked primer can be eliminated. Florida exposures look excellent. Thermoplastic EPDM polymers lack the physical properties required for flexible skins, but significant use for sight shields or filler panels is expected starting on 1975 model automobiles.

ON THE HORIZON

Since the fall of 1967, flexible exterior automotive trim has come a long way — in design, processing and materials. Largely responsible for this rapid progress has been the experience obtained from the development of sight shields or filler panels. What has now emerged is, in the truest sense, an entirely new industry, born of necessity, but holding promise of creating a vast new market for synthetic rubber. It transfers the high speed production techniques characteristic of the Plastics Industry to the Rubber Industry along with the capability of producing parts of almost unlimited size, shape and color. In the future, products made this way need not be confined to exterior trim or even the automotive industry itself. The advantages of this technology and these materials suggest that they will find a variety of new applications in the appliance, recreational, mass transit and many other industries.

COMPATIBILITY ON THE ROAD

ULRICH SEIFFERT
Research and Development Center
Volkswagenwerke AG

INTRODUCTION

The problem of vehicle aggressivity and compatibility of the traffic participants has been well known for several years (1).* In the beginning, the aggressivity of a vehicle was mainly used as a definition of the problem which existed. In the meantime, we have come to talk much more about the compatibility of the actual participants which make up road traffic. A technical definition for the term compatibility is not yet available. The best definition today could be: That the people who make up the traffic population, be they passengers in cars or pedestrians are in a compatible traffic condition if, when an accident occurs, the protection criteria for the human being are not exceeded. It can be seen from the accident statistics first, which accident configuration is most frequent.

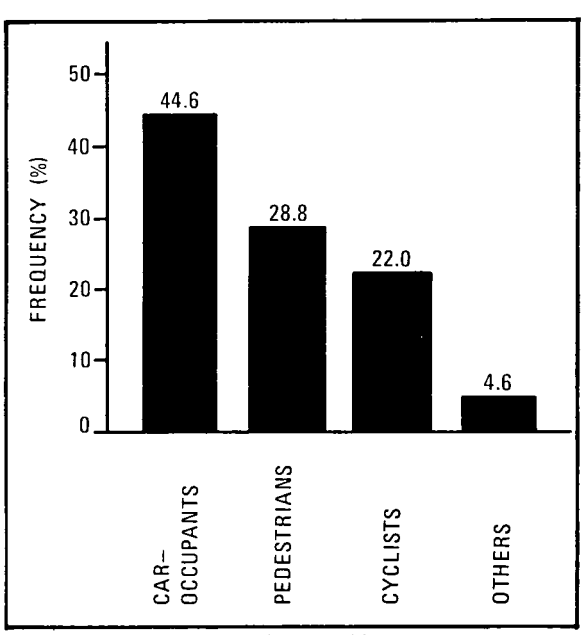


Figure 1. Distribution of Fatal Accidents

*Numbers in parentheses refer to references at the end of this paper.

FINDINGS FROM ACCIDENT STATISTICS

Figure 1 illustrates the distribution of fatal accidents for the different types of traffic participants for Europe. The vehicle occupant group is the largest, followed by pedestrians and cyclists/motor cyclists. A closer study of the data shows the distribution of fatal injuries to traffic participants, including their collision "partners," as illustrated in Figure 2. When a study of the sequence is made, the following frequency is shown:

Fatalities in the collision between:
Pedestrian – car
Car – car
Cycle – car
Car – truck
Car – fixed obstacle
Pedestrian – truck
Cycle – truck
Others

Overall, the death rate is at its highest in passenger cars.

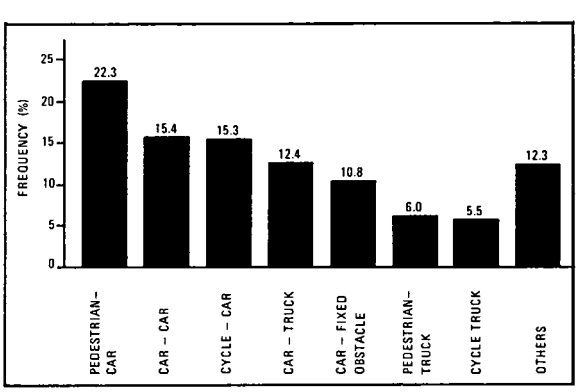


Figure 2. Distribution of Fatal Accidents in Different Crash – Types

PEDESTRIAN PROTECTION

Where pedestrian collisions are being considered, one must separate the town center and suburban collision with a passenger car or truck in West-Germany (approximately 73%) from the collision which occurs outside the built up area (approximately 27% (2),). In the first instance there are essentially three measures which, when implemented, would result in a reduction of fatal accidents.

1. Separation of traffic participants by means of optimal routing.
2. Education of pedestrian and driver.
3. Measures on the car itself.

In the second instance the measures could take the following form:

1. Making the pedestrian more easily visible.
2. Education of pedestrian and driver.
3. Measures on the car itself.

The automobile manufacturer is mainly concerned with the measures to be taken on the car itself, with regard to the pedestrian safety aspect. This includes the making safe of the car outside contour and the optimum shaping of the vehicle front end. As these measures would only be effective up to a specific impact speed it is absolutely essential to have here the evidence of the impact speed including the statistical distribution.

CAR TO CAR COLLISION

The next highest percentage of collision participants is the car to car collision. Together with this group the car-truck and car-fixed obstacle collisions must also be considered. Before considering the different parameters more closely it is essential to undertake a sensible definition of the terms. This was attempted in Figure 3. In the complex area of car to car collisions it is especially essential, to have this standardized use of terms. The most important point is, that for the restrained car occupant, the change in speed during the impact is generally the main factor.

Influencing Factors in the Collision Sequence

The following factors have a decisive effect on the accident sequence, from the beginning onwards:

The impact of the pedestrian with the road after a car collision could be eliminated by a net, although this could result, besides the high costs through the necessary energy support, in injuries on the pedestrian itself.

1. The masses involved.
2. The impact speed entering the collision phase.
3. The direction and location of impact (front, side or rear).

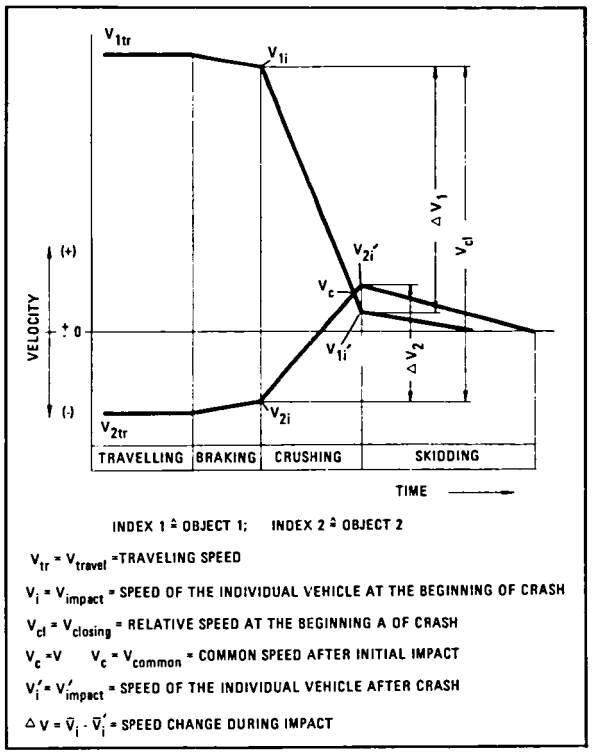


Figure 3. Velocity vs. Time Relationship for Representative Collisions

4. Deformation characteristics.
5. Architecture.

Masses Involved – Apart from the case where a vehicle crashes into a fixed immovable obstacle and only the mass of the car under observation is involved, the masses of two partners are involved in the accident. The larger their differences the greater will be the change in speed of the lighter one, for any given impact velocity. In extreme cases, the speed change of the small car will be so large that the smaller car reaches (twice) the impact speed, Δv will become 2v. As this physical fact cannot be eliminated, one must extract the masses to be observed from statistics. Figure 4 shows the cumulative frequency of the registered cars within the

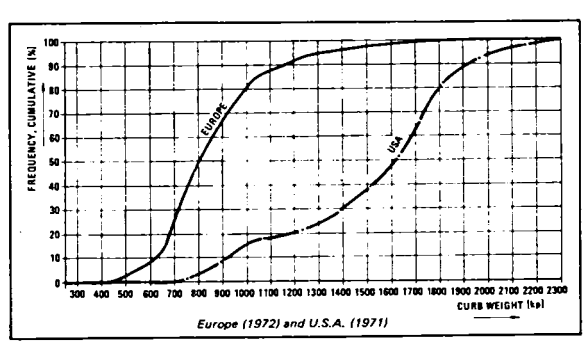


Figure 4. Cumulative Distribution of Vehicle Masses in Europe (1972) and USA (1971)

EEC for the year 1972(3) and for the United States (4). The mass difference of the registered cars can be seen clearly. In the United States, a merging of today's two peak values can certainly be expected, because here a tendency of the larger cars in the direction of the medium U.S. cars can be forecast. As there is today still very little statistical material on collision probability, the cumulative collision frequency shown in Figure 5 was calculated in

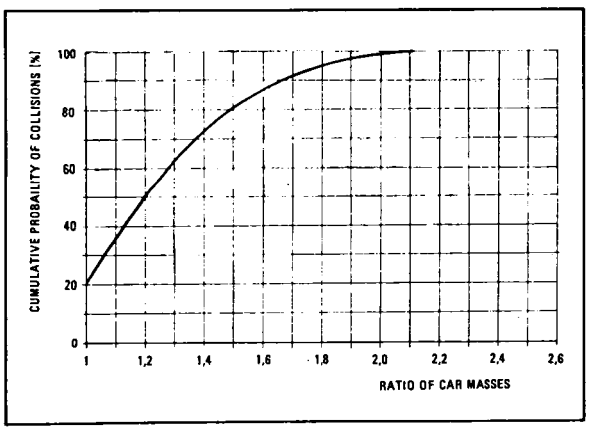


Figure 5. Probability of Collisions

conjunction with the involved mass conditions. As can be seen, in 95% of car to car collisions the mass relationship is approximately up to 1 to 1.8. If one takes this mass relationship as a basis the speed change for the small vehicle with the same impact velocity will be:

$$\Delta v_{1.0m} = v_i - \frac{(mv_i - 1.8mv_i)}{2.8m} = v_i + 0.286\, v_i = 1.286\, v_i$$

$$\Delta v_{1.8m} = v_i - 0.286\, v_i = 0.714\, v_i$$

With a collision speed of v_i = 30 mph per vehicle, the Δv for the small car would be 38.6 mph and for the large car it would be 21.4 mph. A shift of the overall accident probability due to mass changes of future cars can be expected. Even if this is taken for granted, there will be no significant change in accident probability between two cars, as long as the trend of mass change of newly registered cars does not vary.

Impact Velocity – It is clear that the impact velocity between the traffic participants is a significant factor in relation to the force on the car and the car occupants.

Usually, the frequency of deaths, severe and minor injuries from the accident analysis is related to equivalent barrier impact speeds and equivalent test speeds (5). The earlier ESV conferences have discussed this subject in detail and have shown the problems involved in those derived speeds.

We know from the accident analysis, that more than 75% of frontal passenger car collisions are not equivalent to frontal barrier impacts.

If there is no central impact exactly on the vehicle's longitudinal axis, the deceleration-time history and thus $\Delta v = f(t)$ is such, that the speed in the cars longitudinal direction – and consequently the deceleration – is lower.

Extrapolation from this accident analysis in the direction of higher barrier impact speeds (6) of 45 or 50 mph is critical, as will be shown in the following example.

A 1500 kg car develops in a 30 mph barrier impact a mean deformation force of 30,000 kp or 29,400 daN, at a mean deceleration of 20 g and with a deformation distance of 460 mm.

If the speed is increased, for example, to 50 mph, it is necessary for energy absorption purposes to either extend the deformation distance or increase the deformation force. In order to avoid lengthening the vehicle excessively one will frequently go up to the limit of the deceleration level at which the injury criteria are still fulfilled. Assuming that the deformation distance increases to 600 mm, the mean vehicle deceleration would then increase to 41 g, and the mean deformation force to 61,500 kp or to 60,000 daN, that is to say, more than double. In a car-car collision this would mean, that the smaller car is not only subjected to the larger mass, but is also exposed to the larger deformation force, so that the g-level on the small car would exceed the survivable limit.

In addition, the measures used to increase the deformation force would have a very negative effect in a side impact through the reinforcement on the bigger vehicle. Therefore, before one demands considerably higher impact speeds for the barrier impact, v closing of 60 mph for the given mass relationship of 1/1.8 should first be considered as the problem to be solved.

Collision Direction and Impact Area – The accident analysis shows that of all accidents, 60% are frontal collisions, 28% side collisions, 7% vehicle

rollovers and the rest rear end collisions. The concentration of the deformation in the frontal collisions shows furthermore that it is not symmetrical with the vehicle longitudinal axis. More than 75% deviates from the 90 degree frontal barrier impact and represents an asymmetrical impact. In the side collisions it is found, that the impacts are concentrated at the level of the so-called strong passenger cell mainly on the side doors. The impact direction is approximately 75° from the front, based on the longitudinal axis of the vehicle struck. This knowledge must also be taken into consideration with regard to the compatibility.

The rear end collisions are relatively slight from a statistical point of view, so that they don't need to be considered at the moment in connection with the question of compatibility.

Deformation Characteristics – The deformation characteristics of vehicles in the market at present vary considerably. Investigations of more than 30 vehicles between 690 and 1,324 kg produced minimal forces of 23,500 daN and maximal forces of 95,500 daN with deformation distances of from 430 to 885 mm, see Figure 6 (3). By collecting these deformation force-distances one could, with the aid of a computer program, record these values statistically. The large differences existing at the moment could be levelled off in the development of vehicles when the nominal deformation characteristics are known.

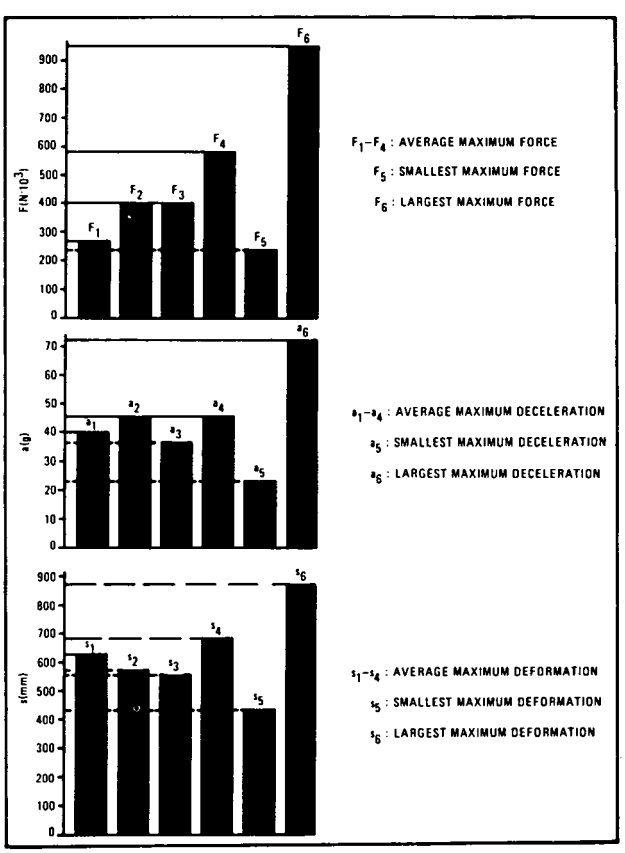


Figure 6. Average Maximum Forces, Deceleration and Deformation for Particular Classes of Masses and Highest and Lowest Values

Architecture of Vehicle Structures – The architecture of vehicle structures has a significant influence on the performance of the traffic collision partners. This applies particularly to the car-truck collision where by optimizing the underride protection a further degree of optimization can be obtained. One can determine the energy absorbing structures of many models in genuine accident simulation tests. However, within the individual companies one can assume that knowledge of these energy absorbing components is available. As indicated in Figure 7 one could attempt to classify this information and evaluate it with computers.

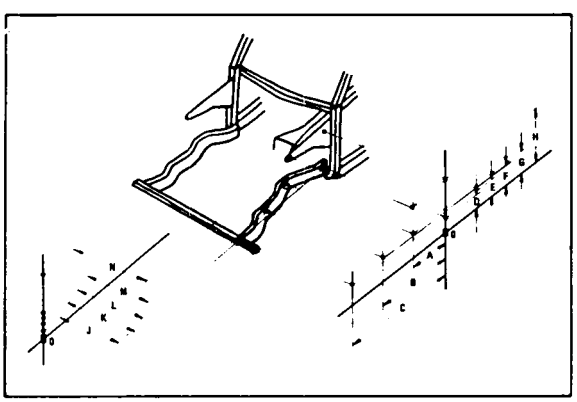


Figure 7. Scheme of Front Structure Architecture

METHODS OF JUDGMENT AND CRITERIA FOR VEHICLE COLLISIONS

When ignoring the vehicle-pedestrian collision one must, in the case of vehicle collisions, mainly consider the collisions of car-car, car-truck and car-fixed obstacle. With the assumption that an adequate underride protection was fitted on trucks an angular collision with a rigid barrier at 60-90° would simulate the frontal collision car-truck as well as car-fixed obstacle. For the frontal collision between two vehicles a representative vehicle could be selected on which the large and small vehicles and also vehicles of similar weight could be measured. This representative car could be a deformable barrier the front contours of which are similar to those of a car. The representative car could also be used for side crashes. For European conditions this vehicle mass should be in the 880 kg range.

Assuming, that the cars to be tested would meet the passenger protection criteria in a fixed barrier collision at a particular speed of say 30 mph and assuming that each car in a car-to-car collision would travel up to V_i = 30 mph (V_{ci} equivalent 60 mph), the smaller vehicle would be tested with regard to the larger ΔV (higher demands on the passenger protection) and up to a certain degree with regard to the aggressivity of the structure (forces and architecture).

The larger car would be tested mainly with regard to its aggressivity by judging the deformation of the representative car and to a lesser degree with regard to its passenger protection due to the smaller ΔV. Cars, whose masses differ from that of the representative car by more than 1.6 times, would have to be given a correction factor in respect of the V_{cl} to ensure that the speed change of the representative car does not exceed 37 mph (this value is reached by $m_1 : m_2$ equals 1 : 1.6).

This applies both to cars which are heavier by more than the 1.6 factor or lighter by more than the 1.6 factor.

CONCLUSION

The problem of compatibility in road traffic is extremely complex. The problems of pedestrian protection and car-car, car-truck, car-fixed obstacle collisions need to be solved with the highest degree of priority.

The automobile designer can contribute more effectively by dealing with the second group. In order to put these further measures into effect it is absolutely essential to learn even more about traffic accidents and about the crash performance of the vehicles of other manufacturers.

Human protection has the highest priority, therefore it is essential to know the change in velocity of the vehicles under crash conditions as a function of masses of the collision partners. This would mean, that a crash recording device with at least two measurement axes (longitudinal and lateral) is necessary, to obtain the precise function in crash situations. It might be feasible, to combine at least the recording device of such a crash recorder with the "pre-crash recorder" proposed by our handling experts. In vehicle-pedestrian accidents, the knowledge of the V_i is sufficient.

The development of a deformable representative vehicle must be done with high priority, so that with further knowledge of new accident data optimal test requirements and conditions can be established.

REFERENCES

1. NHTSA Proceedings of the second international conference on ESV, Sindelfingen.
2. VDA-Jahresstatistik 1972
3. U. Seiffert, Probleme der Automobilsicherheit Dissertation Freie Universität Berlin.
4. James O'Day, A statistical description of large and small car involvement in accidents. HSRI, University of Michigan, United States.
5. U. Seiffert, Simulation of road traffic accidents with barrier impact tests. Report of the 4th international ESV conference, Kyoto, Japan.
6. NHTSA Docket 74-15 Advance notice concerning higher speed protection requirements.

COMPATIBILITY BETWEEN VEHICLES IN FRONTAL AND SEMI-FRONTAL COLLISIONS

P. VENTRE
Head of Crashworthiness Department
The Renault State-Owned Works

IN GENERAL

It has been shown[1] that the effects of a collision on the occupants of a given vehicle depend on the three following parameters:

1. The speed variation that the occupant undergoes;
2. The level of deceleration;
3. Vehicle deformation that may lead to modifications of occupant environment and a decrease in the efficiency of the restraining devices.

It is clear that when a collision occurs, the first of these three parameters governs the others, a significant speed variation generally goes with a high level of deceleration and structural deformation.

It is therefore the "speed variation" parameter that must first be determined when finalizing compatibility tests.

STATISTICAL ASPECTS OF THE PROBLEM

The speed variation that the vehicle occupants undergo in a collision depends on the following two factors:

1. Masses of the vehicles involved;
2. Closing speed at moment of impact.

The first of these two factors is statistically known; it is the rating of vehicles according to their masses in a given car population (Figure 1).

This rating enables the collision probability curve to be plotted as a function of the mass relationship $k = M/m$ (where $M \geqslant m$). It is interesting to note that in the 1972 French car population, 95% of the collisions took place with a mass relationship smaller than or equal to 2 (Figure 2).

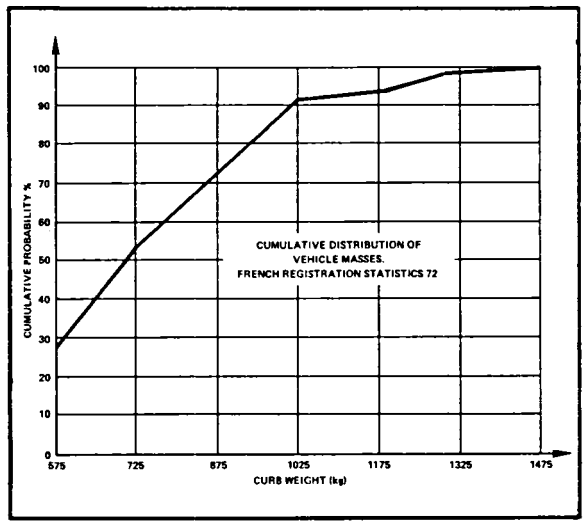


Figure 1

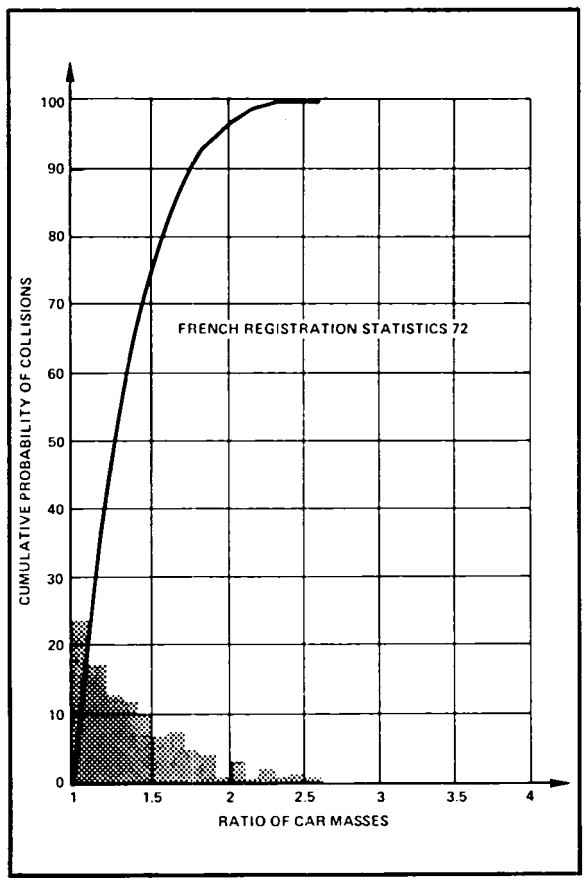


Figure 2

[1] "Proposals for Test Evaluation of Compatibility Between Very Different Passenger Cars" by P. Ventre and J.C. Rullier, Report on the Fourth International Conference on ESV, 1973.

On the other hand, as far as the second parameter is concerned, a hypothesis must be made (closing speed at moment of impact); this hypothesis has been introduced in the calculations in the form of a "vehicle distribution curve of speeds prior to the impact."

First, this curve is common to all vehicles, but it can be modulated as a function of the class of mass (Figure 3).

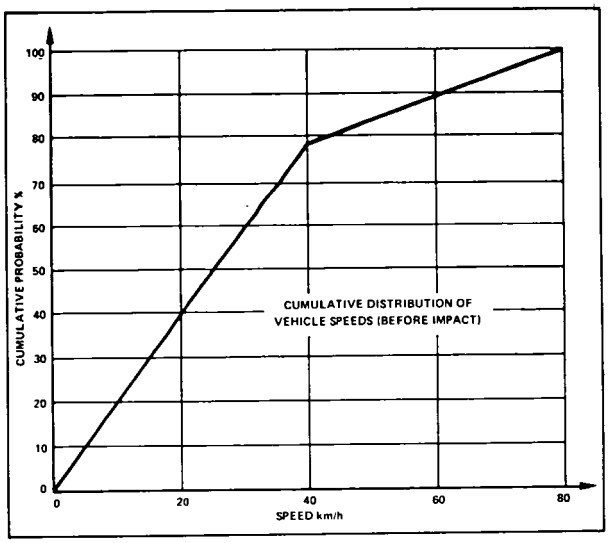


Figure 3

Applying these hypotheses to the seven classes of masses comprising the French car population has enabled us to determine:

- The distribution curve of the occupants involved as a function of the speed variation for each class of mass, on the one hand, and for the car population as a whole on the other hand; this last result, which is very close to the Peugeot-Renault investigation curve corroborates the speed hypothesis described above;
- The distribution curve for the killed and seriously injured as a function of the speed variation for each class of mass, and for the car population as a whole on the other hand; the result (Figure 4) is obtained by introducing in the calculations the probability of being killed or seriously injured as a function of the speed variation (this probability is based on the Peugeot-Renault accident investigation[2]); the curves were also plotted for the occupants wearing belts (Figure 5).

[2]See "Efficiency of the 3-Point Belt in Real Accidents" by Dr. Claude Tarriere, Association Peugeot-Renault, Report on the Fourth International Conference on ESV, 1973.

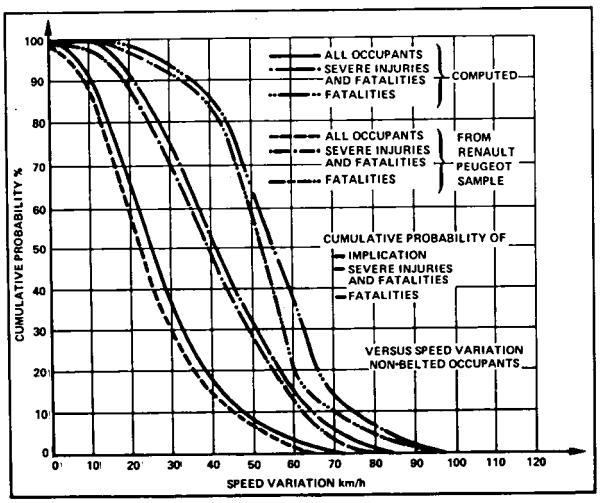


Figure 4

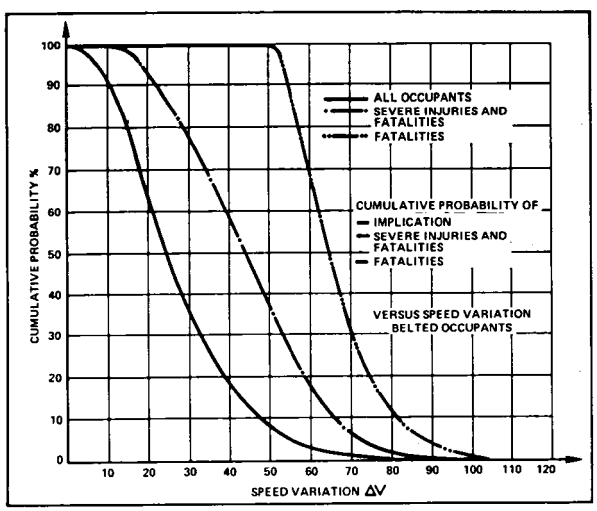


Figure 5

APPRAISING COMPATIBILITY BETWEEN VEHICLES

Implications of the Statistical Study

It is possible to determine a "level of compatibility" concerning the speed variations; to do this, we adopt, for each class of mass, a value of ΔV such that it covers a considerable percentage of the occupants involved. These ΔV taken as a whole and plotted as a function of the mass, give a curve that shows the speed variation to be attained per type of vehicle to obtain a "level of compatibility 95%" (Figure 6).

These curves, as could be expected, have the form $\Delta V_i = a/\sqrt{m_i}$. However, the coefficient is not a constant because of the mass distribution curve being introduced into the entire car population.

This is shown by considerations concerning energy: each vehicle should be capable of dissipating $\frac{1}{2}m_i\Delta V_i^2$.

If the relationship $\Delta V_i = a/\sqrt{m_i}$ were to be verified, $W = \frac{1}{2}m_i(a^2/m_i) = a^2/2$ would be a constant.

The table shows that this theory has not quite been satisfied.

The least frequent types of vehicles will be subject to smaller ΔV than the theoretical ΔV to obtain the same level of compatibility.

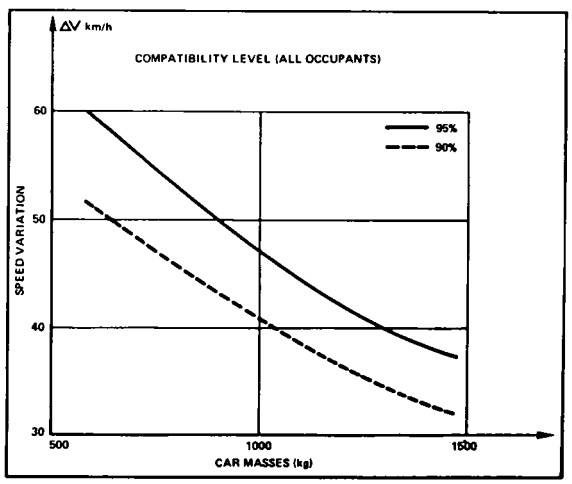


Figure 6

MASS Kilograms	ΔV km/h (95% level)	Vm/s	$a = \Delta V\sqrt{m_i}$	$W_{max} = \frac{1}{2}m_i(\Delta V_i)^2$ Kilojoules	
575	60	16,667	400	79	Most frequent vehicles
725	55	15,278	412	85	
875	50	13,889	411	84	
1,025	46	12,778	410	84	
1,175	43	11,944	409	84	
1,325	40	11,111	404	81	
1,475	37	10,278	395	78	

Vehicle Deformation Stress Laws

The preceding has shown that vehicles should be capable of absorbing the same quantity of energy at the same time protecting their occupants.

This energy dissipation must be directed so that compatibility is attained; this should be done on the deformation-stress characteristics which should comply with the following requirements:

- Not to impose levels of deceleration that make the restraining devices inefficient;
- Should be sufficiently strong as not to entail too high deformation distances.

To determine the deformation-stress characteristics, we have therefore formulated the following hypotheses:

- The restraining devices are efficient up to a vehicle deceleration level γ;
- A vehicle of mass m_o exists which limits the lower end of a given car-mass population;
- In a car-car collision, the structure of the smaller vehicle dissipates a maximum quantity of energy.

According to the last hypothesis, the energy dissipated during the collision, i.e., ΔW

- Will be $\Delta W = \frac{1}{2}[mM/(m + M)]V_R^2$
- Or $\Delta W = \frac{1}{2}[m(m + M)/M]\Delta V_m^2$ if ΔV_m^2 is the speed variation of the smaller vehicle; of this energy, the vehicle will be able to absorb $\frac{1}{2}m\Delta V_m^2 = W$.

The remaining energy will be dissipated through the deformation of the larger vehicle, i.e., $\frac{1}{2}(m/M)(m + M)\Delta V_m^2 - \frac{1}{2}m\Delta V_m^2 = \frac{1}{2}(m^2/M)\Delta V_m^2$ or further Wm/M.

Taking these hypotheses into account, the characteristics will be determined in the following manner:

- If the vehicle of mass M collides with the vehicle of mass m, an energy of Wm/M is to be dissipated with an effort at the most equal to γm; deformation of the larger vehicle will be $x = W/M\gamma$.
- If the vehicle of mass M collides with the vehicle of mass m + dm, it should dissipate $dW = [W(m + dm) - Wm]/M$ with the effort F supposed to be constant for the small variation dm; the extra deformation of the larger vehicle will therefore be $dx = Wdm/MF$; but $F = m\gamma$; $dF = \gamma dm$; hence: $dx = WdF/M\gamma F$; $dF/F = M\gamma dx/W$; $F = CeM^{\gamma x/W}$:

Initial Conditions. When vehicle M collides with the vehicle of mass m_o, it should crush by a quantity $x_o = Wm_o/Mm_o\gamma = W/M\gamma$ and this with an effort at the most equal to

$$F_o = \gamma m_o$$

$$m_o\gamma = Ce^{M\gamma x_o/W} = Ce^{(M\gamma/W)(W/M\gamma)} = Ce$$

$$\Rightarrow C = m_o\gamma e^{-1}$$

hence the general equation for the deformation-stress characteristic of the vehicle of mass M: $F = m_o\gamma e^{[(M\gamma x/W) - 1]}$.

Utilizing this equation has led to the graphs shown in Figure 7.

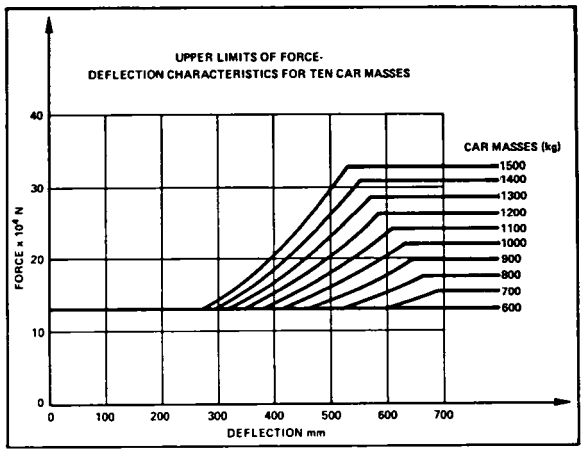


Figure 7

Application to Compatibility Tests

The vehicle deformation-stress characteristics therefore level out (determined by the maximum allowable deceleration γ) this is preceded by:

- A level determined by the vehicle of minimum mass;
- A curved part, the equation of which, as a function of W, m_o, γ and M, has been given above.

Obviously, it is the first part of the characteristic (first level and curved part) that will have to be verified during the compatibility test.

We could envisage this test being carried out in the following way:

1. Given a vehicle of mass m, a mobile barrier of identical mass is built.
2. The mobile barrier must have the deformation-strength characteristic that has been determined for this mass by the above method.
3. The chosen closing speed will be $2\Delta V_i$ where $\Delta V_i = a(m_i)^{-1/2}$; the coefficient a being chosen according to the method stated above.
4. During this test, the following should be confirmed:
   - That the absorber fitted to the mobile barrier is not destroyed beyond the limit corresponding to half the energy dissipated during the collision.
   - That the injury criteria will be satisfied for the occupants of the vehicle to be tested.

This shows that it is possible to verify the compatibility of all vehicles with a barrier whose mass and deformation-stress characteristic depend on the mass of the vehicle to be tested.

RELATIONSHIP BETWEEN VEHICLE FRONT-END DEFORMATION AND EFFICIENCY OF SAFETY BELTS DURING FRONTAL IMPACT

J. ELSHOLZ
Bayerische Motoren Werke AG

INTRODUCTION

The decision in favor of one of the major competing restraint systems for the protection of driver and front seat passenger in typical passenger car accidents is still outstanding.

However, when comparing the relationship between the investments made for the development of the airbag and those made for the development of the safety belt during the last five years, it may be concluded that

1. the safety belt has been able to hold its own very well and that
2. there is still a considerable development potential inherent in this system.

It is the object of this paper to outline one of the possibilities for further activating this potential.

THEORETICAL CONSIDERATION OF CONTROLLED FORWARD DISPLACEMENT OF OCCUPANTS DURING FRONTAL IMPACT

The evaluation of the deformation behavior of the front-end of passenger cars has hitherto been based on the assumption that in an accident the kinetic energy of the vehicle should be transformed into deformation work with a minimum deformation of the vehicle, in order to minimize the deformation of the passenger compartment which constitutes the survival space for the occupants.

In order to achieve this, it has almost universally been tried to give to the characteristic curve which is used for evaluating vehicle front-end behavior, i.e., the acceleration/distance curve, a rectangular shape. There is no doubt that this results in a very efficient use of material properties when this efficiency is defined as the proportion of deformation work to vehicle mass involved in the deformation (mkp/kg rsp ftlbs/lbs) at a given mean acceleration of the vehicle. However, development work performed during the past years with the aim of designing a passenger compartment with optimum rigidity has made it possible for us to introduce much higher forces into this compartment structure without dangerous deformation. Thereby, the undeformed safety zone of the vehicle front-end could be reduced.

This additional deformation zone obtained by structurally improving the passenger compartment may now either be used to increase the deformation work and thereby allow a higher impact velocity, or it may be employed for reducing the effects of the impact on the occupants. We at BMW have decided in favor of the second possibility since it is apparent from a study of accident statistics that the effects of an accident may hardly be reduced, with a reasonable relationship between benefit and cost, by merely increasing the impact velocity which occupants can survive. We, therefore, have adopted the practice of evaluating the deformation behavior of a passenger front-end not with a view of optimum use of material properties, but with the aim of optimum protection for a vehicle occupant who is protected by a restraint system. When evaluating a vehicle front-end, we are making use of three major criteria which are important *for the occupant.*

Relative Velocity Between Belted Occupant And Vehicle After Overcoming Of Belt Slack

This value is of decisive importance for the angular acceleration of the head which necessarily results when the torso of a vehicle occupant is restrained. The higher the differential velocity between occupants and vehicle after the overcoming of belt slack, the greater will be the angular acceleration of the head and thus also the resulting velocity of the head which impacts a vehicle component (steering wheel, instrument panel, windshield, etc.).

Remaining Deformation Distance Of The Vehicle In Which The Occupant Participates After Overcoming Of Belt Slack

This value is decisive for the strain acting on the body of the occupants. The greater the front-end deformation which is *effective for the occupant,* the smaller will be the belt forces, which also contributes to a reduction of the angular acceleration of the head.

Mean Acceleration During Deformation Of The Part Of Front-end which is Effective For The Occupant

This value is also decisive for the strain acting on the occupants.

Various deformation models will now be analyzed, applying the above criteria.

All values are dimensionless and belt slack has been assumed to be constant for all models. The values obtained with the rectangular pulse are used as reference values for the other models.

Rectangular Pulse (Figure 1). As mentioned before, the rectangular pulse provides for optimum transformation of energy, but its technical realization is very expensive. Furthermore, the rectangular pulse is undesirable as far as ease and economy of vehicle repair are concerned, and finally it is found that the rectangular pulse does not represent an optimum when the above criteria are applied to it.

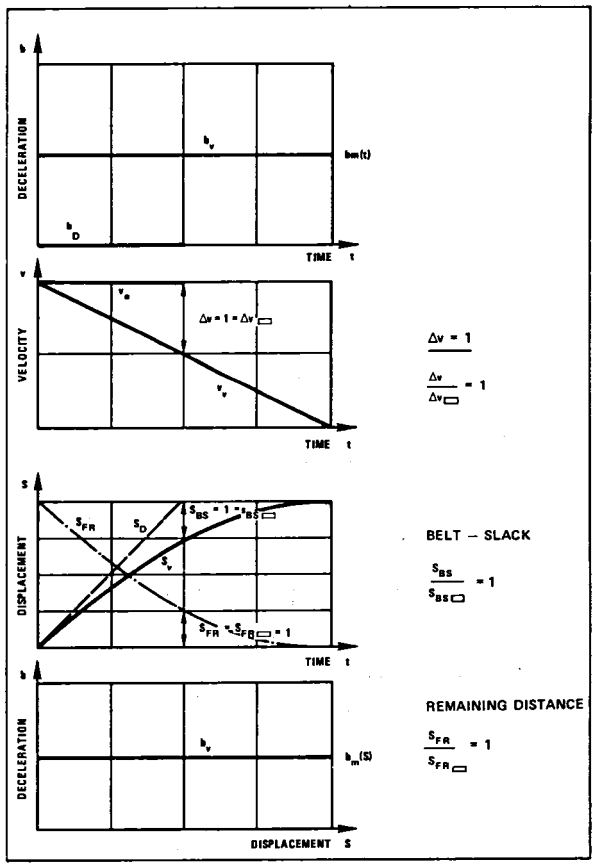


Figure 1

Triangular Pulse (Figure 2). The triangular pulse of vehicle front-end deformation presents serious disadvantages as compared to all other models of deformation. Since vehicle speed has already been strongly reduced at the time of overcoming belt slack, the relative velocity between occupants and vehicle is 28% higher than at the same time in the rectangular pulse when a realistic value for the belt slack is taken as a basis.

The occupant participates only in a small part of the remaining deformation distance of the vehicle, because at this time approximately 75% of this distance has already been used.

Half Sine Pulse (Figure 3). As compared to the triangular pulse the half sine pulse shows a slight improvement when the above criteria are applied to it. However, it presents serious drawbacks compared to all other deformation models. As compared to the rectangular pulse, relative velocity aat the time of overcoming belt slack is 20% higher, and the effective use of front-end deformation is 50% lower.

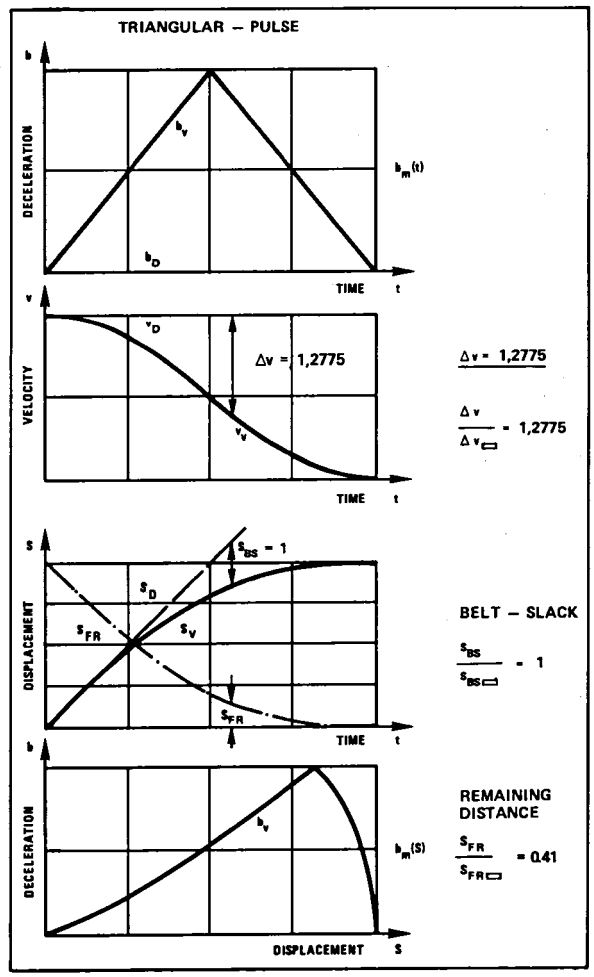


Figure 2

Double Rectangular Pulse (Figure 4). Theoretically, an ideal characteristic may be found which is based on the rectangular pulse but is superior to it under all aspects concerning the adverse effects of the occupants. At the time of overcoming belt slack, the relative velocity is 20% lower and the effective use of the deformation distance shows an improvement of 50%.

However, in view of the limitations already mentioned in connection with the rectangular pulse, this ideal characteristic is not practicable.

Actual Idealized Deformation Characteristic (Figure 5). The experiences gained from model studies of the rectangular and of the ideal characteristic have resulted in a compromise in one of

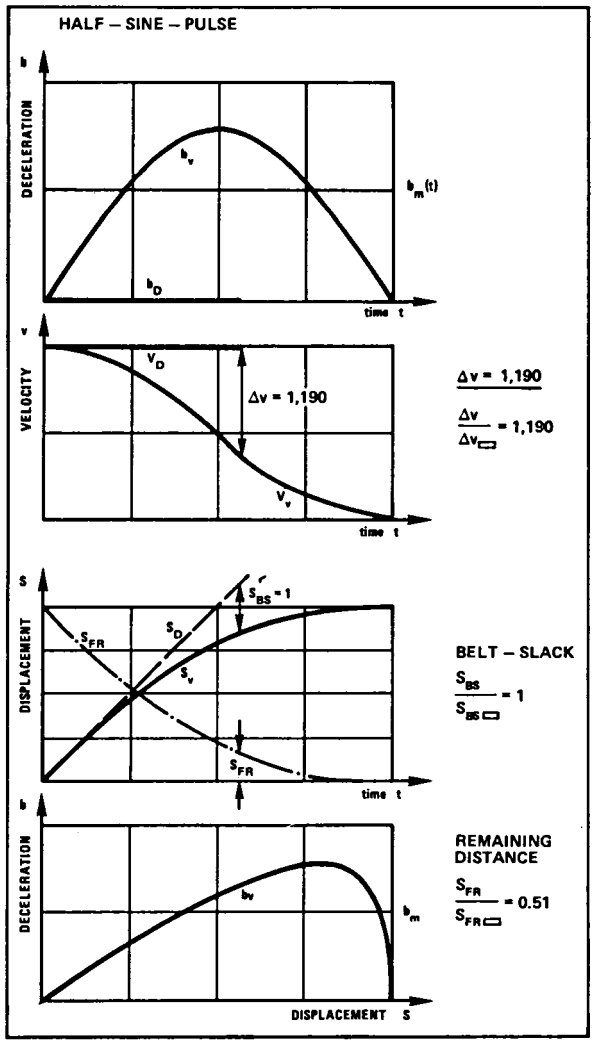


Figure 3

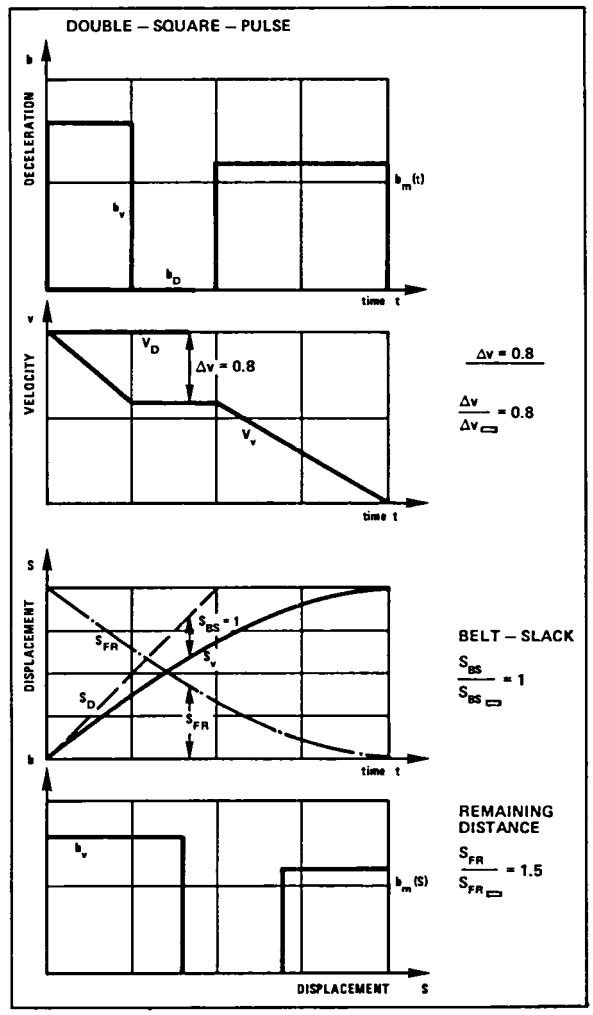


Figure 4

our new vehicle models. This compromise is technically practicable on the one hand while, on the other hand, it shows positive results approaching the ideal characteristic when applying the criteria of evaluation. Typical for this characteristic are two pronounced acceleration peaks and a calm-down valley following the first peak. The first acceleration peak serves to overcome the unavoidable belt slack. Up to this point a relative velocity has been built up which is lower (-17%) than that of the rectangular pulse. While the front-end deformation distance available is slightly below that of the rectangular pulse, it is clearly above the deformation characteristics obtainable in practice. Moreover, the gently rising part of the curve at the beginning of deformation will probably reflect favorably on the economy and ease of repair of the vehicle front-end.

Figures 6 and 7 show the dependency of the criteria of evaluation on the magnitude of belt slack. The shaded area represents the practically feasible range.

The values obtained theoretically have meanwhile been confirmed by experiment. With older models, which do not have the pronounced front-end deformation behavior of the BMW 525, disadvantages for the occupants were clearly apparent.

The fact that the above-mentioned criteria had been correctly applied to the vehicle front-end is also confirmed by the diagrams of vertical head acceleration (Figure 8) which, in case of impact, may serve as a measure for the rotational angular velocity. It may clearly be seen that the jerk, i.e. the differential quotient of acceleration versus time, to which the head is subjected, is less pronounced in vehicles of new design than in the older vehicles. As apparent from the next diagram (Figure 9), the increase of belt forces is also smaller in the newer vehicles.

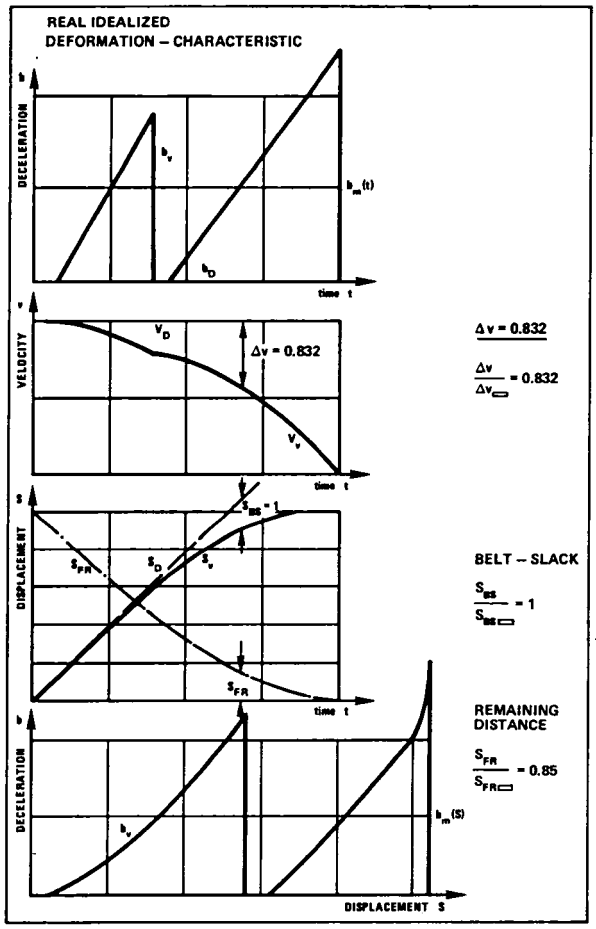


Figure 5

TABLE 1

COMPARISON OF DEFORMATION – MODELS

	Δv RELATIVE VELOCITY	S_{FR} REMAINING DEFORMATION DISTANCE	b_{mR} MEAN DECELERATION OF REMAINING DISTANCE
SQUARE PULSE	1.0	1.0	1.0
TRIANGULAR PULSE	1.28	0.41	1.26
HALF-SINE PULSE	1.19	0.51	1.28
DOUBLE-SQUARE PULSE	0.8	1.5	0.96
REAL IDEALIZED DEFORMATION CHARACTERISTIC	0.83	0.85	1.6

ALL DATA REFERRING TO SQUARE PULSE

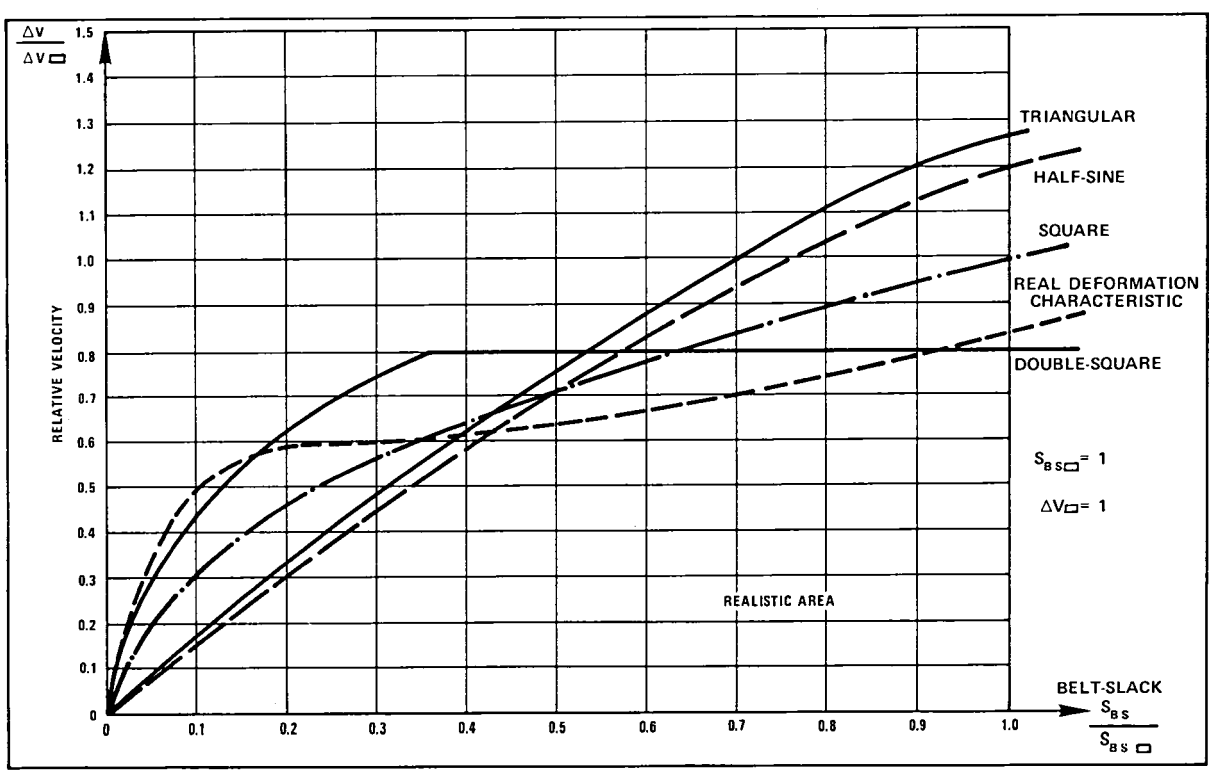


Figure 6

TABLE 2 EFFECTIVENESS OF REAL CRASH-TESTS			
CAR MODEL	Δv [m/s] RELATIVE VELOCITY	S_{FR} / S_{max} [%] RELATIVE REMAINING DISTANCE	b_{mR} [g] MEAN DECELERATION OF REMAINING DISTANCE
TYPE A	5.0	31.3	17.6
TYPE B	3.8	32.8	25.8
TYPE C	3.4	44.7	19.6

CHANGING THE DEFORMATION BEHAVIOR OF VEHICLE FRONT-END

With the theoretical results on hand, we were confronted with the most difficult part of the problem, which was to translate this knowledge into practice. For this purpose, test devices had to be built on which the influence of the most important structural elements on front-end behavior could be studied, first statically and later dynamically. Following this, design modifications had to be carried out in order to produce the desired influence.

For static testing, we designed a hydraulic test device with which the zones of interest could be deformed quasi-statically while measuring force and distance (Figure 11).

In determining the dynamic behavior, we could profit from the understanding that, while there is a considerable difference between statically and dynamically determined force/distance behavior, the dynamic behavior will change but insignificantly up from a certain velocity on. Since this velocity is usually below 25 km/h, our task was to build a test device permitting reproducible tests up to approximately 25 km/h within a minimum of time (Figure 10). Among the many possibilities such as inclined plane, vehicle versus vehicle, or use of the crash test installation, we finally decided on the device shown in the photograph (Figure 12). This is a trapezoidal pendulum device on which individual structural elements may be tested, as well as vehicle bodies and complete vehicles.

Moreover, we found that, with relatively short set-up time, it was also possible to carry out tests on the sensor behavior of passive occupant restraint systems, on repair characteristics of production

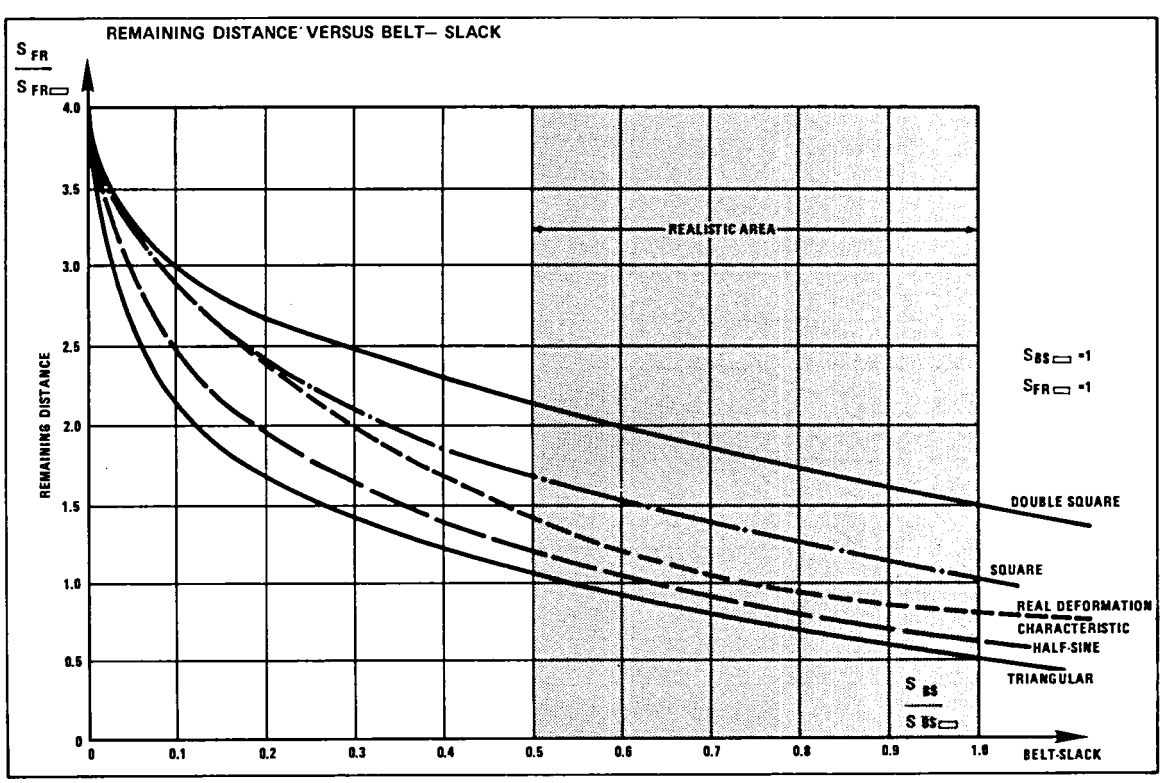


Figure 7

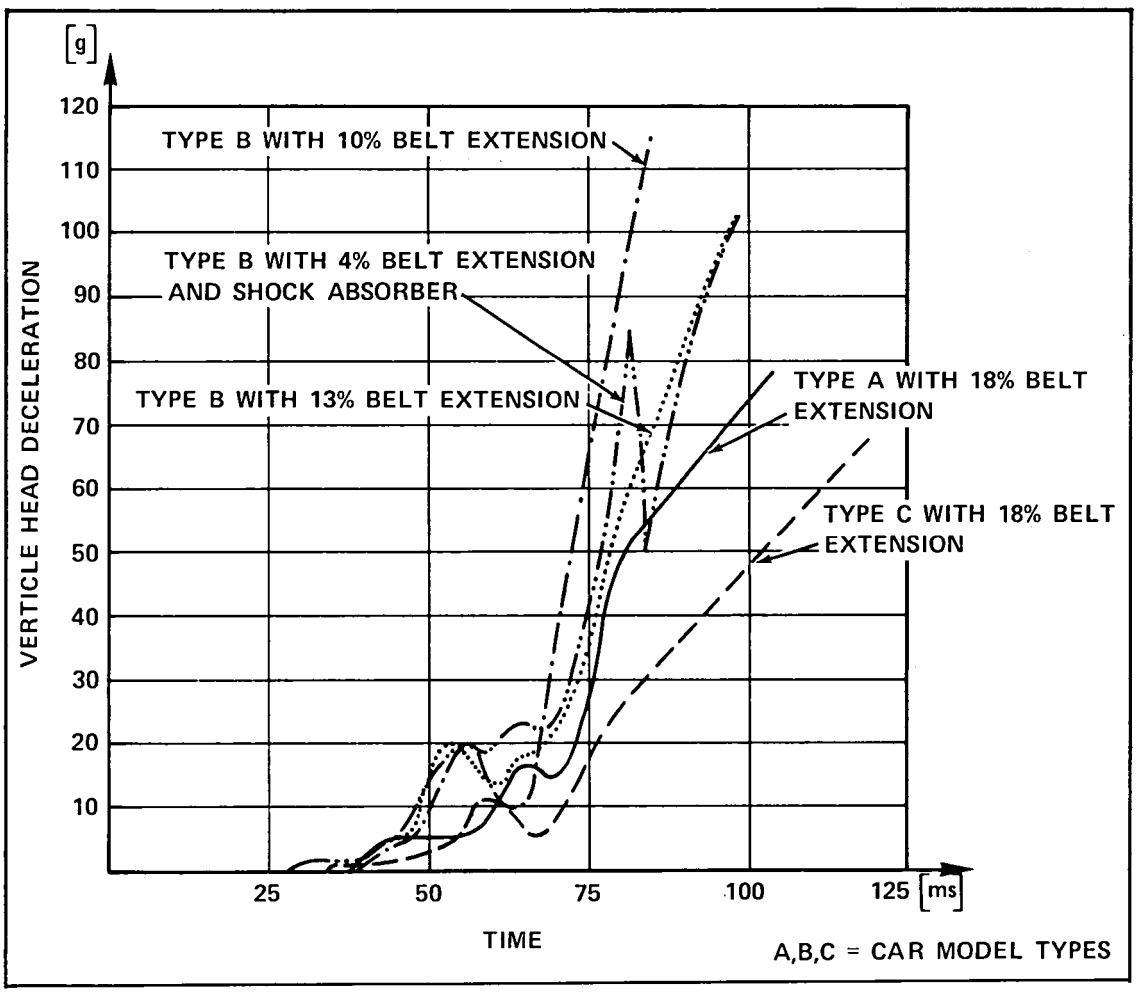


Figure 8

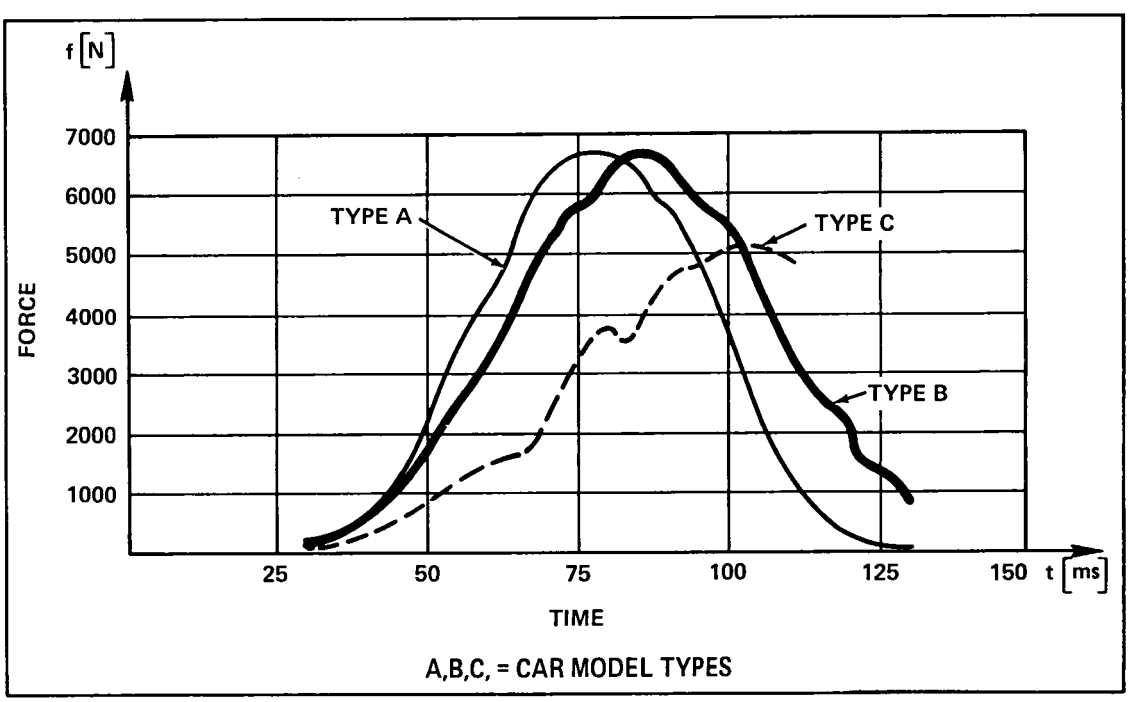


Figure 9

vehicles, and finally also on bumper systems such as those required by FMVSS 215 for the United States.

The test device consists essentially of a concrete barrier, a pendulum frame, and a winch for hoisting up the test object. The barrier has a mass of approximately 27 long tons (60,000 lbs.) and is rendered mobile by a truck axis around which the concrete block is cast (Figure 13).

Movement of this barrier is obtained by hydraulic cylinders mounted on its left and right sides. This makes it possible to adjust exactly the specific point of impact for various types of vehicles. Accidents with various angles of impact may also be simulated on the device.

The pendulum frame is designed for an impact speed of 25 km/h.

The vehicle or other test object is suspended on four pendulum arms by means of special adapters which are mounted in place of the wheels.

Height and parallelity relative to the ground can be adjusted by means of built-in turnbuckles. The pendulum arms which are about 4 m (13 ft.) long are subdivided by universal joints so that it is possible to accommodate different track widths. Inclination of the pendulum arms in the YZ-plane produces oppositely directed axial forces of equal magnitude which exert a stabilizing effect on the freely swinging vehicle.

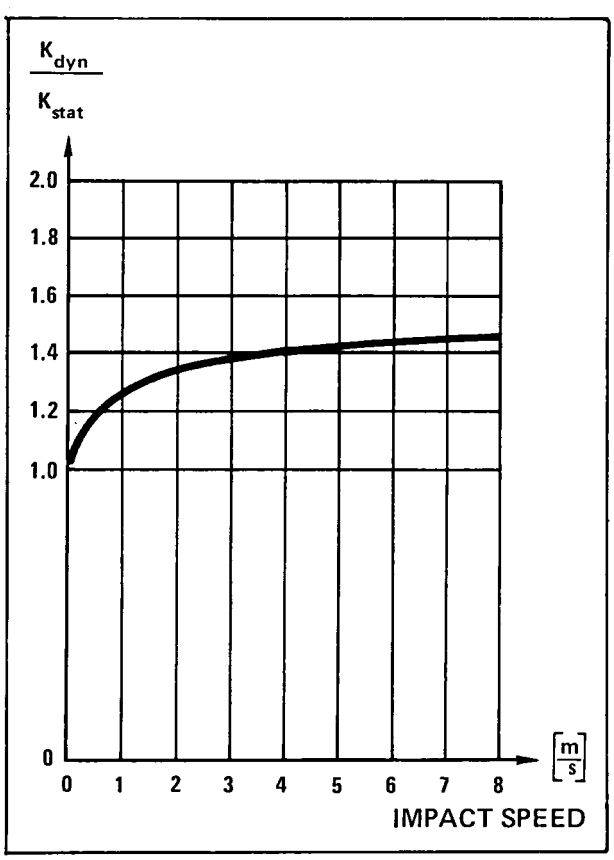


Figure 10

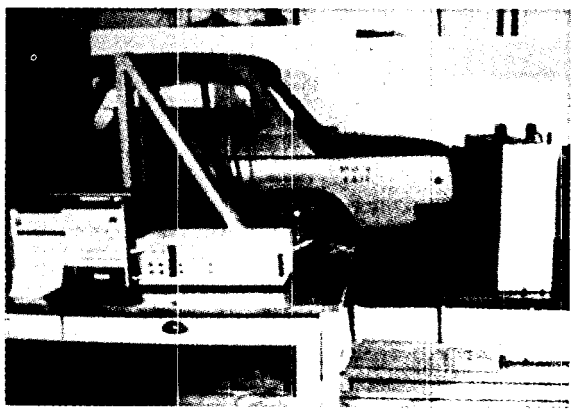


Figure 11

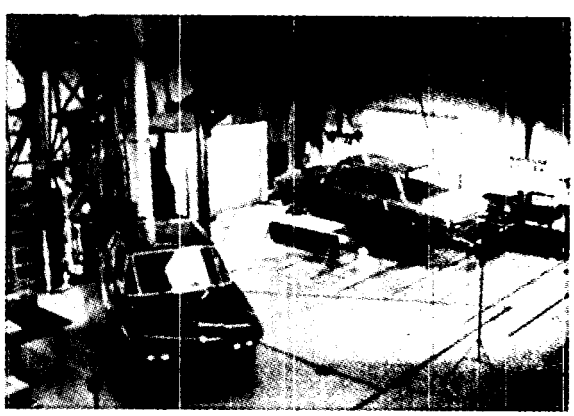


Figure 12

By means of a manually operated winch and a pulley block, the object to be tested is hoisted to the height required for a given velocity. The transmission of forces from the rope to the test object is assured by a tong-like holding device connected to the rope and provided with a centrally acting release mechanism. An electro-magnet closes the holding tongs and permits instantaneous release free of disturbing forces.

This testing device has been used to test, for example, the compression and buckling behavior of engine supports (Figure 14), of engine hoods, and also of wheelhouses with their corresponding side panels.

It was the aim of all these experiments and studies to obtain a behavior in which, after forces have increased to the highest possible level shown by theoretical considerations, buckling of all structural parts is brought about nearly simultaneously; this buckling mechanism finally results in a reduction of acceleration.

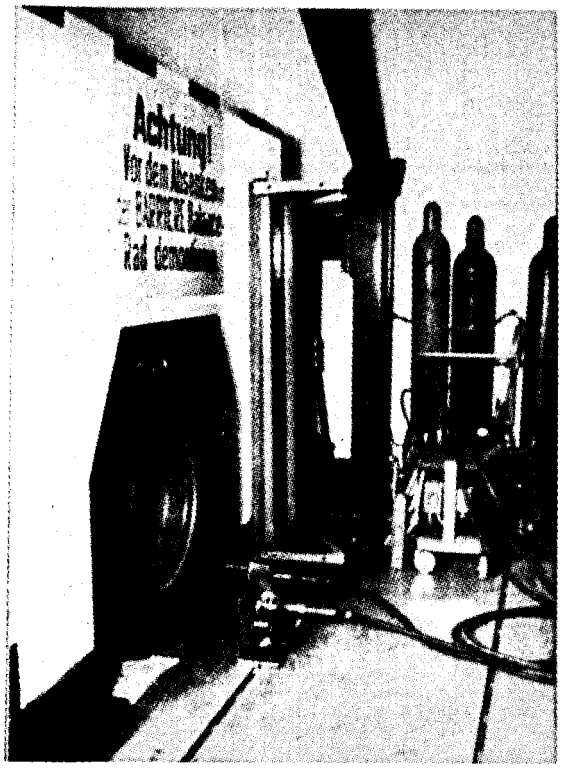


Figure 13

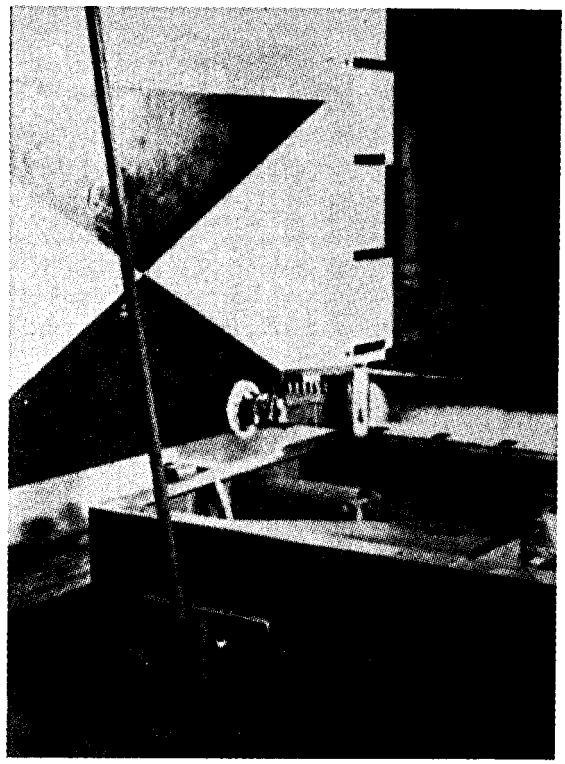


Figure 14

On the design side, this has resulted in the individual structural components being provided with grooves, holes and preprogrammed points of buckling, so that the desired acceleration/distance characteristic could be closely predetermined. That the ideal characteristic previously determined could not be obtained is due to the fact that a vehicle is primarily built for purposes other than sustaining an accident. However, we shall continue our efforts to adapt vehicle front-end behavior as closely as possible to the characteristics of the occupant restraint system.

SUMMARY

Crash tests conducted after theoretical studies confirmed that the efficiency of a safety belt system may be improved by influencing the deformation characteristics of the vehicle front-end.

It was possible to quickly overcome belt slack, to obtain a small relative velocity between occupant and vehicle up to the point of increase of forces, and to assure that belt forces increase slowly in such a manner that the change of rotational angular velocity of an occupant's head is smaller than in comparable vehicles.

Finally, it could be shown how the individual major structural elements could be experimentally evaluated and corresponding modifications in design could be made in order to make better use of the development potential which is still inherent in all safety belt systems.

REFERENCES

1. A. Egli, "Stopping the Occupant of a Crashing Vehicle – A Fundamental Study," SAE Paper No. 670038, Automotive Engineering Congress, Detroit, Jan., 1967.
2. D.F. Moore, "Minimization of Occupant Injury by Optimum Front-End Design," SAE Paper No. 700416.
3. D.N. Renneker, "Basic Study of Energy Absorbing Vehicle Structure and Occupant Restraints by Mathematical Model," SAE Math. Model Conference, Anaheim, California, Oct. 12, 1967.

THE FORD AUTOMATIC SAFETY BELT SYSTEM

EDMUND HELLRIEGEL
AXEL RAUTHMANN
Body Advanced Engineering
Ford Motor Company

Product Development Group presents a new passive operating safety belt system, that offers maximum protection during impacts and requires no activity by the occupant.

INTRODUCTION

Although the automotive industry offers comfortable safety systems for interior occupant protection, i.e., the 3-point harness with inertia reel, road accident statistics note serious injuries based on missing acceptance by occupants to provide personal activity for fastening of safety belts.

Results of extensive analysis of road accidents at front, rear, side and rollover impacts proved that excellent occupant protection can be achieved with a 3-point safety belt system offering the following features:

1. Self-applying function.
2. Entrance and exit of occupants in an unhindered manner.
3. Adequate freedom of movement while driving the car, adaptable to an almost infinite variety of people.
4. Immediate locking of safety belt at normal and emergency braking deceleration.
5. Quickest wind-up of belt at an impact followed by nearly constant energy absorption at lowest level of physical stress to occupant.
6. Easy rescue of occupant by unlocking belt.
7. High level of long-term reliability and easy serviceability.

DESCRIPTION OF SYSTEM

The system is basically a 3-point lap and diagonal belt with fixing point described below:

- At the inner reclining mechanism.
- On the seat back frame (upper).
- On the door still, with a connection to the door inner panel.

The inertial reel is combined with a separate belt retractor and load limiting device. This system is registered as the "Ford Pyro Reel."

The electronic side of the system is a sensor situated on the front end of the vehicle, the seat contact switch, the belt locking attachment and the nonalert control light situated on the instrument panel. The automatic belt transport system locks in the down position on the door sill. The emergency release is located in two places: (1) the door sill; (2) in the belt stem attached to the seat reclining mechanism; and an energy absorbing swing head restraint completes the system.

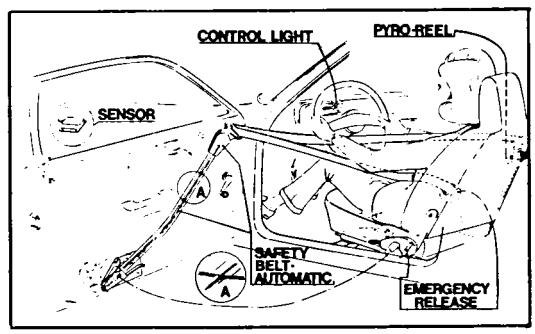


Figure 1.

COMPONENT FUNCTION

Entrance and Exit

When the door is opened the belt system stays in the upper and foremost position on the door waist and is rotated outward – offering unhindered entrance and exit (Figure 2).

Upon sitting on the seat and closing the door the occupant is automatically fastened by the 3-point harness. The belt lock and belt attached is retracted via electric motor and teleflex cable running in the flush fitting guide rail on the door into the locked position at the door sill (Figure 2).

By opening the door from outside or inside, with the motor running or switched off, the belt and belt lock will automatically return to the upper and foremost position at the door waist (Figure 3).

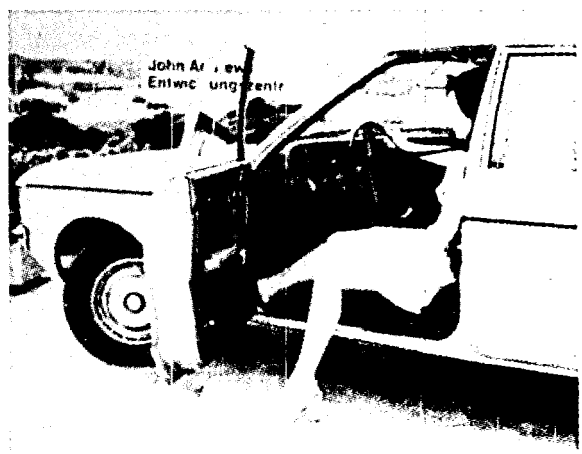


Figure 2.

Figure 3.

Braking Under Normal Driving Conditions

The front end installed sensor identifies the braking and acceleration over 0.4g and activates the inertia reel (integrated within the Pyro Reel). This automatically blocks the belt from unwinding and prevents occupant forward movement.

Front Impact and Multiple Impacts

The electronic sensor identifies impact by g-level and time duration, closes the electric circuit and activates a propellant charge within the Pyro Reel. Expanding gas in the Pyro Reel causes the belt webbing to wind up by approximately 150 mm (Figures 4 and 5), so the occupant is already fully

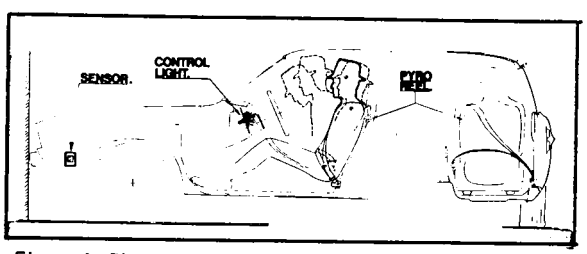


Figure 4. Phases of occupant forward movement.

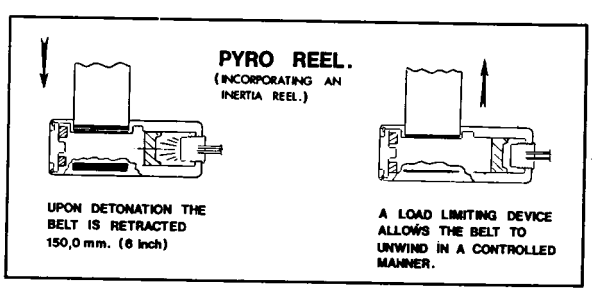


Figure 5. Pyro Reel.

tightened into his seat 18 ms after impact. After completion of gas expansion and the belt wind-up operation the Pyro Reel permits controlled chest-forward movement by unwinding of the belt under the influence of a load limiting device.

The Ford-Werke AG developed Pyro Reel incorporates all such currently known functions as Interia Reel, webbing retractor and load limiting device, within one small housing for fitment into a normal production seat back.

Through the function of the combined inertia reel together with the load limiting device a very much lower body acceleration and a low physical loading for the occupant is obtained (Figure 7).

No energy absorption by the webbing is required for the first impact, so, in the case of multiple impact, remaining energy can be absorbed by elongation of the webbing with a permanent set. The elongation of the belt webbing starts at a load figure above the working level of the load limiting device.

The sensor signal activates the unlocking of a tilting headrest from the seat back frame, the headrest follows the forward head motion for approximately 100 mm. Head whiplash and translatory head movement force the head-rest to return to its previous basic position by absorbing energy via an integrated torsion bar (Figure 6).

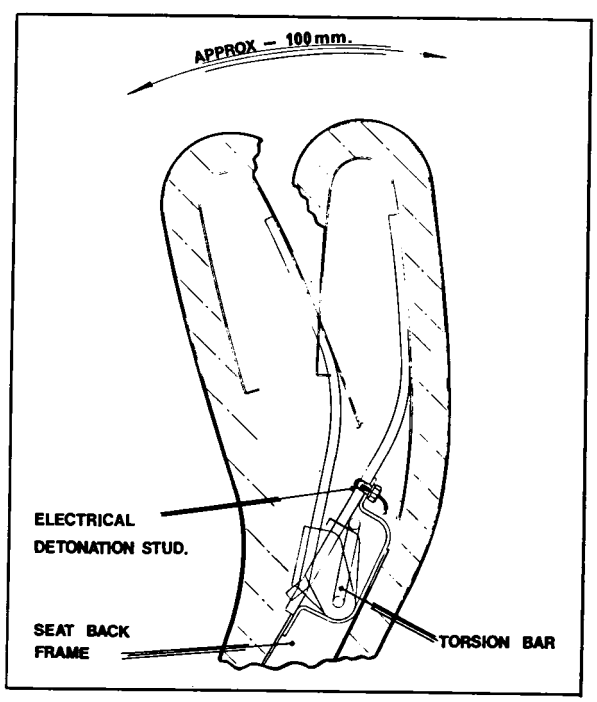


Figure 6.

The graph of a 50 kph front impact achieved with the new belt system on a series Capri 1974 vehicle demonstrates excellent results of dummy behavior during the total duration of the crash. Maximum head g-load = 31g (Figure 7).

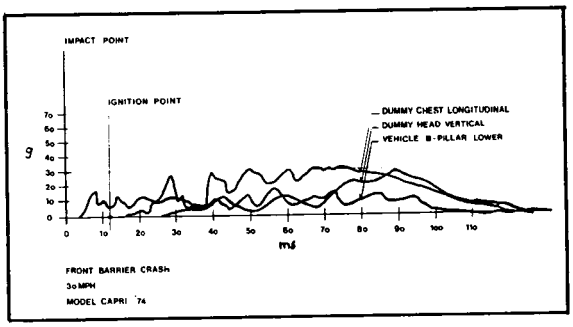


Figure 7.

A further reduction of load figures can be expected with the introduction of the 572 Dummy. The evaluation of acceleration curves clearly demonstrates the distance to the permitted HIC and SI value of 1,000 and shows the following results (Figure 8):

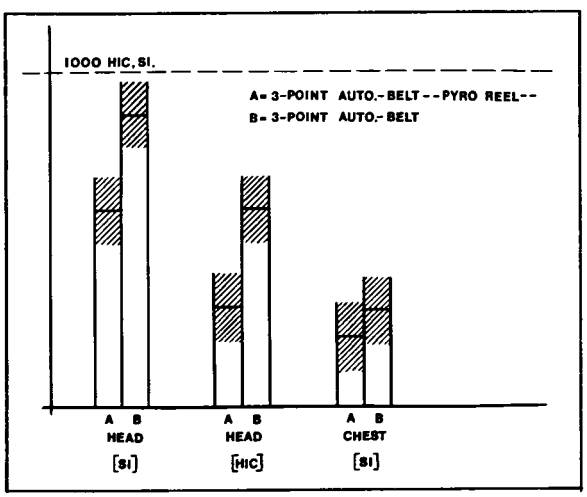


Figure 8. Normal 3-point lap and diagonal versus full passive safety belt.

Driver's Head – SI = 582
Driver's Head – HIC = 290
Driver's Chest – SI = 202

Comparative measurements to the normal 3-point lap and diagonal inertia reel belt show conclusively the advantage the Ford Full Passive Belt System with Pyro Reel has. At front and side impacts and rollover the head had no contact with the steering wheel nor the knee with the instrument panel (Figures 9a-9h).

Side Impact and Rollover

Differentiation of front and side/rollover impact is achieved by a sensor registering the amount of pendulum swing within the sensor. The sensor activates the Pyro Reel for belt wind-up operation and tightens the occupant into the seat. The Solenoid

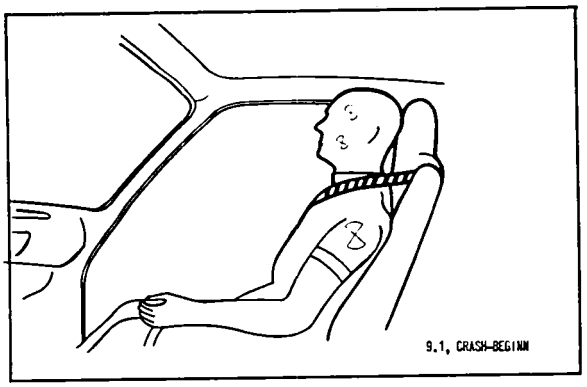


Figure 9a. At impact (side view).

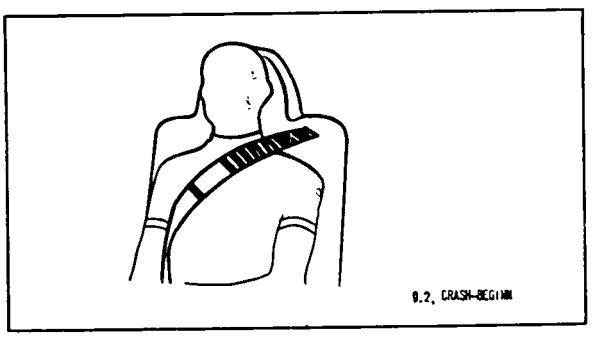


Figure 9b. At impact (frontal view).

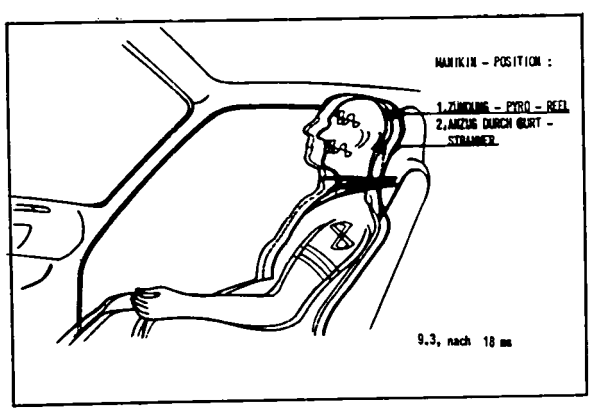


Figure 9c. 18 ms after impact.

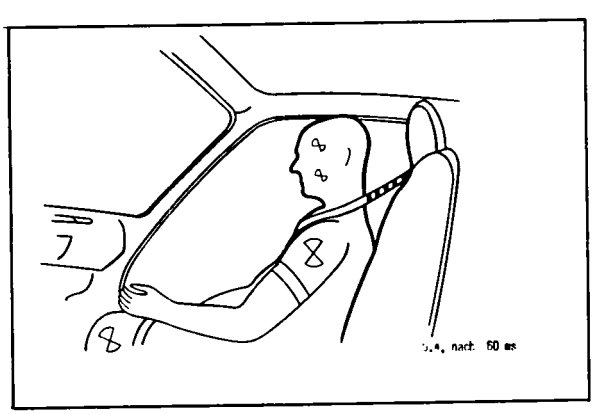


Figure 9d. 60 ms after impact (side view).

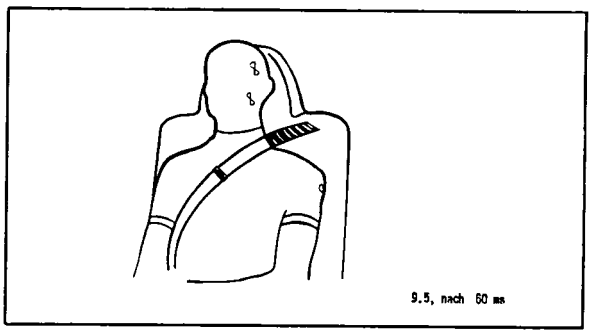


Figure 9e. 60 ms after impact (front view).

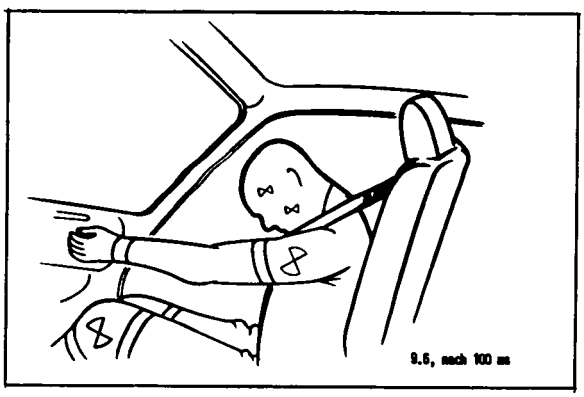


Figure 9f. 100 ms after impact.

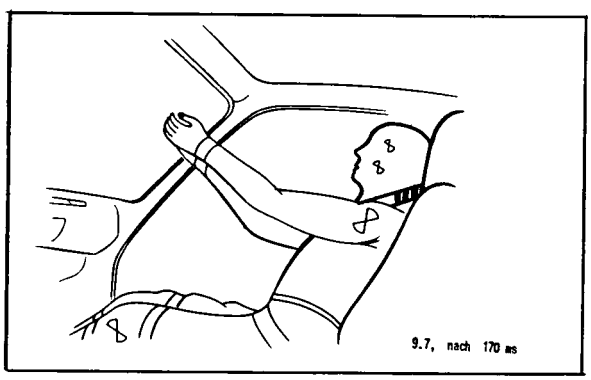


Figure 9g. 170 ms after impact (side view).

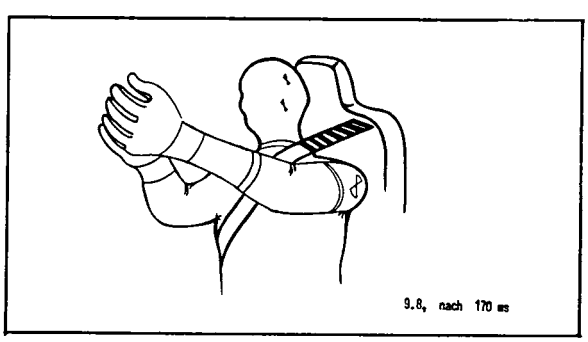


Figure 9h. 170 ms after impact (front view).

magnet, integrated in the Pyro Reel, locks the belt from unwinding over the total impact duration.

Impact energy absorption is obtained by dynamic elongation of belt webbing and within the body structure by the foam-filled door sill, door with side intrusion bar and additional padding at door trim panel, headliner and pillars, etc. Extensive testing was carried out to obtain these results.

The Emergency Release of Occupants After an Accident

A quick and easy unfastening operation of the occupant after the impact is offered for optimal occupant protection. In case of impact, the sensor causes the belt lock to become detached from the door, but to remain locked to the door sill anchorage. The belt can now be released manually by level operation (Figure 10).

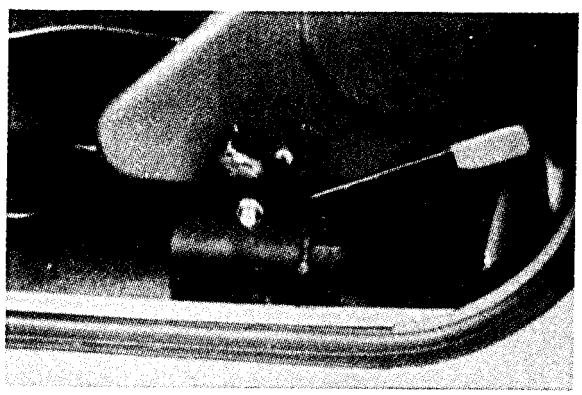


Figure 10. Emergency release in the door opening.

Furthermore, there is a second emergency release attached to the belt stem on the seat. This offers occupant rescue by unlocking belt webbing from the anchorage (i.e., pushing down the red button on top of the stem, which can only be done after an impact) (Figures 11 and 12).

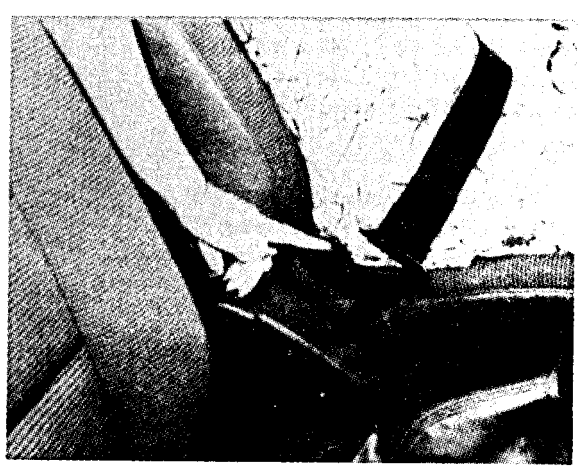


Figure 11. Emergency belt release, center of vehicle.

Figure 12. Emergency belt release, center of vehicle.

Figure 13. Optimum freedom of movement (side).

FURTHER ADVANTAGES

- No restriction of current seating package (Figures 13 and 14).
- Optimal seat belt fastening of occupants irrespective of seat position and occupant size due to integration of two belt anchorage points to the seat.
- Belt system quarantees rapid movements for passengers in any direction permitted by current seating package before the impact.
- Belt webbing release operates proportional to the chest forward movement due to the position of the Pyro Reel in area of headrest and prevents friction with occupant's clothing.
- No disturbing noise level at ignition of propellant charge with Pyro Reel due to low volume of charge.
- Additional protection of occupant during side impact based on the belt anchoring system on the door sill that prevents door ingress into car interior.
- Occupant side movement during any impact is limited by the door and the favorable position of the seat belt at the seat frame inner side.
- High level of long-term reliability for mechanical and pyrotechnical components (i.e., 99.9995% for propellant charge) and easy serviceability due to mechanically operating belt system.

Figure 14. Optimum freedom of movement (lateral).

- Sensor operation is guaranteed by separate condenser in case of loss of normal current supply.
- Protection of occupant even in case of failure of sensor, belt retractor and load-limiting device provided by mechanical operating inertia reel.

CONCLUSIONS

The Ford automatic passive belt system is a far-reaching step forward over today's 3-point belt system – it offers a safety package with high accident protection for the occupant at acceptable economic costs.

STRUCTURAL AND OCCUPANT PROTECTION SYSTEMS OF THE OPEL SAFETY VEHICLE

E. S. KIEFER
Adam Opel AG

As outlined in the previous ESV Conferences, our Opel Safety Vehicle (OSV) program is directed towards the development of a 2,000 pound passenger car concept based on high volume production technology and the utilization of conventional materials to achieve the highest level of safety performance within these parameters. Because requirements for active safety – fulfilled by our production vehicles – are more stringent than the VDA ESV specifications, a new milestone for safety would need to be achieved in fulfilling the occupant protection requirements of our experimental safety vehicle program.

Instead of building a complete new ESV prototype we modified the Kadett according to our OSV program. Choosing the Kadett, because it is a modern, light vehicle and a high volume product, would encourage response from a large marketplace. Since the basic vehicle already meets existing domestic and European legal requirements, the effort for such tests properly could be omitted.

In providing passenger protection for the entire range and variety of accidents, we concentrated on development of basic systems for the Opel Safety Vehicle. These systems, in reality, are combinations of safety components; in this regard, the safety problem eventually would be solved through protection of the occupant as a first priority while design of vehicle reaches its proper functional conformation.

These basic safety systems are targeted in terms of: front impacts, rear impacts, side impacts, and rollovers.

Our presently built vehicles are already designed to manage the kinetic energy of front impacts up to 30 mph and to maximize passenger survivability from these impact velocities when restraint systems are used, Figure 1.

Achievement of performance objectives in terms of higher front impact velocities required development of components for improved occupant protection. A critical analysis of necessary weight and cost increases became vital in terms of overall systems analysis and man/machine concepts.

To attain performance objectives for OSV front impact in terms of improved occupant protection, we placed heavy emphasis on some four components: front structure, steering systems, seats, and restraint system.

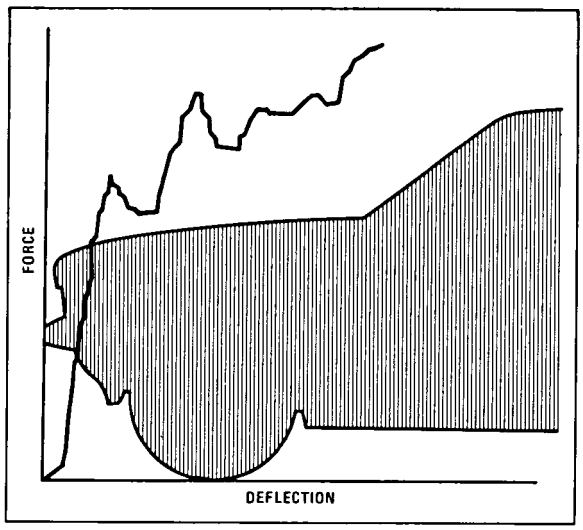


Figure 1

The continuous research work on foam filled structural elements and the step by step development from single foam filled sheet metal elements to simulated front ends showed us a path to the management of higher energies without excessive penalties of weight and cost. Some of the important results were:

- Simplified foam filled front structures showed the same deformation characteristics as single elements, used to study the influence of the various parameters.
- Special surface treatment of the elements before foam filling improved adhesion between foam and metal – a necessary condition for good deformation characteristics.
- A short distance between welding spots prevented bursting of profiles under compression.
- The load deflection curve of foam filled profiles can be modified by varying foam density, sheet metal thickness or shape of profile, Figure 2.

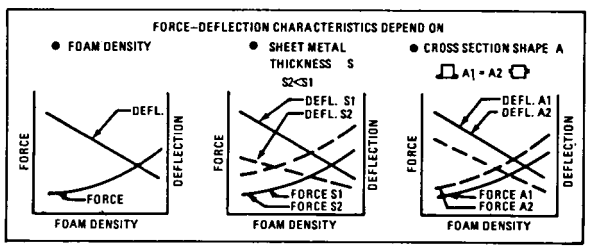


Figure 2

In summary, Opel's research work on foam filled structural elements gave proof that efficiency of such structures is influenced by surface treatment, welding

spot distance, foam density, sheet metal thickness, and profile shape. The knowledge of the interaction of these parameters will enable the designer to predetermine the deformation behavior of a given structure.

We added an upper longitudinal rail to the front structure of the basic vehicle to arrive at good load distribution, Figure 3. With this design, impact forces are transmitted through the upper and lower rails to the A-pillar and from there through door side bars and rocker panels to the rear end. The importance of splitting the front structure in a lower and upper rail is seen in the reduction of vehicle aggresiveness – providing an additional advantage for this design.

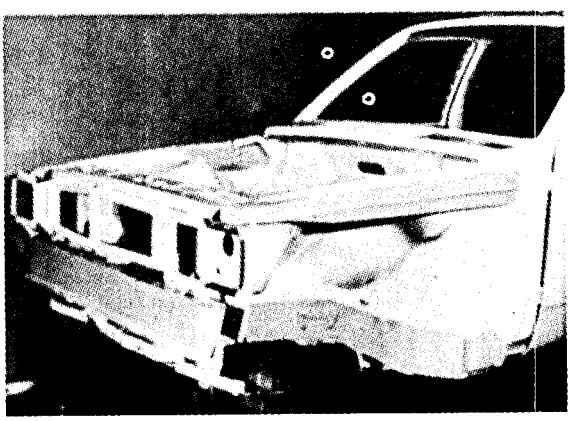
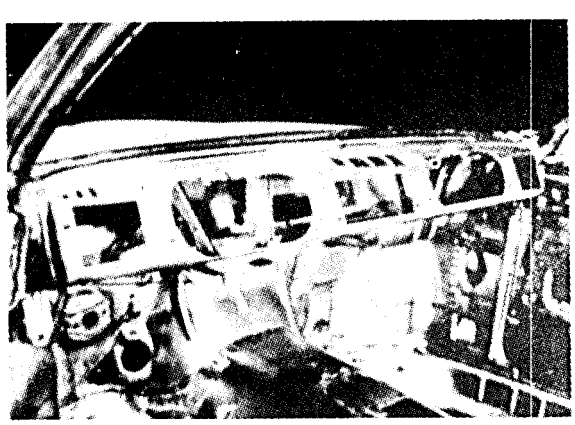


Figure 3

Opel's system for occupant protection in front impacts demonstrates effective use of steering system, restraint system, seats, and passenger compartment design.

To achieve low decelerations on impact, the dash panel and front header were carried over from production Kadett to Opel Safety Vehicle 40 without any modifications; these Kadett features had already exhibited optimum design characteristics for low decelerations on impact, Figure 4.



To the production steering system – designed for impact velocities up to 30 mph – we added a lower collapsible steering spindle to reduce the steering intrusion into the passenger compartment. An added advantageous design feature provided for mounting the steering gear housing so as to allow a slight rotation; to act as a release for forces imparting bending to the lower spindle. These design modifications enable the vehicle to meet 40 mph test requirements, Figure 5.

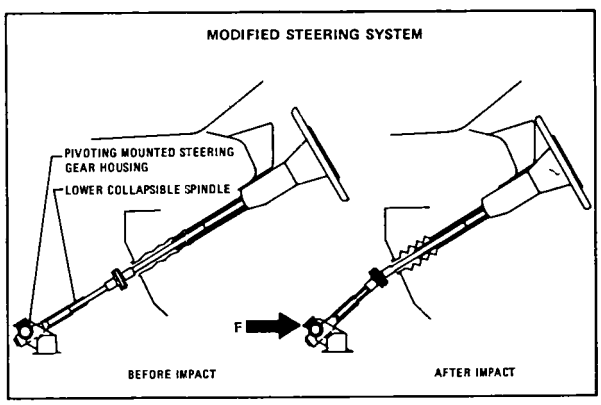


Figure 5

Opel's 3-point restraint system uses a 4 percent stretch webbing, a low-stretch webbing not permitted by ECE regulations. European legal requirements for restraint systems provide for minimum forward motion of dummy and high elongation of restraint system webbing. Significantly, we find test results to be in conflict with such *requirements* since lower injury numbers could be achieved with low-elongation webbing. It is emphasized that all OSV tests were performed with belts preloaded to approximately 50 lbs.

In our opinion, modifications of ECE requirements are also needed for regulations on steering intrusion; particularly, we would recommend that design or vehicle performance be specified instead of injury potential.

To protect vehicle occupants in rear impacts Opel concentrated its work on the rear structure, gas tank location, seats, and headrests. For production vehicles, Opel's extensive research work in the field of rear impacts resulted in design of the controlled energy absorbing rear end structure of the current Kadett model.

Added reinforcements in the OSV 40, required for low speed bumper protection, slightly increase the very low injury numbers of the base car – a necessary trade-off for bumper protection.

The location of the gas tank on the *kick up* – a position remote from the crash zone – provides protection against damage from the rear, Figure 6. In side impacts, the tank is protected by the rear wheels. For construction material, the fuel tank uses light weight high molecular Polyethylene of great pliancy. The tank is enclosed with steel bulkheads, fore and aft.

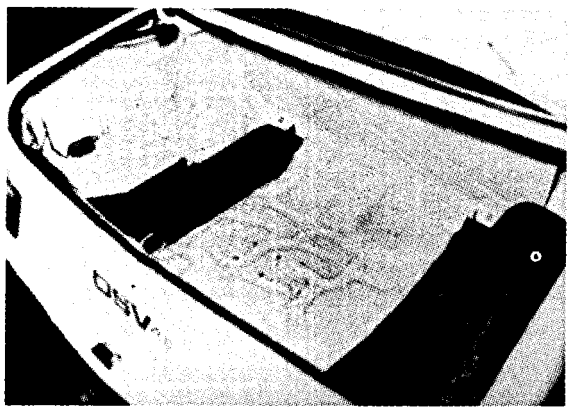


Figure 6

Further passenger protection has been provided by the *front bucket seats*; design features of these seats include energy-absorbing, roof-attached seat backs and integrated headrests, Figure 7. Mounting the seat back to the vehicle roof ensures reduction of (1) forces acting on the seat anchorages, and (2) deformation of seat back and floor pan. Opel used such strength of design to better maintain support of seat back and headrest.

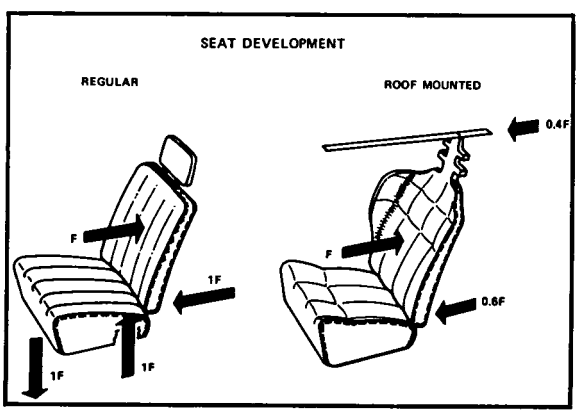


Figure 7

To reduce the risk of head injury to rear passengers, the conventional rear headrest was replaced by a shutter type headrest. This shutter design features passenger protection while minimizing obstructions to driver's rear field of view, Figure 8.

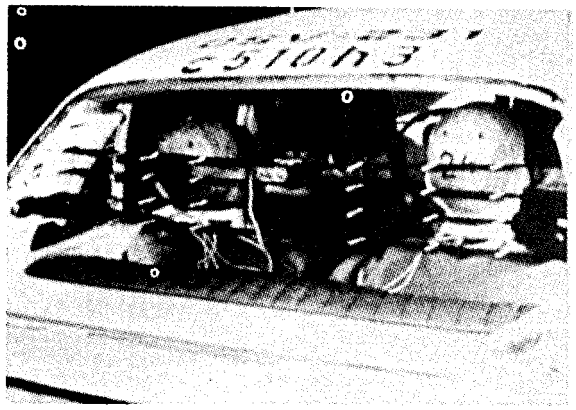


Figure 8

Opel's research on danger to occupants from side impact provided for development of the following components: side structure, shoulder support and interior padding. Such developments confidently look toward the improvement of occupant protection.

In side impacts – car-to-car, as well as side pole – intrusion characteristics were tailored to the interior survivability space. To control the penetration of the passenger compartment, we filled the door side bars and rocker with polyurethane foam. Bending tests with foam filled elements showed that the bending load capability for local pole type loading can be increased drastically, Figure 9.

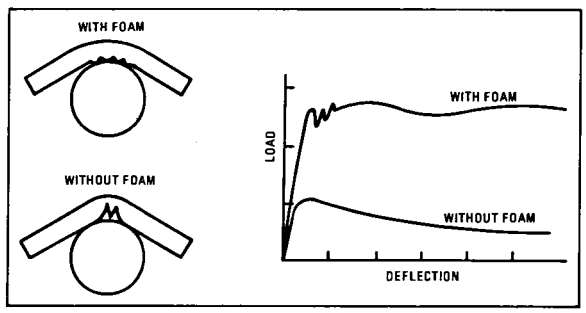


Figure 9

In addition to door bars, we used the hinge-door-lock system of the basic Kadett for providing sufficient strength to withstand consequences of OSV side impact test, Figure 10.

The high location of door beams ensured prevention of override for the bullet-car in angular impacts; such a design serves to improve fore-and-aft strength in front and rear impacts, Figure 11.

For additional improved occupant protection in side impacts, shoulder supports have been incorporated in the front seats in Opel's effort to prevent head knocking and impact between front passengers, Figure 12. Then, to provide for even better occupant protection, we padded interior

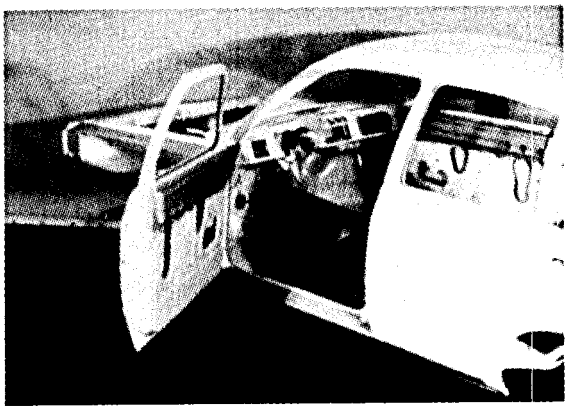


Figure 10

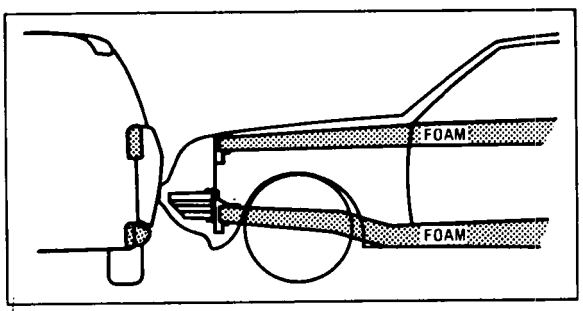


Figure 11

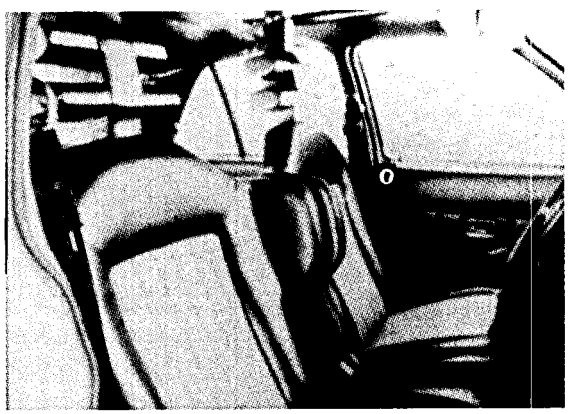


Figure 12

pillars, roof rails, and doors with the best commercially available energy absorbing foam. In a series of laboratory tests on pillar type sections, we found that a foam padding reduced g-levels as much as 20 percent, Figure 13.

The design of our current Kadett pillars and roof rails already complies with OSV test procedures for rollovers; in addition, the seats of OSV 40 contribute to retention of survival space in the vehicle. Retention inside the vehicle for the occupant in the rollover test was a test condition; statistics have

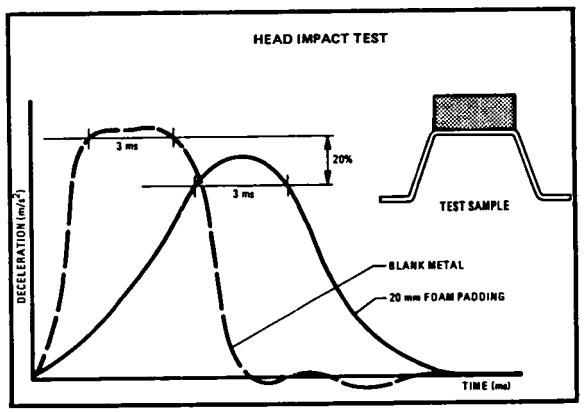


Figure 13

indicated that occupants survive rollovers with relatively low injuries as long as they remain in the vehicle.

When considering any specific future proposal in the field of legal requirements, Opel would prefer a rollover test with fixed dolly velocity, Figure 14. Test repeatability as well as realistic variation in results for different type and weight vehicles, however, make doubtful any rollover compliance procedure.

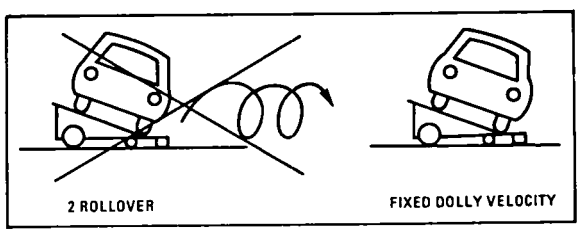


Figure 14

Although the low speed bumper system does not contribute to passenger protection, Opel has developed alternate systems for minimizing low speed damage to vehicles and offering pedestrian protection in these very low speed impacts. In this regard, we consider the presently installed hydraulic bumper system on our production vehicles to be a good approach for low speed impact.

For improved pedestrian protection, Opel developed a "soft nose" – equipping the OSV 40 with soft Polyurethane front and rear fascia, supported by a specially developed honeycomb insert, Figure 15. The smooth rounded shape of the front and rear fascia will cause an impacted pedestrian to roll on the vehicle's hood thus substantially reducing the risk of injury. Furthermore, the pliant fascia diminish the specific impact on pedestrian body parts which first come into contact with the vehicle.

In presenting safety systems as vehicle installation proposals, we do not represent our particular choice,

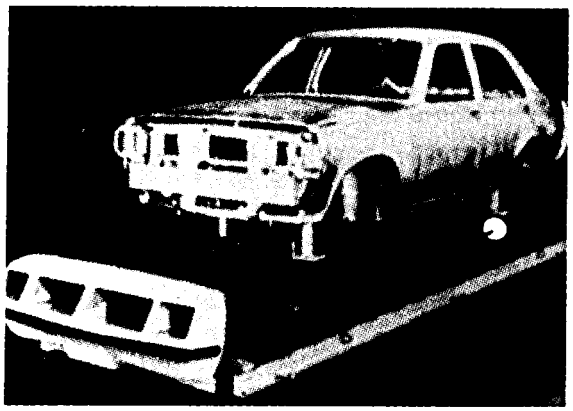


Figure 15

Figure 16, to be a final design for high volume production. Furthermore, we make the following statements in a conservative approach to achieving objectives for safety vehicles:

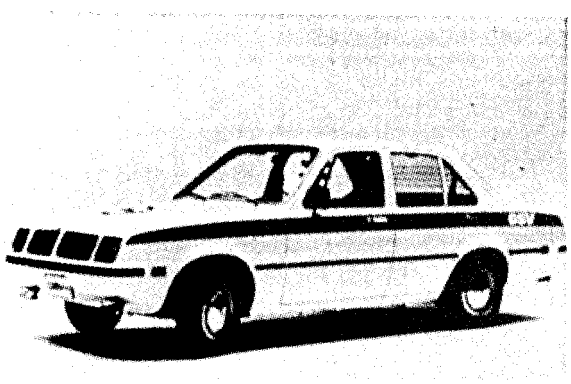


Figure 16

- From our point of view, we have indicated that, sometimes, legal requirements and conclusions from actual test results contradict each other. After careful review, we recommend that legal requirements be updated to comply with precise analysis of test results, and that protection of occupant rather than design of vehicle be specified.
- In order to better judge in real life situations the benefit of our endeavors, more *biomechanical data* and more information on *pre-accident conditions* is required.
- In considering the current state of the economy, and the energy and material resource situation, the objectives of this safety program on structural and occupant protection systems must be reevaluated. Our OSV 40 is adding too much weight; further work is needed in this area.
- It is noted that the automobile industry has shown the solutions and consequences of vehicles *safe* up to 30, 40 or 50 mph barrier equivalent speeds. Now, governments have a larger fund of technical expertise to use in promulgating future regulations.

The selection of the 40 mph barrier capability for our Opel Safety Vehicle has been based mainly on actual accident data; these data show that a 40 mph barrier capability would prevent 74 percent of the fatalities in front impacts. Such vital information is based on data published by HUK (German insurance companies association) and NHTSA.

SOME CONSIDERATIONS OF BODY STRUCTURE CRUSHABILITY IN RELATION TO IMPACT SPEED

DR. L. ROSTI-ROSSINI
Alfa Romeo

INTRODUCTION

The ESV prototypes have given us an idea of dimension, weight, sophistication and cost – those factors needed to satisfy the safety specifications initially proposed. If we keep in mind that such vehicles are the result of optimization, we must believe that the macroscopical differences between them and the vehicles already in use could hardly be reduced.

Since we know that such differences are mainly consequences of the proposed impact test speeds, we have been concerned with identifying the theoretical foundations of such phenomena and, more precisely:

1. Mathematical relation between impact speed increase and dynamic consequences on vehicle structure and passengers.
2. Maximum impact speed (in various collision directions) challenging present production cars, while providing, at the same time, sufficient occupant protection.

The purpose of this paper is to provide some elements for a numerical answer to the proposed questions – on the basis of experimental results and theoretical estimations – and to provide an analysis of the total behavior of body structures, dealing only briefly with restraints (information already presented in earlier papers).

In particular, for the ESV, we have exploited a simplified theoretical method in estimating required variations in dimensions and weight for a car with specified structural performance, while varying the required impact speeds. As a consequence, one can deduce that the minimum crush distance required to guarantee occupant safety during collision increases as the square of impact speed, while total vehicle weight increases even more rapidly. Moreover, applying the proposed calculation method, the structural designer can evaluate the possible advantages in weight and dimensions coming from the adoption of high energy-absorbing elements that are integrated with conventional bodies.

STRUCTURAL CONSIDERATIONS

Considering the structural behavior of a vehicle in a collision from the restrained passenger point of view, structural effectiveness can be substantially judged on the basis of "kinematic" and "energetic" aspects. "Kinematic" aspects are concerned with the influence of the compartment's law of motion on the passenger's law of motion through the connection set up by restraints. "Energetic" aspects are concerned with energy-absorping capacities and strengths of the various structural elements. After any collision, the vehicle presents macroscopically deformed areas (where nearly the entire impact energy is absorbed) and substantially undeformed areas.

To keep the passenger compartment an undeformed area we must:

- Eliminate the danger of passenger injury due to metallic intrusion.
- Furnish the maximum possible internal distance for the motion of the restrained passengers.
- Retain door functionability to allow passenger escape after collision.

Therefore, to make the passenger compartment an undeformed area, the following conditions must apply:

1. The structure between passenger compartment and impact areas must be "more deformable" than the compartment structure ("differential" structure).
2. The structure between the passenger compartment and the impact areas must absorb –the whole impact energy of the passenger compartment and the connected masses (i.e., the structure must have sufficient energy-absorbing capacity).

The "energetic" and "kinematic" aspects are physically mixed and reciprocally conditioned. However, it is possible to differentiate function and effect since the "energetic" aspect appears less binding than the "kinematic."

KINEMATICS

At the ESV conferences held in Washington and Kyoto, we analyzed, among other things, the kinematic aspects of frontal crashes. At the Washington conference, the relation between experimental compartment motion and simulated (but simplified) restrained passenger motion was examined. The experimental "compartment acceleration" function, which coincides with the acceleration function recorded on the restraint anchorages, was developed into a Fourier series.

It was demonstrated that the average value and the first harmonic of the acceleration function expansion had practical effects only on the motion of a simplified dummy. This phenomenon, associated with the filtering effect of the restraint dummy, is always

true if the function, "compartment acceleration," is sufficiently compact.

At the Kyoto conference we presented a paper regarding the simulated dummy motion inside a medium sized compartment, utilizing a simplified acceleration law. This law resulted from superimposing a constant value and a co-sinusoid having a period equal to the crash duration. In this case, the ratio between the accelertation constant value and the first harmonic amplitude was taken as being equal to 1.

The diagram used for compartment acceleration is shown in Figure 1.

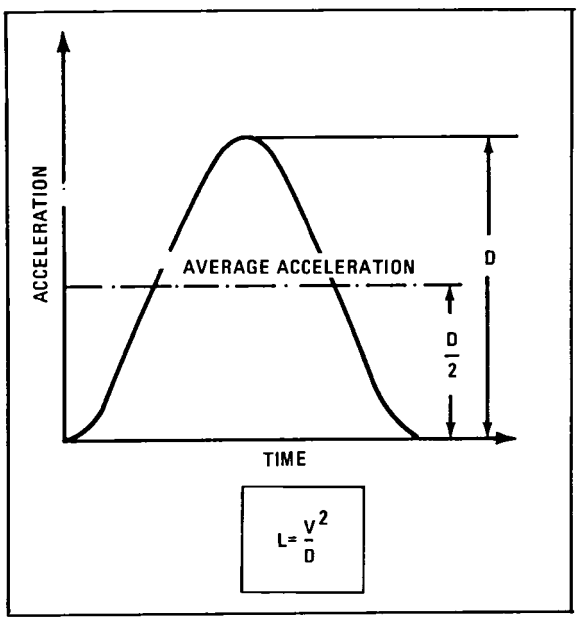


Figure 1

The type of acceleration diagram shown in Figure 1 gives a particularly simple relation between impact velocity (V) against the barrier; maximum acceleration amplitude (D); and compartment crush distance (L). Equation 1 presents the relation between crush distance (L) and impact velocity (V), for various acceleration amplitudes (D). (See Figure 2.)

We had pointed out at the Kyoto conference that, for the purpose of maintaining an acceleration level of less than 60 g on the dummy torso, the compartment acceleration (D) should not exceed 58 g. In this test, the simulated restraints had critical damping and no play, meaning the system was ideal.

In practical cases the admissible level of passenger compartment acceleration (D) must be reduced – if we take into consideration the characteristics of restraints actually on the market and their adjustment to the different passenger sizes – with a consequent reduction of internal space available for their operation.

On the basis of experiments carried out in our laboratories and simulated computer tests, we think that a vehicle having current body structures must comply with crush distances not less than those furnished by the curve D = 40 g as shown in Figure 2, with the purpose of maintaining the occupant torso acceleration level inferior to 60 g. (Restraints, in this case, should be the most perfect manufactured today.) We would like to also point out that, for general application, Equation 1 could be slightly modified by introducing the factor K:

$$L = K(V^2/D) \qquad (2)$$

In cases of constant acceleration, K = 0.5; and the value of K, relative to every type of vehicle, can be estimated by Fourier analysis of the experimental compartment acceleration diagram.

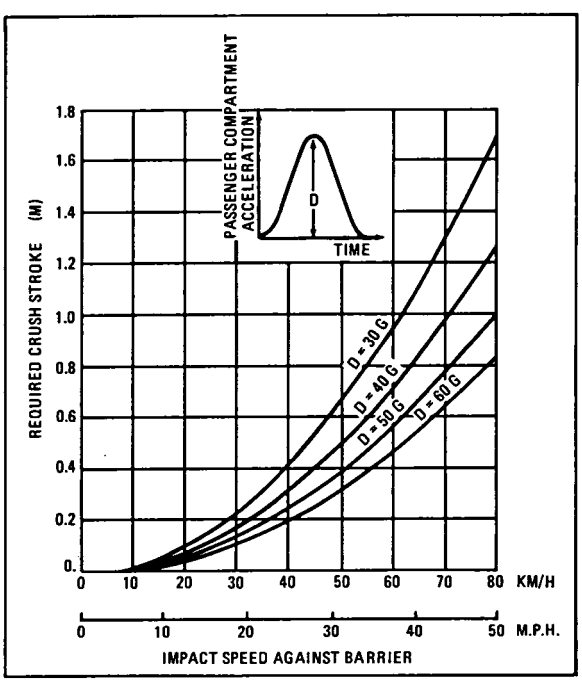


Figure 2

As a result of the "kinematic" structural behavior analysis, we can concentrate on the following conclusions:

1. Only the average value and the first harmonic of the compartment acceleration equation are important in relation to passenger restrained protection.
2. To guarantee a tolerable acceleration on a passenger's torso, it is necessary (even if not optimium) that the vehicle has enough crushable length.

3. The crushable length must increase with the square of impact speed.
4. The crushable length is, within certain limits, dependent on the restraint system used.

ENERGETIC ASPECTS

The energetic aspect of structural behavior during the crash can be analyzed under two conditions:

1. The ratio between the deformability of the passenger compartment and the structure between the compartment itself and the impact area must be such as to keep the compartment undeformed until the whole structure (between the passenger compartment and the impact areas) is deformed. This condition is necessary (but not sufficient) to guarantee compartment structure integrity at any impact speed. (In fact, in order to keep the compartment undeformed, condition 2 must be met.)
2. The structure between the compartment and the impact area must have an energy-absorbing capability which is sufficient to meet the maximum required impact speed.

Condition 1 is not easily expressed in synthetic and general terms; Condition 2, on the contrary, can be represented with a simplified mathematical model.

Let us consider a vehicle which has undergone a generic fixed barrier crash (at a velocity V) and, therefore, is partially deformed. In order to schematize the behavior of the various parts of the vehicle, let us define:

M_o = mass of the parts, without a notable connection to the passenger compartment, which have dispersed their energy directly onto the barrier, without producing noticeably serious structural deformations (i.e., in a front crash against a barrier for a front engine car, M_o includes the engine and the gear box).

M_r = mass of the vehicle practically undeformed, less M_o, which has dispersed its energy on the structural parts which are between M_r and the barrier. (It includes the mass of the passenger compartment which has not been deformed, with all its accessories, and other parts which can vary according to the type of crash.)

M_a = mass of the parts which are seriously deformed, which are between M_r and the barrier.

M_p = mass of the passengers.

The crash energy balance gives the following relations:

$$\begin{cases} \frac{1}{2}M_o V^2 = L(M_o) & (3) \\ \frac{1}{2}(M_a + M_r + M_p)V^2 = L(M_a) + LR & (4) \end{cases}$$

where:

$L(M_o)$ = deformation work of mass M_o
$L(M_a)$ = deformation work of mass M_a
LR = work being absorbed by the restraint systems.

We are particularly interested in developing Equation 4, which contains the passenger compartment, by means of M_r. The work which is absorbed by the restraint systems is only a part of the energy of passengers ($\frac{1}{2}M_p V^2$) and therefore, for the sake of simplicity, it can be neglected (LR = 0). The deformation work of the mass M_a can be evaluated from the energy-absorbing capacity of the various deformed parts. If the material, by which the mass M_a is formed, has an absorbing limit energy per unit of mass (call it C), the maximum energy which can be theoretically absorbed by the mass M_a is:

$$L_{lim}(M_a) = CM_a \qquad (5)$$

In our case, knowing the energy actually absorbed by M_a – and therefore knowing the deformation work $L(M_a)$ – we can define a "structural efficiency" given by the following ratio:

$$\eta = L(M_a)/L_{lim}(M_a) = L(M_a)/CM_a \qquad (6)$$

Under such assumptions we can write in the following way:

$$\frac{1}{2}(M_a + M_r + M_p)V^2 = \eta CM_a \qquad (7)$$

$$M_a = (M_r + M_p)[V^2/(2\eta C - V^2)] \qquad (8)$$

Equation 8, although extremely simplified, is particularly interesting: it contains the impact speed, the material characteristic C (energy-absorbing limit per mass unit), the structural efficiency η, and the masses M_a, M_r, and M_p. From Equation 8 we see that:

1. Impact speed, efficiency and employed materials being equal, it is convenient that M_r, the mass that does not absorb energy, be as light as possible.
2. M_r and material being equal, it is theoretically possible to increase the speed V at which mass M_a becomes deformed, modifying the shape of the structures in order to increase the structural efficiency.

3. There exists a critical speed, V_{cr}, not dependent on M_r and M_p, at which the necessary absorbing mass tends toward infinity:

$$V_{cr} = \sqrt{2\eta C} \qquad (9)$$

It is evident that possible impact speeds will in any case be very much lower than V_{cr}.

4. For every car and every impact direction, there exists a speed V_o, beyond which the passenger compartment begins to become deformed. If we experimentally know that velocity and the characteristic masses, it is possible to evaluate the structural efficiency η_o of the considered car in the given impact direction:

$$\eta_o = [(M_a + M_r + M_p)/2CM_a]V_o^2 \qquad (10)$$

In order to better evaluate the meaning and the possible utilization of these parameters, we will show a series of quantitative data, obtained by experimental or analytical methods.

Energy-Absorbing Capacity

Table 1 shows the comparison between the results of static tests on metal body test pieces subjected to traction and tubes subjected to compression to the static and dynamic tests on parts of the car body. In the last column there is the structural efficiency, calculated as the ratio of the energy absorbed by a given car body part and the energy absorbed by a 5 mm tractive test piece at the position of test piece failure. It is clear that the actual efficiency (defined from the maximum energy-absorbing capacity) will be less or equal to the efficiency shown in the table.

For practical purposes, we can use the calculated values η (which in the preceding formulas are always accompanied by the value C). In this case, C is evaluated for the tractive test piece. A description of the tested parts follows:

A = part of a tractive test piece taken from the area of failure (length: 5 mm, and initial section: 1 mm by 20 mm).

B = complete test piece, rectangular: 80 mm by 20 mm; section: 1 mm by 20 mm, used for test A.

C = smooth tube of 76 mm diameter (wall thickness: 1.5 mm), used for axial compression test.

D = front structure of a production car used for the barrier impact test.

E = production car side-frame with doors, used for static crash test.

F = production car rear end structure, used for test with moving barrier.

Table 1
ENERGY-ABSORBING TESTS ON SHEET METAL AND VEHICLE BODIES CONSTRUCTED WITH THE SAME MATERIAL

Part	Kind of Test	Weight (kg_f)	Absorbed Energy (kg_f m)	Specific Energy (m)	η
A	static	0.000786	3.48	4,427	1.0
B	static	0.012576	19.5	1,550	0.350
C	static	2.8	4,300	1,545	0.349
D	dynamic	65	6,800	104	0.023
E	static	38	1,400	36	0.008
F	dynamic	49	3,900	80	0.018

Critical Speed

From Equation 9, we evaluate the critical speed for the structural efficiency for the same cases in Table 2 as in Table 1.

Table 2
CRITICAL SPEED, V_{cr}

Case	η	V_{cr} (m/s)	V_{cr} (km/h)	Note
A	1	294	1,058	*
B	0.35	174	626	
C	0.349	174	626	
D	0.023	44	158	
E	0.008	26	93	**
F	0.018	39	140	

*For the energy-absorbing capacity per mass unit, the value: $C = 4{,}427 \times 9.81 = 43{,}428\ m^2/s^2$ has been assumed.
**The side-frame has the lowest critical speed.

Relation Between the Impact Velocity and the Deformed Mass

In Table 3, we have masses and efficiencies for a series of experimental tests with various kinds of structures. A description of the tests follows:

1 = production car – front crash against fixed barrier.

2 = production car – rear crash (moving barrier).

3 = rigid sled with collapsible tubes – front crash against fixed barrier.

4 = production car, modified with high absorbing structures – front crash against fixed barrier.

5 = reinforced production car – side crash.

Table 3
CRASH TEST AGAINST RIGID BARRIER OR EQUIVALENT

Test	V_o (m/s)	M_r (kg)	M_p (kg)	M_a (kg)	η_o	Note
1	12.0	725	140	65	0.023	
2	8.5	975	–	49	0.018	*
3	12.5	1,255	–	6.5	0.349	
4	14.5	750	140	70	0.033	
5	5.0	990	–	38	0.008	**

*V_o is the speed energetically equivalent in the crash of the vehicle against the fixed barrier.

**V_o is the energetically equivalent speed; data are derived by a static test.

In the following diagram (Figure 3), the ratio: $M_a/(M_r + M_p)$ for various values of the structural efficiency, is plotted against the impact speed (V) against a fixed barrier.

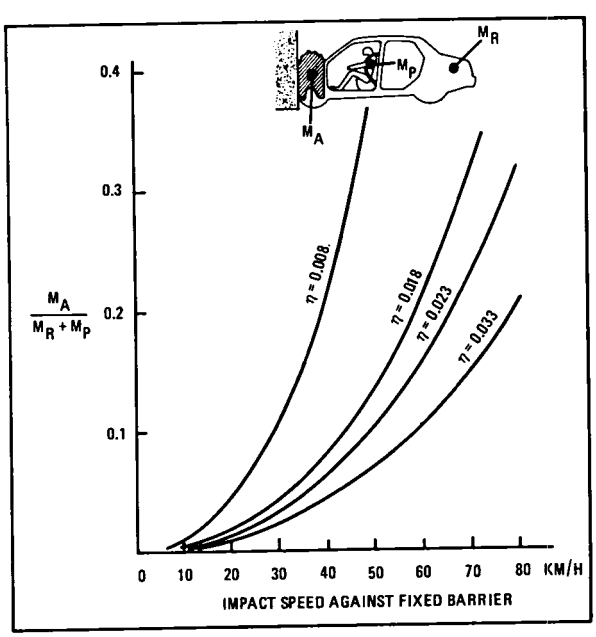


Figure 3

Vehicle Weight Increase versus Barrier Impact Speed

On the basis of experimental data (Figure 3), it is possible to estimate, for any impact speed, the value of the ratio between the absorbing mass M_a and the undeformed mass M_r for a given car and a given impact direction. For a given car, it is possible to estimate the weight increase necessary to absorb the energy of the various prescribed crashes (without deforming the passenger compartment) by assuming the various impact speeds to be proportionate. For example, evaluations made under the hypothesis that the relation between the equivalent fixed barrier crash speed are:

V side crash = 40% V front crash
V rear crash = 50% front crash.

These ratios can be modified as to the various test specifications without compromising the adopted evaluation method.

In our case, we calculated the various ratios: $M_a/(M_r + M_p)$ for a vehicle with $V_o = 5$ m/s in the side crash; $V_o = 8.5$ m/s in the rear crash; and $V_o = 12$ m/s in the front crash – supposing in a first approximation that the structural efficiency does not vary with the velocity increase.

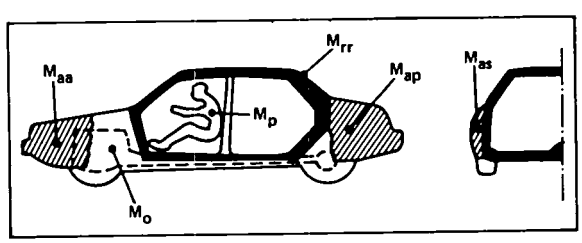
From the diagrams of Figure 3, we get the following ratios for every crash speed:

$$M_a/(M_r + M_p) = \alpha \quad \text{front crash} \tag{11}$$

$$M_a/(M_r + M_p) = \beta \quad \text{side crash} \tag{12}$$

$$M_a/(M_r + M_p) = \gamma \quad \text{rear crash} \tag{13}$$

We now substitute the parameters M_a, M_r and M_p with their actual values in the various crashes and determine the new total mass M of the vehicle for every value of impact speed.



$$M = M_{aa} + M_{ap} + 2M_{as} + M_o + M_{rr} \tag{14}$$

For a front engine car we have:

$$M_{aa}/(M - M_o - M_{aa} + M_p) = \alpha \tag{15}$$

$$M_{as}/(M - M_{as} + M_p) = \beta \tag{16}$$

$$M_{ap}/(M - M_{ap} + M_p) = \gamma \tag{17}$$

Hence:

$$M_{aa} = [a/(1+a)](M + M_p - M_o) \quad (18)$$

$$M_{as} = [\beta/(1+\beta)](M + M_p) \quad (19)$$

$$M_{ap} = [\gamma/(1+\gamma)](M + M_p) \quad (20)$$

Let us take:

$$\epsilon = \frac{a}{1+a} + \frac{\beta}{1+\beta} + \frac{\gamma}{1+\gamma} \quad (21)$$

Then:

$$M = \frac{M_{rr} + \epsilon M_p + M_o[1/(1+a)]}{1-\epsilon} \quad (22)$$

Equation 22 gives the total mass M of the vehicle. Once we know the various ratios: a, β, and γ at the considered speeds, M_{rr} and M_o are constant.

When $\epsilon = 1$, the total mass (M) tends toward infinity.

Of course, for every velocity, Equations 18, 19 and 20 must be verified.

In Figure 4, we have the ratio between the total mass M and the total mass $\bar{M}$ of the same kind of vehicle, without any modification, plotted against the chosen set of impact speeds (solid line). The calculation data is:

$\bar{M}$ = 1,030 kg
M_p = 140 kg
M_o = 233 kg
M_{rr} = 607 kg

The vehicle has a front engine.

We may note that, for a front impact speed up to 43.2 km/h (and also for an equivalent impact speed of 17.3 km/h for side crash and 21.6 km/h for rear crash), the considered vehicle does not require additional masses.

In Figure 5, the ratios between the required front, side, and rear deformable masses, and those which are available in the actual car, are plotted – always – against impact speeds. We note that, as the impact speed increases, the heaviest weight increase is required for the front and side parts of the vehicle.

Utilizing the same procedure, we present a similar evaluation for a reinforced vehicle which has a structural efficiency of the front end increased by 40%.

For the sake of simplicity, we make the assumption that the structural efficiency of the side and rear parts could have the same 40% increase (Figure 4 – dotted line). The data for such a car (always with a front engine) is:

$\bar{M}^*$ = 1,060 kg (for reinforcement of the original car with $\bar{M}$ = 1,030 kg)
M_p = 140 kg
M_o = 233 kg
M_{rr} = 632 kg

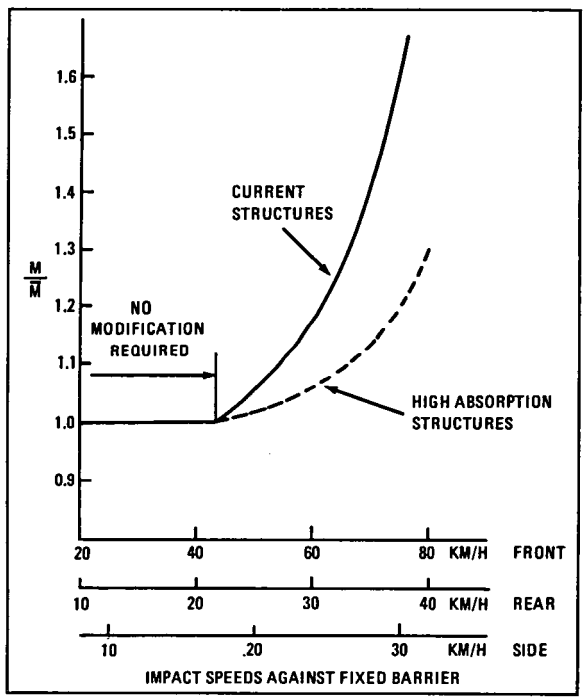


Figure 4

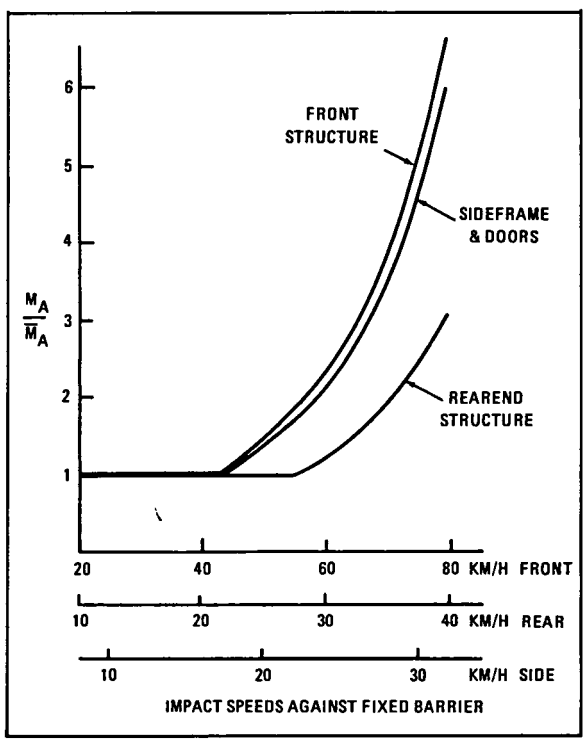


Figure 5

In Figure 6, the ratios between the required weights of the deformable parts and the originally available weights are plotted. Referring to the two curves of Figure 4, we may say that the solid line represents the situation of a normal front engine production car, modified so as to increase dimension and weight, but without appreciable variations of efficiency.

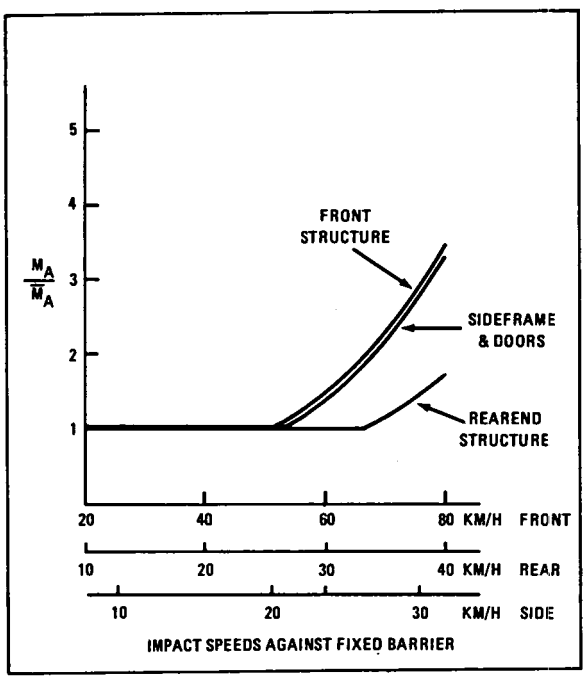


Figure 6

The dotted line, on the contrary, represents an ideal limit (for the same vehicle), totally modified in the absorbing structure and partially modified in the nonabsorbing parts, in order to obtain a major structural efficiency in the various impacts.

CONCLUSIONS

The analysis of the "energetic" behavior of vehicle structures, carried out with the proposed calculation method (simplified, but of general significance) allows the following considerations:

1. The design of current structures, during normal use, does not provide high absorption capacity in a collision. Current structures have a low collapse efficiency, as indicated by our studies of their elements.
2. However, current structures are able to withstand collisions at moderate velocities without dangerous compartment deformation.
3. To increase the capability of given vehicles to absorb higher impact speeds without incurring passenger injuries, the total vehicle mass must be increased in relation to the square – or even more – of the determined, critical impact speed.
4. Mass increase is affected strongly by structural efficiency; therefore, the effect of structural efficiency is more critical at higher impact speeds.
5. In trade-offs between impact speeds and dangerous deformation of the passenger compartment, increase the absorbing mass without trying to change the structural efficiency.

In all cases, structural validity is subordinate in satisfying kinematic conditions relating to passenger safety and protection, otherwise, increased absorbing capacity would be totally useless to the survival of the occupants.

A RESEARCH STUDY FOR AN ENERGY ABSORBING SLIDING SEAT

GAETANO DANESE
Director
Rome Motor Vehicles Testing Center

FOREWORD

With conventional arrangements the occupants of a car are restrained by safety belts anchored to the bodywork. In the event of a collision the occupant dissipates his energy by causing the extension of belt webbings (Figure 1); in other words, he moves relative to the seat that remains fixed to the car body.

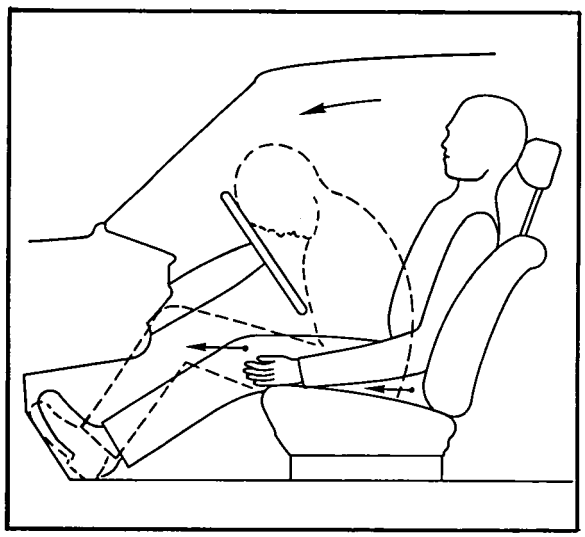


Figure 1

This movement, and the thorax forward rotation in particular, occur in a confined environment where there are obstacles present, like the instrument panel and the steering wheel.

Occupant impact against such obstacles could be prevented by using more rigid belts but, in this case, the necessary motion for an adequate energy dissipation would no longer be available and, consequently, the loads on the occupant would increase.

INVESTIGATIONS AND TESTS

Keeping in mind the above considerations it was then thought of *restraining the occupant directly onto the seat* by belts more rigid than those currently in use and of *having the seat move relative to the car body instead of being fixed* (Figure 2). This gives us the benefit of controlling occupant motions and of exploiting more rationally the available compartment space.

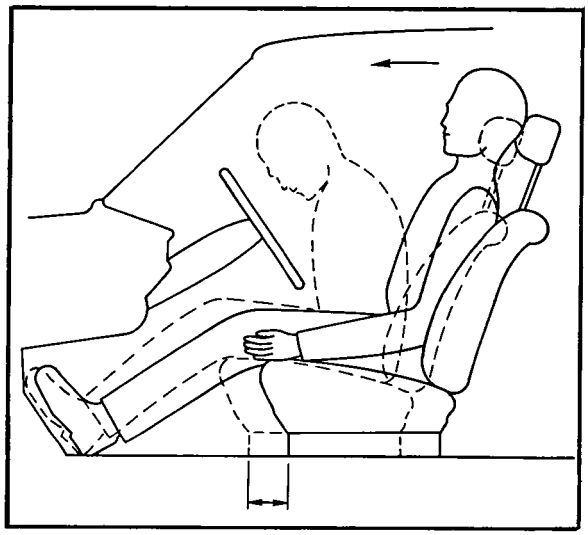


Figure 2

The seat includes a lower frame – anchored to body floor – and an upper frame movable relative to the former. Several motions are possible (Figures 3, 4 and 5):

- Longitudinal
- Rotation in the vertical plane, and
- Lateral

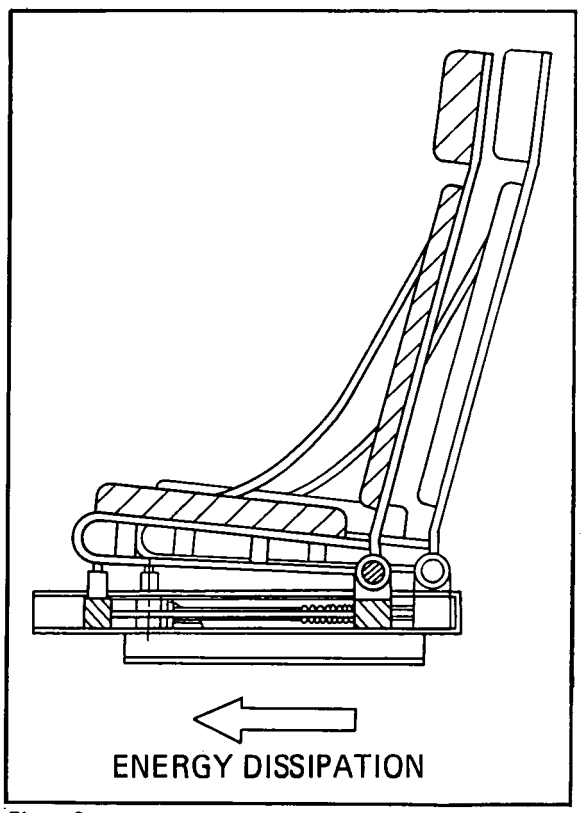


Figure 3

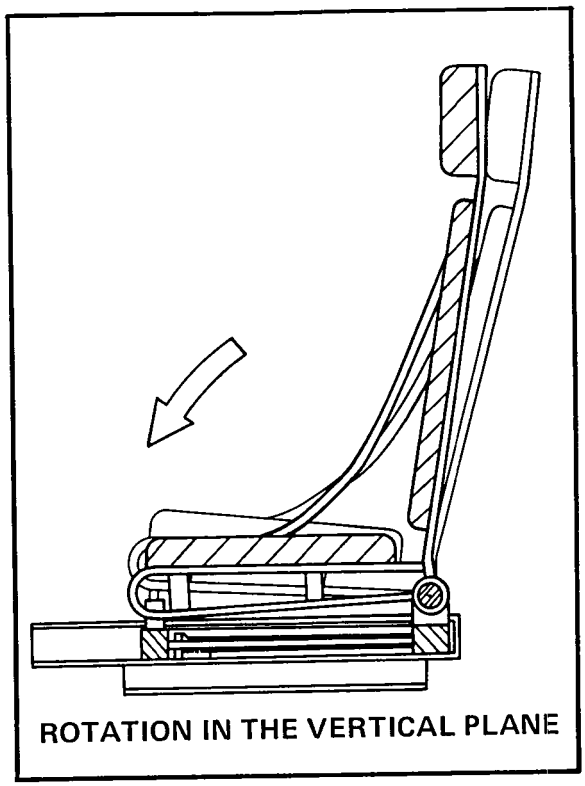


Figure 4

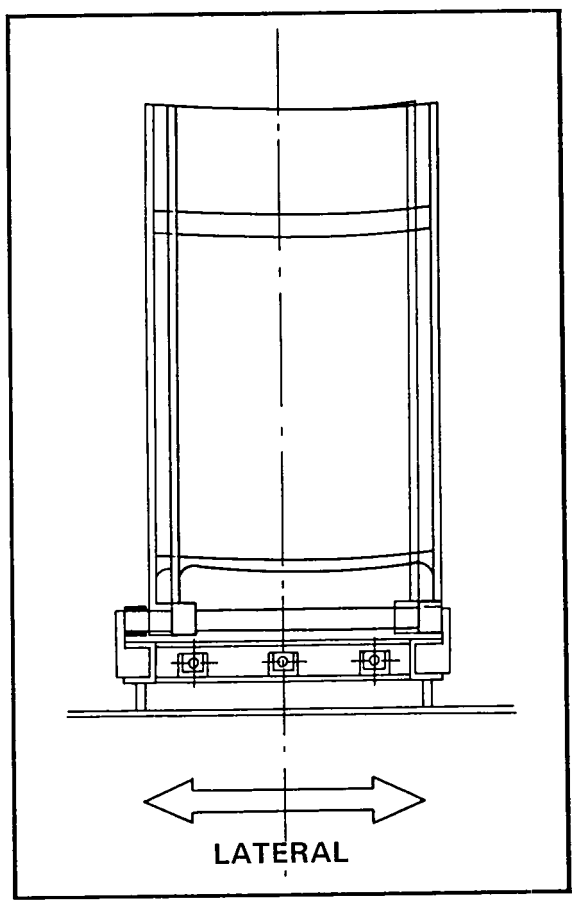


Figure 5

Such motions are counteracted by the consequent reactive forces which are generated in dampers and permit the dissipation of occupant kinetic energy during collisions.

To provide such damping effect, and reduce the consequences of frontal collisions in particular, it will obviously be necessary to have devices in which plastic deformation prevails over elastic. Because of this concept, an early system based on rubber springs (intended to provide some damping through friction, but subsequently proven to be inadequate) was waived and the investigation was concentrated on energy dissipation by fluid dampers or deformable tubes collapsing under axial loads (Figures 6 and 7). (Final choice later fell on the collapsible tubes as these, compared with the fluid dampers, showed more reliable performance with practically constant reactions under axial load having a predominantly plastic character.)

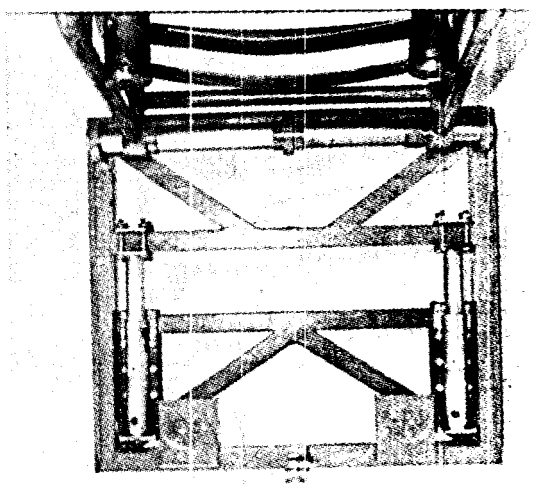


Figure 6

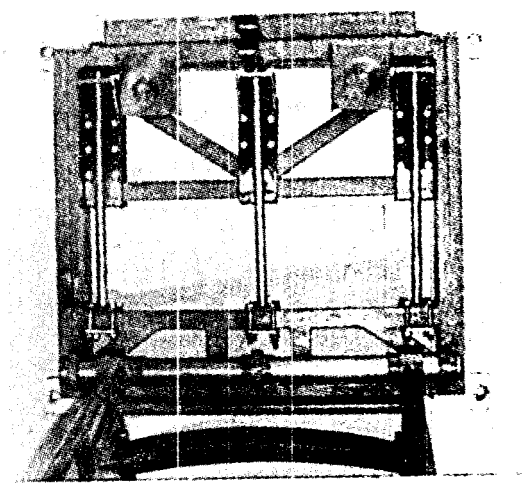


Figure 7

During recent tests conducted in the Fiat Safety Laboratory, the behavior of a dummy on both the standard and sliding seats was investigated.

Sliding Seat

Belt anchorage attachments are fitted on the seat framework (more rigid than usual).

Seat Installed on Dolly (Figures 8, 9 and 10). Energy dissipators consisted initially of fluid dampers and later of collapsible tubes. A 50th percentile dummy was restrained on seat by three-point *double* belts.

TEST CONDITIONS						
Speed	km/h	20	35	50	65	80
	mph	12	22	31	41	50
Dolly Deceleration	g	15	24	32	38	42
Dolly Stopping Distance	cm	10	20	31	44	60
	in.	4	8	12	17	24

Figure 8

TEST CONDITIONS		
Speed	km/h	50
	mph	31
Dolly Deceleration	g	30
Dolly Stopping Distance	cm	33
	in.	13

Figure 9

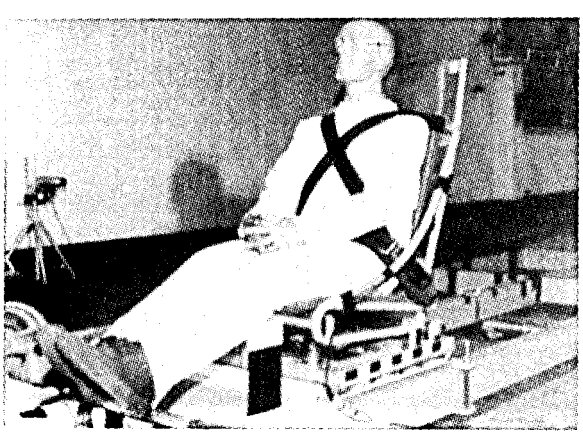


Figure 10

The dolly-seat-occupant system was tested by launching at increasing speeds from 20 to 80 km/h. For the reasons explained earlier, the collapsible tube damping system was preferred.

Seat Installed on Fiat 128 Body Mock-Up (Instrument Panel Replaced by Expanded Polystyrene Knee Pad) (Figures 11, 12 and 13). Energy dissipators consisted of short, pre-deformed

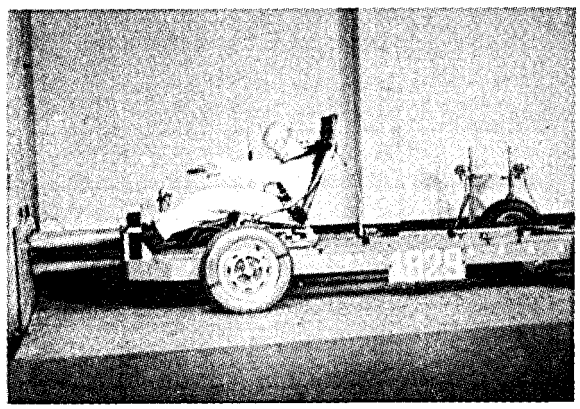


Figure 11

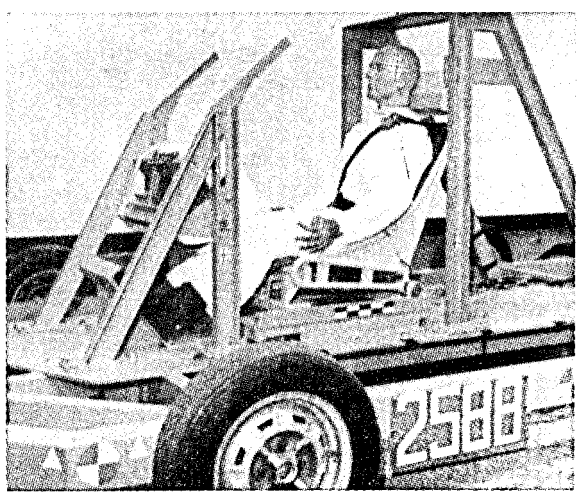


Figure 12

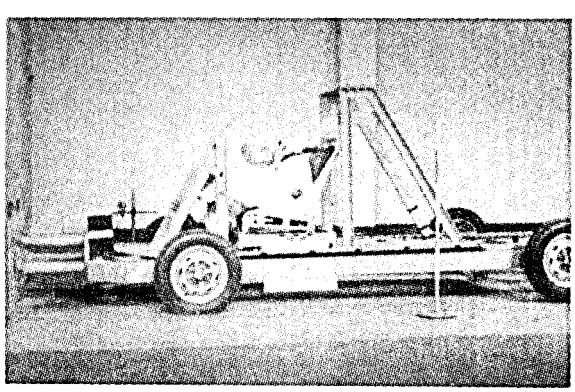


Figure 13

collapsible tubes. To overcome the adverse effects of friction of the lower (fixed) seat frame in longitudinal guide rail channels, the sliding frame was mounted on ball bearings.

The same dummy was used but restrained by *rigid three-point belts*. Tests were conducted at 50 km/h and average deceleration of 30 g.

Conventional Seat (Non-Sliding) Installed on Fiat 128 Body Mock-Up

Dummy was restrained by conventional safety belts.

Tests were conducted at 50 km/h and average deceleration of 30 g.

On comparing the results (Figure 14) from the last two tests mentioned above, it becomes evident that with the sliding seat reductions are obtained in accelerations of both the head (severity index halved and 15% lower decelerations) and thorax (severity index and decelerations cut down by one-third) and in belt loads (less than half).

The increase in femur loads appears to be avoidable by providing knee pads of different design.

Results												
System	Head		Thorax		Femurs				Belts			
	S.I.	Acc. Max	S.I.	Acc. Max	Sin		Des		Max load on lap section			
									Inboard		Outboard	
	–	g	–	g	kg	lbs	kg	lbs	kg	lbs	kg	lbs
Conventional Belts	963	59	328	66	147	324	307	677	1,403	3,097	673	1,485
Sliding Seat	426	52	239	44	660	1,456	785	1,732	650	1,434	285	629

Figure 14

CONCLUSION

In its present form the energy-dissipating sliding seat concept offers notable advantages over the conventional fixed seat-and-belt arrangement; optimization of the system hence appears to be advisable.

In view of this the Italian Ministry of Transportation feels it expedient to continue the studies and experimentations on this particular restraint system.

SOME METHODS OF ABSORBING THE ENERGY OF MOTOR VEHICLES AND THEIR OCCUPANTS

P. WATSON
Transport and Road Research Laboratory

The purpose of this paper is to outline some of the progress which has been made on various methods of absorbing energy to help protect vehicle occupants and reduce damage costs.

The Transport and Road Research Laboratory proposed performance levels for the front of a car to protect pedestrians when struck, reduce the damage repair costs at low and medium speed impacts and to protect the occupants in high speed crashes.

The solution of the problem of protecting pedestrians probably lies in the fields of mechanical design ingenuity and material selection. Research so far has shown that the critical height of the bumper and its relationship to the second point of impact are important in giving an acceptable trajectory to the pedestrian on impact. Happily this requirement is not in conflict with protecting occupants in car to car accidents but indeed promotes vehicle compatability.

The vehicle performance at the three levels of speed are however in conflict as the effective length of a car for protecting occupants in high speed crashes will be shortened by the performance requirements at the two lower speeds. This situation of conflicting design requirements has stimulated research activity in bumper and bumper mounting design to achieve an acceptable compromise solution.

The bumpers themselves have to be strong enough to transmit loads to energy absorption devices, where fitted, and the ideal system would have a force displacement characteristic approximating to a square wave. If a device is force sensitive, this is of advantage in low speed impacts but at high speeds the forces could rise so much before displacement occurs that the frame or bodywork to which the device is attached collapses before any stroke takes place, and no energy is absorbed by the device to protect occupants.

"The Role of Extrusion Devices in Energy Absorption for Safety" by Deutsch, Elson and Moore (1) records the progress in this field by Imperial Metal Industries.

Figure 1 shows the IMI impeder which consists of a simple container, a plastic extrusion cartridge and a ram. On impact, the ram is forced into the cartridge and the plastic extrudes, absorbing energy. The devices are very simple and rugged and have no seals or critical apertures which can degrade with age. They are not recoverable, however, but the device can be simply reset by replacing the plastic cartridge.

The extrusion device is not velocity sensitive, and up to the limit of the available stroke, its free displacement characteristic is square in form with an efficiency of 85% which is maintained over a wide range of force levels, velocities and temperature range, -20°C to +40°C (Figure 2).

These devices used in conjunction with soft fronts offer a means of absorbing energy in low speed no-damage impacts and an alternative means of energy absorption in high speed crashes to that of providing a readily crushable vehicle front end. An example of this approach is the system adopted on the BLMC 1800 ESV.

It has been calculated that two extrusion devices stroking 14 inches behind a conventional recoverable bumper system could absorb all the impact energy of a vehicle of effective mass of one ton striking a barrier at 30 mph while limiting the structure force to 30.00016 per unit.

"Development and Manufacture of Automobile Bumpers" by Thomson (2) outlines the development by Enersorb Ltd. of a system based on an elastomer reinforced in a particular manner by a recently developed synthetic fiber. The effect of this type of reinforcement is to increase the modulus in bending of the elastomer by at least a hundred fold but still retain the ability of the elastomer to undergo substantial recoverable deformations (Figure 3). It was found that in bending the material had a nonlinear spring rate, the effect of this is that the bumper unit when deflected very quickly rises to a high force level which remains fairly constant until the legs start to contact each other when the rubber goes into compression when very high loads are transmitted through the bumper unit. The load displacement characteristic of the Enersorb System with the load increasing as the rubber comes under compression is shown in Figure 4.

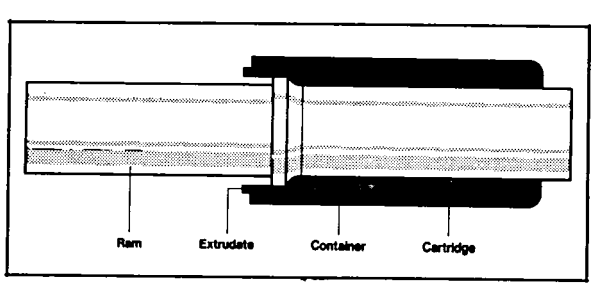


Figure 1. Extrusion Device During Impact

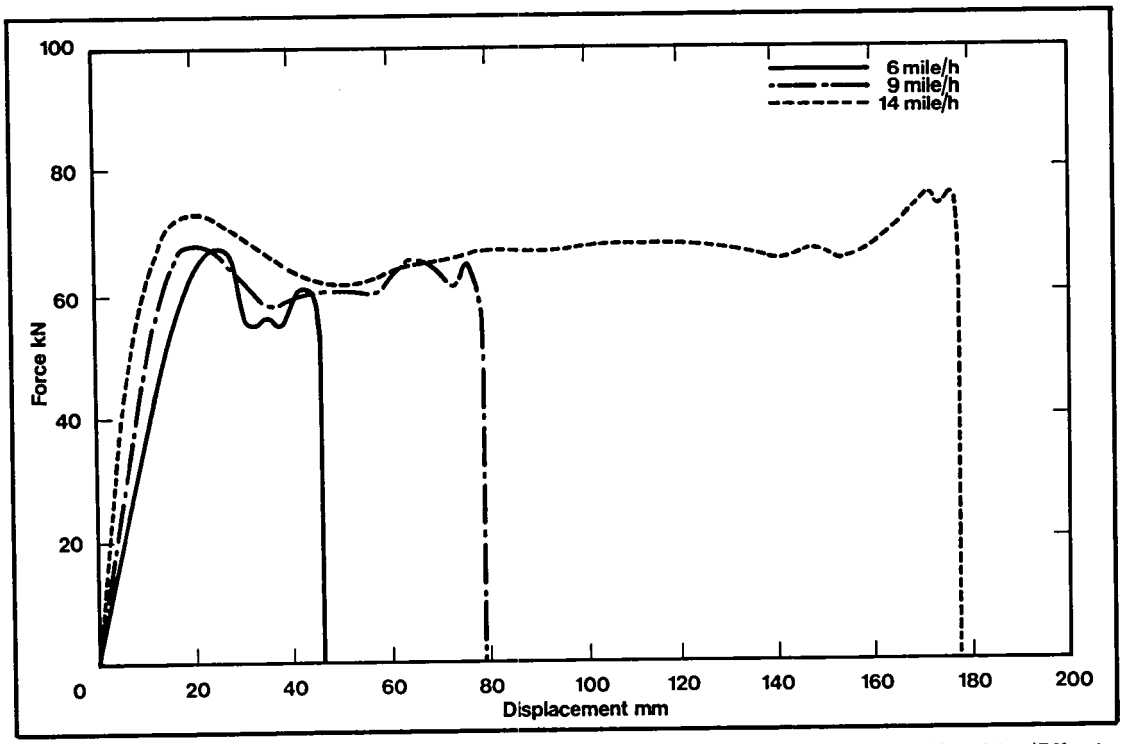


Figure 2. Force/Displacement Characteristics of an Extrusion Device over a Range of Impact Velocities (Effective Vehicle Mass 1.20 Tons with Two Devices)

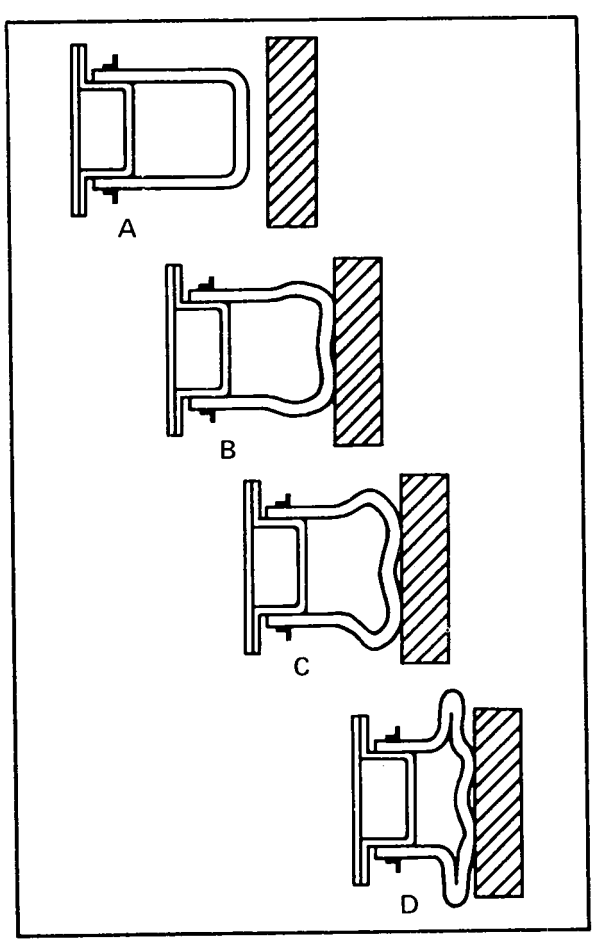


Figure 3

Under test efficiencies between 80% and 85% have been achieved and the performance of the unit over a temperature range of -30°C to +60°C is within a load variation of 30%.

An Enersorb bumper system has been successfully used in Phase I of the British Leyland vehicle developed at Cowley, where it showed resistance to damage when subjected to higher than design impact speeds.

"The Dunlop MK II Composite Energy Absorbing Bumper System" by Newton (3) details the development of energy absorbing devices by Dunlop which are used in conjunction with conventional

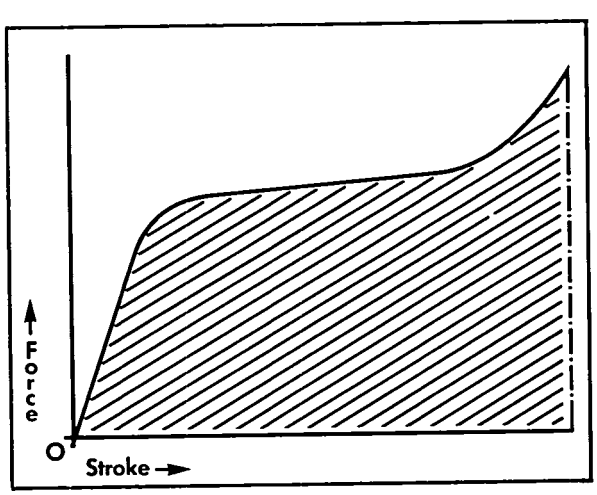


Figure 4

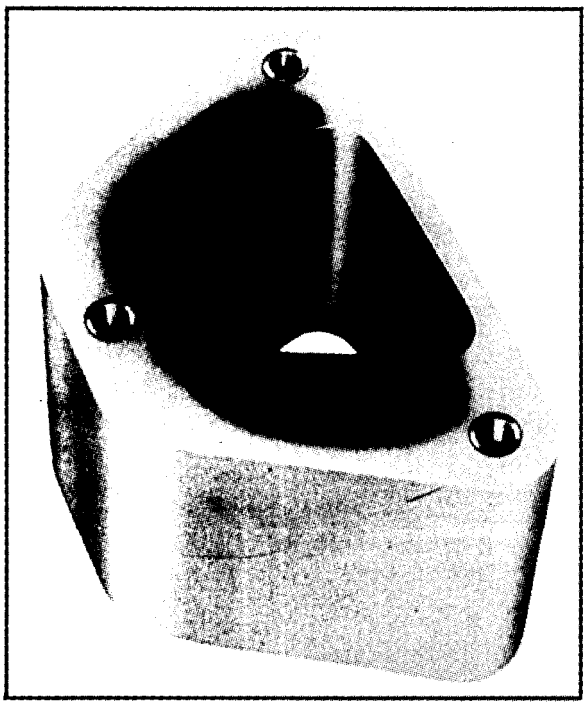


Figure 5. Composite Energy Absorbing Bumper Mounting

bumpers. The design takes advantage of the recent developments in the fusion bonding of runner and plastics by direct injection moulding techniques at the Central Research and Development Division of Dunlop. The composite construction of the energy absorber (Figure 5) minimizes the damage in low speed collisions and controls the deceleration loads experienced by the vehicle structure.

The preferred method of mounting is shown in Figure 6 with the wider end attached to the bumper to spread the impact load and reduce the bending moment in the bumper. Under impact the mounting will initially absorb energy by buckling and then by continued bending of the composite arms until the predetermined maximum deflection has occurred. In the event of an impact above the design specification excess energy will be absorbed in direct comparison to the rubber and the force transmitted to the vehicle structure will be a minimum.

During deflection, the high resilient rubber constituent stores sufficient energy to return the mounting immediately to its original shape and the geometry of the system provides maximum protection in the case of angled impacts.

A typical load/deflection characteristic is also shown in Figure 6 with an efficiency of 86%.

The concept of using high energy absorbing rubber/plastic composites is now being extended in the development of full flexible bumpers and soft front ends.

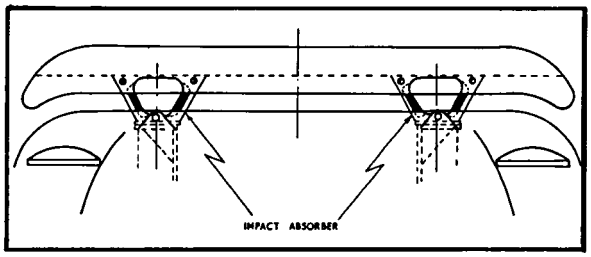


Figure 6a. Method of Mounting Car Bumper

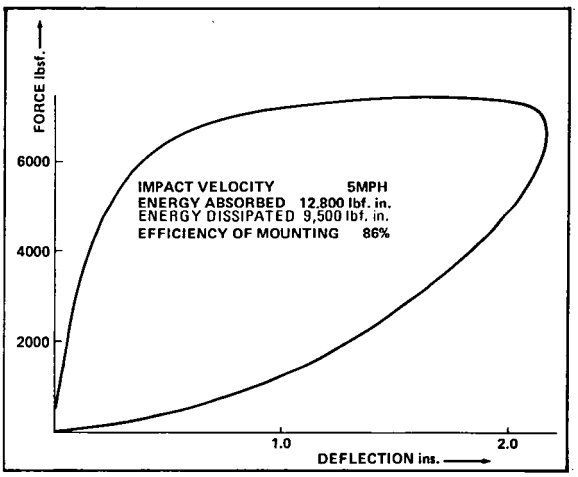


Figure 6b. Force/Deflection Characteristics of MK II Impact Mounting

"A Practical Approach to the Protection of Motor Vehicles by the Absorption of Impact Energy" by Persicke and Child (4) describes the development of a hydraulic shock absorber which will recover with no damage at impact speeds up to 5 mph and be effective in protecting occupants at impact speeds up to 40 mph.

Figure 7 shows the design used which is a patented combination of a specially designed liner to a cylinder, supported by an elastic member and supplemented by a yielding member which deforms to prevent the cylinder bursting at the highest rated speeds as the piston strokes. The liner has a machined slot which forms the orifice and by variation of the slot profile over its length the force level can change over its stroke length to suit the characteristics of the vehicle to which the device is fitted.

A typical force/displacement characteristic is shown in Figure 8 for an impact speed of 5.5 mph for a vehicle of effective mass of one ton.

Laminated car windshields are themselves an energy absorption system when struck by occupants during vehicle impacts and the paper by Pickard, "Reduced Laceration from a New Laminated Windscreen" (5), records improvements of the new "TRIPLEX Ten Twenty Windscreen."

Figure 9 shows the makeup of the Ten Twenty laminate. An outer glass, lightly stressed for optimum

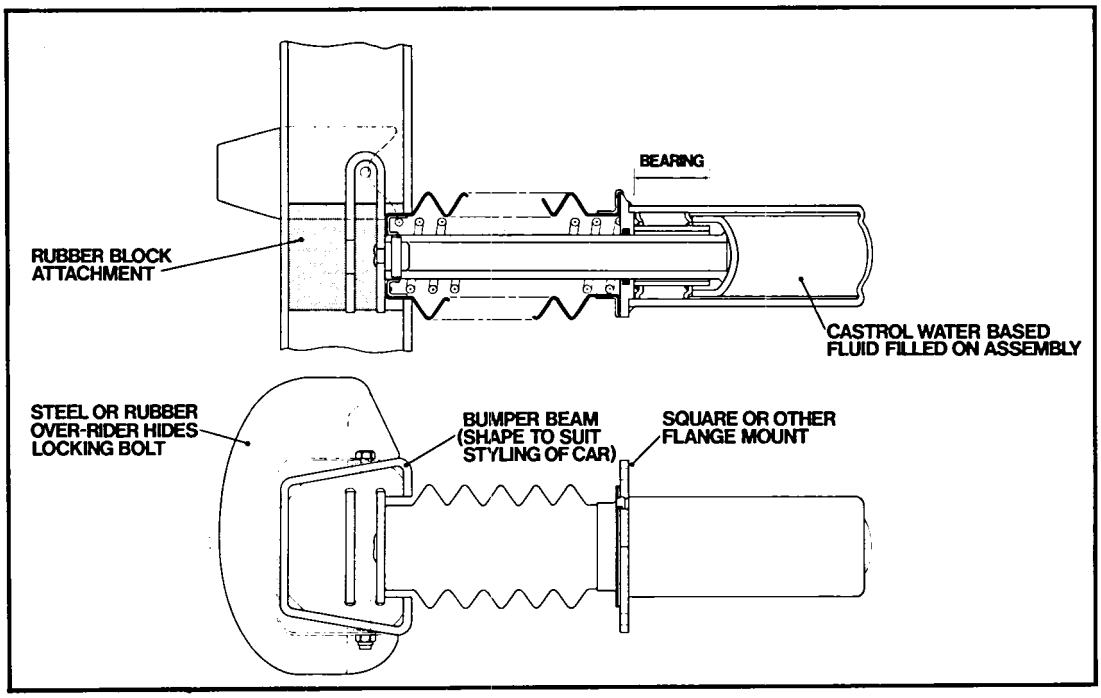


Figure 7. Hydraulic Shock Absorber and Bumper Attachment

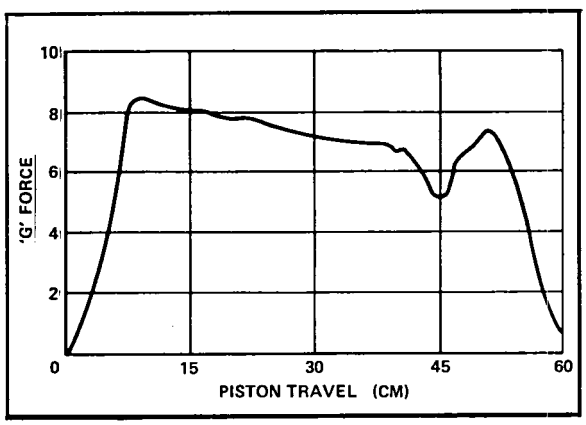


Figure 8. Performance Graph of Shock Absorber

road stone performance, a standard H.P.R. vinyl interlayer which has already proved its excellent deceleration and containment qualities, and a high stressed thermally toughened inner glass which greatly reduces facial laceration. This stress level in the thermally toughened inner glass can be accurately controlled.

Figure 10 shows that, on impact, the "Ten Twenty Windscreen" does not exhibit any significant difference in the Gadd Severity Index, compared with conventional laminated windshields. Tests have shown, however, that there is a two logarithmic unit reduction in facial laceration with these new windshields – shown by the lower line, compared with conventional laminated windshields (Figure 11).

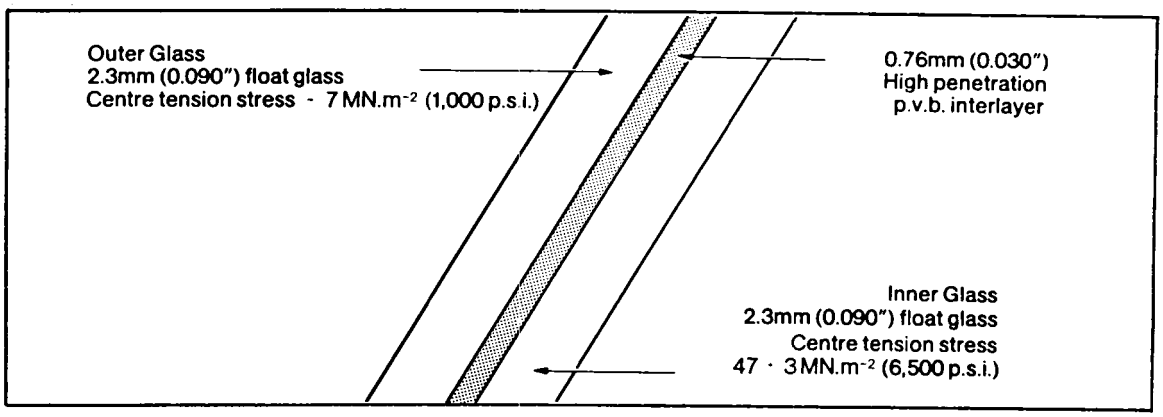


Figure 9. Construction of the "Ten Twenty Windscreen"

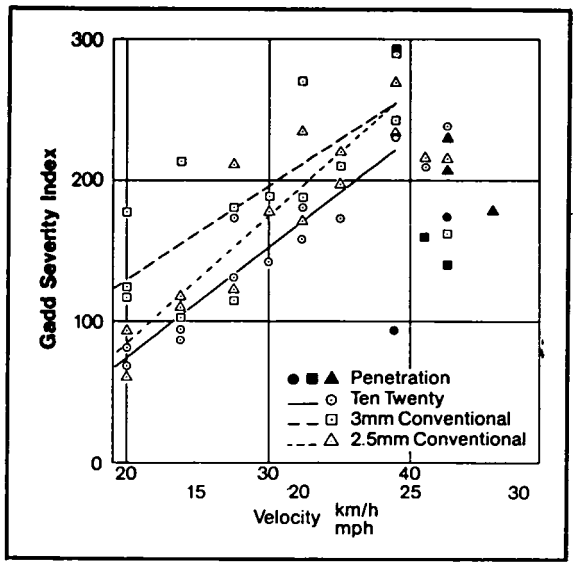


Figure 10. Gadd SI as Function of Impact Velocity by Dropping Headform Test (Triplex Test)

The Vertical Axis is the Triplex Laceration Index, which is an objective measure of windshield performance for facial lacerations calculated from this formula. This is a function of measured lengths and depths of cuts obtained on a facial mask covered with two layers of leather (Figure 12).

Independent confirmation of this two logarithmic unit reduction in laceration is shown by the lower line in Figure 13, showing results obtained by Patrick at Wayne State University.

Figure 14 shows the characteristic bulge of a conventional laminated windshield (on the left) compared with the TRIPLEX Ten Twenty windshield, both impacted at 19 mph. The finer and more extensive fragmentation and smoothness of the bulge in the case of Ten Twenty causes much less

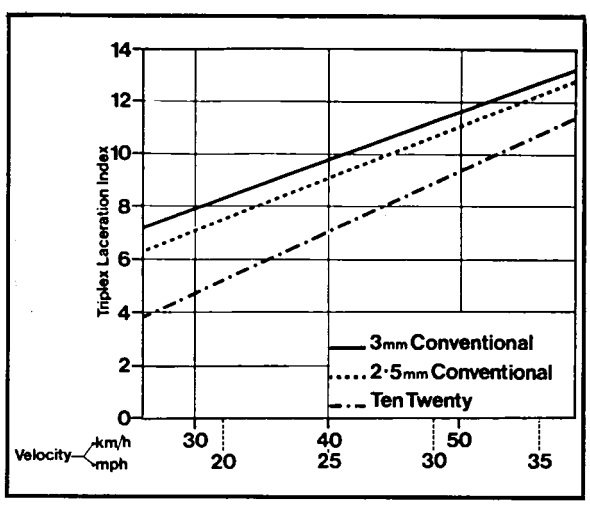


Figure 11. Laceration Index (TLI) as a Function of Velocity by the Skull Impact Test at 20°C

$$2+\log_{10}\left[1+1.16D_1+50.8D_2+16{,}500D_3\right]$$

Where:

$$D_1=\sum_{s=1}^{s_{max}} n_{1s}\,s^2$$

$$D_2=\sum_{s=1}^{s_{max}} n_{2s}\,s^2$$

$$D_3=\sum_{d=1}^{d_{max}}\sum_{s=1}^{s_{max}} n_{3sd}\,s^2\,d^3$$

Figure 12. Triplex Laceration Index (TLI)

severe lacerations as the face moves down the windshield.

Triplex and the TRRL are now starting a program to establish the performance of various glasses when impacted from the exterior by a pedestrian.

"Energy Absorption by Variable Shear Strength Duplex Materials" by Millman and Chappell (6) details the novel developments at the Wolfson Institute in the University of Nottingham.

Figure 15 shows a simple model device of a crimped flat strip of helically crimped wire which is pulled through an orifice. Under the action of the applied load, the core tends to straighten out to produce self-regulating pull-out characteristics, while the elastically strained core, pressing against the surface of the orifice, generates a frictional force. The system is designed to operate at a load just below that which would cause fracture of the core and the force/displacement characteristic of both wire and strip cores shows a near ideal square form (Figure

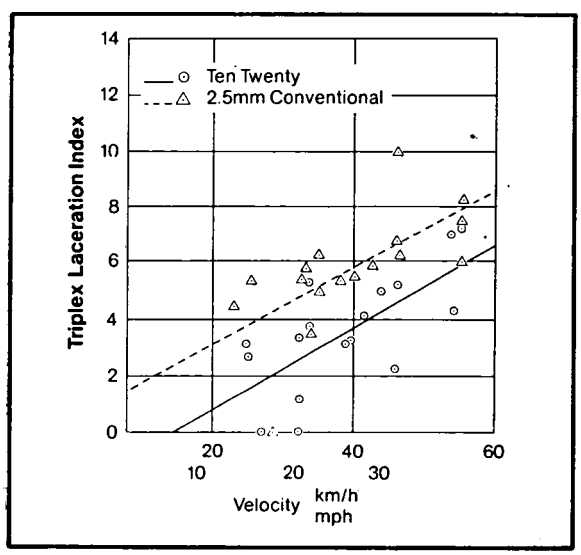


Figure 13. TLI as Function of Velocity by WHAM III Test Rig (Wayne State University Tests)

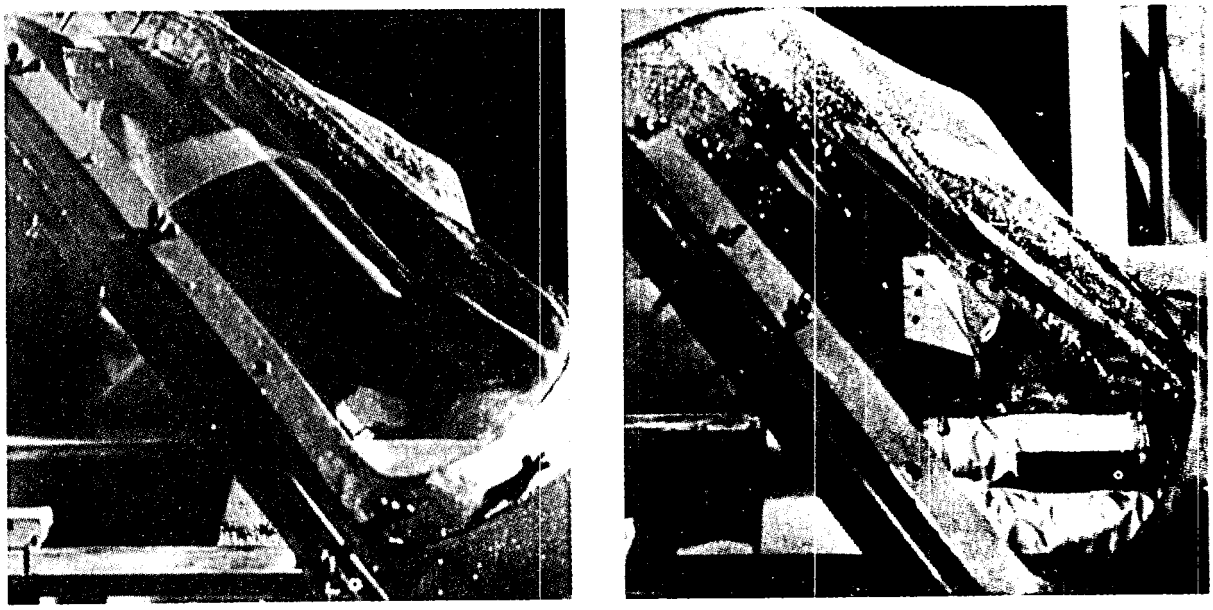


Figure 14. Side View of Characteristic Bulge in Conventional Laminated and Triplex Ten Twenty after Sled Headform Impact at 19 mph. Note: Finer and More Extensive Fragmentation and Smoothness of the Bulge in the Case of Ten Twenty Causing Much Less Severe Lacerations as the Face Moves Down the Screen

16). Energy will continue to be absorbed at a constant rate, depending on the length of core which is used in the device. By changing the geometry of the system and the number of core elements, the characteristics of the device can be tailored to suit the design requirements.

Materials that exhibit high strength properties normally have poor ductility – with the result that fracture would occur after a short extension, while materials with good ductility will only resist low loads. These duplex devices offer characteristics of both high strength and ductility for energy absorption.

A potential application for these devices is providing yielding in seat belt installations where they can be designed to operate under the required loads and extensions and already prototype designs are under construction for controlling the deformation of steering wheels and columns.

I trust that this brief resume of these six papers gives an outline of the activities in the fields of energy absorption which will be available for future vehicle designs.

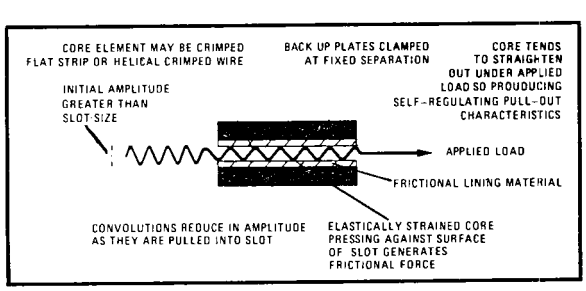


Figure 15. Duplex Energy Absorbing Devices

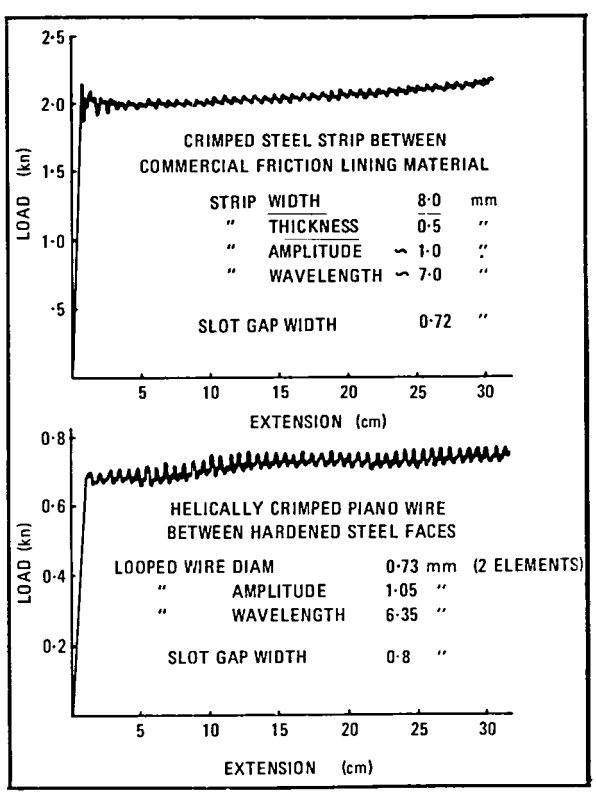


Figure 16. Force/Displacement Characteristic of Wire and Strip Cores

REFERENCES

1. K. Deutsch, M. G. Elson, and R. W. Moore, "The Role of Extrusion Devices in Energy Absorption for Safety."
2. I. M. Thomson, "Development and Manufacture of Automobile Bumpers."

3. D. A. Newton, "The Dunlop MK II Composite Energy Absorbing Bumper System."
4. G. Persicke and J. R. Child, "A Practical Approach to the Protection of Motor Vehicles by the Absorption of Impact Energy."
5. J. Pickard, "Reduced Laceration from a New Laminated Windscreen."
6. R. S. Millman and M. J. Chappell, "Energy Absorption by Variable Shear Strength Duplex Materials."

VEHICLE COMPATIBILITY IN CAR-TO-CAR SIDE IMPACTS AND PEDESTRIAN-TO-CAR FRONTAL IMPACTS

PETER M. FINCH
Manager, Vehicle Safety
British Leyland Motor Corporation, Ltd.

You have already seen the two British Leyland films briefly outlining our work to date, and have received copies of the relevant progress reports.

At the risk of being considered repetitive I would like to explain in a little more detail that facet of our investigations concerned with car-to-car compatibility in side impacts, and suggest a relationship between this situation and pedestrian-to-car compatibility in frontal accidents.

You may remember from the Marina film, that our standard vehicle suffered intrusion and occupant injury problems, when impacted at 90° by a similar vehicle travelling at 30 mph.

The apparent solution was to reduce the degree of penetration by stiffening the side structure, and the severity of the dummy impact by the addition of energy absorbing interior panels.

Two different models were investigated, but it is only the 2 door coupé that I wish to discuss today.

It would be tedious to explain all of the modifications incorporated into this vehicle, since details are given in the appropriate progress report. But as well as the addition of door guardrails, improved crossmembers and strengthened 'B' posts, the major modification was to increase the beam stiffness of the sill section, and by the creation of an inner wall, provide a barrier against the inward collapse of the door assemblies.

Each proposed modification was individually assessed, and those selected combined in an experimental body shell, for final analysis prior to the resumption of dynamic testing.

At the time a computer simulation programme was not available, so we adapted one of our crush rigs as an impact simulator. A shaped barrier form designed to reproduce the post-impact profile of the bullet car, was mounted on the face of the crush rig.

This was forced under load into the side of a standard body shell. When we had achieved the same degree of penetration as resulted from the dynamic test, we calculated the amount of 'work-done'. Then by applying this same procedure and 'work-done' to a modified vehicle, it was possible to obtain a reasonable prediction of the amount of intrusion to be expected under dynamic conditions.

A similar approach was successfully adopted for car-to-pole tests.

Interior occupant protection was the subject of separate investigations. One aspect of these was the study of energy absorbing padding. The majority of the materials tested proved unsatisfactory for our requirements. Homogeneous foams either resulted in high rebound velocities or tended to collapse into solid blocks of material, with resulting low penetration and high decelerations.

Our best results were obtained from sheet steel covered with a layer of force distributing foam and slotted or perforated to encourage the desired collapse pattern and low decelerations.

An alternative approach was to mould foam pads, so as to produce a series of ribs or cones within the periphery of the moulding.

This encouraged the collapse of the material at the desired loads and prevented solidification of the cellular structure, enabling us to achieve satisfactory dummy loads, at levels not attainable with a solid mass of foam of the same density.

The Phase 1 side impact vehicle was produced by marrying these structural and trim ideas into one unified design. When impacted at 90° by a standard Marina the intrusion was reduced from 14.25 inches to a maximum of 5.25 inches; with improved, but not entirely satisfactory injury levels.

A second test was then conducted, identical to the former except that the standard Marina bullet car was replaced by a modified vehicle of the same weight. This vehicle incorporated all the structural modifications found necessary to achieve the 40 mph frontal barrier impact and low speed requirements.

This time the maximum intrusion was reduced to .94 of an inch, in addition all the dummy loads and accelerations were reduced to insignificance.

From analysis of the high speed films the difference in results appears to have been due to three main factors. Firstly, the front end of the Phase 1 bullet vehicle incorporated soft headlamp surrounds and an energy absorbing bumper; these were responsible for dissipating approximately 20% of the impact energy prior to contact of the metal structures, and the elimination of hard contact points. Secondly, the bumper was centred 14½ inches above ground level at kerb weight. In its lower position it contacted the overlapping lower door and sill panels, whose combined rigidity restricted penetration to a minimal degree. Finally, because the bullet car, with its low bumper position struck the target car well below its centre of gravity, the latter tended to roll towards the bullet car and so delay impact of the occupants' head and upper torso with the interior. This delay contributed to the reduced occupant impact velocity.

In the previous test the standard bullet car, with its conventional bumper height backed by almost vertical frontal panels, spread the impact loads over the target vehicle's relatively weak side panels. Although this produced an even distribution of load, the deformation resulted in the dummies being struck by the collapsing inner panels at a higher velocity than in the subsequent test. As a result, the recorded dummy loads were also higher.

These results illustrate the advantages to be gained by making the two cars compatible in this particular accident situation.

They show that a bullet car having a low, energy absorbing bumper coupled with an angled front end profile, can materially effect the results of a side impact even when impacting a reinforced safety vehicle.

We believe that a car incorporating this frontal design also offers a further advantage, in that it can be adapted to provide increased pedestrian protection in frontal impacts. In these situations, which account for nearly 90% of pedestrian casualties, initial contact is generally made by an angular metal bumper; designed for vehicle, rather than pedestrian protection. Impact from the hard unyielding surface results in the high incidence of leg fracture common with this type of accident.

The same low bumper which afforded car-to-car compatibility, when fitted with a 'soft' load distributing leading edge, can reduce the possibility of leg fractures, and by striking below the knee, the likelihood of incurring clinically serious joint injuries, with their unfortunate long term after effects.

At low to moderate speeds the struck pedestrian falls towards the impacting vehicle; a secondary impact against the usual high profile bonnet can cause injuries to the pelvis and thorax, these may be followed by further injuries as the victim is projected onto the road surface.

A car having a low frontal profile rising to a smooth bonnet area, similar to our Phase 2 Marina, will strike the pedestrian below his centre of gravity causing him to fall flat across the bonnet, where preferably he will be retained by some mechanical means until the car can be brought to rest. An example of this has been shown in the Longbridge film and demonstrated at Crowthorne on the Austin 1300.

In summary therefore, we offer for consideration these two aspects of our investigations, which in the provision of a degree of compatibility, are related through the provision of the low bumper and modified frontal design. We urge that in the establishment of national and international bumper heights, more consideration be given to the possible effect that bumper location has on other road users particularly those less protected than the occupants of the striking vehicle.

Finally we hope that these initial investigations may encourage further work on this subject.

THE APPLICATION OF COMPUTER SIMULATIONS IN VEHICLE SAFETY

W. C. EMMERSON
Manager, Engineering Development
J. E. FOWLER
Project Leader
British Leyland Motor Corporation Limited

INTRODUCTION

In the development of a new vehicle numerous experimental tests have to be carried out to ensure acceptable performance in all respects, and this requires the production of many expensive prototype vehicles. Fortunately much of the performance testing is non-destructive and therefore a particular prototype can be used for a number of different purposes. However, in the case of safety requirements many of the tests are destructive in nature, and each new requirement can seriously effect the development lead time and cost. It was therefore decided that there was a need to develop techniques which were basically non-destructive, so that assessments could be made of various design schemes to ensure that new models would comply with future safety requirements.

The approach adopted at British Leyland in 1968 was to develop computer systems which simulate the various safety performance requirements. These systems would enable an assessment to be made of the survival criteria for pedestrians, vehicle occupants and the vehicle structure. Some of the computer programs developed have been used in the joint work undertaken with the Transport and Road Research Laboratory (T.R.R.L.), and reference is made to this work in the paper to be presented by P M Finch (Ref. 1). The present paper elaborates on the various simulation models which have been developed, and illustrates their application to the safety projects undertaken with T.R.R.L.

All the computer programmes for assessing the structural performance of the vehicle have been developed in British Leyland and include the analysis of Front, Rear and Side impacts. Using the various vehicle simulation models, assessments can also be made of vehicle to vehicle impacts under any combination of these three impact conditions. In the analysis of occupant and pedestrian survival criteria it has been found that the Calspan program (Refs. 2 and 3) is suitable, and in the case of occupant analysis the input information used is the output from the vehicle impact simulation programmes in terms of deceleration/time characteristics.

The project work undertaken jointly between T.R.R.L. and British Leyland, which is described in (Ref. 1) was all based on developments of the Morris Marina. The simulation models and results presented in this paper are also based on this vehicle, although other vehicles have been analysed in a similar way.

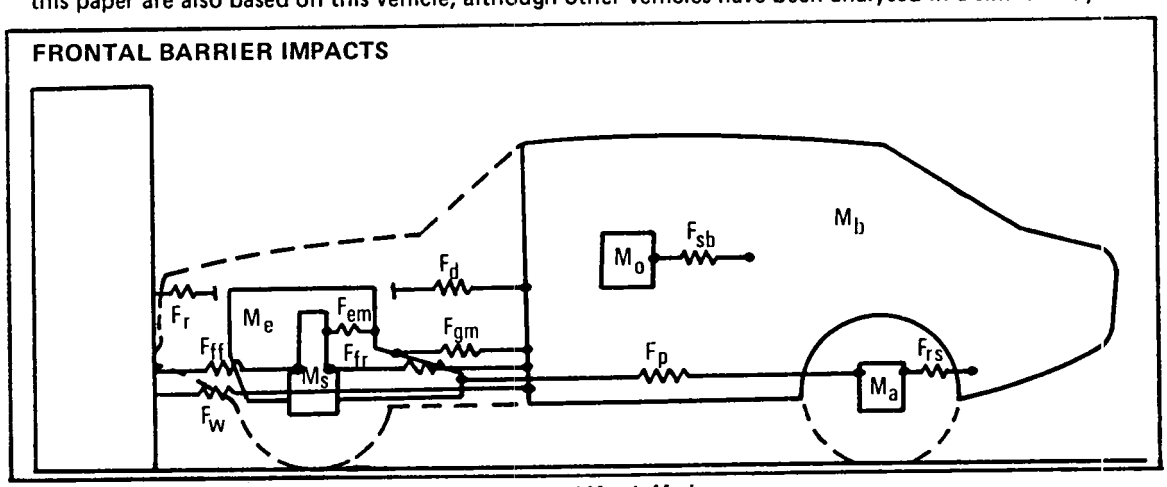


Figure 1. Frontal Barrier Test. Simulation model of standard Morris Marina

The mass distribution of the Marina was analysed and divided into a number of well defined and where possible discrete masses. These masses were considered to be connected by non-linear, plastically deforming "spring" elements representing parts of the structure. The idealized model of the standard Marina is shown in Figure 1. (The notation used is described in the Appendix). For this model the equations of motion, on which the simulation programme is based, are as follows:—

$$M_b\ddot{x}_b + F_w + F_d + F_{gm} + F_{fr} + F_{rs} + F_{sb} = 0 \quad . \quad . \quad . \quad (1)$$

$$M_o\ddot{x}_o - F_{sb} = 0 \quad . \; . \; . \quad (2)$$

$$M_e\ddot{x}_e + F_{em} + F_r - F_d - F_{gm} - F_p = 0 \quad . \; . \; . \quad (3)$$

$$M_a\ddot{x}_a + F_p - F_{rs} = 0 \quad . \; . \; . \quad (4)$$

$$M_s\ddot{x}_s + F_{ff} - F_{fr} - F_{em} = 0 \quad . \; . \; . \quad (5)$$

The forces in the equations (1) to (5) represent the dynamic forces produced instantaneously during the impact, and are functions of the relative displacement between the masses (i.e. relative crush) and the rate of crush. The following empirical relationship is used to relate the dynamic forces to the static crush characteristics:–

$$F_{dynamic} = F_{static} \left[1 + \lambda \frac{\dot{x}}{Vo}\right] \qquad \text{where } \lambda \text{ is}$$

the "dynamic magnifier", $\dot{x}$ is the rate of element deformation and Vo is the reference velocity at which λ has been determined. The value of λ varies for different types of construction and for different parts of the structure. The actual values used are usually based on previous comparisons between theory and experiment.

In the case of the standard base line Marina the static stiffness of the "spring" elements were determined experimentally by slow crushing in a specially designed crush rig (Ref. 1). Figure 2 shows a typical stiffness characteristic which in this case is for the Marina front end forward of the front wheel centre line.

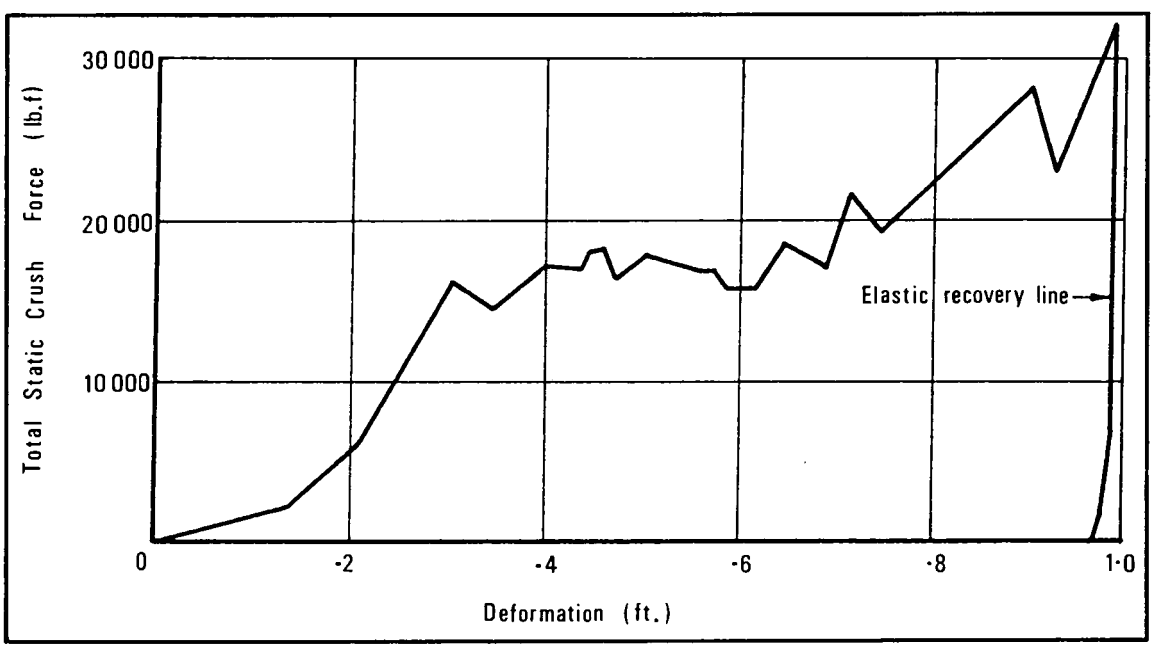


Figure 2. Stiffness Characteristic of Morris Marina Front Half of Front End

This graph clearly shows the inherent non-linearity of the structure, which is the main reason why normal analytical techniques are not adequate, and why a mathematical model of the system must be made for use in a computer simulation.

A flow diagram for the complete programme is shown in Figure 3. The most critical part of the programme, and the part that largely determines its success, is in the calculation of the dynamic forces. The measured static characteristics are input to the computer in a tabular form. The instantaneous static force is then calculated from the relative displacement and modified according to the rate of crush. The rebound characteristic is equally important and its position varies according to the maximum displacement reached. The rebound point is detected by the point at which the vehicle velocity is zero. Each component force is determined in a subroutine (Macro). A series of these routines have been written to cover most of the different types of structural elements that could be encountered. These were developed independently to the main programme. Furthermore, these subroutines can be used for any type of vehicle impact simulation and are not restricted to frontal impacts.

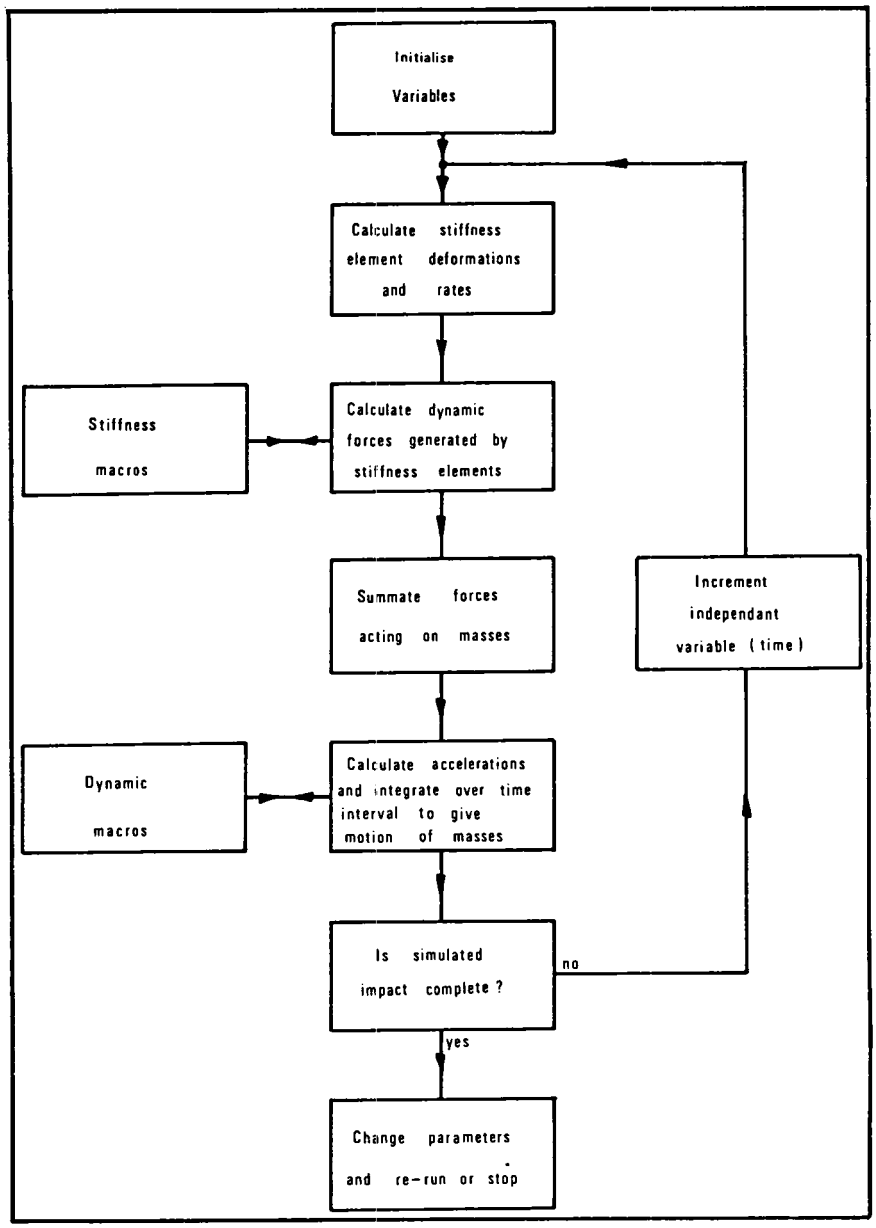


Figure 3. Basic Program Flow Diagram

The output from the programme can be any of the displacements, crush distances, velocities, accelerations or forces specified as variables and these can be plotted if required.

The occupants are represented by a single mass restrained by a single stiffness element as shown in Figure 1. This is obviously a simplification of real occupants, but its purpose is to approximate the effects of the occupants on the crashing vehicle and not vice versa. For accurate simulation of the effects on the occupants a separate occupant simulation programme is available which uses the vehicle simulation results as input data. The simplified representation of the occupants does give, however, an indication of how near to the optimum is the shape of the vehicle deceleration envelope. The occupants' peak acceleration being regarded as a vehicle "severity index".

A more detailed and technical description of the simulation techniques employed can be found in (Ref. 4).

Static crush tests were conducted on the standard Marina and the simulation model validated against the results of a barrier test. A comparison of the computer simulated and experimental test deceleration plots is shown in Figure 4. Once satisfactory results had been achieved the proposed changes to the standard Marina could be evaluated with the programme.

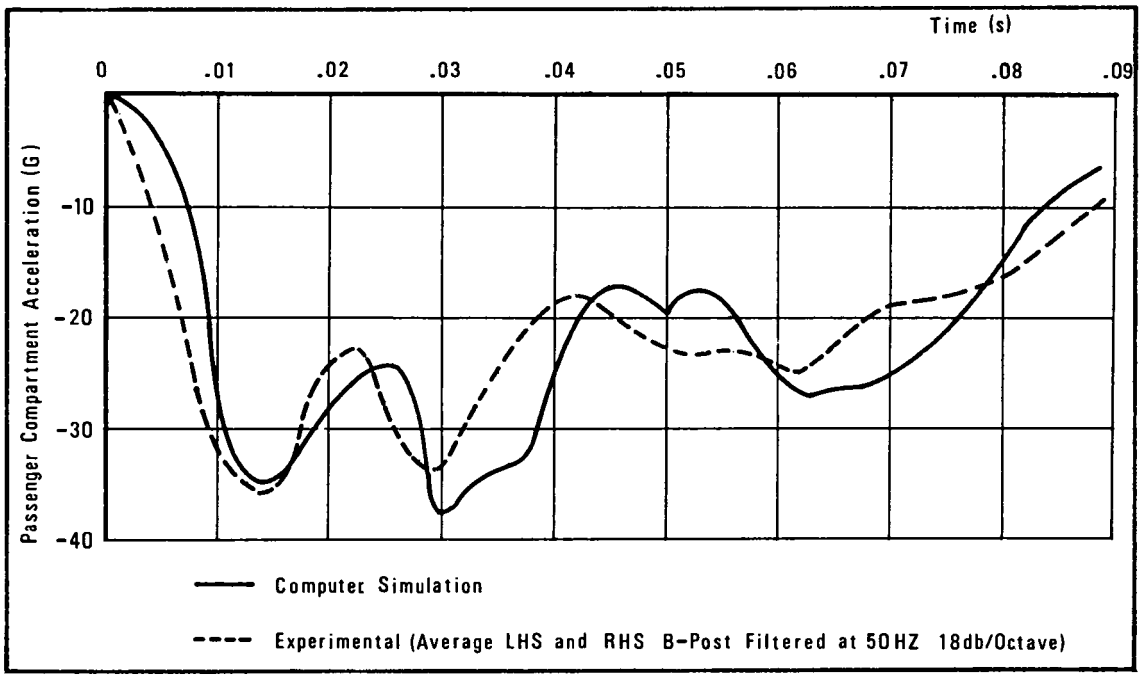


Figure 4. Comparison of Results for a 40 mph Frontal Barrier Test on a Standard Morris Marina

The main vehicle modifications considered were:—

a Front end structure incorporating curved longitudinals.
b Increased engine to dash clearance.
c Increased dash and tunnel stiffness.
d Hydraulic low speed bumper system.
e "Metalastik" and deformable cylinder bumper system.
f "Cable Belt" bumper system.
g Independent rear suspension.
h One-piece versus two-piece propeller shaft.
i Engine restraint systems for low speed impacts.
j Wheelbase increase by moving front wheels forward.
k Waist level longitudinal rails in doors.

Various combinations of these changes were simulated, in some cases the advantages gained by one modification would be reduced by the interaction with the effects of another. Based on these simulation results it was decided to build the first modified vehicle (Phase I) to include modifications (a), (b), (c), (f), (i) and (k) given above. It should be noted that the effects of the modifications evaluated were specific to the modified Marina and may not be desirable for vehicles with different stiffness or mechanical layouts. The simulation model representing the vehicle in the Phase I condition is shown in Figure 5, and the comparison of the simulated and experimental results is shown in Figure 6. The close agreement between these results indicates the usefulness of the simulation technique during the design of the vehicle. An important advantage is the exact repeatability obtained and the facility to change one variable at a time while the others remain exactly constant. This is extremely difficult with experimental testing due to the inherent scatter in results.

The above list of modifications (a) to (k) include both high speed and low speed design features. The high speed requirements are basically concerned with occupant survival, whereas the low speed features are to minimise vehicle damage. It is of interest to describe further some of the work on low speed impacts.

Bumper system evaluation was simulated using a model similar to that shown in Figure 5, but with a freedom for engine pitch included. With low speed no damage impacts the restraint of the engine is equally as important as the design of the bumper and back-up system itself. Forward acting engine restraints are necessary to prevent the engine penetrating the radiator and possible damage to the engine mountings. The type of restraint and its point of reaction on the engine is of prime importance and this

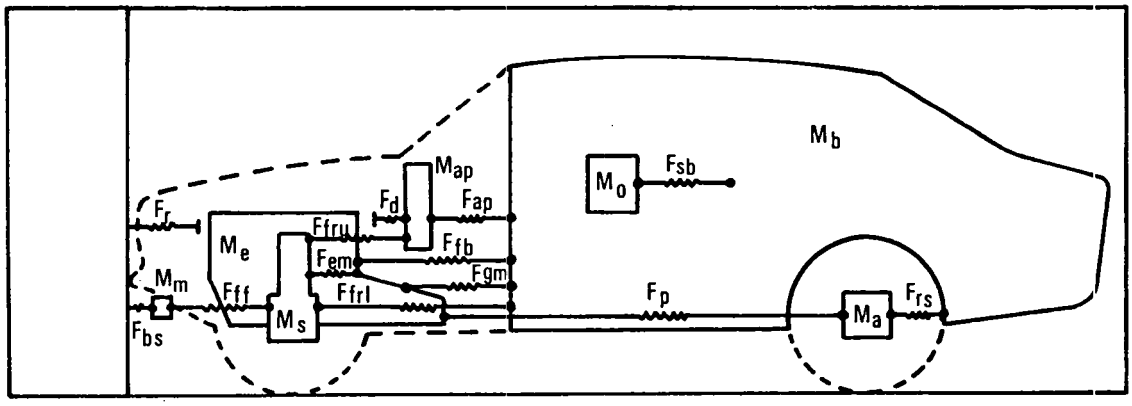


Figure 5. Simulation Model of Modified (Phase 1) Marina

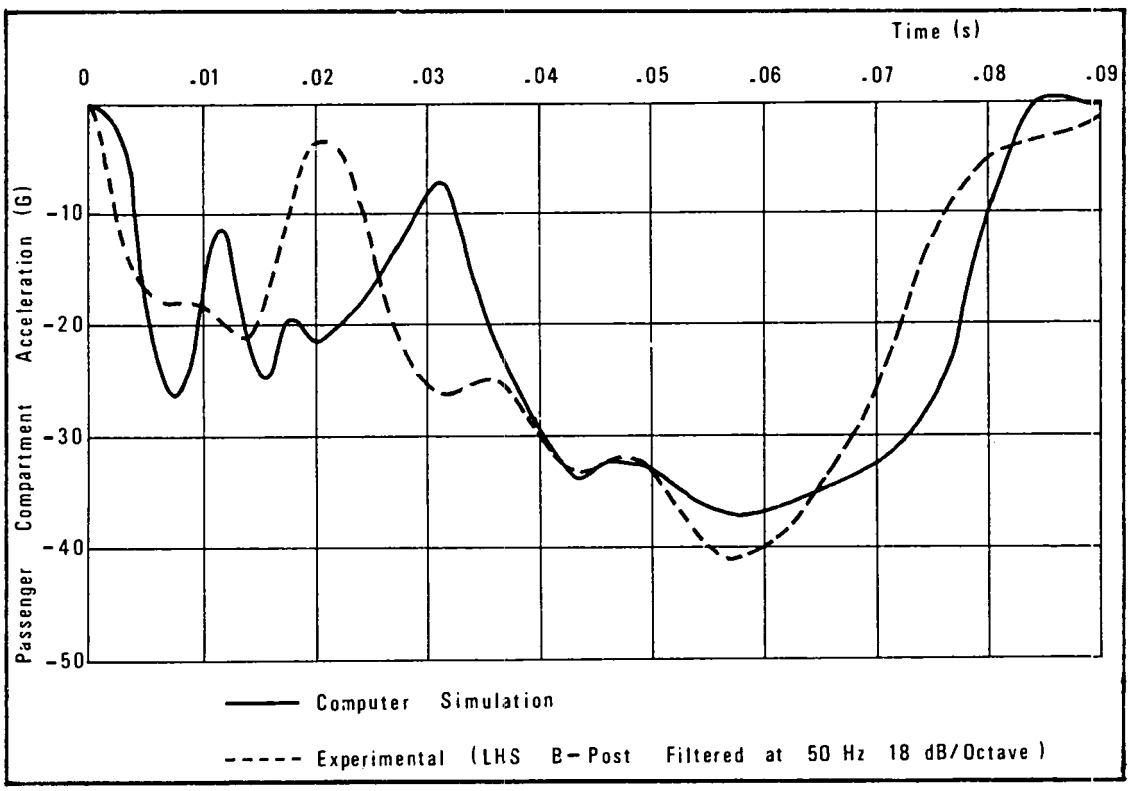


Figure 6. Comparison of Deceleration for Modified (Phase 1) Marina

was studied with the model. It may be thought that a simpler model could be used, but the effective mass of the vehicle, as seen by the bumper system, increases during the impact because a large proportion of the vehicle mass is flexibly connected to the main structure. The simulation, therefore, approximately compensates for these effects which are more pronounced at higher speeds.

SIDE IMPACT

There are three separate types of 90 degree side impacts that need to be considered; mobile barrier to car, car to pole and car to car. One model covering the first two cases has been developed and is shown in Figure 7. The nomenclature for this model is described in the Appendix. For the pole impact the mobile barrier mass, M_{bar} is replaced by a rigidly fixed pole and the initial velocity conditions changed to achieve the correct relative motion.

Several simple models were investigated before the model shown in Figure 7 was developed. The earlier models showed that it was necessary to separate the mass and crush characteristics of the side structure from the main structure of the target car. In reality the effective mass of the separate side structure is

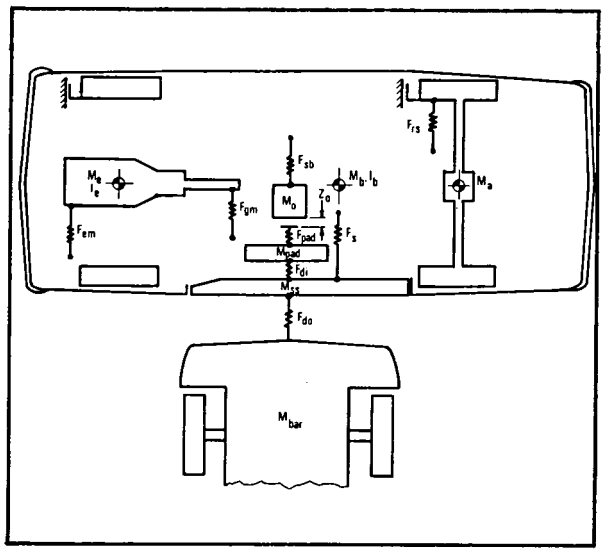


Figure 7. Basic Side Impacts Simulation Model

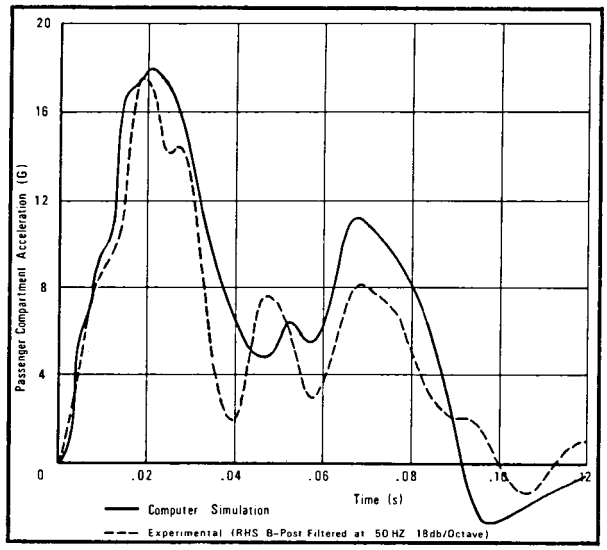


Figure 8. Comparison of Results for a 30 mph Mobile Barrier Side Impact Test

initially zero but progressively builds up as the structure collapses. Separate mass and stiffness elements have also been included in order to represent proposed foamed plastic trim pads. The clearance between the occupant and interior surface (Zo) is also included as a variable parameter. For the vehicle programme the occupant torso is represented by the single mass Mo, and the occupant simulation programme is used for a detailed analysis of the effects upon the occupant.

The simulation programme is formulated using the same techniques as described for the frontal impact programme. Due to the different nature of pole and mobile barrier side impact tests, separate crush tests need to be conducted to provide the input data.

Side impact tests were conducted on a standard base line Morris Marina and the results used to validate the side impact simulation models. Most of the stiffness input data was again measured experimentally by using static crush tests. The dynamic magnifiers for the foam door pads were determined by conducting dynamic tests in addition to the static ones.

Figure 8 shows the acceleration predicted by the mobile barrier impact simulation compared with that for the equivalent experimental test. The peak accelerations are close, but thereafter the simulated acceleration decays less rapidly than the experimental curve. This can be partly accounted for by the fact that the experimental method used for measuring some of the static stiffness characteristics was not completely satisfactory. However, for an initial attempt at side impact simulation the results were considered useful enough to warrant the use of the model to predict the effects of changing three basic variables. These were, the occupant to vehicle free space, overall stiffness of side structure, and interior padding.

The distance between the interior side surface and the occupants torso, Zo, was varied between practical limits, i.e. 0–0.8 ft. From this it was found that there was a general trend for the Gadd severity index to slightly reduce with increasing free space. This is due to the fact that the velocity of the door inner surface, after being accelerated by the barrier, reduces as the crush of the target vehicle increases, until they are both moving at a common velocity. Contact with the occupant torso at this time will therefore produce the lowest impact velocity. However, the realistic increase in free space that can be achieved does not bring about any worthwhile improvement.

The stiffness of the side structure was varied within practical limits and its effect upon the occupant determined. This showed that the design condition was close to the optimum. Little improvement being gained by any increase in stiffness, but considerable deterioration in the conditions was evident if the stiffness was reduced.

The application of four inches of normal foam padding to the interior surface of the door produced the most dramatic improvements in the thorax severity index, reducing it by as much as 90% relative to the unpadded condition. A peak in thoràx acceleration may still occur due to the foam being fully compressed, but this could be overcome by increasing the depth and/or stiffness of the padding.

The side impact target car model is at present under development to improve the accuracy and to generalise its application. This includes the incorporation of the vehicle bending stiffness in a horizontal plane. This type of distortion is very noticeable in the pole tests and to some extent in car to car tests. To facilitate the simulation of impacts at angles other than 90 degrees longitudinal components of the

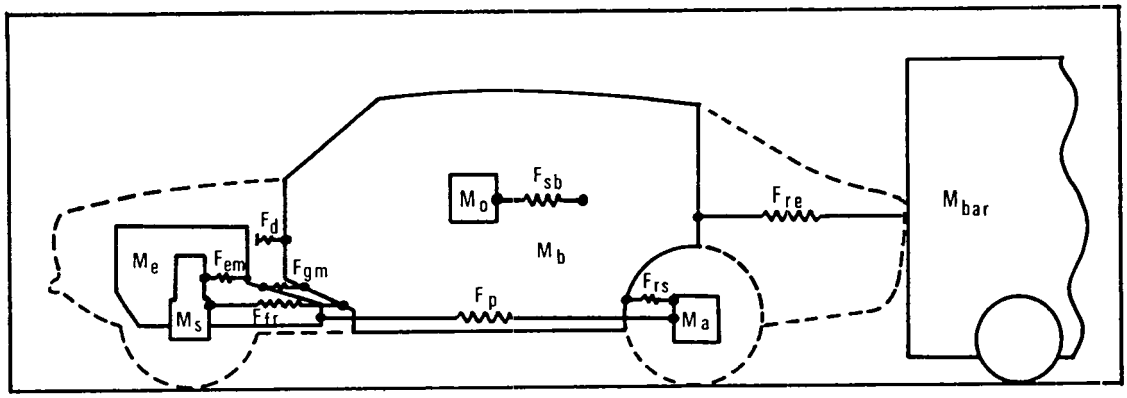


Figure 9. Rear End Mobile Barrier Impact Model

stiffness element forces are being incorporated. Friction of the occupant on its contact surfaces is also being considered.

This type of simulation model, for conventional type vehicles, has been developed directly from the frontal model and is shown in Figure 9. The engine, front suspension, rear suspension and occupants have been included as separate masses since these effect the motion of the vehicle, especially the engine which can penetrate the passenger compartment. The equations of motion are formulated, and the structural stiffness characteristics measured, by similar methods to those used for the frontal model.

The rear end structure is represented by a single stiffness characteristic, since this has been found to be satisfactory due to the general simplicity of the rear end structure and the limited amount of information required from the simulation. This model has been mainly used to check a design for conformance with U.S. fuel system integrity regulations. Again the occupant has been represented by a single mass and the occupant simulation programme may be used to check conformance with the proposed dynamic testing of seat and head restraints.

VEHICLE TO VEHICLE IMPACTS

It is possible to combine the previously described models to represent simple cases of vehicle to vehicle impacts.

Computer simulation of head-on car to car impacts has been considered by assuming that the interface between the vehicles remains effectively plane. That is, there is no local intrusion of one vehicle into the other. Although this is a simplification of the actual conditions, the difference in the effects upon the occupant and the passenger compartment would appear to be small. This simplification can only be justified for vehicles which are not grossly dissimilar. Consideration is being given to the modifications of the crush testing procedure to provide input data that will enable the model to simulate the "hard" and "soft" spot interaction and off-set impacts. The car to car model has been used for an investigation into vehicle aggressiveness during frontal impacts.

The frontal and side impact models are also currently being combined to simulate car front to car side impacts. In addition it is planned to develop this model to accommodate angled side impacts.

CRASH VICTIM ANALYSIS

Initially British Leyland started to develop an occupant simulation programme from first principles. A simple programme was completed but because of the enormous amount of effort that would be required to develop a 3-dimensional version it was decided to acquire a commercially available programme which was completely developed.

The model finally chosen was the 3-dimensional crash victim simulation developed by Calspan Corporation. This model is a generalised 3-dimensional model enabling an impact between a victim and a set of vehicle contact surfaces and restraint systems to be simulated. The victim can therefore be a vehicle occupant in any initial position or a pedestrian being struck by a vehicle. A detailed description of the Calspan programme can be found in (Refs. 2 and 3).

The vehicle deceleration pulses produced from any of the previous vehicle simulation models can be used as input data, together with the geometry and stiffness of the vehicle contact surfaces. Conventional belt restraints and air bags can be represented. The programme will calculate victim trajectory, points of contact, forces, and the usual injury criteria, enabling a complete design to be evaluated with all the advantages attributed to simulation techniques.

GENERAL COMMENTS

The application of computer simulation techniques has proved to be an extremely useful design aid. It ensures that maximum value is obtained from experimental data, and enables design studies to be undertaken in which many alternative schemes can be considered without the need to produce numerous expensive prototype vehicles. For example, in the case of the front end structural design of the T.R.R.L./British Leyland experimental vehicle some 52 schemes have already been considered. These schemes covered both high speed and low speed impact requirements.

Comparison of actual barrier test data with the predicted results has shown that very good correlation is obtained. The accuracy of the predictions is largely due to the selection of a simulation model which adequately describes the vehicle configuration, and the use of realistic stiffness characteristics which are determined experimentally.

A considerable amount of stiffness data has been collected during the past few years, and this makes it possible to undertake further design studies without the need for lengthy experiments. One area of work in which initial investigations have given interesting results concerns the compatibility of light and heavy vehicles when in frontal collision. It is possible to see the effect of large mass differences and the significance of the structural stiffness/mass ratios. This is an important consideration in view of the fact that many collisions occur between vehicles with large mass differences.

The use of computer simulation techniques, to help analyse the many problems encountered in the design of new vehicles, will ensure that the best possible solutions are obtained. This will clearly result in vehicles being designed which will provide better protection for pedestrians and vehicle occupants in real life accidents.

REFERENCES

1 Finch, P M 'British Leyland/T.R.R.L. Experimental Safety Systems Contract'. Fifth International Conference on Experimental Safety Vehicles 1974.

2 Bartz, J A 'A Three-Dimensional Computer Simulation of a Motor Vehicle Crash Victim'. Phase I—'Development of the Computer Programme'. Calspan Corporation Report VJ-2978-V-1, July 1971.

3 Bartz, J A and Butler, F E 'A Three-Dimensional Computer Simulation of a Motor Vehicle Crash Victim'. Phase II—'Validation Study of the Model'. Calspan Corporation Report VJ-2978-V-2, December 1972.

4 Emmerson, W C and Fowler, J E 'Simulation of Frontal Vehicle Impacts'. Conference on 'Vehicle Safety Legislation—Its Engineering and Social Implications' organised by The Institution of Mechanical Engineers and School of Automotive Studies, Cranfield. July 1973.

APPENDIX

The masses shown on the models are defined as follows:

M_b : passenger compartment

M_o : occupants

M_a : rear axle and unsprung mass

M_e : power unit

M_s : front suspension, wheels and part of front end structure

M_{ap} : dash and 'A' Post structure

M_m : bumper system back-up structure

M_{bar} : mobile barrier

M_{ss} : side structure

M_{pad} : internal padding

The forces developed by the stiffness elements are defined as follows:

F_{sb} : occupant restraint system

F_{rs} : rear suspension

F_p : propeller shaft

F_{fr} : rear half of front end; F_{frl} and F_{fru} refer to upper and lower halves

F_{ff} : front half of front end

F_d : dash structure
F_{em} : engine mountings
F_{gm} : gearbox mountings
F_r : radiator and grille structure
F_w : wheels
F_{ap} : 'A' post and front door structure
F_{bs} : low speed bumper system
F_{re} : rear end structure
F_{do} : side skin panels
F_{di} : side inner panels
F_{pad} : internal padding
F_s : main side structure
F_{fb} : engine tie bar

The displacements of the masses in equations (1) to (5) are defined as follows:

x_b : absolute displacement of passenger compartment
x_o : absolute displacement of occupants
x_a : absolute displacement of rear axle
x_e : absolute displacement of power unit
x_s : absolute displacement of front suspension etc.

THE ROLE OF EXTRUSION DEVICES IN ENERGY ABSORPTION FOR SAFETY

K. DEUTSCH
M.G.J. ELSON
R.W. MOORE
Imperial Metal Industries Limited
Kynoch Works, Witton

INTRODUCTION

For many years Imperial Metal Industries Limited, formerly the Metals Division of Imperial Chemical Industries Limited, but now a separate public company, has been concerned with the manufacture of wrought forms of nonferrous metals, while diversification has since led the Company into the field of plastics technology. It was this background which led one of its research scientists, Dr. Karel Saxl, to the idea of utilizing the energy absorbed in the extrusion process as a means of removing the unwanted kinetic energy released in impacts. The energy-absorbing device or impeder which stemmed from this idea (Figure 1) consists of a simple container, a plastics extrusion cartridge and a ram. On

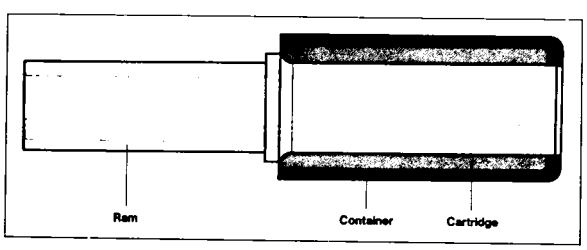


Figure 1. Extrusion Device Before Impact

impact (Figure 2), the ram is forced into the extrusion cartridge and the plastic extrudes, absorbing energy. The devices are very simple and rugged and have no seals or critical apertures which can degrade with age. They are therefore eminently suitable for applications, such as on motor vehicles, where considerable periods of time may elapse between installation and use. They are not recoverable, however, and are unsuited to those applications where repeated operation without attention is essential.

DEVELOPMENT WORK

Following initial privately funded development work, IMI entered a collaborative contract with TRRL early in 1973 to design, develop and evaluate an energy-absorbing bumper mounting device for use

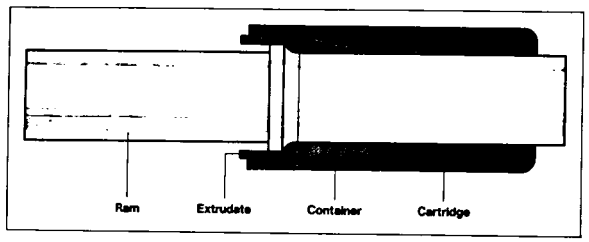


Figure 2. Extrusion Device During Impact

on experimental safety vehicles. A series of prototypes was developed and tested; this involved the evaluation of extrusion-based devices, together with recoverable units based on the barrelling of a plastics tube. Work on the latter has recently been discontinued owing to its low efficiency.

Extrusion devices were designed and constructed for use on the BLMC 1800 experimental safety vehicle on which they are used to back up a "soft front" bumper to give protection at higher impact velocities (up to 10 mph). In this composite system, the bumper provides recoverable protection in impacts up to 5 mph. At higher velocities, the energy removed by the bumper is small and the extrusion device operates to absorb the major part of impact energy.

MAIN CHARACTERISTICS OF THE PLASTICS EXTRUSION ENERGY ABSORBER

The force-displacement characteristic of the plastics extrusion device approximates to a square wave (Figure 3) and the energy-absorbing efficiency of the device is therefore high (approximately 85%). This characteristic is maintained over a range of force levels (Figure 4) and velocities (Figure 5), and for a given device the force level is effectively velocity

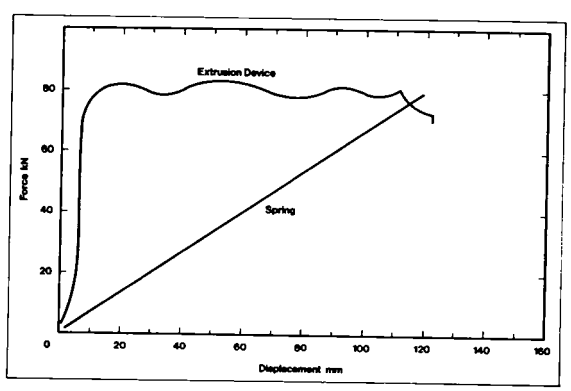


Figure 3. Comparison of Force/Displacement Characteristic of an Extrusion Device and a Spring

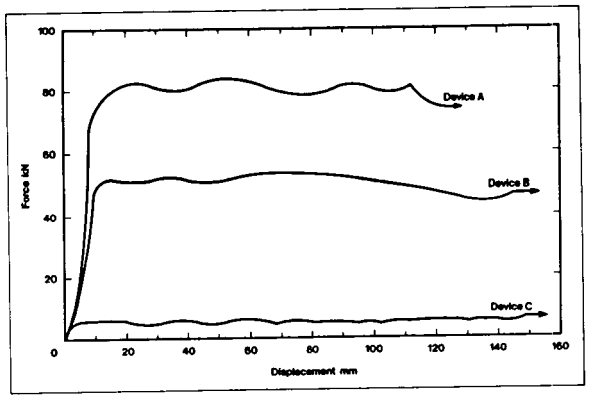


Figure 4. Force/Displacement Characteristics for a Range of Plastics Extrusion Devices

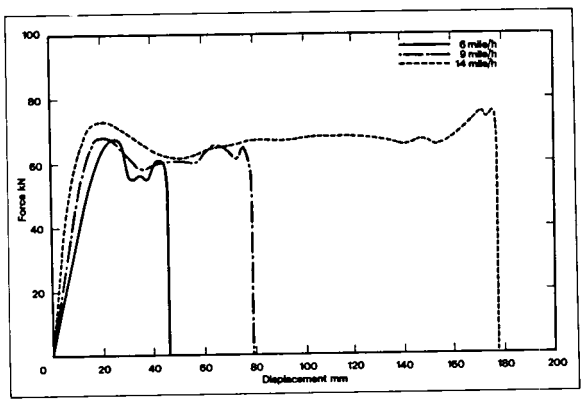


Figure 5. Force/Displacement Characteristics of an Extrusion Device over a range of Impact Velocities (Effective Vehicle Mass 1.20 tonnes with Two Devices)

insensitive. With correct choice of plastics material, the temperature sensitivity is not unduly restrictive and the force level for a given device is sensibly constant over the temperature range between −20° C and +20° C, reducing by only approximately 10% as the temperature increases to 40° C (Figure 6).

HISTORY OF THE REQUIREMENT FOR ENERGY-ABSORBING DEVICES

Pressure for legislation to make energy-absorbing bumpers mandatory in the U.S. began in 1965, but it was not until September, 1972, that the first cars so fitted reached the public. Current requirements demand that bumpers should protect safety-related equipment during collisions, front or rear, at 5 mph. Corner impacts at 3 mph must also be survived

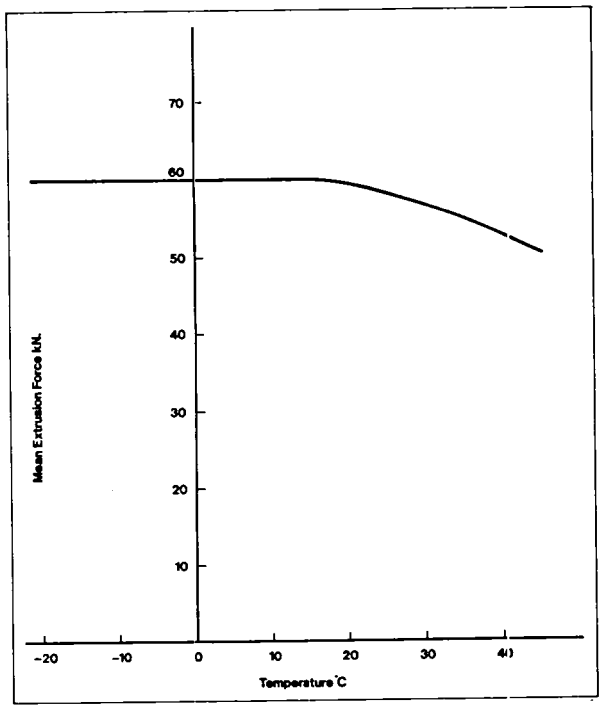


Figure 6. Variation of Mean Extrusion Force with Temperature

without significant damage. In all cases, protection must be afforded during repeated test collisions and adjustment to, or replacement of, the device is not permitted between tests. Requirements in the near future will specify that the car must be completely undamaged after such impacts and it is possible that the speed of impact will eventually be raised, e.g., to 10 mph. Indeed, some states in the U.S. have already suggested this.

Numerous solutions emerged in response to this mandatory requirement, but these can be divided into main categories:

1. The soft front and its variants.
2. The rigid strong bumper attached to the vehicle frame through energy-absorbing units.

The soft front may be a complete vehicle front end moulded in a suitable plastics material or may be in the form of a bulky bumper unit backed by a rigid armature. The former is an expensive solution and demands adequate room behind the frontal structure of the vehicle for energy-absorbing collapse to take place. The second alternative is an elegant but still expensive solution to the problem.

There is little published information on the dynamic behavior of such systems, but Eckhard

Boigk of Bayer AG in an article in Plastvärlden (No. 9, 1974) presented graphs showing the force-displacement characteristic of a typical bumper system for different impact speeds; this work indicated that such systems are somewhat velocity sensitive, i.e., they harden up when impacted at high velocity.

The alternative solution, that of a rigid bumper attached to the vehicle frame via energy-absorbing units, also has a number of variants. Bumper design will not be considered, as this is a major problem area in its own right, but the bumper must be strong enough to transmit the impact energy to the absorbers wherever struck without permanent deformation. The energy-absorbing units may use simple springs, rubber in shear, hydraulic shock absorbers or solid elastomer devices.

The spring has a linear force-displacement characteristic (Figure 3) which is inefficient; the ideal device will have a force-displacement characteristic approximating to a square wave (Figure 3). Some of the hydrualic or solid elastomer devices approach this and have efficiencies in the region of approximately 85%. Simple devices are, however, force sensitive (Figure 7), the force usually rising significantly with increasing velocity of impact. While this is of advantage in very low speed impacts because the frame forces are correspondingly low, it does mean that at high speeds the force rises so much before displacement occurs that the frame or bodywork to which the device is attached collapses before any stroke takes place. Thus, at high speeds no energy is absorbed by the device and the effect on the occupants is as if no protective device had been fitted. Thus, the use of such energy absorbers or soft-front devices does not lessen the need for a crushable vehicle structure to minimize the "g" forces on the occupants during high speed crashes.

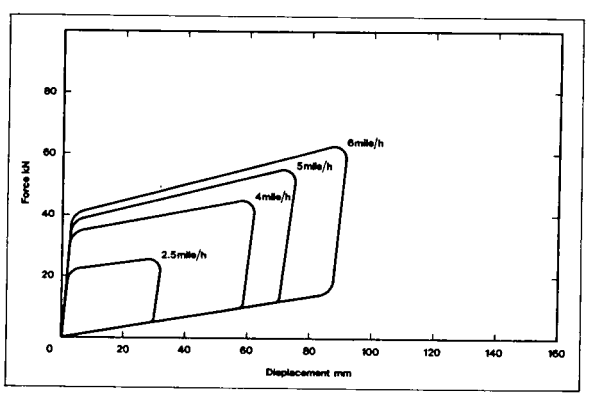


Figure 7. Notional Force/Displacement Characteristics for a Hydraulic Device at Different Impact Velocities

AN ALTERNATIVE SOLUTION

The extrusion device does not suffer from the disadvantage of being velocity-sensitive; up to the limit of the available stroke, the force remains sensibly constant over a wide range of impact velocities (Figure 5). Thus, the use of extrusion devices to back up soft fronts or hydraulic energy absorbers offers an alternative means, less costly to the owner, of energy absorption in high speed crashes, to that of providing a readily crushable vehicle front end. An example of this approach is the system adopted on the BLMC 1800 ESV mentioned above, in which impact energies of up to 10 mph can be absorbed.

It is worth emphasizing the relative energies involved in crashes of various speeds. Because of the square law relationship, the impact energy taken out by a 5 mph device (even if it strokes at all) is only one quarter of the total energy of a 10 mph impact, one sixteenth of that of a 20 mph crash and only one thirty-sixth of that of a 30 mph impact (Figure 8).

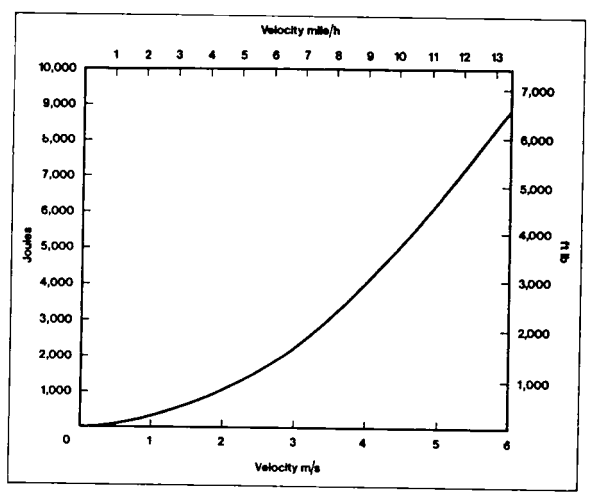


Figure 8. Variation of Kinetic Energy with Impact Velocity

However, it has been calculated that two EA devices stroking 14 inches behind a conventional recoverable bumper system could absorb all the impact energy of a vehicle of effective mass one ton striking a barrier at 30 mph while limiting the structure force to 30,000 pounds per unit. Obviously, stroke, force and vehicle

velocity can be permutated to give the required characteristic.

Clearly, the adoption of such an approach would require integration into the total vehicle design to provide room for the necessary stroke. This need not, however, require a vast bumper overhang; it would, for example, be theoretically possible to design novel forms of vehicles in which the entire front end of the vehicle can slide back under the rear half of hood and fenders in a severe impact with minimal resultant damage. More important, perhaps, is that such controlled reduction in vehicle length should lead to greater safety for the occupants.

Other contributions to the reduction of "g" forces on the occupants in high speed impacts can be provided by suspending heavy vehicle subsystems (such as the engine/gearbox assembly) by means of individual energy-absorbing devices. These can be designed so that they only operate and give benefits in very high speed crashes, and thus play no part in, and hence require no attention after, low speed impacts.

OTHER APPLICATIONS FOR EXTRUSION DEVICES

Extrusion energy-absorbing devices are also well suited for use in advanced seat belt restraint systems (Figure 9) where the constant force characteristic is of value. Current belt systems rely on the stretch in the belt when loaded heavily to permit the body to decelerate gradually. Although very effective, safety belt induced injuries are not unknown and means are being sought to improve belt design. One approach envisages the use of wider, less extensible belts, to reduce chest pressures with energy-absorbing devices at the belt attachment points to provide the controlled travel necessary.

A form of extrusion device may also be considered to permit the controlled collapse of steering columns (Figure 10). Devices may either be mounted coaxially in the steering wheel tube – as shown in the diagram – or alternatively mounted between the steering column and the bulkhead to absorb energy by longitudinal motion. The latter solution eliminates all chance of a coaxially mounted device jamming due to bending of the column in impact.

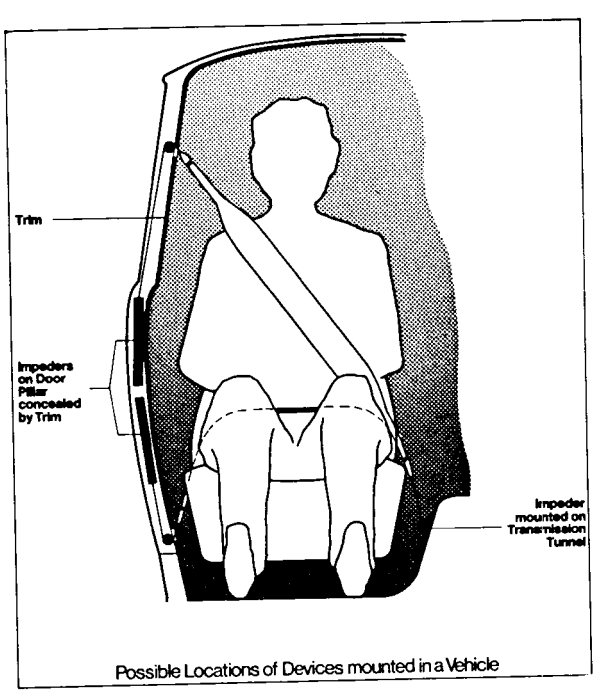


Figure 9. Extrusion Devices Used as Seat Belt Force Limiters

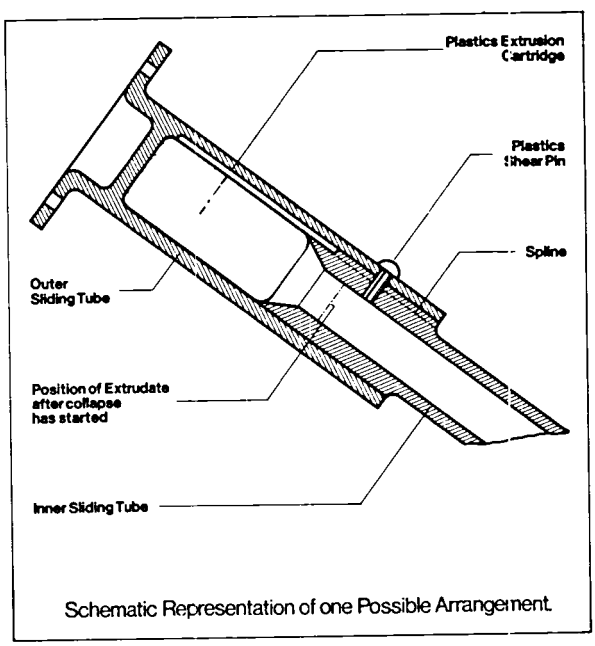


Figure 10. Steering Column Device

SUMMARY

Although the current U.S. legislation for bumper energy-absorbing systems requires the use of recoverable systems, there are many areas where one-shot devices may be very cost effective in improving the safety performance of the vehicle. In particular, an extrusion device behind a soft front or recoverable shock-absorber system may offer the cheapest means of providing vehicle and passenger protection in high speed impacts.

THE DUNLOP MK II COMPOSITE ENERGY ABSORBING BUMPER SYSTEM

DR. D. A. NEWTON
Manager, Advanced Products Group
Dunlop Limited

SUMMARY

Legislative action on safety standards for motor vehicles undertaken in recent years has resulted in considerable emphasis being placed on the reduction of the damageability of automobiles under low speed impacts. The effective control of the impact energy in such low speed impacts also results in the minimization of the deceleration experienced by the vehicle occupants.

The MK II composite energy absorbing bumper mountings developed by the Polymer Engineering Division of Dunlop Limited are designed to rigidly support a conventional bumper bar, and to deflect under impact in such a way as to absorb the energy associated with low speed collisions with optimum efficiency over a wide range of operating temperatures. The design takes advantage of recent Dunlop Research Division developments in the fusion bonding of plastic and rubber in order to produce light weight and cost effective units which will eliminate vehicle damage without significantly increasing the vehicle weight or affecting the vehicle operating costs.

This paper describes in detail the development and performance of the Dunlop energy absorbers in relation to current and anticipated safety standards for motor vehicles. Further developments concerning energy absorbing flexible bumper bars and "soft front ends" are also discussed.

INTRODUCTION

Consideration of vehicle safety and operating economics have highlighted the desirability of designing motor vehicles to be capable of withstanding low speed impacts with minimal damage. The vehicle should withstand such low speed impacts with no damage to components which may affect its safety in operation.

For example, the U.S. Federal Motor Vehicle Safety Standard 215, 1974, requires the automobile to withstand barrier and pendulum impacts at 5 mph, and a pendulum impact at 3 mph on the corner of the bumper at a 30° angle, without damage to safety related components, e.g., lighting, fuel systems, door locks, exhaust, steering and braking systems. It is likely that the 1975 Federal regulations will specify no vehicle damage whatsoever under these impact tests. Certain European automobile manufacturers have also recognized the desirability of protection against vehicle damage under low speed impacts.

These impact standards involve more kinetic energy than can reasonably be absorbed without permanent damage to the vehicle structure, and there is consequently a requirement that additional energy absorption capability be incorporated in the vehicle bumper system.

This additional energy absorption can be provided either by mounting discrete energy absorbing units between the rigid bumper bar and the vehicle structure, or by the development of fully flexible bumper systems.

As a first step, Dunlop Polymer Engineering Division has developed the composite MK II energy absorbing bumper mounting. This mounting is designed to rigidly support a conventional bumper bar. Under low speed impacts, the mounting will deflect and absorb the impact energy with optimum efficiency over a wide range of operating temperatures. The impact loads transmitted to the vehicle structure and occupants are therefore minimized, consistent with the space made available for the deflection of the bumper bar by the requirements of the vehicle styling.

The design takes advantage of recent developments by the Central Research and Development Division of Dunlop Limited in the fusion bonding of rubber and plastics. These fusion bonding techniques have permitted the development of light weight and cost effective units which will allow the elimination of damage without significantly increasing the vehicle weight or affecting the vehicle operating costs.

DESIGN REQUIREMENTS FOR AN ENERGY ABSORBING BUMPER MOUNTING

Let us consider the 5 mph fixed barrier impact test specified by U.S. FMVSS 215, 1974. This represents a total of 30,000 pound-inches of impact energy that must be dissipated for a typical 3,000-pound car.

Of this impact energy, a proportion ranging from 15 percent to 30 percent is dissipated in recoverable deformation of the vehicle structure and systems, while the remainder must be dissipated through the two energy absorbing bumper mountings.

As an example therefore, the bumper mountings must absorb a total of approximately 24,000 pound-inches for a typical 3,000-pound vehicle.

This energy must be absorbed with the minimum level of force transmission through to the vehicle

consistent with the permitted bumper deflection. This implies that the mounting should operate with an optimum dynamic load/deflection characteristic. The area under this curve represents the energy absorbed by the mounting, and should be maximized over the necessary operating temperature range of, for example, –25° C to + 50° C.

In addition the mounting must be capable of adequately supporting the bumper bar under normal driving conditions, withstanding necessary towing and jacking loads, and not adversely influencing the vibration or handling characteristics of the vehicle.

OPERATING PRINCIPLES OF THE MOUNTING

The Central Research and Development Division of Dunlop Limited has recently developed methods of manufacturing low cost rubber/plastic bonded components by direct injection moulding techniques.

Certain combinations of polymers and elastomers will form high strength bonds if the polymer is directly injected into a mould containing a blank manufactured of the elastomer. The moulding process must be carried out under predetermined injection temperatures and pressures in order to obtain the correct mould conditions to form good material bonds.

This production technique has been used for the latest development of the energy absorbing bumper mountings designed and manufactured by Dunlop Polymer Engineering Division.

The rubber/plastic composite construction of the mounting is shown in Figure 1. The preferred method of mounting the bumper on the energy absorbing bumper units is shown in Figure 2.

The smaller end of the unit is attached to the front longitudinals of the vehicle chassis and the wider end is attached to the vehicle bumper; this arrangement helps to spread the impact load and reduce the bending moment in the bumper. The plastic side walls give added strength and stiffness, thus allowing both towing and jacking operations to be performed without excessive deflection occurring in the unit. The width of the mounting is designed to match the bumper inside width and allow the mounting to fold neatly inside the bumper when under maximum deflection.

Under frontal impact the mounting will absorb energy, initially by buckling, and then by continued bending of the composite arms until the predetermined maximum deflection has occurred. The mounting under full deflection is shown in Figure 3.

During deflection, the high resilience rubber constituent stores sufficient energy to return the mounting immediately to its original shape. The "V"

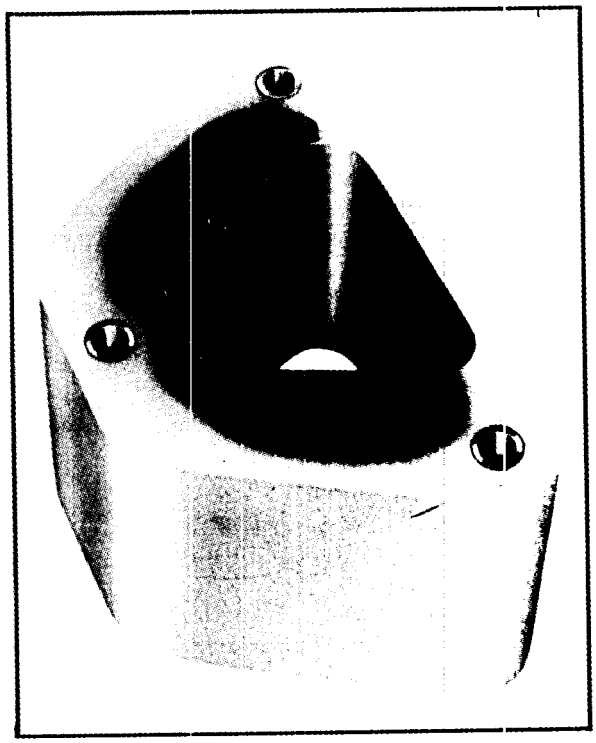


Figure 1

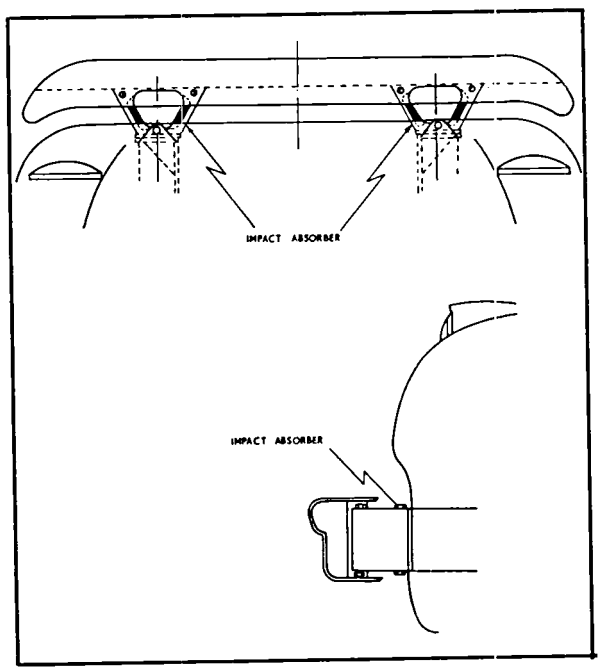


Figure 2

geometry of the unit provides maximum protection in the case of angular impacts, such as the 30° corner impact specified in the U.S. FMVSS 215, 1974.

In the event of an impact above the design specification of 8 km/h (5 mph), excess energy will be absorbed in direct compression of the rubber. The force transmitted through the mounting to the vehicle structure is designed to be a minimum, and

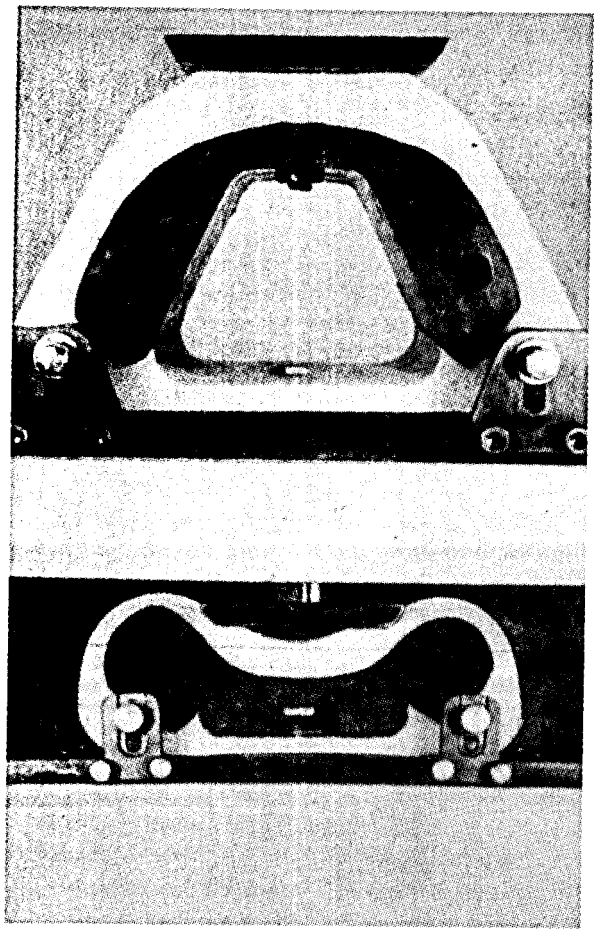


Figure 3

the mounting will therefore operate at maximum efficiency.

A typical dynamic load/deflection characteristic is shown in Figure 4. This characteristic can be adapted to particular vehicle specifications by modifying the ratio and type of polymer and elastomer, and/or modifying the dimensions and overall geometry of the functional components.

The combination of rubber and plastic is carefully designed to provide the minimum variation in performance of the mounting over the necessary temperature range. The range of performance is shown in Figure 5, a graph of efficiency versus temperature, and Figure 6 which shows the

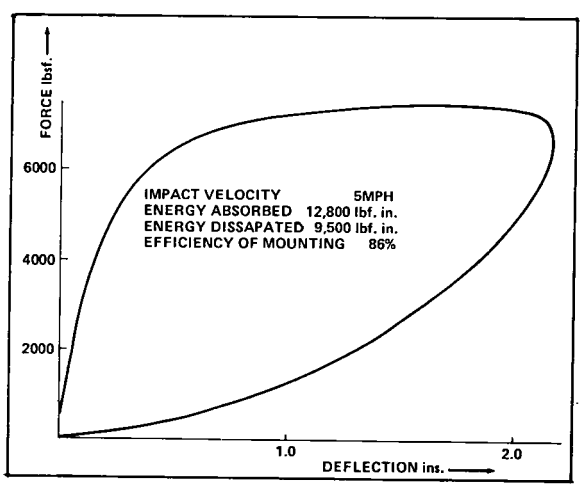


Figure 4

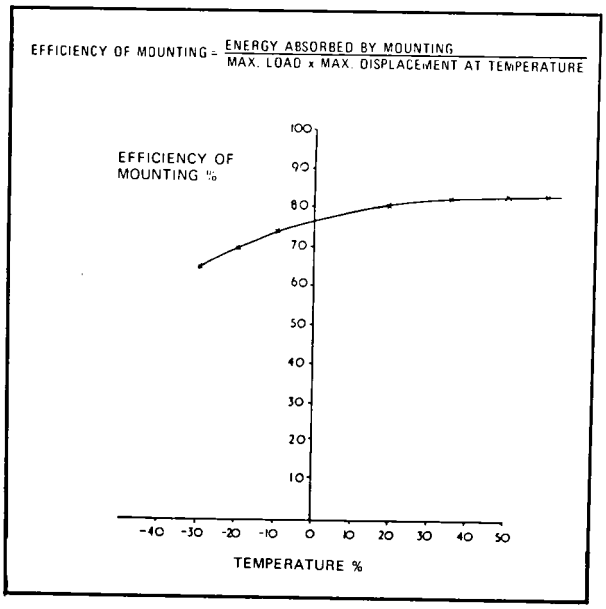


Figure 5

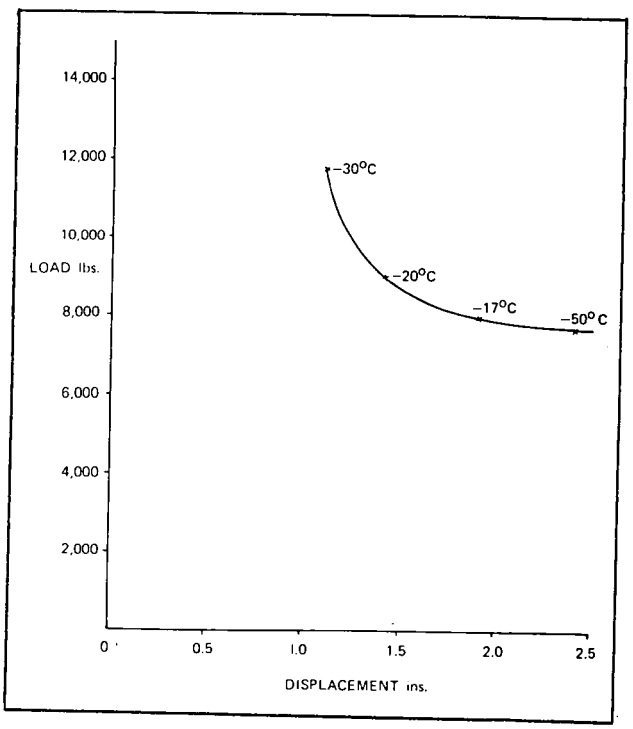


Figure 6

temperature variation in the maximum transmitted load and maximum deflection experienced under impact.

The arrangement of the composite unit allows convenient assembly into the vehicle such that the vertical stiffness of the bumper installation is sufficient to avoid significant vibration of the bumper system during normal driving conditions. The typical stiffness of the mounting in various planes is shown in Figure 7.

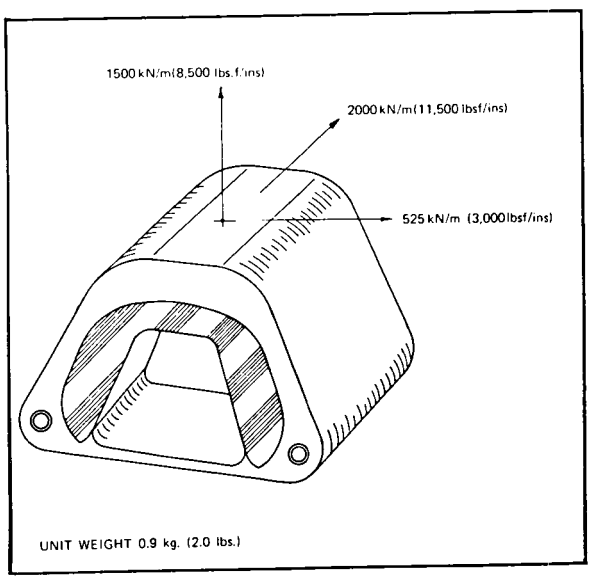


Figure 7

FURTHER DEVELOPMENT INTO RESILIENT BUMPERS AND "SOFT FRONT ENDS"

Such a bumper bar combined with composite laminates making up the front panelwork of the vehicle constitutes an alternative cost effective

The concept of using the high energy absorbing capability and resilience of rubber/plastic composites can be applied to the development of full flexible bumpers and soft front ends. An example of a fully moulded section of flexible bumper is shown in Figure 8.

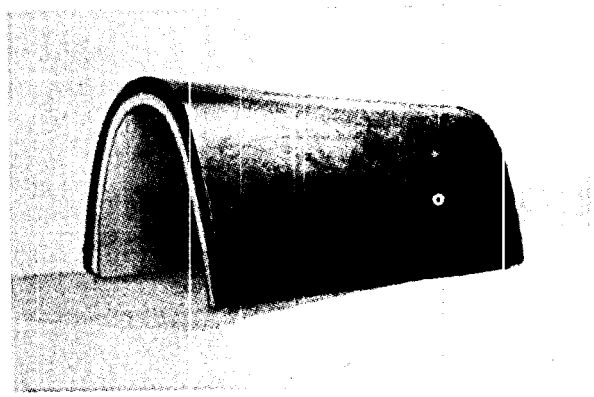


Figure 8

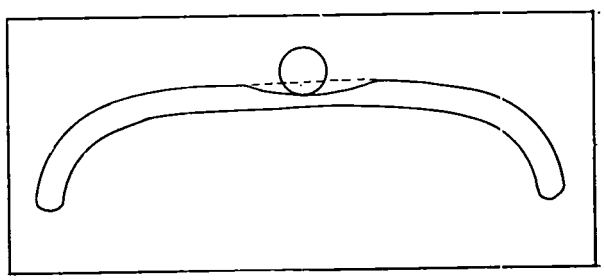


Figure 9

method of overcoming the low speed safety and damageability problem of motor vehicles.

In principle, the fully flexible system is more effective for point impacts since the load is absorbed by the bumper system deflecting over a wider area, as shown in Figure 9.

The operating properties of the composites described in relation to the discrete energy absorber demonstrate that a flexible bumper system, which is fully recoverable after impact, can be developed. This development is now in progress within Dunlop Polymer Engineering Division.

ACKNOWLEDGEMENTS

I wish to thank my colleagues at Polymer Engineering Division, notably Mr. P.J. Cocks, and the considerable contribution by the team at Dunlop Central Research Division led by Mr. I. Cheetham.

I wish to express my thanks to the Directors of Dunlop Limited for permission to publish this paper.

ENERGY ABSORPTION BY VARIABLE SHEAR STRENGTH DUPLEX MATERIALS

R. S. MILLMAN and M. J. CHAPPELL
Wolfson Institute of Interfacial Technology
University of Nottingham

ABSTRACT

Duplex elements for the reinforcement of polymer or metal matrix composites have been produced consisting of convoluted high strength core wires in small bore stainless steel tubing. The interface between the core and sheath is purely frictional and its shear strength is controlled by the local tensile load in the core. As the load on the core increases the shear strength of the interface decreases until the core may be withdrawn at a constant load determined by the geometrical and physical properties of the two components. Experimental composites consisting of duplex elements in a polymer matrix have been made and tested statically and in impact up to 600 ft/sec. A simplified form of duplex device suitable for constant load linear extension has been produced consisting of either crimped steel strips or helically crimped wires clamped in a parallel slot between flat steel plates. Such devices have been produced capable of extending more than 12 inches at substantially constant load at up to 50 ft/sec. Some effects of different frictional materials on the dynamic and static performance of these devices have been investigated. In an alternative form of duplex element, the extension is produced by the controlled tensile collapse of a high strength tube. Experimental composites of this type have also been tested in the laboratory. A number of potential applications, both in the road transport industry and elsewhere, have been examined and a variety of designs for such devices are described.

INTRODUCTION

There are a number of locations within a vehicle structure where tensile extending energy absorbing devices such as those based on the duplex principle (Reference 1) would improve the safety of the occupants in the event of an accident. One of the most obvious examples is the conventional seat belt in which an energy absorbing link could be incorporated to reduce wearer deceleration during impact. Deforming members which essentially operate in a tensile mode could be incorporated within doors as reinforcement against side impact and as general protection in the form of wrap round bumpers; similar devices could also be employed in situations such as that encountered in steering wheel to column attachments where a compressive load can conveniently be converted into its tensile counterpart. In the more general field of energy absorption duplex devices could provide a significant contribution in such diverse areas as the design of motorway crash barriers, aircraft engine containment rings, lightweight body armor, pressure vessels and load and pressure sensitive bolts. Lightweight structural materials with high works of fracture and novel crack stopping properties are also under investigation at the present time (References 2 and 3).

THE DUPLEX PRINCIPLE

A variety of devices and composites capable of extending and absorbing energy in a tensile mode has been developed from the concept of the duplex reinforcing element as proposed by Morley (References 4, 5 and 6). As the name implies, duplex reinforcement consists of two parts, an inner or core element and an outer or sheath element. Two forms of reinforcement have been produced so far, both operating on the principle of the variable shear strength interface but utilizing different mechanisms, which will subsequently be designated as either duplex or dupile.

Duplex Elements

In the form most extensively investigated and referred to simply as the duplex system (Reference 7), the core takes the form of a convoluted strip or wire which is arranged to be in frictional contact with the inside of a tube or sheath element (Figures 1a to 1c). The shear strength of the interface between core and sheath is purely frictional and is maintained by elastic strain in the two components. Because of the convoluted form of the core, the elastic force and hence frictional force will be reduced if a tensile load is applied to the core. The effect of this reduction in interfacial friction with an increase in tensile load is that the pull-out load cannot exceed a limiting value $L_{(max)}$ determined by mechanical and geometrical properties. Since the shear strength of the interface is inversely proportional to the tensile stress in the core, the relation between the pull-out load and the length of interfacial contact with the tubes takes the form of an exponential (Figure 2). It follows from this that unless the coefficient of friction increases drastically the core cannot be broken however long it is or however far it is pulled out.

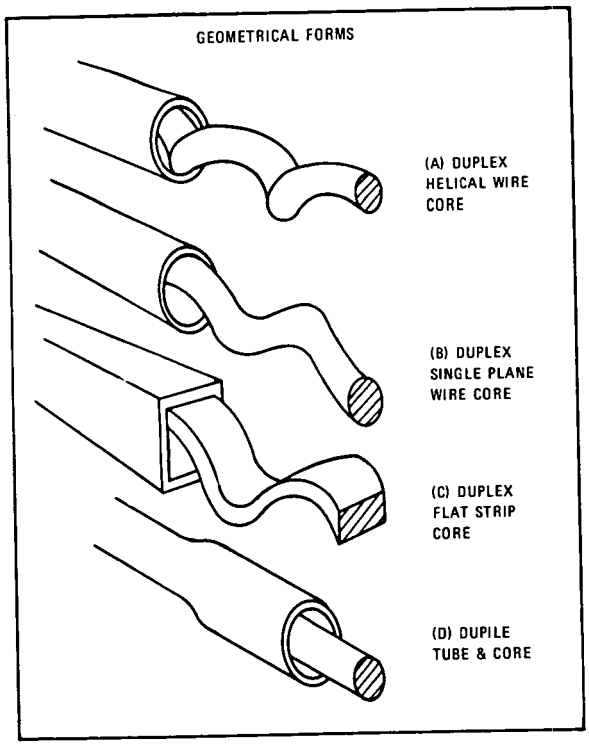


Figure 1

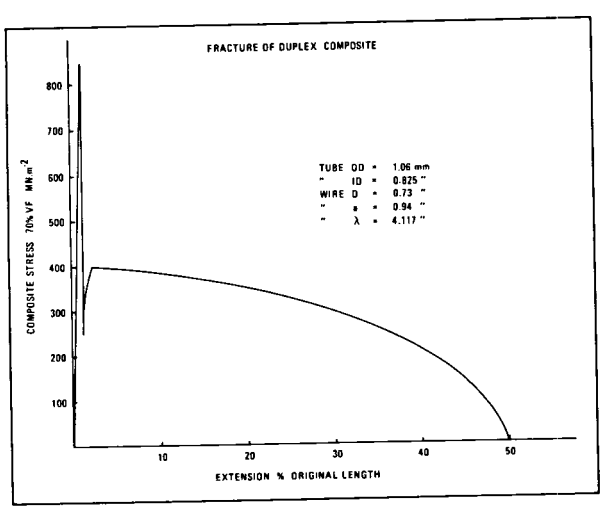


Figure 2

"Dupile" Elements

An alternative form of duplex combination which relies on the limited ductility of a high strength steel tube to provide the debonding mechanism has been examined and is referred to as the "dupile" system (Reference 8) (Figure 1d). In this configuration the main load bearing element is a metal tube with a core wire or tube loosely running through it (Figure 3). The outer tube may be a reinforcing element of an orthodox composite or it may be a large self-contained energy absorbing unit with attachment points at each end (comparable with an invertube). In

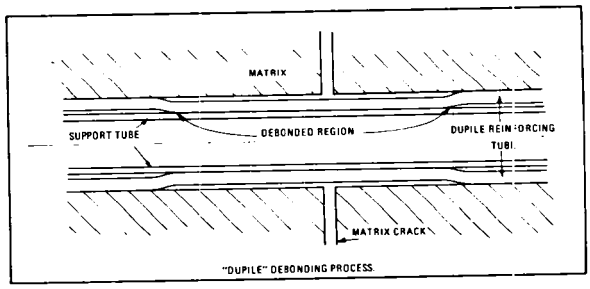


Figure 3

all cases, however, the geometry and material of the load bearing outer tube must be so arranged that when a tensile load is applied to it, it collapses onto the loose core element before rupturing.

The presence of the core element prevents complete rupture of the outer tube by initially limiting radial collapse and then by carrying some of the tensile load across the frictional interface produced between the two components. Further application of a constant extending load causes the collapsed region to spread out from the point of initial failure until all the available tube length has deformed. A typical load extension curve for a dupile tube system in a resin matrix is shown in Figure 4.

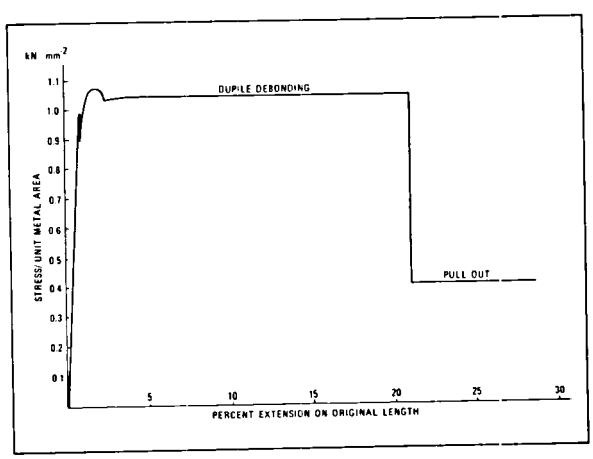


Figure 4

The dupile extension process takes place at constant load whenever there is tube and core material available for deformation. As the extension of a dupile element in a resin matrix proceeds, eventually a point is reached at which the area of tube resin interface is too small to support the tube deformation load and consequently it fails in shear. The area of tube where this takes place is still in frictional contact with the resin and so extension continues at a reduced load determined by the frictional constants until the element is pulled out (Figure 4).

APPLICATION OF DUPLEX MATERIALS TO ENERGY ABSORBING DEVICES

Although conceived initially as a means of improving the anistropic properties of composites, it was soon realized that duplex materials would be capable of absorbing energy very effectively where stresses might be applied to them in tension. Where the absorption of energy is the primary consideration, duplex materials may be conveniently described as either "composite" or "model" devices as illustrated in Figure 5.

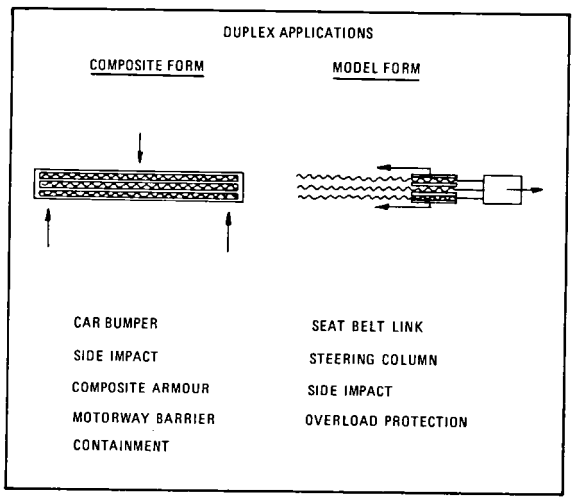


Figure 5

Composite structures are those in which both tubular elements and cores run the whole length of the device. In the case of duplex composites, both tubes and cores carry load elastically up to a predetermined value, after which the tubes break and load is carried by the cores through the frictional interface between core and tube. For dupile composites the tube alone carries the load until the elastic limit is reached, after which the tube and core share the load over the collapsed region during the course of extension.

The load-extension relationship for duplex material would be of an exponential form whereas the dupile material would extend at constant load. In the duplex case extensions of up to 50% total original length would be possible; the corresponding figure for dupile might be approximately 20%. In both systems the effective extensions measured over the loading length could be far more than these values since material not initially loaded could be drawn into the stressed region. In the case of duplex composites, the load to initiate extension would probably be higher than that to continue pull-out because of the load required to rupture the sheath elements. This property confers an almost unique combination of high yield point and long elongation at high stress on composites made from duplex elements. Potential applications of such composites might include tensile loaded bumpers, door reinforcement, motorway barriers, etc.

In many situations it may be possible to predict exactly the direction in which a device will be required to extend and hence to use a simplified duplex form, here referred to as a model system. The main feature of this system is that, while the load may be applied initially over a very short gauge length, the device is still capable of extensions much greater than this initial length. This feature which is shared in different degrees by all duplex devices arises from the fact that material not initially within the loading length may be drawn into it and made to carry stress.

In addition, since not all the core element is situated within the sheath initially, it may be made to undergo plastic as well as elastic deformation during pull-through thereby operating at a higher load than would otherwise have been possible. Furthermore, it is possible to control the load extension characteristics during the course of pull-through by altering the geometry of the system or the number of elements in operation at any given time. An additional feature which makes possible the construction of a compact device is the fact that the loads may be applied directly to the core rather than through a relatively low shear strength frictional interface. Some examples of the forms of pull-out characteristics which might be available from different duplex or dupile systems are shown schematically in Figure 6.

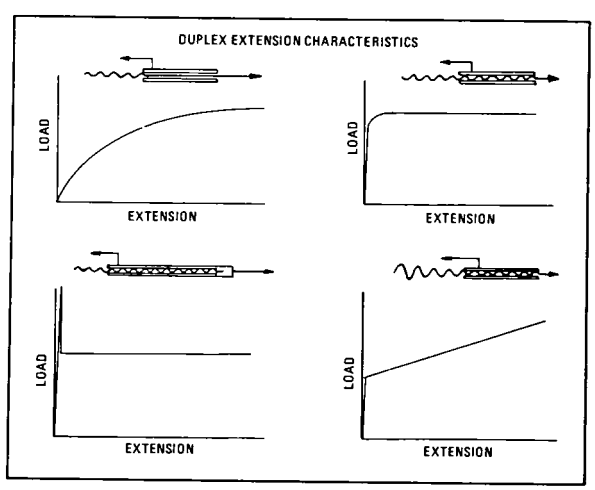


Figure 6

EXPERIMENTAL DUPLEX COMPOSITES

Static Tests

Simple duplex composites (Table 1) were made by first casting a single layer of hypodermic tubes in an epoxy resin matrix. Lengths of piano wire were crimped between cogs for just less than half their length (Figure 1b) and then pulled into the tubes using the uncrimped half as a lead in. Since this method of production resulted in the core interacting more strongly with the tube at one end than the other, some of the crimped core was left protruding from the pull-in end of the tubes and some of the

TABLE 1

	Wire	Tube
Mean OD	0.73 mm	1.06 mm
Mean ID	-	0.825 mm
U.T.S.	2530 N/mm^2	1530 N/mm^2
Wavelength	4.117 mm	-
Amplitude	0.94 mm	-
Pull-out load	840 N	
Composite pull-out Stress at 70% V.F.	650 N/mm^2	
Matrix	Ciba epoxy resin type CT200 / HT901 hardener	

lead in wire was crimped at the other end. This expedient ensured that after breaking the tubes at their mid point and extending a few centimeters, the pull-out load was approximately the same in both directions. The problem of the extending load varying with direction of pull-out could be overcome by drawing tubes down onto a crimped wire, for example; in fact specimens made in this way have been tested and do exhibit the same pull-through characteristics in both directions. These specimens have so far not been tested dynamically.

The majority of static and dynamic tests carried out took the form of a modified bend test in which the ends of the specimen were fixed in freely rotating grips so that they were maintained at a constant distance of separation. This test configuration was chosen because it approximates to the conditions which might be encountered by a protecting panel subjected to a normally incident impact. The behavior of specimens in this test was essentially tensile, after a crack had been initiated at the mid-point by bending.

The relationship between penetrating load and distance is shown in Figure 7, where the dotted curve illustrates the optimum performance of the systems assuming that the maximum pull-out load was realized from the start of deformation. In fact most of the specimens only attained this value after some length of core wire had been pulled in from outside

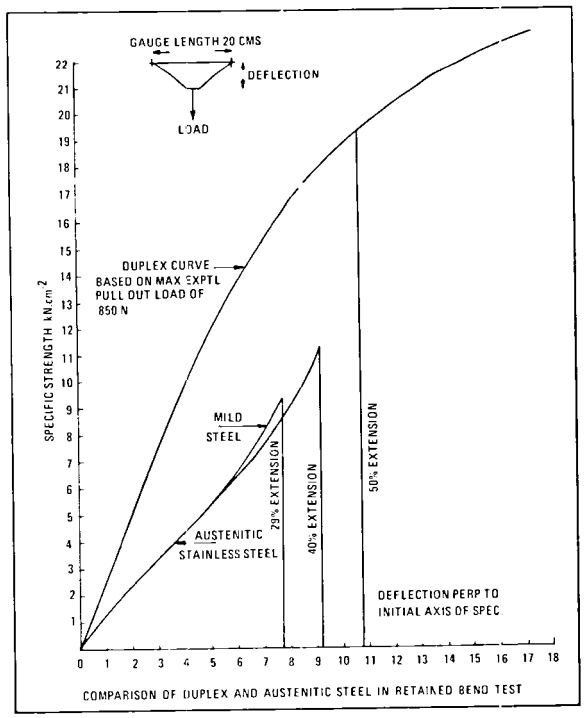


Figure 7

the ends of the tubes (see previous section). Also included for comparison in Figure 7 are various other conventional materials of which the most effective is seen to be austenitic stainless steel. It should be pointed out that difficulties always arise when attempting to make comparisons between duplex and conventional materials in terms of extension. The reason for this is that ductile metals show a reduction in extension with increased dimensions whereas duplex materials can often be made to extend more than the original loading length by drawing in core material from regions of the specimen not originally under load.

Dynamic Tests

Dynamic tests were carried out on duplex composites suspended in the same way as for the static tests. The retained bend test rig was modified so that a cylindrical aluminum missile could be projected at the mid point of a strip specimen and maintained in contact with it by means of a split guide tube until brought to rest. The missiles were projected at up to 230 m/sec from a high speed impact "air gun" made available by Rolls Royce Limited, Hucknall. The gun consisted of a 25 mm ID barrel 5,000 mm long connected to a gas receiver which could be pressurized up to 700 kN/m^2 with nitrogen.

The gas was confined to the receiver by means of a melinex diaphragm separating the receiver from the barrel. The missile was accelerated by firing a

detonator taped to the diaphragm which allowed the compressed gas in the receiver to escape into the barrel and propel the missile along it. The velocity and energy of the missile could be varied independently by altering the weight of the missile and the gas pressure accelerating it.

Two sets of cylindrical aluminum missiles 34 mm long and 24 mm diameter were available; one set was solid weighing 53 gm and the other set had been drilled out to weigh 14 gm. Tests were carried out on duplex materials and on strips of austenitic stainless steel of comparable dimensions; additionally tests were fired using missiles with a 3.8 mm radiused end.

The results for duplex materials when impacted by a cylindrical sharp ended missile are summarized graphically in Figure 8; here a comparison has been made between energy absorbed in the impact test and that which would have been absorbed for the same deflection in a static test. The dynamic energy has been taken to be the same as the kinetic energy of the missile calculated from its velocity as measured just prior to impact. After the tests the specimens were removed and the extension measured. An estimate of the energy which would have been absorbed in a static test producing the same extension was then obtained by comparison with the average result of 5 static tests carried out on similar specimens. The results shown in Figure 8 indicate a reasonable agreement between static and dynamic behavior. In dynamic tests there would appear to be a reduction in energy absorbed per unit extension compared with that which would have been absorbed in static tests.

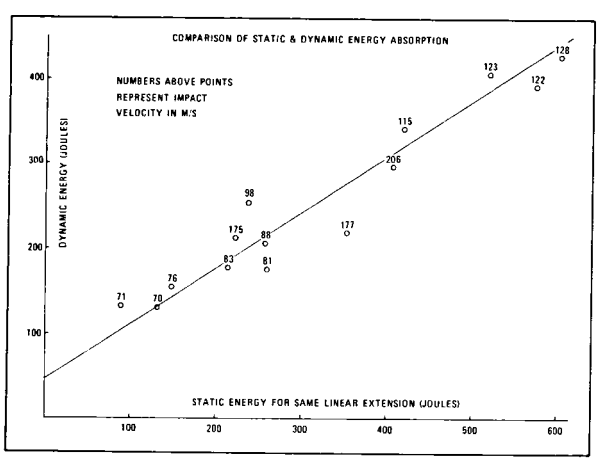


Figure 8

It may be possible to account for this in some cases by the fact that not only did the specimens fail at the point of impact but also near the clamps, evidently because of the inertia of the clamping system. The result of this would have been to reduce the pull-through load to not more than 70% of the maximum value since there was in this situation only about 5 cms of tube wire interface providing the frictional force.

The type of missile, rectangular or radiused, used in the tests had a completely different effect on the two types of material examined (Table 2). The round ended missile which produced good elongation and high energy absorption in the austenitic steels was hardly retarded by the duplex material, since the missile shattered the resin, pushed the wires and tubes aside and then passed straight through. On the other hand, the stainless steel was unable to absorb much energy when struck by the sharp missile because the stress concentrating effect of the sharp end severely limited extension and reduced the energy absorbed by a factor of 0.38 compared with the radiused case.

TABLE 2

Specimen	Missile	Missile Velocity m/sec	Just contained energy Joules	Energy per unit x section J/mm	Energy per unit wt. J/gm
Austenitic Steel	Sharp	70	135	14.4	1.98
Austenitic Steel	Radiused	116	360	38.7	4.90
4 Element Duplex	Sharp	88	200	40.0	7.65
10 Element Duplex	Sharp	127	425	34.0	6.40

When duplex material was struck by the sharp ended missile, the resin was first shattered after which the hard wires cut into the edge of the missile and held the two together until the missile had been brought to rest. Since the problem of maintaining contact between missle and duplex elements is not a fundamental one nor does it shed any light on dynamic pull-out behavior, the majority of tests on duplex samples were carried out using a sharp ended missile. In this case, the performance of duplex material was most impressive, the unique combination of hard wires and long elongation giving up to 1.5 times more energy absorption per unit weight than the best stainless steel.

Comparisons between duplex materials and ductile metals in terms of energy absorption are complicated by the effect of gauge length and geometry on maximum available extension. For small scale tests this is particularly noticeable where, for duplex devices, material outside the clamped region may be drawn into the stressed region producing extension of more than 50%; the maximum extension available from ductile metals under these conditions is, however, limited to the stressed region alone. With much larger specimens this effect would be reduced

in both cases, when the maximum extension of duplex materials would not exceed 50% and that of ductile metals would probably be less than 30%, particularly when loaded under impact conditions.

The main purpose of the dynamic tests just described was to establish whether the duplex process was capable of deformation at high velocity and, if so, whether duplex materials offered any advantage over conventional ones. It seems clear that duplex composites do have some advantages over conventional materials, even in the crude form so far tested, and it would appear that by making the best use of their unique combination of properties that they are potentially very important in energy absorbing structures.

DUPLEX MODEL SYSTEMS

The model form of duplex material was developed and tested with the specific application to a seat belt energy absorbing link in mind. For this reason, the investigation was confined to devices engineered in a way likely to be acceptable to the motor industry. Although systems utilizing hypodermic tubes were not completely ruled out, a simplified arrangement was thought to be necessary for this work.

The standard device adopted consisted of ground mild steel plates clamped together to produce a parallel slot through which convoluted elements could be pulled. The plates were approximately 75 mm long, 50 mm wide and 6.5 mm thick; they were clamped together by 6¼ inch B.S.F. screws at a fixed separation which was controlled by spacer shims. The convoluted elements took two forms, one produced by drawing out tightly coiled piano wire into the form of an open helix and the other by rolling flat spring steel strip between a pair of crimping rollers.

A series of experiments was carried out to investigate the effect of wavelength, amplitude and slot gap width on the steady load required to pull the convoluted elements through the slot between the plates. Each experiment was carried out with a fresh wire or strip and using a fresh area of the slot. The plates were reground from time to time when necessary.

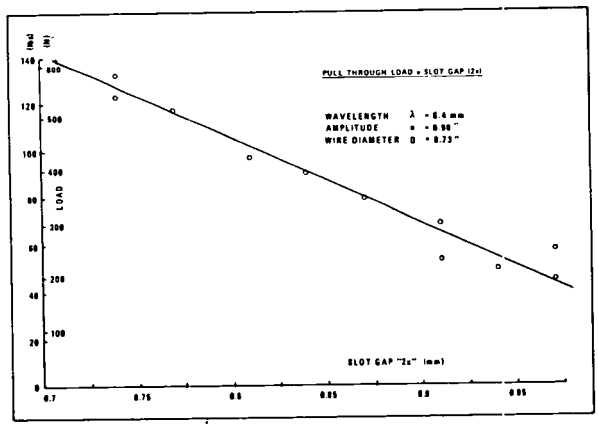


Figure 9

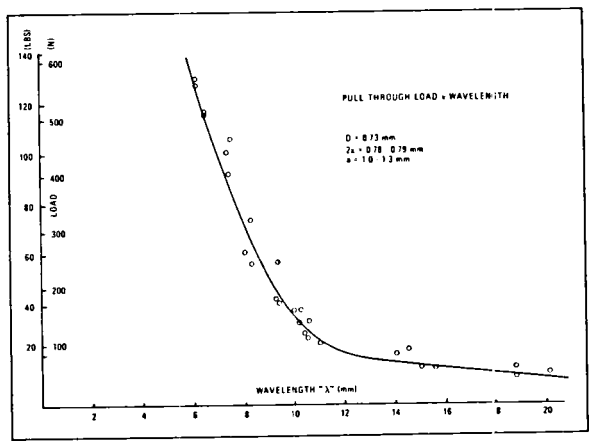


Figure 10

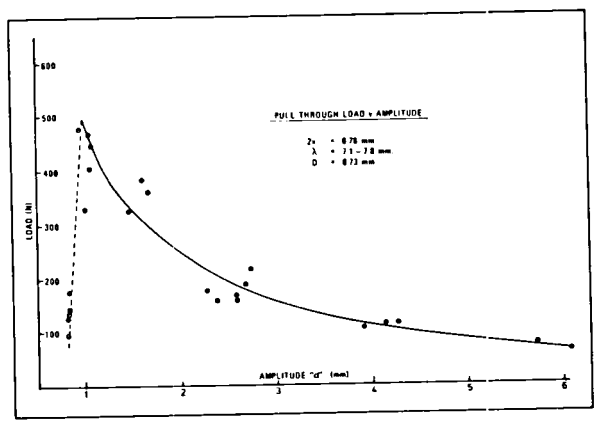


Figure 11

HELICAL WIRE CORES

Some of the results of this investigation, in the case of helical wire cores, are shown in Figures 9, 10 and 11. It can be seen from these results that the relation between slot gap width and pull-through load is fairly linear, with a reduction in gap predictably giving an increase in pull-through load. The relation between pull-through and wavelength is, however, far from linear, with wavelengths of 6 mm producing pull-through loads approximately eight times those obtained from a 20 mm wavelength core.

Unfortunately, it is not possible to make a quantitative comparison between the pull-through loads generated by different wavelengths under these conditions, since the theoretical treatment at present available only refers to the maximum pull-through load for a given system. In most cases the

pull-through load obtained with 75 mm long plates is unlikely to be as high as the maximum theoretical value $L_{(max)}$; this is particularly so for the long wavelength cores where the interfacial contact length to reach $L_{(max)}$ might be many times longer than the length of the plates. A further factor which makes comparison more difficult is that the short wavelength elements suffered some degree of plastic deformation during pull-in, whereas the interaction of the long wavelength elements was purely elastic. The effect of plastic work in the short wavelength system would probably be to reduce the maximum pull-through load since plastic deformation would limit the elastic strain developed in the core.

The pull-through versus amplitude relationship (Figure 11) is again complicated by plastic deformation taking place in the core. For large amplitudes the core tends to be straightened out as it is pulled into the gap, so increasing the wavelength and giving a lower pull-through load than would have been expected had the wavelength remained constant. As the amplitude is reduced, the plastic deformation and consequent increase in wavelength is reduced, with the effect that the pull-through load increases sharply. The maximum pull-through load is obtained when the initial amplitude is about 20% larger than the slot gap. Further reduction in amplitude reduces the elastic interaction, frictional force, and hence pull-through load.

A number of the experiments were repeated using hard steel inserts (VPN ~ 470) between the plates as frictional surfaces for the wires. The results collected in Figure 12 show exactly the same qualitative effects as were obtained with mild steel frictional surfaces except that in the case of the hard steel faces the loads were different since the frictional coefficient was different.

Convoluted Strip Cores

In the case of convoluted strip elements, the investigation of the effect of geometry on pull-through load was rather more limited than that of helical wire elements. Only three sizes of crimping cogs were available, giving 4, 7 and 10 mm wavelength respectively; of these the 4 mm cogs were not capable of crimping the 0.5 mm strip and very little amplitude variation could be obtained using the 7 and 10 mm cogs without breaking the 0.5 mm strip. Although the main factor investigated was the effect of gap width on pull-through load, the effect of wavelength and strip thickness was also examined. The general trends which were identified from this work were similar to those observed in the case of the helical wire cores.

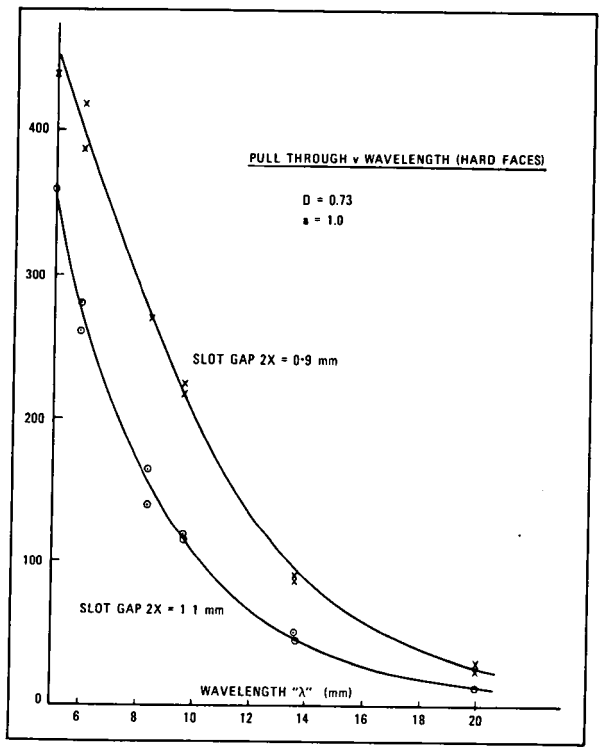


Figure 12

The behavior of the convoluted strips between mild steel plates was frequently very erratic. On some occasions the pull-through load could increase by a factor of two during the course of a long pull-through. As a result of this behavior and on the rather poor results obtained in initial dynamic testing, the investigation was extended to other materials for use as frictional surfaces. Three other materials were examined as working surfaces for use with the convoluted spring steel strip; these were aluminium, brass and a commercial clutch plate lining material, type DM1, supplied by Ferrodo Limited. For each of these materials a comparison of pull-through load versus slot gap width was made and the results are shown in Figure 13. The chief differences between these curves may be attributed to the different coefficients of friction existing between the various materials and the convoluted steel strip.

Although not shown in Figure 13, there was considerable scatter on the results obtained with mild steel, brass and aluminium, whereas the results obtained using Ferrodo blocks as frictional surfaces were much more reproducible though significantly lower in value. It is felt that one of the most important areas for further work on long extension duplex devices is the behavior of different frictional surfaces under static and dynamic conditions.

Dynamic Testing

A number of dynamic tests were carried out using an Izod machine suitably modified for tensile impact

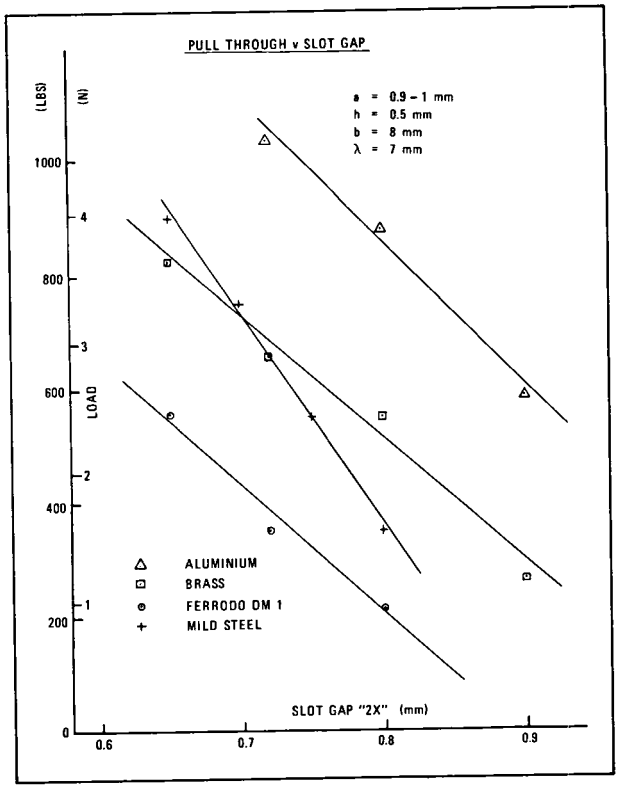


Figure 13

testing. Results obtained using single and looped wires (see "Core End Fixing") at impact velocities of approximately 4 m/sec suggested that there was very little difference between the amount of energy absorbed at this speed and that absorbed under static conditions. Both mild steel and hard steel faces were used but again there was little significant difference between static and dynamic behavior. The range of results obtained is shown in Table 3 where it can be seen that the agreement between load calculated from dynamic energy and that obtained from static tests was quite good.

TABLE 3

Frictional Material	Dynamic Energy (J)	Pull-Through Distance (mm)	Load from Energy (N)	Load from Static Test (N)
Mild steel	149	317	480	545- 635
Hard steel	160	178	915	900-1040
Hard steel	158	191	865	900-1040

(Strips were not tested under these conditions because of gripping problems)

In order to carry out similar tests at velocities up to 15 m/sec (considered adequate for vehicle testing), a pendulum was fitted to the end of the 50 mm ID air gun used by Rolls Royce, Hucknall, for material impact testing. This rig permitted a combined mass of approximately 17 kg (missile 6.8 kg and pendulum 10.5 kg) to be applied to a specimen at up to 15 m/sec.

Two sets of experiments were carried out with this apparatus, one using either wires or strips operating against mild steel faces and the other with the wires operating against hard steel and the strips against Ferrodo type DM1 clutch plate linings (Table 4). The erratic behavior of mild steel faces which had been observed in the static tests particularly with the convoluted strips seemed to cause serious problems in

TABLE 4 Details of Materials and Geometry

	Strip		Wires	
	Preliminary	Final	Preliminary	Final
Material	0.8%C steel	0.8%C steel	12 mwg wires	12 mwg wires
Number of elements	1	3	8	17
Thickness or diameter (mm)	0.50	0.50	0.73	0.73
Width (mm)	8.0	8.0	-	-
Wavelength	7.0	7.0	7.0	10.0
Amplitude (mm)	0.97-1.9	0.97-1.0	0.93-0.97	1.3
Form of crimp	single plane	single plane	helical	helical
UTS Material (N/mm^2)	1500	1500	2500	2500
Slot width (mm)	0.64	0.72	0.79	0.80
Pull-out load/element (N)	~ 4090	~ 1590	~ 500	~ 270
Frictional material	Mild steel	Ferrodo friction lining type DM1	Mild steel	Spring steel 0.8%C steel

the high speed tests. In a number of tests both strips and wires broke although the pull-through loads expected from static experiments were less than 70% UTS and 50% UTS for strips and wires respectively.

In the second series of tests in which hard steel faces were used for the wires and Ferrodo linings for the strips, the pull-through loads were further reduced to about 25% UTS to minimize the danger of rupture.

The high speed tests were all recorded by a high speed camera which filmed each shot at approximately 3,000 frames/sec. The films were subsequently analyzed to obtain values of missile impact velocity and pendulum initial velocity. After each test the pull-out distance was measured in order to calculate the energy which would have been absorbed in a similar static test. The pull-out load required for this calculation was measured on the Instron testing machine before each test in the case of wire specimens, but was taken as an average value of previous static tests in the case of strip specimens. The pull-out distance, load, and calculated static energy are given in Table 5, which also includes values of the dynamic energy obtained from the initial velocity of the pendulum.

Examination of the high speed films revealed that although the missile eventually came to rest at the same time as the pendulum, the two were not in contact the whole time during deceleration. Values

TABLE 5

Test Number & Type		Gun Pressure (kN/m²)	Vel. of Missile (m/sec)	Vel. of Pendulum (m/sec)	Pull-through Extension (mm)	Static pull-Through (kN)	Static Energy (Joules)	Pendulum dynamic energy (Joules) min.	max.
1	3 strips	275	12.8	7.0	43	6.5*	280	256	423
2	3 strips	345	16.7	10.0	100	6.5*	650	522	863
3	3 strips	412	18.5	11.2	100	6.5*	650	655	1084
4	3 strips	480	22.8	14.0	155	6.5*	1002	1024	1692
5	3 strips	550	26.5	14.0	210	6.5*	1365	1024	1692
6	3 strips	689	29.6	16.7	292	6.5*	1898	1457	2408
7	16 wires	275	17.4	9.5	128	4.5	576	471	779
8	17 wires	345	20.7	10.9	146	5.2	760	620	1026
9	17 wires	412	23.4	14.0	176	5.3	935	1024	1692
10	17 wires	480	24.6	14.6	197	5.5	1080	1113	1840
11	17 wires	550	27.1	14.0	237	5.5	1304	1024	1692
12	17 wires	550	27.4	12.2	198	5.9	1168	777	1285

* Mean of static tests

for the dynamic energy were obtained, a minimum value assuming the pendulum to be acting alone and a maximum value assuming the combined mass of missile and pendulum to be brought to rest simultaneously. Comparison of energy derived from static measurements and that absorbed in dynamic tests suggest that both the wire and strip specimens could be expected to absorb as much energy under dynamic conditions as that predicted from static tests giving the same extensions.

In conclusion, it would appear to be quite feasible to employ duplex devices of the types described as energy absorbing links for seat belts where energy dissipation of the order of 1,000-2,000 Joules is required at impact velocities up to at least 16 m/sec.

APPLICATIONS AND PRACTICAL CONSTRUCTION

Core End Fixing

One of the chief problems encountered in the testing of both wire and strip devices was that of gripping one end of the convoluted element; usually this was done by casting a resin block around the ends of the wires or, in the case of strips, by clamping them between a pair of blocks in whose surfaces convolutions had been machined to coincide with the crimping on the strips. By far the simplest and most effective method discovered was to loop the elements around a circular rod and feed them through the working gap twice. This method of attachment appeared to be equally satisfactory either for strips or wires and it had the great advantage of being very compact compared with other methods. If a device incorporating the duplex principle were to be considered for mass production, a looped fastening would be attractive since it does not involve an expensive joining process.

"Model" Devices

The most obvious potential application of model duplex devices is as a seat belt energy absorbing link. Duplex devices should be particularly attractive for this application since they may easily be constructed to operate at the required loads and extensions under dynamic conditions while still having comparable dimensions to existing seat belt hardware. They could conveniently be installed in any of the anchorage points or even in the buckle and they can be designed to give almost any load extension characteristic that might be required.

A potential application for either composite or model duplex devices exists in the reinforcement of car doors to withstand side impact. Duplex extending devices or composite strips could be fixed to the inside of door panels and very useful characteristics could be obtained from the combination since, in this configuration, the resistance of the door panel itself tends to decrease with horizontal penetration whereas the resistance of the duplex member can be made to increase (Figure 7).

Steering columns and wheel to column attachments are positions at which energy absorbing deformations may be designed to take place in collision situations. A possible way in which a collapsible column incorporating a duplex device could be constructed is shown in Figure 14. In this system the duplex cores would be connected by one end to the lower end of the wheel column and would form an anulus of elements around it.

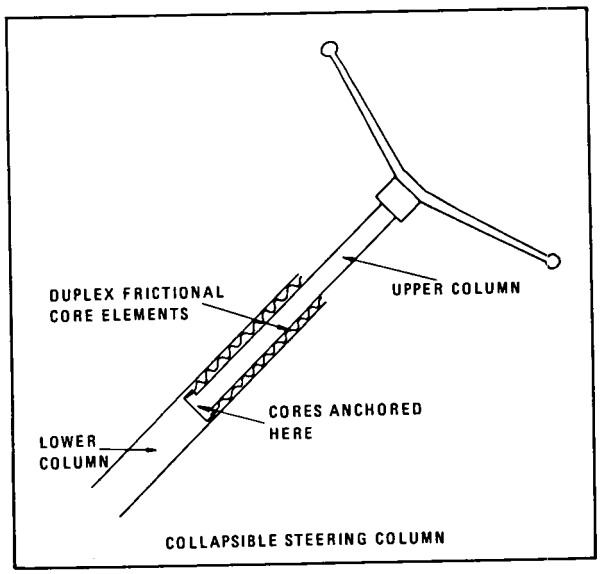


Figure 14

Although the elements would be in frictional contact both with the upper and lower tubes, they would only be able to slide against the inside of the lower column and so would be, in effect, single sided. During collapse, the upper column, together with the cores, would slide into the lower tube at a substantially constant load. No guides, bushes or splines would be necessary since the elements would be capable of carrying the normal loads of driving without deflecting irreversibly. Such a system could conveniently be assembled and tested simultaneously by simply forcing the upper tube down into the lower and measuring the load required.

In some situations it might be advantageous to incorporate a deformable steering wheel to column attachment as illustrated in Figures 15a and 15b. In the arrangement shown in Figure 15a, the wheel is held against the column boss by means of the duplex strips, with the normal driving loads, steering torque, etc., carried by the strips without any pull-out. In the event of an accident, however, should part of the driver's body contact any part of the rim of the wheel, the duplex elements on the other side of the boss will pull out in a predetermined manner. During the course of impact it should be possible to arrange for the resistance of the wheel rim to deflection to vary, in such as way as to minimize the forces imposed on the impacting body. In this way a contribution could be made to the energy absorbed during ride down while at the same time maintaining the load to the face or chest below that necessary to cause serious injury.

An alternative geometrical arrangement in which the duplex elements are located only on the forward side of the column is shown in Figure 15b. In this layout, the duplex elements do not rotate and the

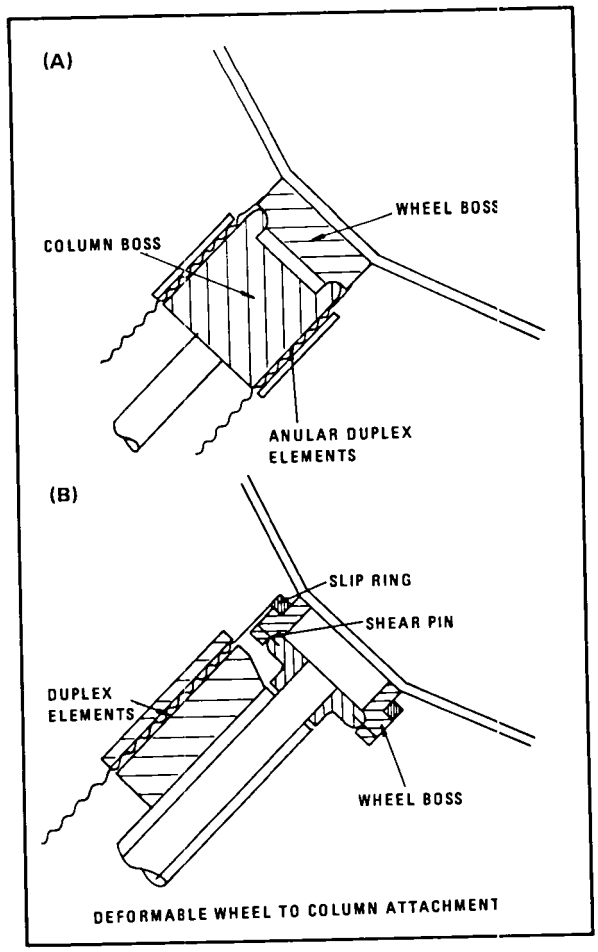


Figure 15

wheel is held in position by separate means. Under impact, the wheel breaks away and its movement is transmitted to the duplex links by means of a slip ring. The deflection of the wheel is then controlled by pull-out of the duplex link.

Composite Applications

Of the potential applications for duplex composites illustrated in Figure 5, the first to be examined was that of containment rings for aircraft gas turbines. At present shields are fitted to large turbo-fan engines around the fan assembly in order to protect the rest of the aircraft structure from damage in the event of a fan blade failure. The rings are usually made from austenitic stainless steel and they represent a heavy and expensive part of the engine which is rarely, if ever, called into operation. It would appear from preliminary work that fiber wound duplex composites could significantly reduce the weight and cost of containment rings without loss of energy absorbing capacity.

A material which is required to behave in a similar way to a containment ring is lightweight composite armor plate. This usually consists of a substantial

sheet of GPR to the front surface of which is attached a layer of special ceramic tiles. Under impact from small arms fire, the tiles break into fragments, at the same time breaking up and slowing down the missile so that a large number of relatively slow moving particles is produced. These in turn are contained by the GRP which absorbs their energy through delamination and fiber pull-out. It would seem quite feasible to use a resin matrix duplex composite as the back-up layer for the tiles in which case one would expect the combination of wire hardness and long elongation at high stress to be very effective in absorbing the energy of the shattered particles.

In the road transport industry, an obvious potential application exists in the construction of vehicle bumpers capable of absorbing large deflections at very high loads. Such bumpers could be constructed in the form of a wrap round sheet of duplex composite running from the front of one wheel arch over compressive energy absorbers at the forward corners of the body, and back to the other wheel arch. The system could be so arranged that in a 2.2 m/sec (5 mph) impact the compressive struts would absorb the energy with the duplex composite strip deforming elastically to accommodate the deflection. In a serious impact, however (with for example, a telegraph pole), the tube elements would fail near the point of impact and the wires would carry the load by pulling out from the tubes at either side. Even in an impact occurring near to the corner of the car it is likely that at least 500 mm of pull-out length would be available.

A further application in which an impact load is applied to a long tensile extending member is the motorway crash barrier. It could be very advantageous to construct these barriers from duplex material particularly in dangerous situations such as on bridge parapets and on bends in the road. By incorporating different duplex devices in series, it would also be possible to vary the characteristics so that, for example, the barrier would increase its resistance to extension with continued penetration and so allow for different vehicle weights.

ACKNOWLEDGMENTS

The work described here forms part of a program supported by NRDC and the Wolfson Institute. The authors are indebted to Professor J. G. Morley for his advice and encouragement and to Messrs. D. M. Braddick and R. Stevens for invaluable technical assistance. Thanks are also due to the Executive Committee of the Wolfson Institute for the provision of laboratory facilities, and to Rolls Royce Ltd. for provision of high speed testing equipment.

REFERENCES

1. "Duplex Energy Absorbing Devices," J. G. Morley, British Patent No. 4061/72.
2. "Transverse Crack Propagation in Fiber Reinforced Composites," J. G. Morley, Faraday Special Discussions of the Chemical Society, **2** (1972), 109.
3. I. R. McColl and J. G. Morley, to be published.
4. "The Design of Fibrous Composites Having Improved Mechanical Properties," J. G. Morley, Proc. Roy. Soc. Lond., A.319 (1970), 117.
5. "Fibrous Composites with Multiple and Variable Shear Strength Interfaces," J. G. Morley, Composites, **2** (1971), 80.
6. "Composite Materials," J. G. Morley, J. British Interplanetary Soc., **25** (1972), 431.
7. "The Duplex Concept," J. G. Morley, British Patent No. 1331431.
8. "The Dupile Concept," J. G. Morley, British Patent No. 35107/71.

REDUCED LACERATION FROM A NEW LAMINATED WINDSHIELD

JOHN PICKARD
Research and Development Manager
Triplex Safety Glass Co., Ltd.

SUMMARY

A new car windshield has been developed by the Triplex Safety Glass Co., Ltd. of England which promises a substantial reduction in facial laceration without adverse consequences in other respects. The new windshield is called Triplex Ten Twenty.

Of laminated construction, using two pieces of 2.3 mm thick float glass and a standard HPR polyvinyl butyral interlayer of 0.76 mm thickness, the two glass components are thermally stressed to different levels. The outer glass is lightly stressed to about 7 $MN.m^{-2}$ for optimum stone impact resistance and the inner glass to 47 ± 3 $MN.m^{-2}$ (center tensile stress) for major reduction in laceration.

Test programs have been carried out at Triplex using a dropping head form and using a skull impactor sled and by Wayne State University in Detroit, U.S.A., using a 50th percentile anthropomorphic dummy on their Wham III crash simulator.

The results of these evaluations are given in terms of Head Injury Criterion, Gadd Index and various laceration scales including a newly developed objective scale – the Triplex Laceration Index, or TLI.

The TLI is a formula based on the number and the measured length and depth of cuts in the dummy head coverings. When compared with conventional laminated windshields Triplex Ten Twenty has been demonstrated to give a reduction in laceration of 2 units on this logarithmic scale.

A reduction of TLI units is equivalent to a reduction of:

99% in the number of cuts of constant size,
or 90% in the length of cuts of constant depth and number,
or 78% in depth of cuts of constant length and number,
if the differently weighted attributes (depth, length and number) are considered separately.

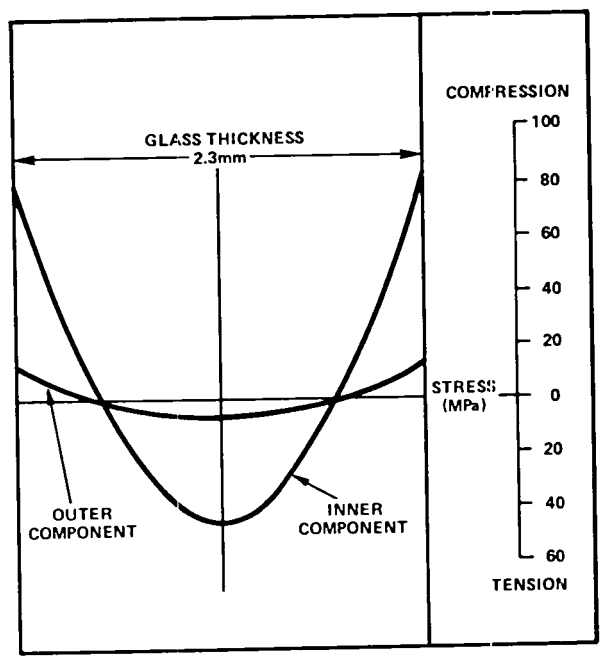


Figure 1. Stress Profiles For Two Components of Ten Twenty Windshield

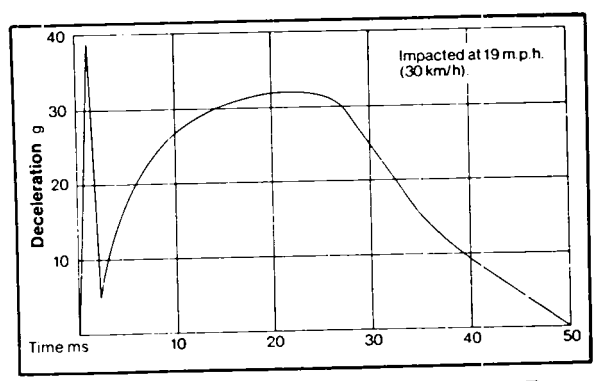


Figure 2. Typical Deceleration Trace for Ten Twenty Windshield

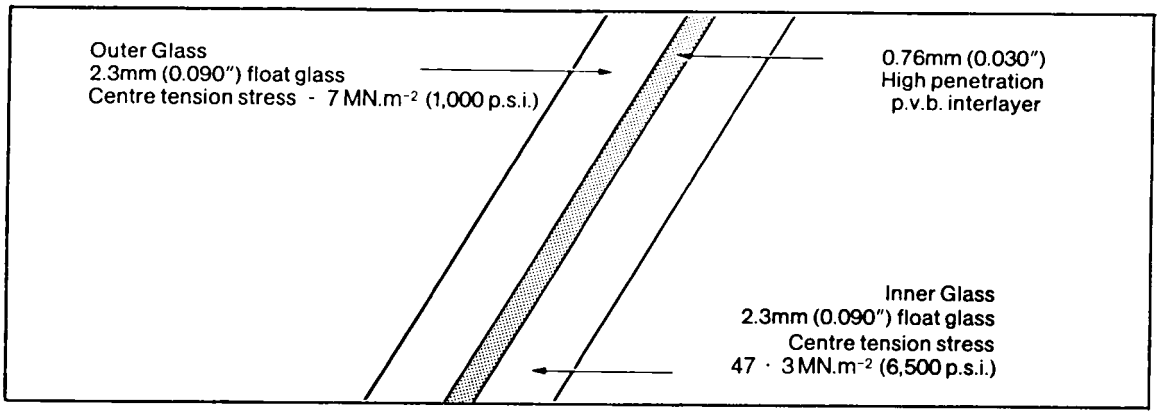


Figure 3. Construction of the Ten Twenty Windshield

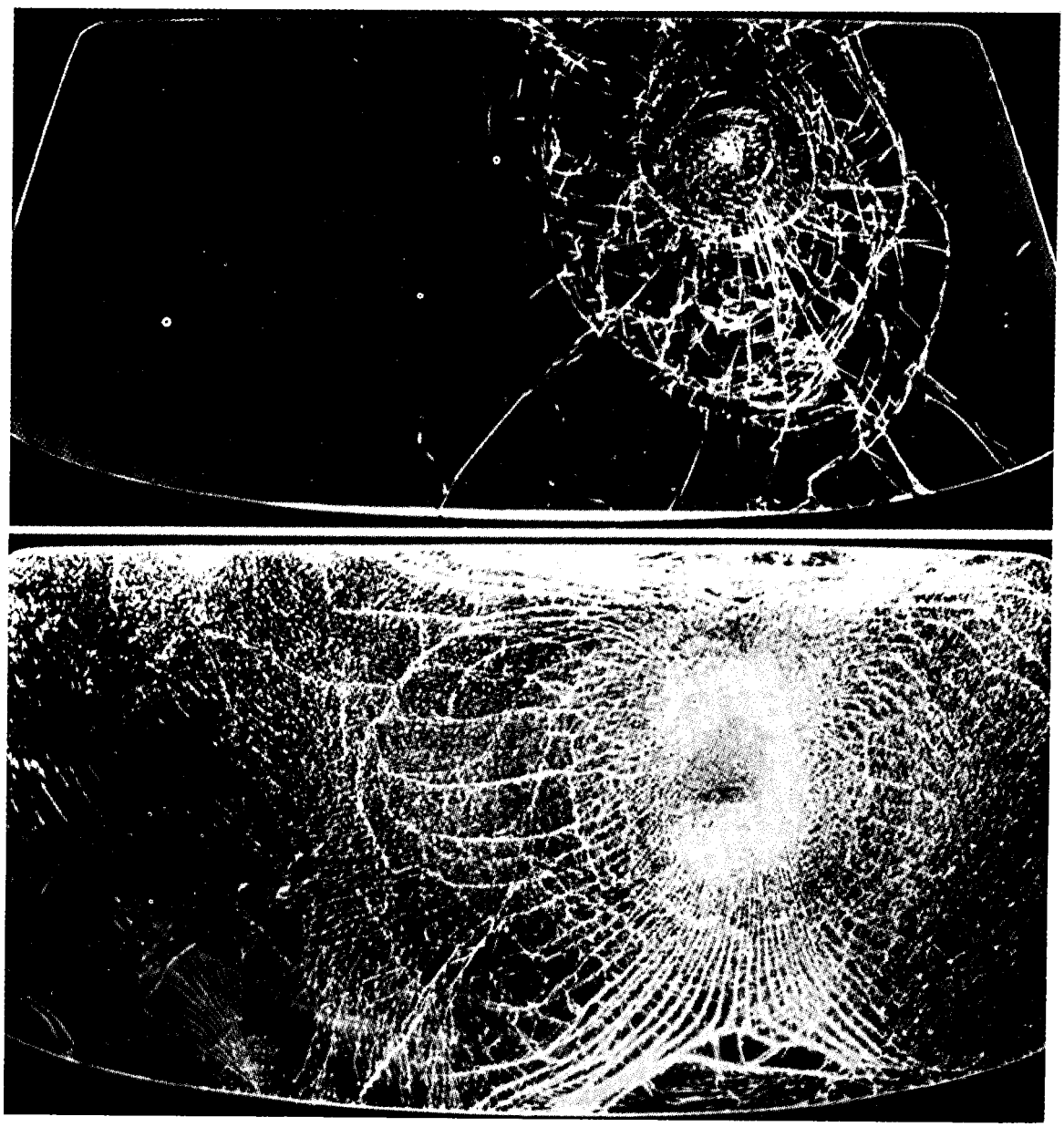


Figure 4. 3 mm + 3 mm Laminated Glass and Ten Twenty after 19 mph (30 km/h) Sled Impact

Figure 5. 3 mm + 3 mm Laminated and Ten Twenty Side View after 19 mph (30 km/h) Sled Impact

In practice all three attributes contribute simultaneously and a case where half the average length, half the average depth and one third of the number of cuts occurred would rate 2 units TLI.

THE OBJECTIVE

A car windshield is required ideally to possess a number of performance characteristics apart from its normal transparency. It is expected:

1. To retain its transparency after normal road stone impact.
2. To retain weather protection after such an impact.
3. To retain an occupant thrown against the glass at an acceptable deceleration rate to avoid permanent brain damage.
4. To retain the occupant within the car.
5. To retain the occupant while causing a *minimum of lacerative injury.*

The test program described briefly here (Reference 1) shows that the first four features are met by Triplex Ten Twenty in the same way as by conventional HPR laminated windshields, while providing the important additional advantage of a major reduction in laceration.

TRIPLEX TEST PROGRAM

For deceleration measurement a 10 kg dropping headform was used falling onto windshields mounted normally to the direction of impact. Measurements were made in terms of Gadd Severity Index (SI) and Head Injury Criterion (HIC) at speeds up to 38 mph (60 km/h).

The results are shown in Table 1 comparing Triplex Ten Twenty windshields with conventional 3 mm + 3 mm and with conventional 2.5 mm + 2.5 mm laminated windshields at 20°C.

The results are plotted on Figure 6 and show superior performance of Triplex Ten Twenty in terms of Gadd Severity Index. No significant difference was found in terms of HIC (Figure 7) nor in penetration velocities between the three types.

For laceration measurement a skull impactor, similar to that developed by Corning and shown in Figures 8 and 9, was used.

Windshields were clamped in an aperture set at the usual car mounting angle. The headform was covered first with a 6 mm thick cap of PVC to represent flesh and to enable the depth of "flesh" lacerations to be measured. Over this were placed two 1 mm layers of moist chamois leather to represent skin.

Table 1

RESULTS OF 10 kg HEADFORM NORMAL DROP-TEST ON TEN TWENTY 2.5 mm AND 3 mm WINDSHIELD

Test No.	Windscreen Composition	Velocity, km/h	Deceleration: Max Deceleration Due to Glass, g	Deceleration: Duration of Deceleration Due to Glass, ms	Deceleration: Max Deceleration Due to Interlayer, g	Deceleration: Total Duration of Impact, ms	Gadd SI	HIC	Total Interlayer Tears, mm	Pene-tration
1965	3mm	38.9	56	2.2	45	111	240	187	230	No
1966	3mm	38.9	114	3.9	41	82	293	173	–	Yes
1967	2.5mm	38.9	55	2.2	39	57	234	193	180	No
1968	2.5mm	41.0	59	1.9	38	119	213	179	254	No
1969	2.5mm	43.0	63	1.2	41	114	212	172	252	No
1970	2.5mm	46.0	98	1.5	40	43	181	132	–	Yes
1971	Ten Twenty	38.9	37	1.4	40	55	233	198	130	No
1972	Ten Twenty	41.0	40	1.2	40	60	209	176	150	No
1973	Ten Twenty	43.0	42	0.9	38	148	172	141	–	Yes
1974	3mm	34.9	53	2.2	40	51	210	175	95	No
1975	2.5mm	34.9	58	1.7	35	61	193	168	105	No
1976	2.5mm	34.9	75	1.4	38	51	220	181	100	No
1977	2.5mm	34.9	38	1.9	35	120	167	140	262	No
1978	3mm	32.0	61	1.7	38	51	183	153	0	No
1979	2.5mm	32.0	57	1.2	40	51	234	192	100	No
1980	Ten Twenty	32.0	40	0.9	34	55	175	153	0	No
1981	3mm	27.8	45	1.7	31	51	115	95	15	No
1982	2.5mm	27.8	85	2.4	34	51	207	163	5	No
1983	Ten Twenty	27.8	39	1.2	33	50	131	113	55	No
1984	3mm	24.0	60	2.4	28	51	104	84	0	No
1985	2.5mm	24.0	53	1.9	30	51	117	98	30	No
1987	Ten Twenty	24.0	51	1.2	28	50	98	85	40	No
1990	2.5mm	20.0	32	1.4	23	51	64	57	0	No
1991	3mm	20.0	94	3.9	21	53	124	78	0	No
1992	Ten Twenty	20.0	49	2.2	24	51	68	59	0	No
1993	3mm	43.0	52	2.7	43	52	161	137	–	Yes
1994	Ten Twenty	43.0	45	1.4	41	60	232	193	–	No
1995	2.5mm	43.0	51	2.9	45	71	227	170	–	Yes
1996	3mm	41.0	43	2.2	43	71	160	141	–	Yes
1998	Ten Twenty	29.9	39	1.4	33	50	138	121	50	No
1999	2.5mm	29.9	86	2.4	36	49	176	138	50	No
2000	3mm	29.9	84	2.2	32	61	185	140	0	No
2001	2.5mm	20.0	77	2.9	23	52	95	56	0	No
2002	2.5mm	24.0	63	1.4	30	55	112	89	0	No
2003	2.5mm	27.8	52	1.4	33	48	123	103	50	No
2004	2.5mm	32.0	43	1.4	36	50	166	143	80	No
2005	2.5mm	38.9	65	2.2	45	56	273	215	190	No
2006	2.5mm	43.0	46	1.2	43	109	208	154	–	Yes
2007	3mm	20.0	112	3.9	18	51	172	122	0	No
2008	3mm	24.0	90	3.2	33	43	208	136	0	No
2009	3mm	27.8	88	2.9	34	47	179	136	0	No
2010	3mm	32.0	96	2.2	43	46	264	202	0	No
2011	3mm	38.9	65	2.4	48	47	292	244	100	No
2015	3mm	20.0	90	3.7	21	53	116	57	0	No
2017	Ten Twenty	20.0	31	1.7	27	52	76	63	0	No
2018	Ten Twenty	27.8	66	1.0	34	51	175	145	0	No
2019	Ten Twenty	24.0	29	1.7	29	50	91	78	0	No
2020	Ten Twenty	32.0	37	1.0	35	75	153	128	150	No
2021	Ten Twenty	38.9	36	1.2	38	42	93	72	–	Yes
2022	3mm	43.0	81	2.0	38	43	139	97	–	Yes

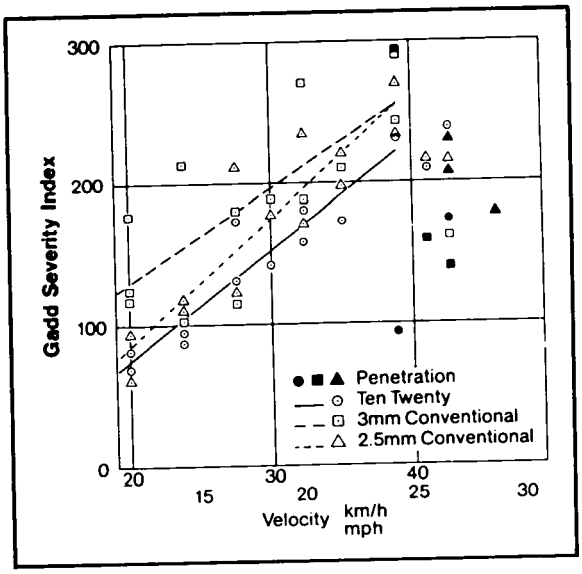


Figure 6. Gadd SI as Function of Impact Velocity by Dropping Headform Test

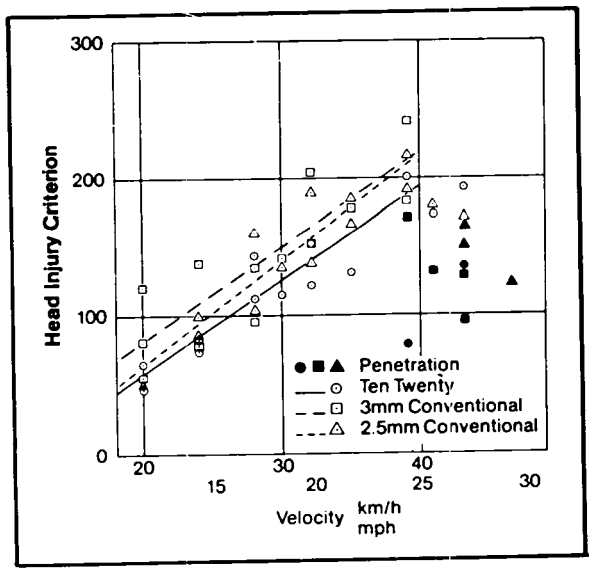


Figure 7. H.I.C. as Function of Impact Velocity by Dropping Headform Test

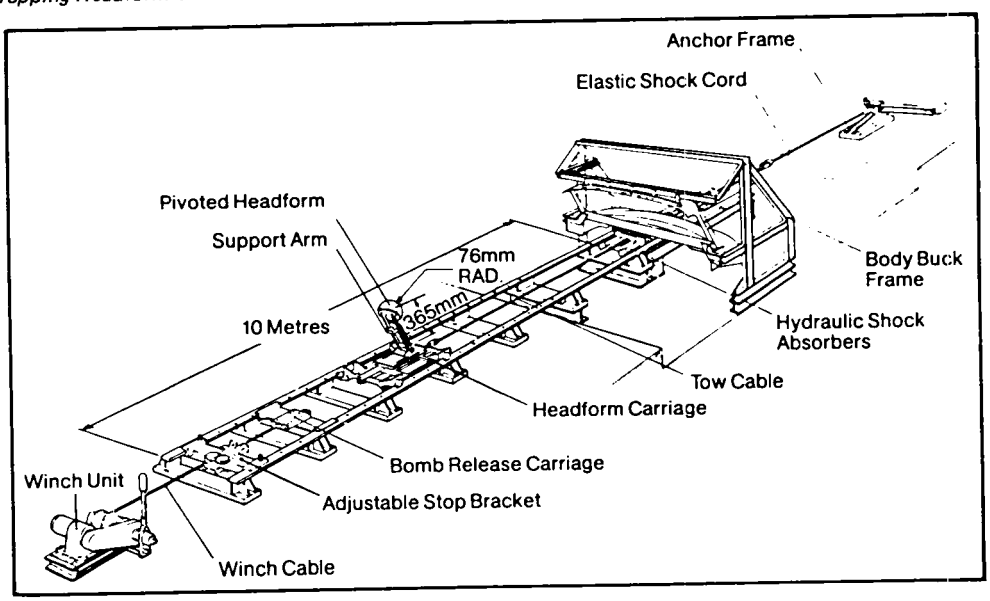


Figure 8. Skull Impact Test Rig

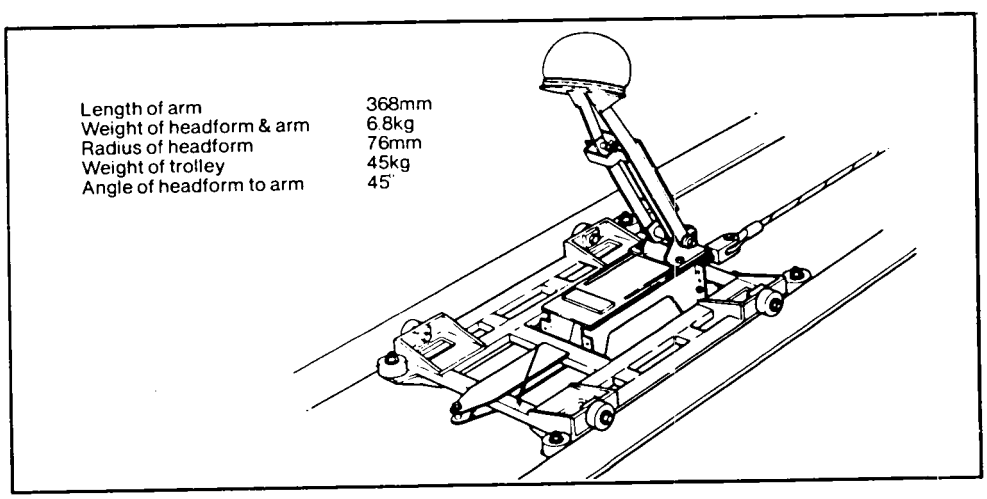


Figure 9. Carriage and Headform Assembly of Skull Impact Test Rig

A high degree of subjective assessment was found to be the weakness of existing laceration indices. A new objective index was therefore evolved (The Triplex Laceration Index, or TLI) based solely on measurements of cuts produced in the leathers and PVC facemasks. Full details of this index were the subject of a separate paper presented to the American Association of Automotive Medicine (Reference 2).

Mathematical analysis of experimental results led to the logarithmic formula:

$$TLI = 2 + \log_{10}(1 + 1.16\,D_1 + 50.8\,D_2 + 16{,}500\,D_3).$$

Where D_1, D_2 and D_3 are the lacerative damage in the two layers of chamois and PVC cap respectively. The lacerative damage in each layer is computed as a function of the length, depth and number of cuts.

The results given in Table 2 and plotted on Figure 10 show a 2.7 (logarithmic) unit improvement over conventional 3 mm + 3 mm annealed laminated and a 2.0 (logarithmic) unit improvement in TLI over conventional 2.5 mm + 2.5 mm.

These results indicate reduced laceration by Triplex Ten Twenty of 5 X 100 and of 100 times respectively without penalty in other directions.

Table 2

DATA FROM SKULL IMPACT RIG AT 20°C (TRIPLEX SAFETY GLASS)

Test No.	Type of Windscreen	Velocity of Impact, km/h	TLI	Total Interlayer Tears, mm
80	2.5mm	57.1	10.3	250
81	Ten Twenty	55.3	9.8	290
82	3mm	33.8	8.9	20
84	3mm	50.8	12.1	250
87	3mm	27.4	5.7	–
88	Ten Twenty	52.9	8.7	30
89	2.5mm	27.5	5.9	–
90	3mm	51.5	12.5	390
91	2.5mm	41.5	9.4	112
93	2.5mm	28.3	4.8	–
94	Ten Twenty	28.0	2.9	–
95	2.5mm	50.8	10.8	250
97	3mm	41.5	9.7	230
98	3mm	40.7	10	–
100	3mm	33.6	9.8	–
102	2.5mm	33.8	9.8	–
104	Ten Twenty	27.4	0	–
106	3mm	26.7	8.4	–
107	2.5mm	58.1	12.9	350
108	Ten Twenty	57.1	11.9	207
109	2.5mm	33.1	9.0	3
110	Ten Twenty	40.4	7.6	134
113	Ten Twenty	33.1	6	–
114	2.5mm	50.7	12.7	235
115	Ten Twenty	40.4	8.1	14
120	3mm	54.5	12.3	190
122	Ten Twenty	31.5	9.2	–
123	2.5mm	37.8	10.1	52
124	Ten Twenty	46.8	8.2	50
125	3mm	54.5	12.1	500
127	Ten Twenty	26.1	6	–
134	Ten Twenty	48.8	10	148
142	2.5mm	58.1	12.2	221
143	Ten Twenty	26.1	5.4	–
145	Ten Twenty	48.8	10.5	61
147	2.5mm	49.4	12.4	400
148	Ten Twenty	54.2	12.8	–
149	Ten Twenty	32.7	3.1	22
151	Ten Twenty	55.0	8.7	161
152	Ten Twenty	32.3	5.5	–
153	Ten Twenty	41.0	5.8	12
154	2.5mm	54.9	12.2	190
157	2.5mm	26.1	5.7	–
159	Ten Twenty	39.3	5.7	–

WAYNE STATE UNIVERSITY TESTS

The Deceleration and Lacerative performance of Triplex Ten Twenty and conventional annealed 2.5 mm + 2.5 mm laminates was compared on the Wham III crash simulator at Wayne State University using a Sierra 1050 anthropomorphic dummy in a modified Volvo car.

Comparisons were made at speeds between 16 and 56 km/h head impact speeds and the results are shown in Table 3 and on Figures 11 (TLI) and 13 (HIC).

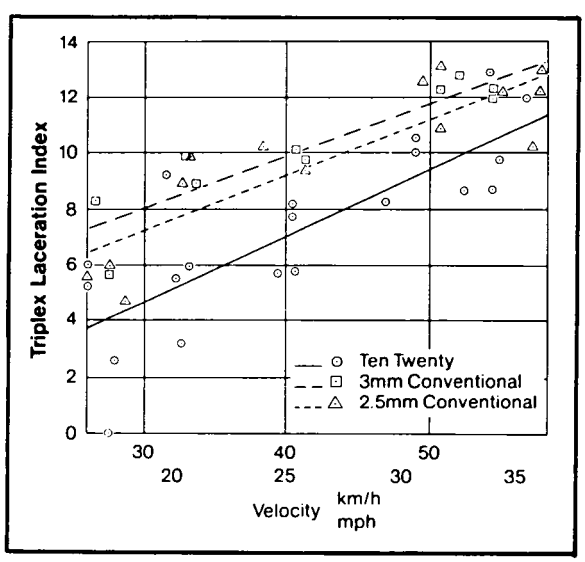


Figure 10. TLI as Function of Velocity by Skull Impact Test at 20°C

Though the Wham III and Triplex laceration tests showed a difference in absolute levels of TLI the Wayne Test program confirmed the improvement of 2 logarithmic units TLI for Triplex Ten Twenty compared with 2.5 mm + 2.5 mm annealed laminate.

At the same time Wayne found that HIC was not significantly increased and remained well below the accepted danger level of 1,000 (U.S. MVSS 208).

MECHANISM

The very large reduction in laceration is attributable to the fine fragmentation on head impact of the highly stressed inner glass of Triplex Ten Twenty.

When the head strikes a Triplex Ten Twenty windshield the energy of the impact plus the stored stress in the inner glass component causes the inner glass to break into a toughened type of fragmentation over the depth of the glass down which the head moves after initial impact. With the characteristic finer and more blunted fragments this causes much less laceration damage than when the head has to pass over the coarser and sharper fragments of a conventional annealed laminated windshield (see Figures 4 and, particularly, 5).

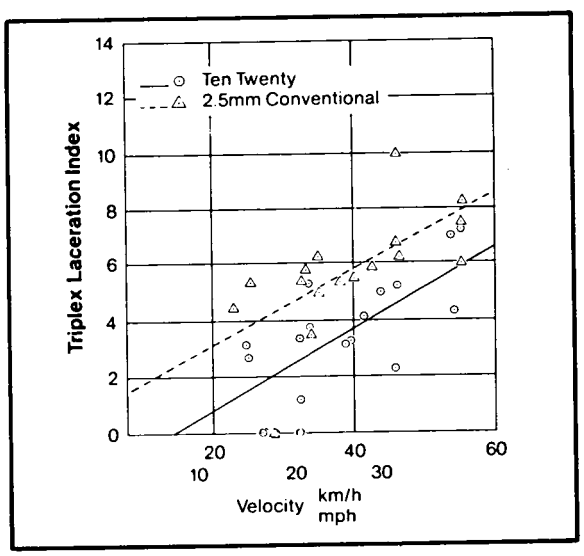


Figure 11. TLI as Function of Velocity by WHAM III Test Rig

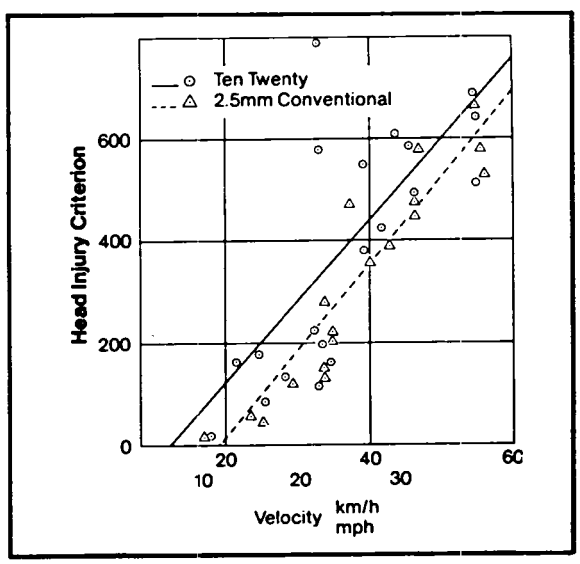


Figure 13. HIC as Function of Impact Velocity by WHAM III Test Rig

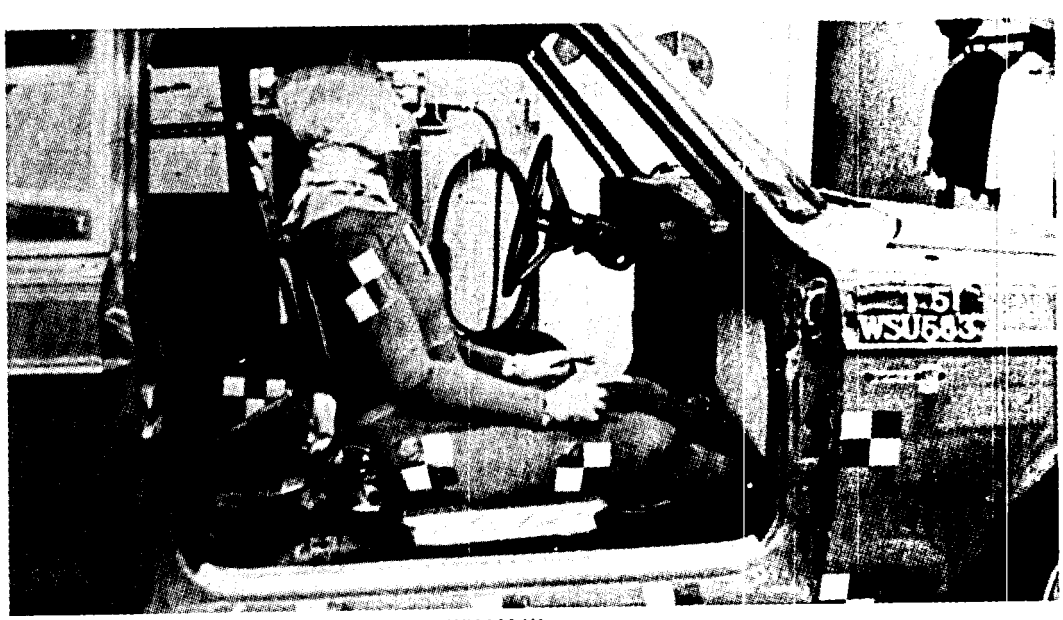


Figure 12. Vehicle with Dummy Installed on WHAM III

Table 3

SUMMARY OF DATA ANALYSIS FOR TRIPLEX WINDSHIELD LACERATION STUDY AT WSU

Run No.	Type of Windscreen	Velocity, km/h	Stopping Distance, mm	Head Index Gadd	Head Index HIC	Laceration Index WSU	Laceration Index Corning Scale	Laceration Index T.L.I.	Laceration Index Pummel	Remarks
T- 4	2.5mm	34.1	410	253	220	3	4	4.9	3	Abrasion on forehead
T- 5	Ten Twenty	33.5	410	200	164	1	4	5.3	3	Light abrasion on forehead
T- 6	2.5mm	34.1	410	248	202	3	5	6.2	4	Windshield slightly pulled out, abrasion on forehead
T- 7	Ten Twenty	32.8	410	249	196	1	3	3.8	5	Abrasion on forehead, one cut through outer layer on nose
T-10	2.5mm	16.3	300	11	9	0	0	0	5	Head broke outer glass, no cuts
T-11	Ten Twenty	16.9	200	13	11	0	0	0	5	Head broke outer glass, no cuts
T-14	2.5mm	24.8	300	67	55	1	4	5.2	4	Light abrasion and few small cuts on forehead and nose
T-15	Ten Twenty	24.9	300	131	82	1	3	3.2	3	Abrasion on forehead
T-16	2.5mm	36.7	500	576	482	3	4	5.4	3	Severe abrasion and a few cuts on forehead, buttocks slid off seat, legs jammed under instrument panel
T-17	Ten Twenty	41.5	500	677	423	1	3	4.1	4	Severe abrasion on forehead and nose, no SI acceleration
T-18	2.5mm	42.2	500	465	393	3	4	5.9	3	Lacerations and severe abrasion on forehead, abrasions on nose
T-19	Ten Twenty	27.5	330	371	121	0	0	0	5	–
T-20	2.5mm	28.3	330	172	113	0	0	0	3	–
T-21	2.5mm	33.1	410	174	143	3	4	5.4	3	Abrasions and cuts on forehead
T-22	Ten Twenty	32.0	410	278	222	1	2	3.4	4	Scratches on forehead
T-23	2.5mm	33.0	410	229	161	1	3	3.6	4	Scratches on forehead
T-24	Ten Twenty	32.3	410	175	131	1	1	1	4	Scratches on forehead
T-25	Ten Twenty	43.9	610	787	606	3	4	4.9	3	Severe abrasions on forehead, dummy slid off seat
T-27	Ten Twenty	45.9	610	582	494	3	4	5.2	3	127mm tear (vertical), severe abrasions with few cuts on forehead and nose
T-28	2.5mm	46.0	610	570	482	5	5	6.2	5	50mm tear, several cuts on forehead and near right eye
T-29	2.5mm	45.9	610	593	463	6	6	6.8	3	More severe abrasions on forehead, cuts on forehead, eyes, and nose
T-30	Ten Twenty	54.1	710	871	523	6	6	7.1	3	267mm jagged tear, extensive lacerations through outer chamois and couple of cuts through inner chamois
T-31	2.5mm	55.0	710	747	584	6	7	8.3	3	50 x 152mm tear, severe lacerations through both layers of chamois, 11mm cut on chin in rubber flesh
T-32	Ten Twenty	55.0	710	798	643	6	6	7.3	3	216mm jagged tear, severe lacerations through both layers of chamois, no cuts in rubber flesh
T-33	2.5mm	55.0	710	905	673	6	6	7.4	3	Severe lacerations through layers of chamois, no cut in rubber flesh
T-35	Ten Twenty	22.5	330	258	167	0	0	0	3	No fracture
T-36	Ten Twenty	32.2	410	1380	811	0	0	0	3	No fracture
T-37	Ten Twenty	39.3	500	836	543	1	2	3.1	3	–
T-38	2.5mm	23.3	330	114	68	1/3	3	4.4	4	–
T-45	Ten Twenty	32.2	410	1069	568	0	0	0	4	No lacerations
T-47	2.5mm	33.1	410	329	268	3/6	5	5.8	4	–
T-[illegible]	Ten Twenty	25.4	330	330	185	1/3	3	2.7	4	–
T-50	2.5mm	40.1	500	487	369	3	4	5.4	4	–
T-51	Ten Twenty	39.4	500	534	381	1	2	3.1	4	–
T-54	2.5mm	47.0	610	689	581	10	9	10.0	3	Severe lacerations, cuts (15 and 4mm) in flesh of dummy
T-55	Ten Twenty	45.9	610	1054	585	1	2	2.3	3	–
T-58	2.5mm	55.0	710	629	528	6	5	6.0	4	63 and 140mm tears
T-59	Ten Twenty	54.9	710	1106	689	3	3	4.5	4	–

REFERENCES

1. Improved Laminated Windshield with Reduced Laceration Properties, S. E. Kay and J. Pickard (Triplex) and Professor L. M. Patrick (Wayne State University), presented at 17th Stapp Conference, Oklahoma, Nov. 1973.
2. An Objective Method of Assessing Laceration Damage to Simulated Facial Tissues, The Triplex Laceration Index, J. Pickard, P. Brereton and A. Hewson (Triplex), presented at AAAM Conference, Oklahoma, Nov. 1973.

OCCUPANT PROTECTION IN FRONTAL IMPACTS: A STATIC, PASSIVE RESTRAINT SYSTEM

W. EGGLESTONE
G. D. SUTHURST
Ford Motor Company Limited

INTRODUCTION

Accident reviews and controlled crash test investigations have established that benefits can be obtained from an efficient occupant restraint system. The reluctance of the majority of the motoring public to participate actively in self protection, e.g., by wearing seat belts, has led to demands for a passive restraint device.

Various passive systems exist or are under development; the most prominent being the inflatable air bag. The U. S. automobile industry has done a considerable amount of work in this field and the advantages and disadvantages of the system are well known. The major disadvantages are related to the activation function, e.g., reliability, noise, one-shot system, ageing effects, pressurized gas cylinders, etc.

Ford Engineering Research, recognizing the problems associated with the air bag system, have analyzed the feasibility of using a subjectively acceptable "static" passive restraint for each passenger position. The basic concept is in the form of an energy absorbent shape in front of each passenger, capable of meeting specific occupant injury criteria under frontal impact conditions. This approach eliminates many of the activation-related problems associated with the air bag concept.

An occupant basically requires a restraint that spreads the deceleration forces over as large an area of the body as is practical. The aim, therefore, has been to design and test a restraint which employs impact panels to decelerate the occupant's chest and knees and, in the case of the front passenger, the windshield to restrain the head.

This use of the windshield has been necessitated by the average European package, where the minimum front occupant deceleration distance places the head beyond the windshield. Standard laminated windshields are acceptable in their capacity to restrain an occupant within the passenger compartment, but unrestrained occupants sustain quite severe facial lacerations when the head rotates forward down the broken windshield. One of the project aims therefore has been to control effectively occupant kinematics to ensure that excessive forward rotation of the head does not occur, thereby reducing facial lacerations. We have also maintained close liaison with the glass producing industry to keep abreast of technological advances in overcoming the laceration problem.

The present state of the art, with regard to human tolerance to impact and to the use of anthropomorphic dummies has not progressed to the point where one can be dogmatic regarding quantitative injury criteria. Nevertheless, it is necessary to establish guidelines to assess effectively restraint performance, and therefore the injury criteria from FMVSS 208 have been used for this project.

It is acknowledged that customer acceptance could be a limiting factor with this type of restraint and efforts have been made to provide maximum occupant freedom within a specific protection level and for similar reasons, basic investigations have been concentrated on four door vehicles only.

The initial investigation was concentrated on the front passenger position and preliminary dynamic tests were conducted using Ford facilities.

During this initial development period successful discussions with the British Government, through the Transport and Road Research Laboratory, resulted in Ford's taking up a safety contract for a passive restraint system.

This was basically our intended program which covers the development and test of front passenger and rear passenger restraints. Performance to be checked in both direct frontal and 30% frontal impacts would be conducted using the Motor Industry Research Association facilities.

By January 1974 effective restraint of rear seat passengers at 48 km/h had been achieved and although special problems exist for the front passenger, related to vehicle package, these are well on the way to being overcome.

Among the proposals intended for future study is a combination of the "static" passive restraint with seat belts for protection at higher impact velocities and also a combination of seat belts in the front and passive restraint in the rear to provide passive protection for the less frequently occupied positions.

This paper reports on the progress of the project up to January 1974 and outlines the future plans and expected accomplishments.

INITIAL INVESTIGATION

Information derived from the extensive Ford of Europe dynamic crash test program established that the friction between the unrestrained occupant and the seat in crash conditions is negligible. Therefore

the occupant maintains the immediate pre-impact velocity of the vehicle until brought to rest by striking the internal vehicle structure. Figure 1 illustrates this by showing displacement for both the vehicle and the unrestrained occupant for a standard frontal barrier impact on a time basis.

From this, it is established that the pre-impact distance between the occupant and a "static" restraint pad determines:

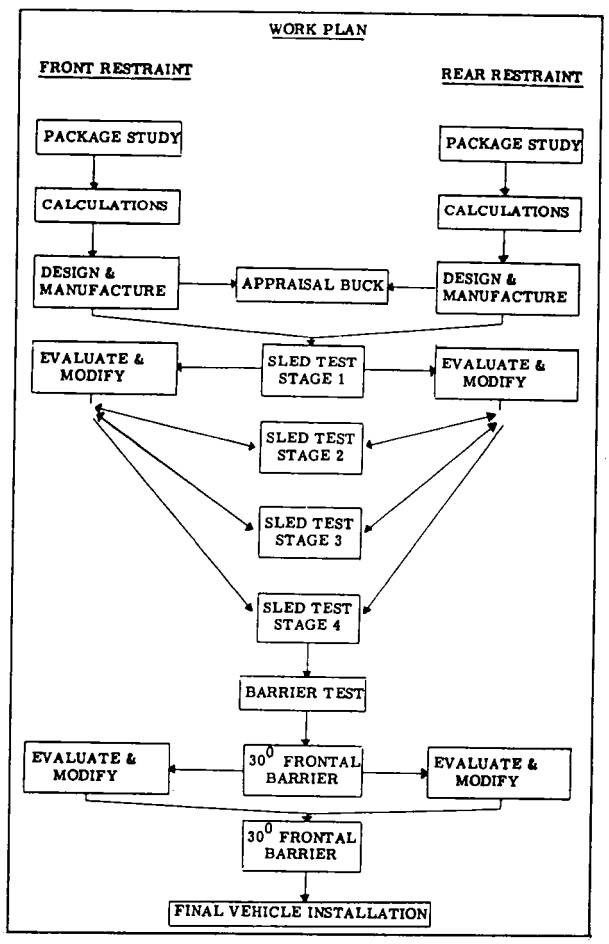
- the velocity at which the occupant impacts the restraint;
- the vehicle displacement remaining after the occupant to restraint contact;
- the depth of restraint available to decelerate the occupant.

Figure 2 shows the occupant and vehicle displacement and the derived occupant and vehicle velocities during barrier impact. From this plot the relative velocity of occupant to restraint impact can be devised for any initial distance between the chest of the occupant and the restraint. For example, for an initial distance of 100 mm between occupant and restraint, the impact velocity is 17.6 km/h.

Further sample distances of 100mm, 250mm and 400mm are analyzed in Figure 3 specifically demonstrating that any increase in the pre-impact occupant to restraint distance:

- increases the occupant to restraint impact velocity;
- decreases the occupant ride down on vehicle displacement;
- decreases the available restraint depth.

Although the occupant to restraint distance should be at a maximum for customer acceptability, it must be restricted to provide the required level of occupant protection. For this reason, plots were made of:



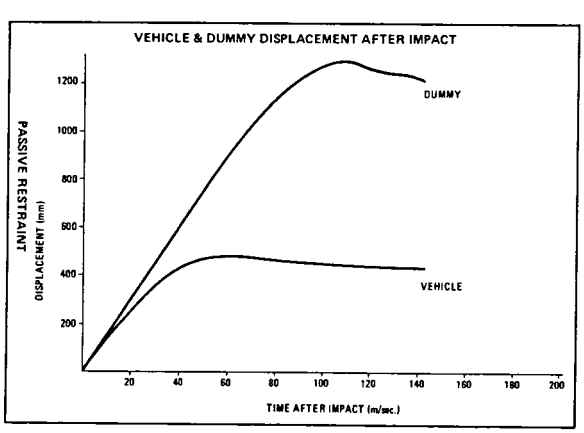


Figure 1

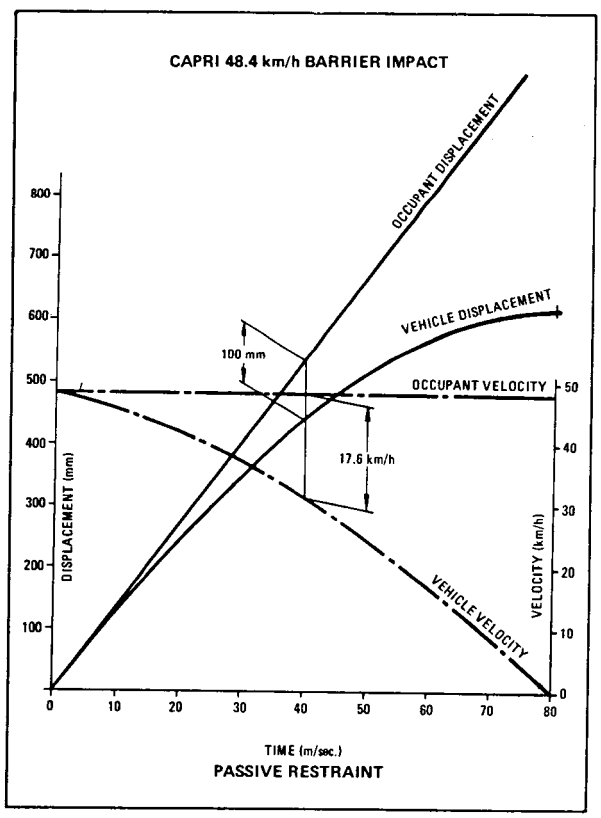


Figure 2

- occupant impact velocity against occupant to restraint distance;
- occupant impact velocity against restraint collapse length.

The occupant impact velocity, which is dependent on the occupant to restraint distance, was obtained as shown in Figure 2 and the restraint collapse length was obtained by assuming a given protection and that the restraint decelerates the occupant with a half sine wave pulse shape. The actual restraint collapse distance is 80% of the restraint length.

These two plots are overlaid in Figure 4 identifying the optimum positions of the restraint in the available package distance of a Capri at half sine chest deceleration levels of 40g and 60g. In actual test conditions, it is anticipated that a half sine deceleration wave form or better will be achieved.

A similar exercise for the Cortina package showed the occupant to restraint distance as 316mm, the restraint length as 289mm and a restraint collapse distance of 231mm (80% of the restraint length) for a 40g peak chest deceleration level. This places the chest portion of the restraint in the same position as the steering wheel. Figure 5 shows this relationship and also the anticipated collapse characteristics where the occupant remains fairly upright during the restraint collapse. The head is restrained by the windshield with laceration inducing rotation kept to a minimum.

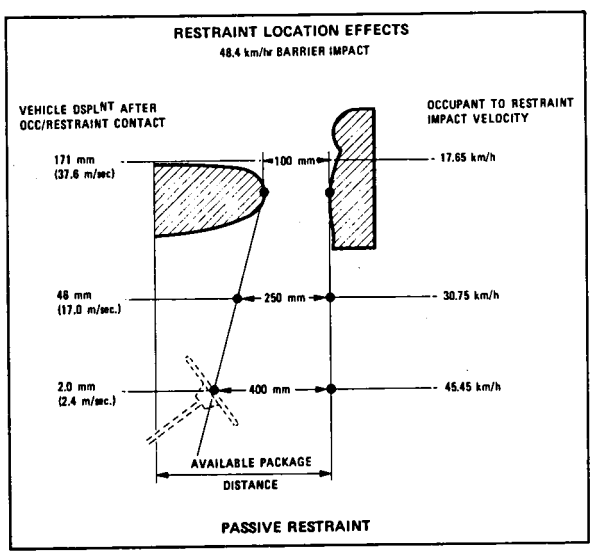


Figure 3

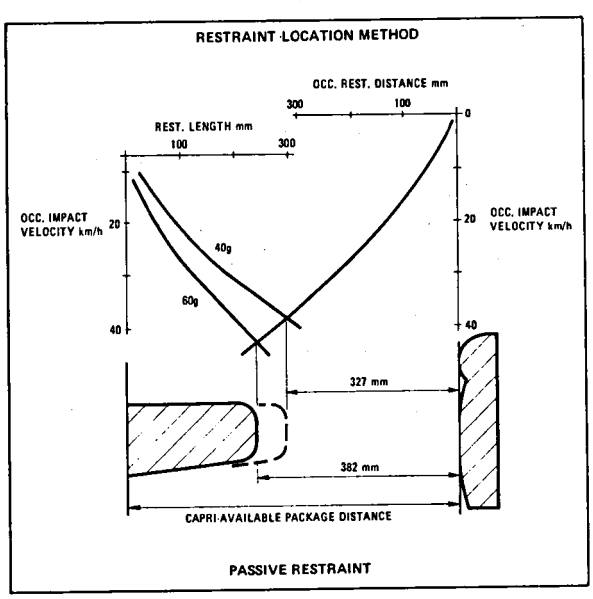


Figure 4

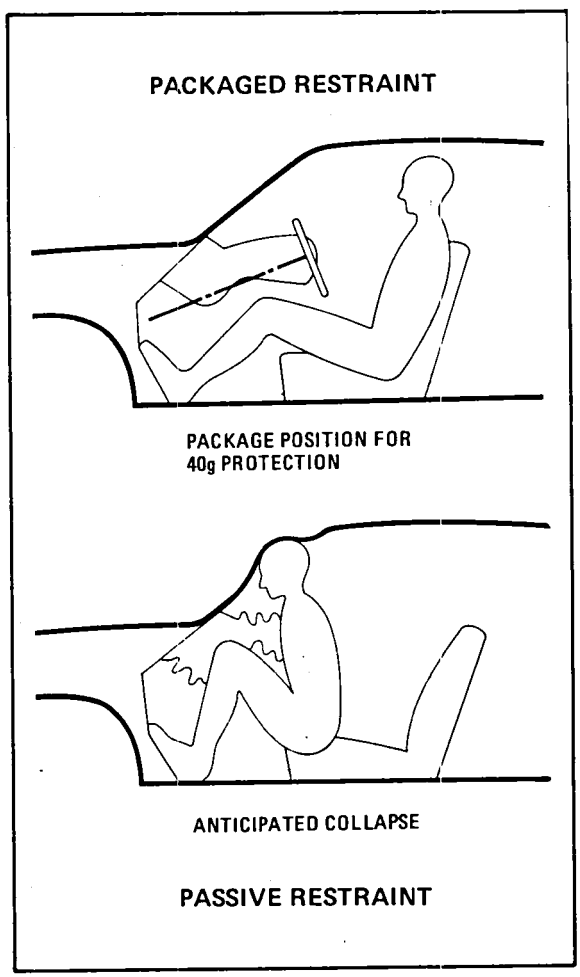


Figure 5

All calculations, to determine the position of the restraint in the passenger compartment, are computed with the front seat in the mid position of the fore and aft adjustment and a 50th percentile dummy. It is accepted that there will be different conditions for the 5th and 95th percentile occupants. In the former case a forward adjusted seat will result in:

- a decrease in the occupant to restraint impact velocity that would improve protection in vehicle impacts over 48 km/h,
- a slight increase in occupant deceleration levels because of the smaller masses of the 5th percentile.

In the latter case, i.e., 95th percentile occupant, the rearward adjusted seat will result in:

- an increase in occupant to restraint impact velocity,
- probable bottoming out of the restraint causing an unacceptable late deceleration peak.

Limitations do exist in the current vehicle package, the main problem areas being the windshield and the header rail. The tendency for the low roof line and the high rake screen in the small European car means interference in the space required for the occupant deceleration displacement relative to the vehicle.

RESTRAINT DEVELOPMENT

The initial investigation into suitable materials has been restricted to formed mild steel and rigid plastic foam. It is well known that mild steel sheet has excellent energy absorbing properties and well established processing methods. The choice of phenolic foam (CIBA-GEIGY Appendix) was similarly based on energy absorbent properties with added advantages over other foams of lower cost and low flammability with no toxic hazard.

Early development work identified the importance of testing restraints of actual size with the correct shape for the effective impacting mass. Therefore, the test series has consisted of sled tests, which are also suited to controlling test cost and repeatability, and barrier tests for final appraisal of the most promising restraint designs, in both cases using 50th percentile test dummies.

Static tests have been conducted at stages throughout the development for instant element stiffness comparisons after design changes.

To assess effectively the restraint dynamic test performance, the injury criteria laid down in FMVSS 208 for the post 1976 period have been used, which are:

Head The resultant acceleration at the center of gravity of the head shall not result in a GADD severity index value exceeding 1,000. (SAE Class 1,000)

Chest The resultant acceleration at the center of gravity of the upper thorax shall not exceed 60g except for intervals whose cumulative duration is not more than 3 milliseconds. (SAE Class 180)

Femur The force transmitted axially through each upper leg shall not exceed 1700 lbft. (7.56 kN) (SAE Class 600)

Five initial restraints were manufactured and sled tested to establish dummy kinematics and restraint collapse characteristics, Figure 6. Four were made from 0.75 mm sheet metal of varying configuration and one of rigid phenolic foam, 24 kg/m^3 density, with convoluted metal inserts and a PVC cover. Due to limitations in the sled test facility available at the time and to prevent test equipment damage, sled tests were conducted at an impact velocity of 38 km/h without a windshield.

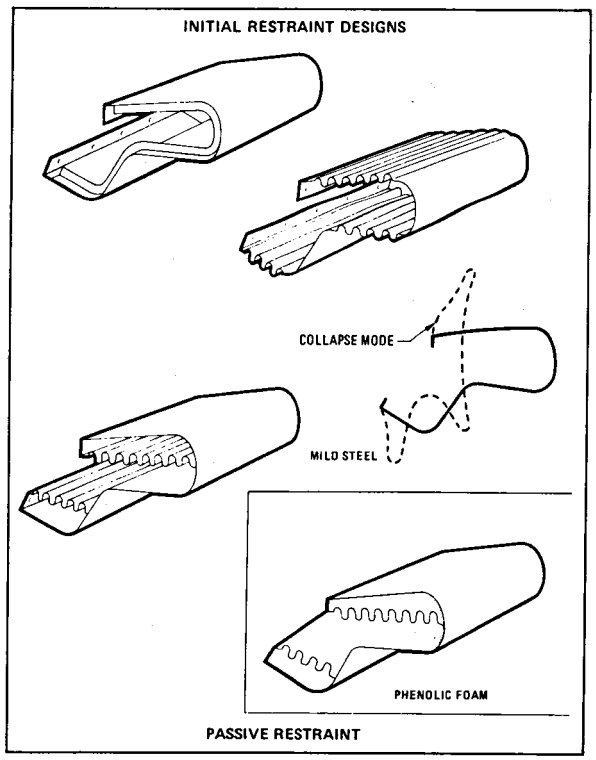


Figure 6

Analysis of results highlighted four main points:

- Early knee contact on the metal restraints resulted in the chest portion being displaced before restraint/chest contact. This indicates that the knee and chest portions should be separated. (This did not apply to the foam restraint as it has an inherent separation quality.)
- Collapse control is difficult using small convolutions – they tend to bend rather than collapse axially.
- With all the metal restraints, the top and bottom surfaces of the chest area contained a large amount of material that was difficult to control under collapse conditions, and resulted in sharp, hazardous folds occurring.

- The foam element, with inserts, proved to be too rigid with insufficient collapse and insufficient support for the foam in the knee area.

Figure 7 shows a redesigned metal restraint to eliminate the problems outlined. This has the metal removed from the upper and lower part of the chest portion to eliminate the sharp folds occurring under impact conditions and also to prevent the knee restraint collapse influencing the chest portion of the restraint. This results basically in two beams, restrained at the ends, absorbing energy in bending and tension. The modification to the foam restraint was to remove the convoluted inserts and increase the support for the foam in the knee area. These elements were sled tested with the impact velocity raised to 48 km/h and a standard high impact laminated windshield fitted. The rubber weatherstrip was adhered to the windshield and the fixing flange.

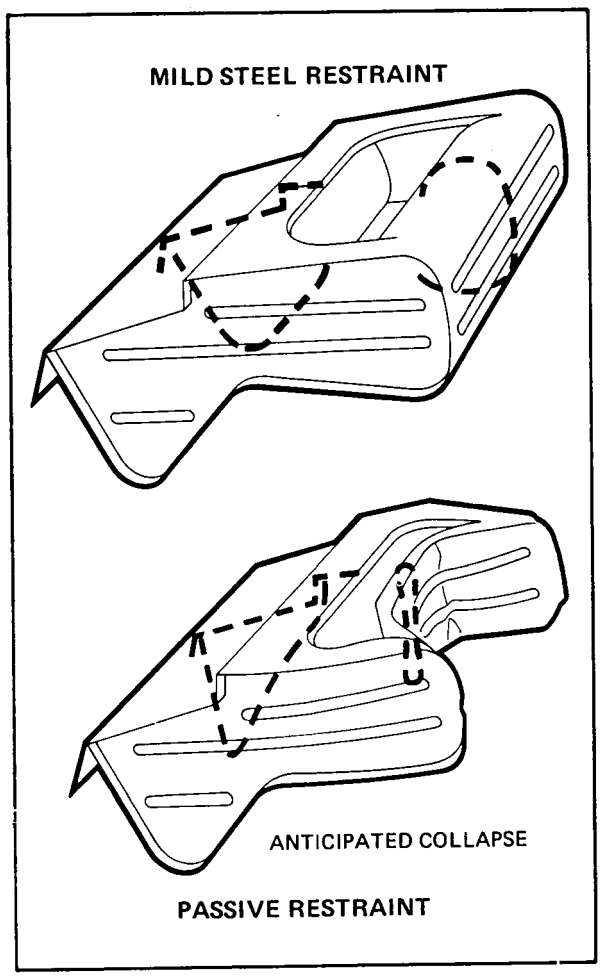


Figure 7

Analysis of test results indicated that there was still insufficient collapse with the foam restraint and that further development was required to increase the occupant's deceleration distance. The metal restraint had excellent collapse characteristics and the dummy kinematics were ideal with a very low rebound velocity, indicating efficient occupant energy absorption. The chest resultant deceleration was within 2½% of the calculated figure and the femur loads were acceptable.

After this successful sled test a 48 km/h barrier impact test was conducted using an identical front restraint. Analysis of results however showed that the required occupant protection levels had not been met and highlighted limitations with an all metal restraint. These limitations included:

- Sharp and extremely stiff areas remaining after deformation.
- Apparent increase in the load up to maximum deformation of the restraint which, when combined with the vehicle rebound, caused an unacceptable late peak deceleration. The effect on test results of the vehicle rebound was expected, but is difficult to simulate with the sled facility.
- Difficulty of phasing the collapse loads of the beam, and of the end supports of the restraint to obtain optimum deformation.

The barrier test also identified the problem of possible head to header rail contact instead of the required head to windshield contact. The header rail moved down 25mm, relative to the dummy, due to the "nose down" of the vehicle during barrier impact, aggravating the problem inherent in the test vehicle package.

COMPOSITE RESTRAINT SYSTEM

To study more closely the collapse characteristics of the restraints and to assess modifications and new concepts, a static test program has been conducted. The sheet metal restraints were modified with designed-in collapse control but although improvements were obtained, elimination of the main disadvantages was not achieved.

At this stage, an investigation into the use of materials other than sheet metal identified the potential of a composite restraint constructed of expanded mild steel sheet and phenolic rigid foam.

An expanded metal beam concept restraint of "C" section was manufactured and static tested. The results show that expanded metal in this form has good collapse characteristics and close correlation with the ideal load/ deflection square wave but with a low load peak value.

When the expanded metal is used as a core in a moulded phenolic rigid foam, with the whole contained in an outer cover, static test results show a vast improvement in the restraint function (Figure 8). The square wave form is maintained at an acceptable peak load level without the tendency found in sheet

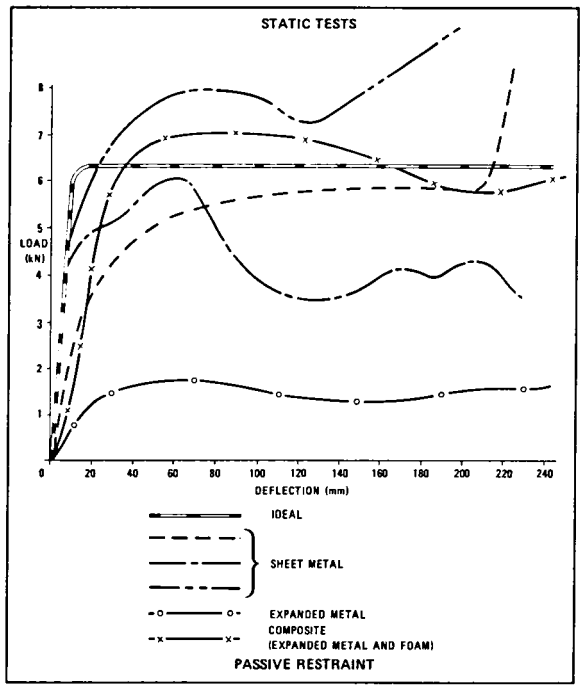


Figure 8

metal restraint to peak sharply within the anticipated restraint deformation.

Figure 9 shows the current test sample restraint, comprising the expanded metal ("C" section) core within a phenolic rigid foam moulded shape and an outer cover of flexible foam and PVC trim. The outer trim cover, in containing the foam on impact, plays a major role in the efficient functioning of the restraint.

Advantages of the composite restraint over the all sheet metal restraint are:

- elimination of extremely stiff injurious areas remaining after deformation;
- The "soft feel" of the restraint before and after impact;
- The more acceptable collapse characteristics with controlled deformation in the chest beam and support legs of the restraint;
- Weight saving.

At this stage in the development only one problem is evident in the prototype restraint manufacture. Surface cracks appear in the skin of the foam after moulding which are caused by shrinkage due to the evaporation of volatiles in the period after moulding. Investigations to eliminate this problem are being conducted in conjunction with CIBA-Geigy.

A number of variations are possible if modifications are required to alter restraint performance under dynamic conditions:

- CORE: shape, metal thickness and mesh size.
- FOAM: density and volume.
- COVER: material elasticity.

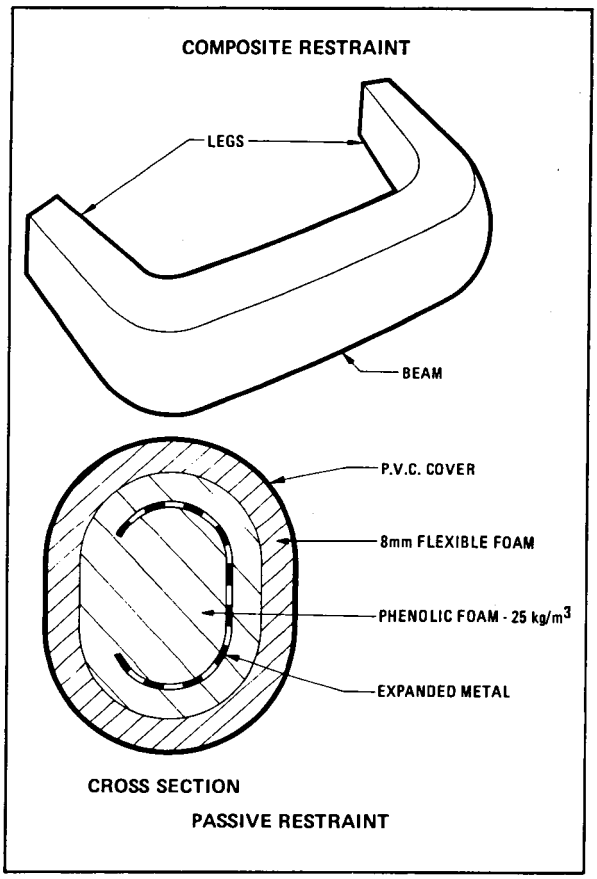


Figure 9

A hard controllable thickness skin forms on the outside of the moulded rigid foam which aids handling and *in situ* service. The effects of ageing and vehicle vibration are unknown at this stage but investigations to explore these effects are in hand.

After successful static tests, using the composite restraint, 48 km/h sled tests were initiated to investigate the restraint collapse characteristics under dynamic conditions.

A test sample was mounted in the front passenger position of a Cortina body shell which was fitted with a standard high impact laminated windshield. An Alderson VIP 50A test device was used as the impacting mass. An independent mild steel knee restraint was introduced at this stage.

Analysis of the results confirmed dynamically the high success potential shown in the static tests. Though the head, chest and femur levels were within the injury criteria, lower values would be more acceptable considering the expected test scatter of repeat tests.

The restraints, therefore, were modified to reduce the overall stiffness by varying the thickness and mesh size of the expanded metal core. Sled tests were conducted after each modification.

Following the success of the composite restraint for the front passenger, a restraint for the two rear passengers was considered using the same form of construction. It was decided to provide separate restraints for each passenger, mounted on the "B" pillars with a common central mounting using the back of the front seat as the knee restraint. A single beam to accommodate both rear passengers would have to be a complex structure in order to accommodate the impact of either one or two passengers.

Further sled tests and one barrier test have been conducted with front and rear restraints fitted. The test method and test results are discussed under their respective headings.

WINDSHIELD GLASS INVESTIGATION

In the static restraint concept, the average European vehicle package places the windshield in the head displacement zone during impact, thus the windshield is involved as an integral part of the restraint system.

Laminated windshields have been found acceptable in their capacity to retain the unrestrained occupant within the passenger compartment. However, severe facial lacerations may occur due to the relative motion of head to broken windshield caused by head rotation.

The energy absorbing restraint pad has been successfully designed to control the occupant kinematics in such a way as to eliminate the forward rotation of the head, thereby reducing facial lacerations.

The present state of glass technology is that various laminated windshields are now under development with improved impact characteristics regarding facial lacerations and head severity indices. This has been achieved by the development, by different glass manufacturers, of interlayers; applied plastic coatings to the inner glass; and the controlled tempering of the inner and outer glass laminates of the windshield as in the Triplex Ten-Twenty.

A test program has been initiated to test these, when available, with respect to dummy head impact. Performance will be compared using head severity and laceration indices. To ensure repeatable and meaningful results, the tests will be conducted with the screen in the package position, impacted by a dummy head mounted on a pendulum rig. Layers of chamois leather on the head will establish a laceration comparison.

APPRAISAL BUCK

A full scale seating buck has been manufactured with front and rear passive restraint units and padded areas exactly simulated in clay. This has allowed a study of interior styling and will be used to assess customer acceptance of the passenger compartment environment on a subjective basis, Figure 10.

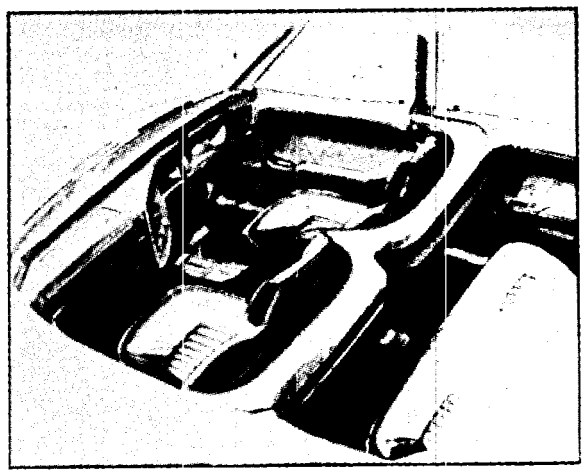


Figure 10

A complete vehicle has been fitted with front and rear composite restraints which are representative of the units at this stage in the development. A four door vehicle has been used to allow the present development approach of mounting the rear restraints on the "B" pillars. This vehicle will allow the subjective appraisal of the system to be extended to actual driving conditions.

RESTRAINT TEST METHOD

Static Tests

Conducted using an "Avery" slow action hydraulic ram machine with an automatic pen trace of the load and deflection.

The ram stroke was controlled to the expected restraint deformation under dynamic conditions. Still photographs were taken at frequent selected intervals during the cycle to record the sequence of restraint deformation, see Figure 11.

Dynamic Tests

Except for the initial metal restraint tests which were conducted within Ford, all dynamic tests have been carried out using the facilities at the Motor Industry Research Association's proving ground.

For all single and multi dummy tests the Alderson VIP50A test device is fitted with tri-axial accelerometers in the head and chest and load cells in the femur. Event markers are attached to the knee, chest and head to record the time of contact with the restraint and windshield.

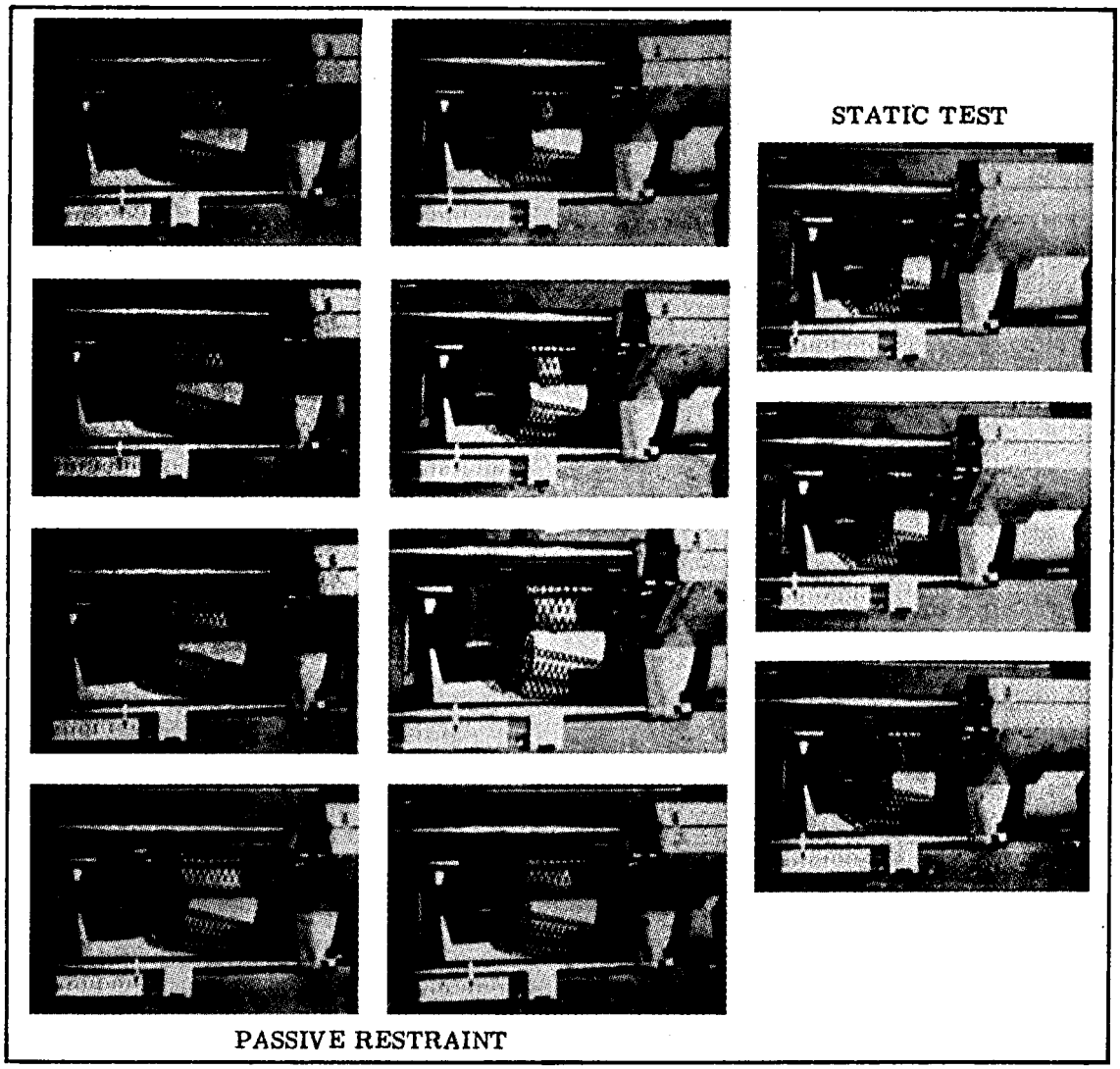


Figure 11

Vehicle instrumentation consists of two accelerometers, one mounted at the base of each "B" pillar.

Accelerometer and event signals are fed by an umbilical cable from the vehicle to the instrumentation recording units consisting of duplicated 1 inch wide magnetic tape recordings and a UV back up system. All instrumentation complies with SAE J211a recommendations.

High speed cine cameras record the vehicle and dummy kinematics for analysis. The vehicle in the barrier test, and the trolley carrying the body shell in the sled tests, are accelerated to 48 km/h by a linear motor and impact a concrete block. The sled trolley is arrested by crumple tubes.

For each test the front seat in located in the mid-position of the seat fore and aft adjustment and the rear seat is used in its unmodified package position.

TEST RESULTS

Figures 12 and 14 show that before test situation with front and rear restraints in package position and the front seat in the mid adjustment location. Figures 13 and 15 show the after test deformation. The large chest to restraint contact area, giving an ideal load distributing condition, and the head to laminated windshield activity can be identified.

Figure 16 shows the results of key dynamic tests using the expanded metal/phenolic foam composite restraints. The stiffness numbers identify the variations in the restraints achieved by modifications to the core section, stiffer restraints as the number increases.

Sled Test MC1

Stiffness 3 passive restraint fitted in the L.H. front passenger position with an independent knee

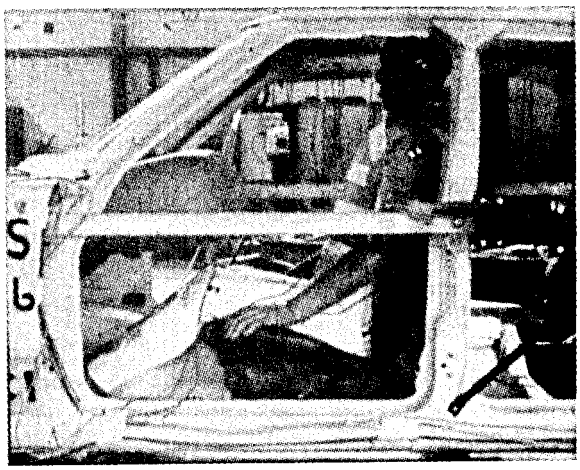


Figure 12

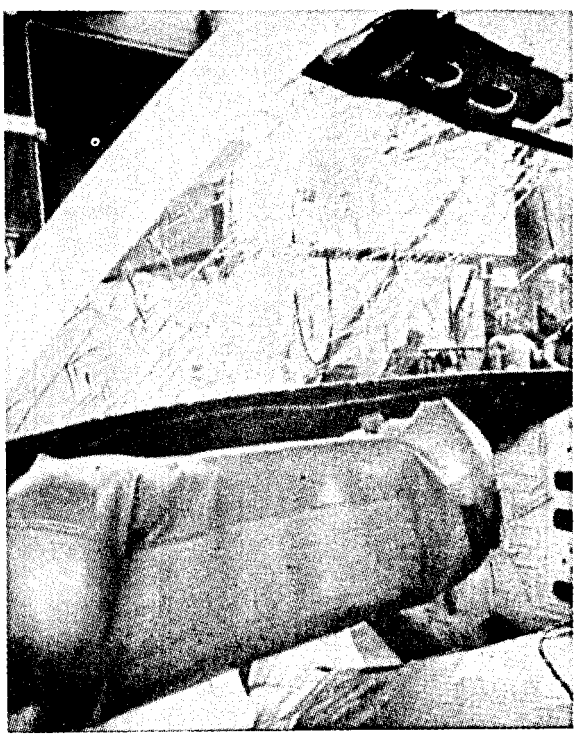


Figure 13

Figure 14

Figure 15

restraint. Test results indicate that although the head, chest and femur injury levels were acceptable, reduction in their values would be necessary to ensure acceptability within an expected test scatter envelope. A direct head to header rail blow occurred with no windshield contact and indicates the possibility of using either a padded header rail or the windshield as the head restraint.

Action: Restraint stiffness reduced to achieve more acceptable injury levels. No modification to the knee restraint.

Sled Test MC2

Stiffness 1 passive restraint fitted in the L.H. front passenger position. Results show an increase in head and chest values with a slight reduction in the femur load.

Film analysis showed that the weakened restraint had affected the control of the dummy kinematics. The restraint bottomed out causing an increase in the head to windshield impact velocity. The head contacted the windshield after a glancing blow on the header rail.

Further study of the test results identified the inter-relationship of the functions of restraining the head, chest and knees. Any modification to one restraint area will influence the overall values.

Action: Increase the restraint stiffness to be more stiff than MC2 but less than MC1. No modification to knee restraint. Introduce a rear restraint for test.

Sled Test MC3

Stiffness 2 restraint fitted in the L.H. front passenger position. Stiffness A restraint fitted in the R.H. rear passenger position.

Results show that the front restraint was still too weak. There was an improved chest value compared with MC2 but the head level indicated that the

DYNAMIC TESTS							
FRONT							
TEST NO.	RESTRAINT STIFFNESS	DUMMY	HEAD (GADD)	CHEST Peak g	CHEST Time Over 60g (msec)	FEMUR (kN) R. H.	FEMUR (kN) L. H.
SLED MC1	3	ALDERSON	791	58	–	7.0	7.0
SLED MC2	1	ALDERSON	1,266	150	3.30	6.5	6.5
SLED MC3	2 2	ALDERSON	1,250	73	1.30	4.3	4.3
SLED MC4	2	ALDERSON	780	74	3.47	3.30	9.7
BARRIER MB1	3	ALDERSON	1,620	41	–	6.5	6.5
REAR							
SLED MC3	RH A LH –	ALDERSON –	525 –	73 –	1.3 –	3.0 –	3.0 –
SLED MC4	RH A_1 LH –	ALDERSON –	850 –	54 –	– –	3.0 –	3.0 –
BARRIER MB1	RH A_2 LH A_2	ALDERSON ALDERSON	983 778	48 32	– –	1.75 2.7	1.75 2.7

Figure 16.

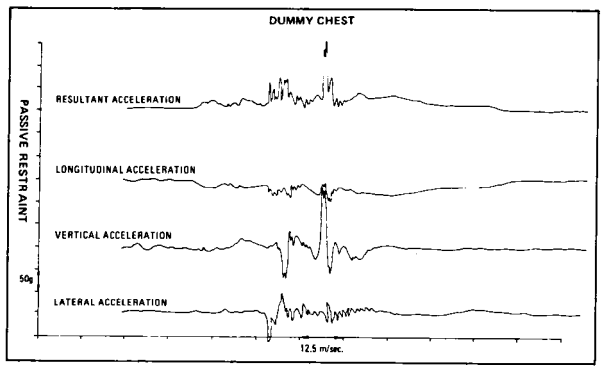


Figure 17

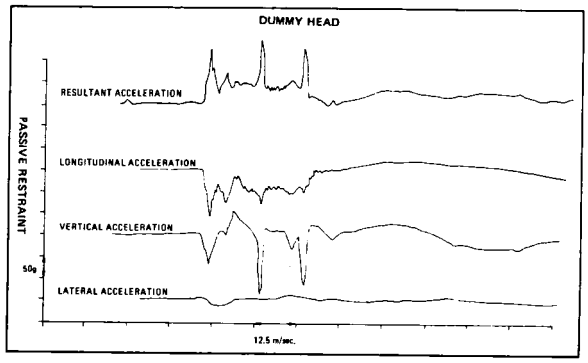


Figure 18.

head-to-windshield impact velocity was still too high. The femur load was of a much lower order.

Detailed magnetic tape analysis of MC2 and MC3 tests showed that unusually high head and chest vertical deceleration peaks were evident on all traces when the dummy neck support wires were set in the lateral position. Figures 17 and 18 show the offending peaks and the influence on the resultant values.

The rear restraint performed well, the results being within the specified occupant protection level. Although the results were encouraging, improvements in performance could be achieved by increasing the stiffness of the restraint. This would have a two-fold effect.

1. More efficient energy absorbtion by reducing the displacement and peak load – present late peaks indicate bottoming out.
2. The reduced displacement would eliminate any front and rear dummy interaction.

Action: Revert to the stiffness 3 front restraint with a modified knee restraint, intended to make a more acceptable contribution to the dummy deceleration. Fit a stiffer rear restraint. Remove the neck wires from both dummies.

Sled Test MC4

Stiffness 3 restraint fitted in the L.H. front passenger position. Stiffness A_1 restraint fitted in the R.H. rear passenger position. It is evident that the stiffer front restraint gives more acceptable protection levels than the less stiff ones. Although results show an acceptable head level, repeatable results cannot be guaranteed due to the various types of head contact that are possible, i.e.: header only, glancing header blow into windshield, etc.

The knee restraint bottomed out on the L.H. side resulting in an unacceptable load on the left femur.

Although the stiffened rear restraints met the specified protection level only limited success was achieved in reducing the restraint displacement. However film analysis showed that it was unlikely that the rear dummy could contact the front one.

The removal of the dummy neck wires eliminated the spurious high acceleration peaks experienced on previous tests.

Action: Confirm the favorable results of the four sled tests by conducting a frontal barrier test at 48 km/h, fitted with one front restraint, a modified front knee restraint and two identical stiffer rear restraints. Identical rear restraints in the same test allow a check on restraint performance repeatability.

Barrier Test MB1

Stiffness 3 front restraint fitted in the L.H. front passenger position. Two stiffness A_2 rear restraints fitted in the rear passenger positions. All values were of an acceptable level except the front head which, due to the vehicle "pitching" on impact, contacted the header rail with no windshield contact. This highlights the problems of packaging the front restraints into a low roof line vehicle.

The rear restraints had reduced displacement with acceptable head, chest and femur values but further tests will be required to establish repeatability.

Film analysis showed that there was no contact between the front and rear dummies during the deceleration period.

CONCLUSIONS TO DATE

- It is technologically possible to provide occupant protection in a 48 km/h frontal barrier impact using the type of static passive system described.
- The restraint collapse characteristics and the large chest contact area can result in acceptable occupant protection levels.
- To provide optimum head protection for the front passenger, modifications are required to reposition the windshield and the header rail.
- With correct dummy kinematics and vehicle package, the windshield can be an effective head restraint although improvements in laceration index need to be achieved. Such improvements seem likely in the near future.
- Test results show that the rear restraints have been consistently successful in meeting the specified occupant protection criteria.

FUTURE WORK

- Continue the development of the composite restraint to improve its collapse characteristics during impact and its acceptability during normal service.
- Investigate the feasibility of using other energy absorbing materials.
- Continue the windshield evaluation tests and keep abreast of advances in windshield technology with respect to head to windshield contact.
- Investigate the use of belts with the passive restraint system with a view to survivability at higher impact velocities and in multiple impacts.
- Develop the restraints to extend acceptable occupant protection levels to larger and smaller occupants by testing with 95 percentile and 5 percentile dummies.
- Investigate the effects of ageing, and vibration, on the restraints.
- Establish the level of customer acceptance by conducting a subjective appraisal of the system under actual service conditions.

ACKNOWLEDGEMENTS

The authors would like to express their thanks to the staff of MIRA and TRRL, and colleagues at the Ford Motor Company Research and Engineering Center for their cooperation and assistance in this project.

APPENDIX
(CIBA-GEIGY (UK) LTD)

PHENOLIC FOAMS

Phenolic foams are produced by the exothermic acid curing of resole resins in the presence of a low boiling point liquid blowing agent.

Properties of Phenolic Foams

Phenolic foams can be produced with cell sizes ranging from very small to large when compared with other rigid foams e.g. polyurethane, and with open and closed cells. The type of open celled foams can also be varied from foams giving only very slight water pick up to those which rapidly absorb large quantities of water, e.g., foams for floral arrangement. These changes in structure also have an effect on compressive strength, friability etc., as also does the final density of the material. The careful control and choice of materials are used to produce foams of various types and properties, usually at a density of 30-80 kg/m^3 (2-5 lb/ft^3) although lower and higher densities can be obtained with some systems.

The compressive strength of phenolic foams is comparable to that of polyurethane at similar density.

The load deformation characteristics of phenolic foams, which show the load deflection curve to have a square wave form, are especially suited for certain applications such as crash panels in automobiles.

While phenolic foams are more friable than polyurethane foams when measured according to the ASTM C367 and C421 methods, these foams have been shown to last the lifetime of fabricated finished articles without showing such evidence of friability as settling or dusting. Testing of phenolic foam used to insulate refrigerated trailer trucks in the U.S. showed that the materials had withstood over 90,000 miles and 3 years of service with no evidence of mechanical failure, such as settling.

Phenolic foam is outstanding in dimensional stability and can be used from liquid nitrogen temperatures to a continuous 150°C. The thermal insulation is good, especially with closed cell foams and the sound absorption coefficients are very high in comparison with those of other rigid plastic foams. Phenolic foam is insensitive to moisture and other potentially corrosive materials.

The fire properties of phenolic foams are exceptional for an organic based cellular material.

They are not readily ignitable (by all available tests) and meet FMVSS-302 flammability requirements. They show Class O fire propagation (BS476 Part 6), Class I, nil spread in the surface spread of flame test (BS476 Part 7) and show excellent fire resistance properties (BS476 Part 8). They show high weight retention in tests such as the Butler Chimney Test with no after burning and produce very little smoke on burning (BS476 Part 9, XP2 and NBS smoke chamber tests). They also show a high oxygen index (ASTM 2863) of 36 and due to their slow rate of burning when exposed to a flame do not produce rapid high levels of toxic gases such as carbon monoxide. Because of these excellent fire properties flame retardant additives are unnecessary as their addition would in fact have a deleterious effect on certain properties such as smoke evolution.

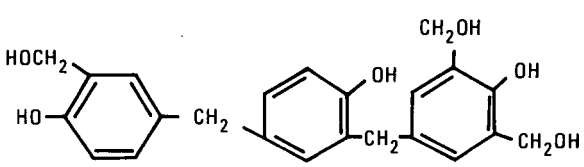
Materials Required for Production of Phenolic Foams

For production of phenolic foams the following materials are required:

- Resole resin
- Acid (Curing Agent, Hardener): to produce the exotherm and cure the resin
- Blowing Agent: to expand the system
- Surfactant: to regulate cell structure
- Additives and Modifiers (Optional)

The final type and properties of the foam required dictate the types of materials to be used and careful formulation is required.

Resole Resin – Resole resins are prepared by the alkaline addition and condensation of phenol and formaldehyde, employing a molar excess of formaldehyde. They are thermosetting and thus of limited shelf life. The resins are complex mixtures of mononuclear and polynuclear phenolic molecules containing reactive methylol ($_CH_2OH$) groups. A typical resole structure is:



On addition of acid further reaction occurs, generally by condensation of methylol groups with phenolic nuclei to produce methylene ($_CH_$) linked structures, giving finally a cured, i.e., crosslinked, material.

The resole resin used for phenolic foam production is a high solids content, low water content material of carefully controlled degree of reaction. On foaming the resin it is essential that a known and controllable exotherm will occur on acid addition followed by final gelation and cure at the correct time. The viscosity of the foam mix is important and this is generally controlled by the resin viscosity.

A highly exothermic resin will give a foamed system of uneven structure with large blow holes. A resin of very low exotherm will not expand sufficiently when foamed. Should the resin gel before full expansion has occurred rupture of the system will occur and should the resin not gel until after full expansion foam collapse can occur.

Acid (Curing Agent, Hardener) – The type and quantity of acid hardener is important as different acids behave differently, e.g., some give a rapid exotherm on addition to the resin while others give little exotherm although leading to a crosslinked structure. Thus tailoring of the acid type can be used to modify the exothermicity of the system to some extent. Acids can have an effect on both cell size and the closed/open cell ratio of the foam.

Blowing Agents – The blowing agents which can be used fall into two main categories, (a) partial resin solvents, e.g. methylene chloride and (b) non-solvents, e.g., pentane, Arcton and Freon chloro-fluorinated hydrocarbons. The partial resin solvents generally produce larger celled foams, although there is the advantage that the resin/blowing agent mix/emulsion is of a lower viscosity which aids the final mixing process, whether batch or continuous.

Surfactants – The type of surfactant used controls cell size and type, i.e. closed or open in conjunction with the other raw materials. Nonionic surfactants which are either ethoxylated products or silicone polymers have been found to be the most suitable. Acid and heat stability are required for the production of consistent foam.

Additives and Modifiers (Optional) – As the foam system depends on a carefully controlled exotherm, temperature control of input materials is necessary. The exothermicity of the system, however, may be controlled to some extent by addition of certain chemicals, e.g., furfuryl alcohol. Mineral fillers may also be used.

D. BURLAND
Ford Motor Company Limited

INTRODUCTION

This program was conceived in conjunction with the Transport and Road Research Laboratory early in 1973. The objective was to provide improved rear impact protection for occupants of a vehicle with due regard to minimizing both initial cost and repair costs.

A complete study of the cost effectiveness aspects was outside the scope of the project and it was decided the best initial approach was through the retention of conventional materials and build techniques. A constraint that the program be based on an underfloor fuel tank installation was established to satisfy the needs of estate cars and give maximum freedom to future design trends.

Although it is not an occupant protection requirement, it was also decided that minor accident damage, especially of safety related components should be eliminated at very low speeds by the use of a "no damage" bumper system. The significance of pedestrian accidents (26% of all serious injuries in road accidents in Britain) influenced the bumper design with respect to height above ground.

Accident data were also reviewed to determine the performance needs of the design. Hazards to occupants are generally considered to come from two sources, (a) deformation and deceleration during impact and (b) fire occurring in association with fuel leakage. In practice rear impacts are responsible for 7% of serious and fatal car occupant casualties, Reference 3 and Figure 1.

Mackay shows rural and urban rear impacts to account for only about 10% of all car impacts, Reference 1. The percentage of car accidents where fire occurs is around ½% and only a proportion of these are due to rear impact. The most common source of fire is believed to be under the hood.

Further detail breakdown of rear impact car accidents involving fire is not available, presumably because of the thankfully low incident rate. It is to be remembered that for a fuel fire to be started, not only must fuel be leaked but an ignition source and correct air/fuel ratio must be simultaneously available at the same place.

Background Ford expertise of rear impact testing vehicles contributed to establishing performance targets, worthwhile occupant protection measures and vehicle design.

TABLE FROM TRRL LEAFLET

CAR OCCUPANT CASUALTIES INVOLVED IN CAR REAR IMPACTS

Damage to Car Occupied	Numbers of casualties in 1971. Percentages of totals in ( )							
	Drivers		Front Passengers		Rear Passengers		All Occupants	
	Fatal	Serious	Fatal	Serious	Fatal	Serious	Fatal	Serious
Symmetrical rear impact (B, BLR or FB *damaged)	30 (1.8)	664 (3.2)	13 (1.6)	364 (3.3)	12 (3.6)	179 (3.6)	55 (2.0)	1207 (3.3)
Asymmetrical rear impact (BL, BR, FBL or FBR damaged)	58 (3.5)	778 (3.8)	22 (2.7)	393 (3.5)	18 (5.3)	274 (5.6)	98 (3.5)	1445 (3.9)
Total with damage to Back B	88 (5.3)	1442 (7.0)	35 (4.3)	757 (6.7)	30 (8.9)	453 (9.2)	153 (5.5)	2652 (7.2)
Total occupant casualties	1651	20652	811	11229	338	4906	2800	36787

*Side damaged Front – F Left – L
Back – B Right – R

Figure 1

PROGRAM

The project work program was established as shown in Figure 2. Basically the program is in two stages, the first leading to a dynamic crash test of the Phase I prototype to assess at an early stage validity of design. Phase II incorporates refinements to optimize or improve the design. This paper reports on work carried out up to January 31, 1974, by which time Phase I had been completed and Phase II commenced.

DESIGN CRITERIA AND PERFORMANCE OBJECTIVES

Under impact conditions, the structure of the vehicle has several functions:

1. to limit the occupant acceleration levels,
2. prevent excessive distortion of the passenger compartment,
3. protect the fuel system from damage that could cause leakage.

To meet the objective of maintaining fuel tank integrity with an underslung tank, a space for the fuel tank had to be provided in the crash situation.

The test condition was determined from accident data and feasibility studies. Legislation and ESV proposals have been made with quite severe requirements, e.g., 48 km/h fixed barrier rear impacts, dissipating 118,000 Joules in Cortina. We have compromised on a level of 80,000 Joules, by testing at 48 km/h using a mobile barrier of mass 2720 kg.

We estimate that this would provide protection for 80% of European rear impacts by interpretation of Mackay, Reference 9.

The test was to be completed at MIRA where 48 km/h is a standard impact speed and therefore more accurate. No test method is perfect but the

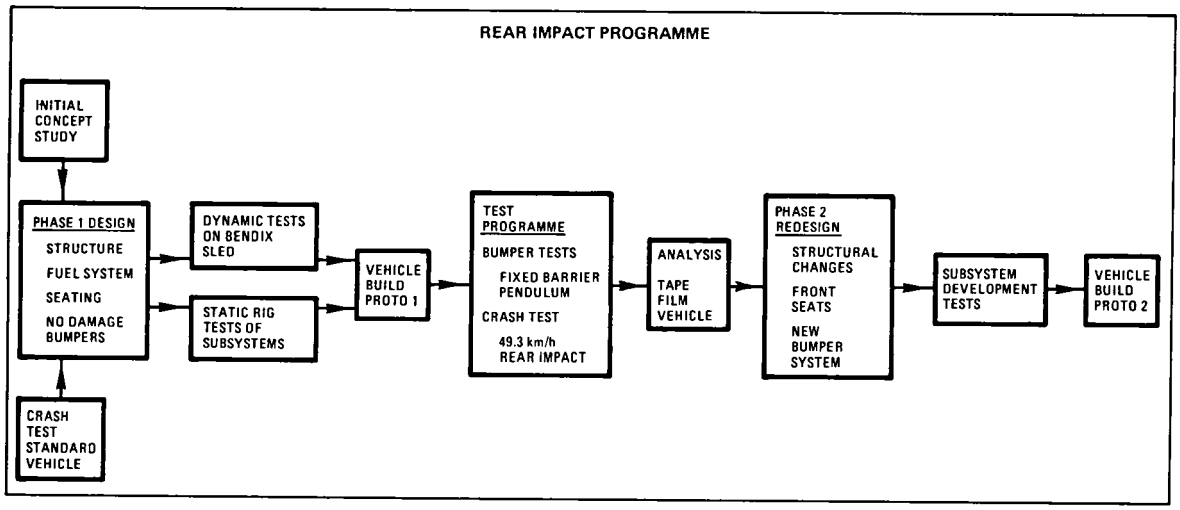


Figure 2

mobile barrier test does avoid the destruction of an extra vehicle and the additional variables introduced by it. The relatively flat front of the mobile barrier is paralleled by the load spreading action of the heavy duty bumper system assumed to be fitted to both vehicles, although the extreme rigidity of the MB face has to be taken into account when calculating equivalent impact speeds.

BASELINE TEST

The first necessity was to establish the performance of a standard car under the extremely severe crash conditions proposed. These data were combined with our existing experience of rear impact testing of the Cortina.

The baseline test revealed that for these high crash speeds, additional reinforcement would be necessary to reduce passenger compartment deformation and improve rear end energy absorption characteristics.

Photographs, Figures 3-5, show the vehicle before and after test; notice the distortion of the door frames and roof structure. Rear end collapse was 520 mm and fuel leakage occurred due to contact of the tank wall with the differential housing and shock absorber mounting.

This test also provided input data for a computer model, set up to determine seat back strength requirements.

Test results from the baseline test indicated that the car occupants would have survived the impact. Figure 6 summarizes results. Improvements considered worthwhile to reduce "injury" (acceleration) levels in dummies were:

1. reinforcement of the front seat to prevent contact with a possible rear seat occupant,
2. introduction of a rear seat head restraint to

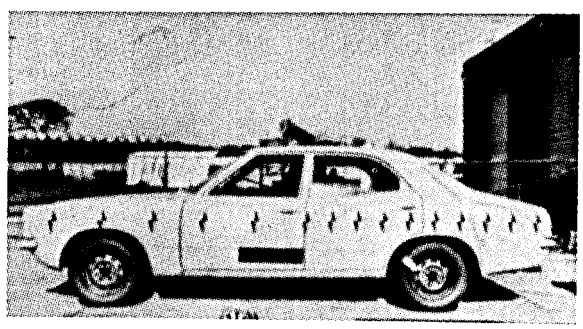


Figure 3

Figure 4

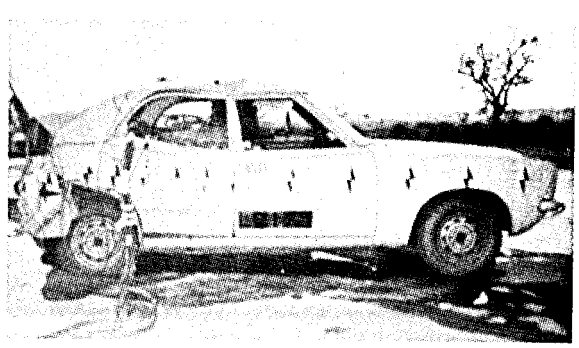


Figure 5

BASELINE CORTINA TEST				
	DUMMY OCCUPANT INJURY LEVELS			
	FILTER CLASS	PEAK RESULTANT 'g'	HEAD INJURY CRITERIA	GADD SEVERITY INDEX
FRONT DUMMY HEAD	1000	99	363	478
FRONT DUMMY CHEST	180	31		93
	FILTER CLASS	PEAK RESULTANT 'g'	HEAD INJURY CRITERIA	GADD SEVERITY INDEX
REAR DUMMY HEAD	1000	82	240	561
REAR DUMMY CHEST	180	53		177
Note: Survival limits for HIC and Gadd Index are considered to be 1000 Filter Class based on SAE J211				

Figure 6

cushion head impact with seat frame or parcel shelf and reduce rearward head rotation.

DESIGN IDEAL

The ideal collapse characteristic, providing the occupant with a long and constant acceleration pulse is impractical. Apart from the problem of producing a design to give a constant acceleration level, the distance available for collapse would only be optimum for one particular speed. It is therefore necessary to compromise between design feasibility, desirable pulse shape and maximum impact speed.

Experience has shown that current techniques for building mass production cars produces structures whose collapse force is suited well to producing tolerable passenger acceleration levels. High frequency acceleration levels are attenuated by restraint systems, seat cushioning and human response characteristics. In the rear impact case the "restraint" system is the vehicle structure and seat assembly that must provide the desired pulse. Assuming the seat structure to be rigid, the onus falls upon the vehicle structure.

It was calculated that with sufficient structural modification, the dynamic collapse of the body structure could be reduced to 400 mm allowing sufficient space for a new relocated fuel tank to remain intact between the rear axle and back panel.

The structure surrounding the fuel tank installation had to preserve this collapse distance without the likelihood of puncture or rupture by distortion. Protection of filter pipe, petrol pipe and sender unit also had to be provided.

The use of heavy 8 km/h "no damage" bumpers would be valuable in protecting the fuel tank installation by completing the box of structural members around the tank, see Figure 7. The ISO Exterior Protection Standard was set for bumper performance, but the speed was increased from 4 km/h to 8 km/h.

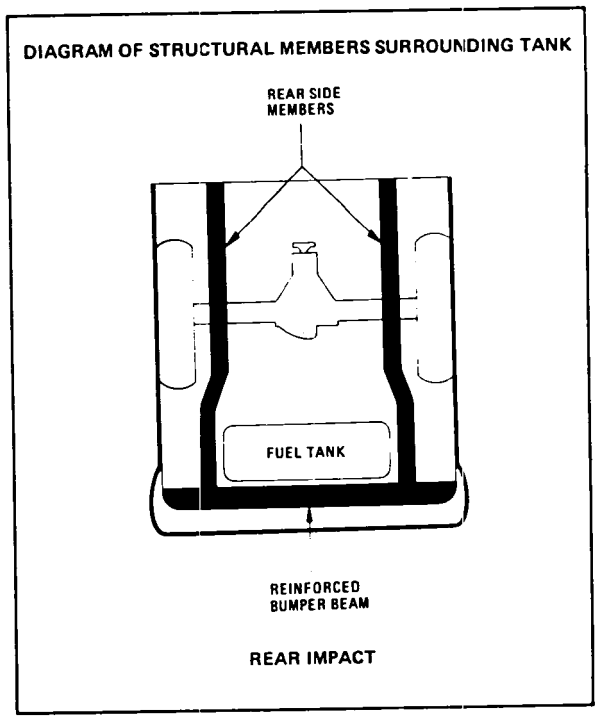


Figure 7

SUMMARY OF DESIGN OBJECTIVES

- Maintain fuel system integrity in a 48 km/h rear impact by mobile barrier of mass 2720 kg.
- Ensure dummy acceleration levels are below accepted injury criteria.
- Minimize passenger compartment intrusion.
- Meet performance objectives with conventional materials and build techniques.
- Incorporate "no damage" bumper system to meet the ISO recommended practice, but at an increased speed of 8 km/h.

VEHICLE MODIFICATIONS

The major issue was location and layout of the fuel tank and filler pipe. The alternatives available for underfloor tank location with rear wheel drive, are:

1. tank positioned immediately behind the rear axle with sufficient energy absorbing structure behind the tank to prevent its being crushed.
2. tank positioned to the rear of the car such that under crash conditions sufficient space be available to accommodate the tank (see Figure 8).

The structure of Cortina folds naturally behind the rear wheelarch due to sidemember bends and panel joint lines. It was considered a major task to change this collapse mode to ensure collapse behind the tank and would necessarily incur a large weight penalty.

The choice of tank location was therefore dictated as position (b) with a rear filler pipe. Protection of

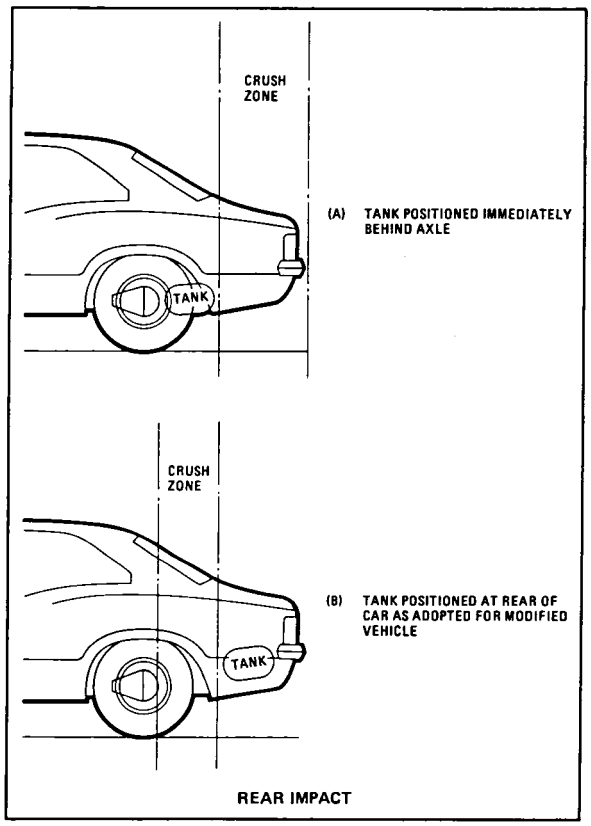


Figure 8

the tank and filler pipe was to be provided by the reinforced beam structure needed to support the energy absorbing bumper system.

Structural Body Modifications

The baseline test revealed the need to strengthen the glasshouse, making it compatible with the reinforced rear structure and confirming major deformation to the rear of the vehicle.

This was basically accomplished by spot welding the inner flanges of the roof rails to the roof panel, thereby closing the open "C" section of the roof rails.

Torque boxes connecting the rear side members to the sills were introduced to stiffen door frame and strengthen the rear sidemember.

The rear sidemembers were doubled in strength along their length –

- to provide sufficient strength to withstand the low speed bumper tests,
- increase energy absorption.

The rear quarter panels were double skinned to ensure that the sheet metal contributed more to energy absorption.

Strengthening of the propshaft tunnel and rear bulkhead were also included. Figure 9 summarizes the major modifications.

Total sheet metal additions including bumpers

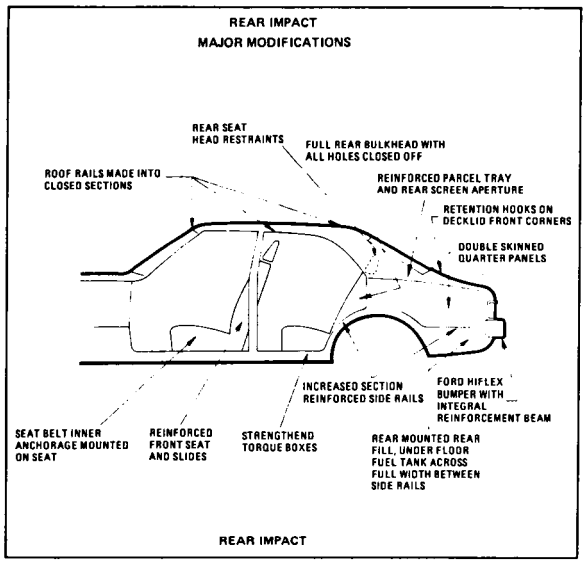


Figure 9

increased body weight by 55 kg of which 25 kg was for high speed impact and 30 kg for no-damage bumpers. However, there is little doubt that this weight could be greatly reduced by –

- further test and development work,
- redesigning the rear end completely rather than modifying an existing vehicle.

Fuel Tank Installation

A review of fuel tank construction techniques led back to the accepted pressed steel type. Flexible tanks proved to be expensive and unsuited to mass production techniques. Plastic tanks (high density polyethylene) are an open issue both on cost and performance. It was therefore decided to continue with the conventional steel tank, but to include safety features as illustrated in Figure 10 as follows:

1. Filler tube with flexible piping to accommodate body distortion during impact.
2. Recessed sender unit and petrol pipe pick up tube to avoid fracture or disconnection.
3. Strengthened tank retention straps with captive end fittings.
4. Elimination of all protrusions likely to contact the tank.
5. Anti-surge flap valve in filler pipe to prevent fuel being forced out through filler cap by inertia effects.

No Damage Bumpers

There are many systems available to meet the 8 km/h "no damage" regulation required by U.S. Federal law. In order to make maximum progress on Phase 1 of the project, initial work was based on a reinforced bumper bar mounted on production

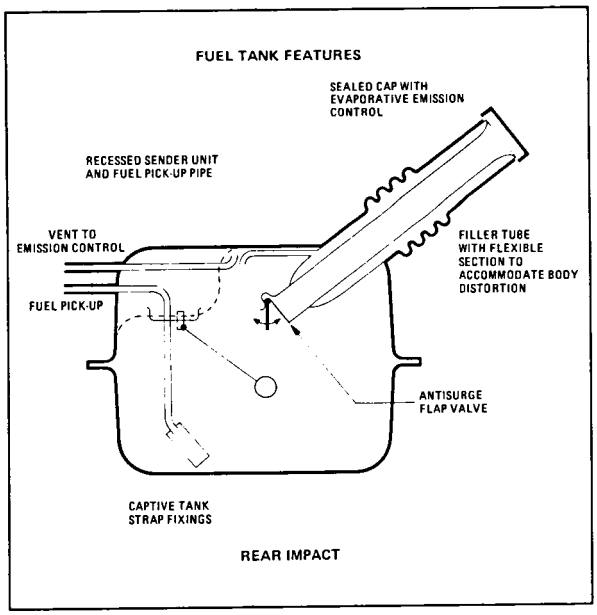


Figure 10

rubber in shear units. This performed satisfactorily but was inefficient in that it in no way utilized the increased strength of the rear of the vehicle as a whole. For Phase II the approach was to package the Ford Hiflex system, but with a new integrated beam to save weight and cost.

Changes to sidemember configuration were necessary to support the bumper at a reduced height. The mid-height of the bumper being established at 380 mm by the TRRL as the maximum desirable height from the pedestrian collision point of view.

Section depth of 127 mm was agreed with TRRL as desirable to ensure bumper to bumper contact when vehicles collided under various dynamic height conditions, and to spread pedestrian contact area.

The Value of "No Damage" Bumpers

Tests on the initial system highlighted important issues that must be carefully considered before vehicles are fitted with these units.

"No damage" protection can obviously be provided at much higher levels than the 8 km/h suggested. But the problem is not simply a question of energy absorber technology.

The loads generated amount to several thousand kg, and are transmitted to the body structure in very localized areas. With structural reinforcement and front/rear energy absorbing bumper systems, the weight added will be equivalent to carrying around an extra person. This will affect not only original cost and cost of repairs, but also performance and economy. In order to absolutely prevent over-ride bumper impact conditions, expensive suspension modifications will need to be introduced to maintain

Figure 11

ride height. The aggressive tank-like front structures that would be developed may have a more devastating effect when in collision with the more vulnerable side structures of other vehicles.

Low-speed no damage bumpers do not help significantly in higher speed impacts; in a 48 km/h frontal impact with a lorry or bridge pillar, the energy absorbed by the bumper system will be less than 3% of the total energy of deformation!

The relevant ISO standard proposes a similar test, but at the lower speed of 4 km/h. It would seem prudent therefore to obtain experience at a lower speed than 8 km/h on the cost effectiveness of improved bumper systems. This speed could then be readily increased if it was proved to be worthwhile.

Boot Lid Retention

It was concluded that more energy absorption might be gained from the boot lid if it was retained at the hinges, instead of simply moving forward under the impact.

Modifications were made to the hinge as shown on Figure 11. The adjacent parcel shelf was stiffened to take the extra load and help keep the rear window and its frame intact and avoid penetration of rear screen.

INTERIOR OCCUPANT PROTECTION

In rear impacts the occupants can be almost ideally cushioned to receive the blow, spread over a

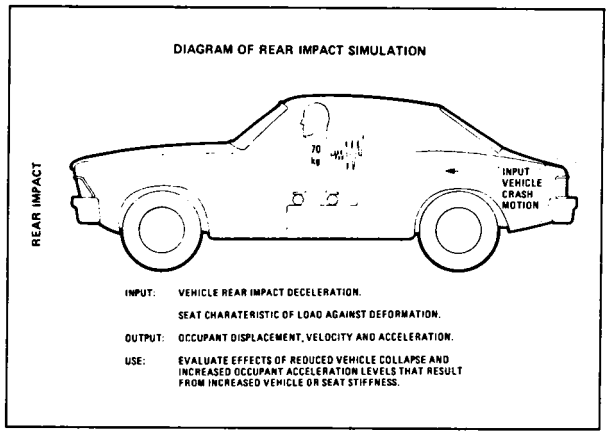


Figure 12

large body area. However, two problems arose with the standard car:

1. the reclining action of the front seat may cause the seat back and its occupant to interfere with the rear passengers.
2. the rear passenger head had no restraint.

It is important that the strength of a normal seat back without head restraint should not be too great. By allowing the back to yield and recline under rear impact conditions, the severity of neck whiplash injuries can be reduced.

However, to eliminate whiplash of the neck, minimize the risk of upward movement of the occupant up a reclined seat back and to eliminate interaction between front and rear seat occupants, a stiffened seat with head restraint was devised.

The design catered for a 20 "g" dummy loading condition determined from the maths model, Figure 12. Initial static tests reached a load of 12 kN applied at torso center of gravity, this being approximately eight times the load capacity of a standard seat. Yield under this load was around 100 mm measured at the loading point. The dynamic seat performance was confirmed by test on the Ford Bendix Accelerator, Figure 13.

A simulated vehicle acceleration was applied to the sled and the seat withstood the inertia load produced by a 50 percentile dummy.

The rear seat head restraint was based on designs for vehicle front seats. This amounted to a simple roll of sheet metal spot welded to the top of the rear seat frame and finished with foam padding.

Height of the restraint above the occupant "H" point was 700 mm, measured parallel to the torso line. Further height increase was not possible without compromising rearward visibility.

The build included inertia reel seat belts front and rear, as these are considered the most effective safety features yet devised. Although designed fundamentally for frontal accidents, seat belts can

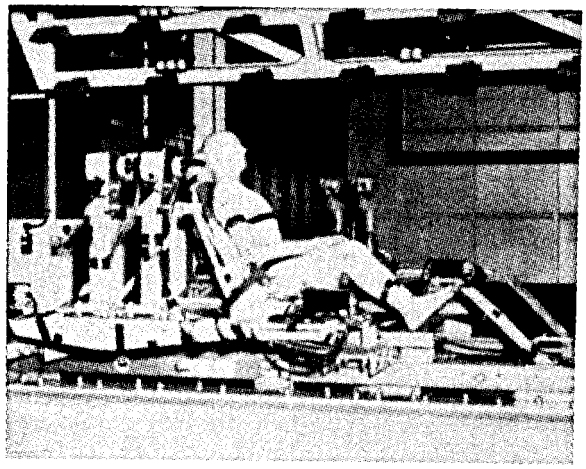


Figure 13

help in rear impacts to prevent occupants sliding up the seat back.

TEST PROCEDURE

The crash performance was assessed at the MIRA facility using a mobile barrier similar to SAE 972, but of increased mass.

The mobile barrier of weight 2720 kg struck the rear of the stationary vehicle at 48 km/h. The car was fully instrumented with two anthropometric Alderson VIP 50a dummies aboard; one seated in the front and one in the rear.

Instrumentation characteristics followed the recommendations of SAE J211 for both occupant and vehicle acceleration measurements.

High speed cine cameras recorded the event in minute detail. Camera layout shown diagrammatically in Figure 14.

RESULTS AND DISCUSSION

The vehicle structure followed its design requirements and collapsed mainly above and behind the rear axle. Figures 15-20 show the modified vehicle before and after test and Figure 21 shows vehicle and barrier acceleration measurements. The fuel tank remained intact although acceleration levels in excess of 21 "g" were encountered. The RHS of the tank was depressed by the lower mounting bolt of the rear shock absorber but due to the special protective nut, no puncture was made. The sender unit and fuel pipes were protected in their recessed position. Contact with the differential housing was just avoided. The anti-surge valve was successful in stopping fuel leaking from around the filler cap.

Passenger compartment deformation was minimal despite 468 mm dynamic collapse of the rear body structure (not including bumper). Both dummies

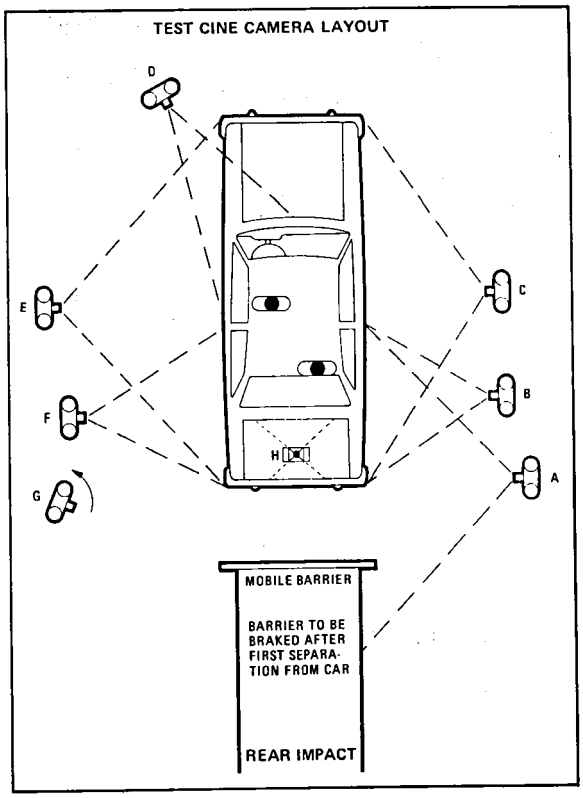


Figure 14

easily survived this impact based on the injury criteria of FMVSS 208, (see Figure 22.).

The front seat frame withstood the high "g" loading but some yield was evident of the floor pan itself. This allowed the seat to tilt 20° rearwards under the dummy inertia load, although not enough to trap the rear seat passenger.

Figure 15

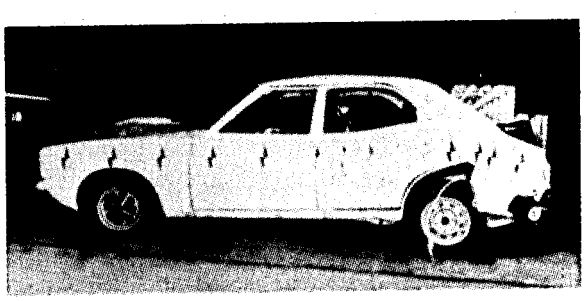


Figure 16

Figure 17

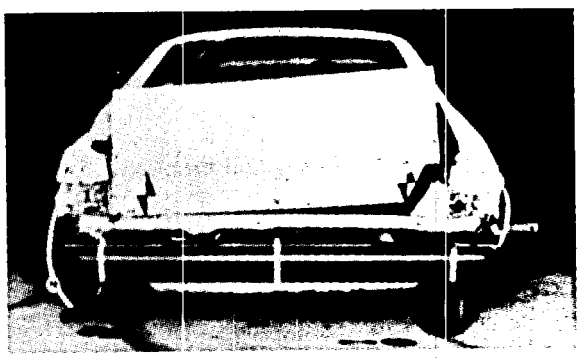


Figure 18

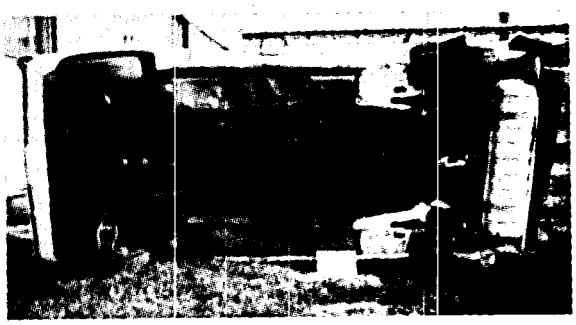


Figure 19

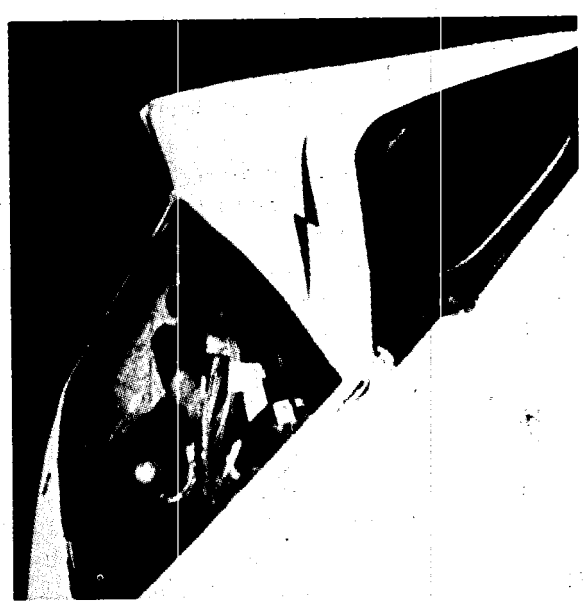


Figure 20

The seat was designed to demonstrate the strength level required to prevent total collapse and is not a representative unit for manufacture. The value of the head restraint could be increased by widening it to cater for angled impacts or by adopting the recommendation of References 4 to 8. These reports suggest high seat backs as the answer, despite reduction in side and rear visibility. The intention is for the front seats to have a performance similar to a high back rear seat in preventing excessive hyperextension of the neck.

It is clear that the risk of "hyperextension" of the neck can be minimized by the use of a sufficiently high and rigid head restraint. However, at the present time this type of injury does not manifest itself in great numbers due to the fact that in most vehicles the front seat will collapse under the inertia of its occupant before neck injury occurs.

The rear seat head restraint was highly successful in cushioning the dummy head impact shown by the Head Injury Criteria of only 85 compared to the nominal tolerance limit of 1000.

The retainers fitted to the boot lid hinges proved satisfactory, Figure 20. The forward edge of the boot lid was held fast while the main body of the lid folded in a useful, energy absorbing manner.

CONCLUSIONS

Although a substantial weight penalty has been introduced it has been shown possible to provide occupant protection and maintain fuel tank integrity in a rear impact approximately equivalent to an 80 km/h vehicle to vehicle.

The following key items are noted:

- The principle of providing "collapse space" for the fuel tank was shown to be successful.
- Elimination of protrusions close to the fuel tank reduced the risk of puncture.
- Recessing the sender/fuel pick up pipe in the fuel tank protected them from impact damage.
- Captive ends to the petrol tank straps prevented the tank being released from its body mounting.
- Fitting an anti-surge valve in the filler pipe was effective and there was no leakage from the spring loaded petrol cap.
- Retaining the leading edge of the boot lid prevented intrusion through the rear screen.
- The simple rear seat head restraint was effective in reducing dummy head accelerations.

With respect to the validity of some of the test targets, it should be noted that:

- Further accident and injury data is required to determine the cost effectiveness of reinforcing the front seats.

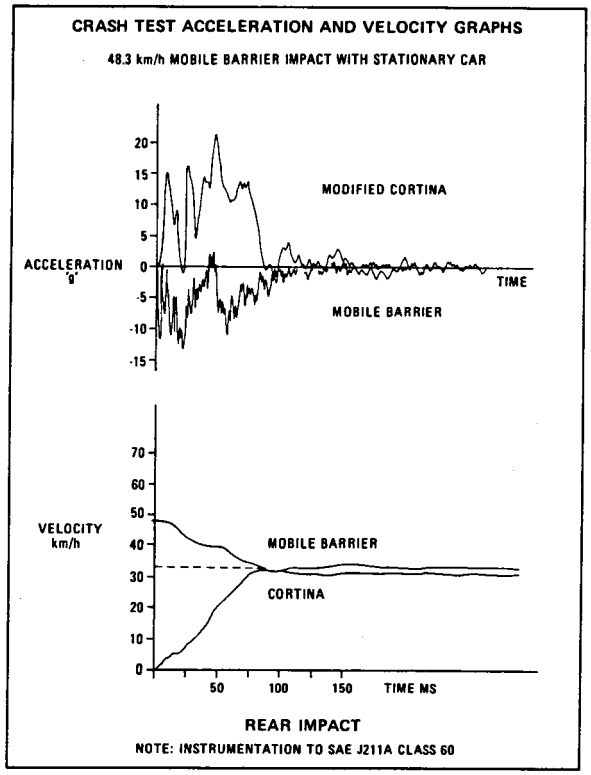


Figure 21

- The 8 km/h no damage bumpers imposed a high penalty in terms of cost and vehicle performance. It is recommended that no higher-level of bumper requirement be specified than the relevant ISO Exterior Protection practice.

FUTURE WORK

Complete build of Phase II prototype which differs from Phase I as follows:

- Bumpers – Installation of Hiflex Polyurethane bumper system with integral reinforcement beam.
- Seat Slides – Reinforced seat mounted on adjustable strengthened slides.

MODIFIED CORTINA					
		DUMMY CRASH TEST RESULTS			
		FILTER CLASS	PEAK RESULTANT 'g'	HEAD INJURY CRITERIA	GADD SEVERITY INDEX
FRONT DUMMY	HEAD	1000	44	193	303
	CHEST	180	19	-	79
		FILTER CLASS	PEAK RESULTANT 'g'	HEAD INJURY CRITERIA	GADD SEVERITY INDEX
REAR DUMMY	HEAD	1000	32	85	153
	CHEST	180	34	-	142

Note: Survival limits for HIC and Gadd index are considered to be 1000
Filter Class based on SAE J211

Figure 22

- Boot Lid Retainers – Reduced weight units.
- Structure – Minor changes to Phase I panels.

Phase II vehicle to undergo low speed bumper tests and final 48 km/h rear impact test at MIRA.

ACKNOWLEDGEMENTS

The authors would like to express their thanks to the staff of MIRA and TRRL, and colleagues at the Ford Motor Company Research and Engineering Center for their cooperation and assistance in this project.

REFERENCES

1. Dr. G. M. Mackay, "Causes and Effects of Road Accidents" Vol. 3. Report from University of Birmingham, Department of Transportation and Environmental Planning.
2. A. M. Naburn, A. W. Siegel, et al., "Injuries to Rear Seat Occupants in Automotive Collisions." SAE Paper 670918.
3. Transport and Road Research Laboratory, "Car Rear Impacts and the Need for Headrests." Leaflet number LF359, April 1973.
4. D. M. Severy, H. M. Brink, et al., "Backrest and Head Restraint Design for Rear End Collision."
5. D. M. Severy, H. M. Brink, et al, "Safer Seat Designs." Proceedings of the 13th Stapp Car Crash Conference, 1969 (SAE Paper 690812).
6. Roger P. Daniel, "Vehicle Interior Safety Constraint Systems," 1970, SAE Paper 700423.
7. R. J. Berton, "Whiplash, Tests of the Influential Variables," 1968, SAE Paper 680080.
8. H. J. Mertz and L. M. Patrick, "Investigation of the Kinematics and Kinetics of Whiplash." Proceedings of the 11th Stapp Car Crash Conference, 1967. SAE Paper 670919.
9. G. M. Mackay, "Field Studies of Traffic Accidents in Europe." Paper from the Kyoto Fourth International Technical Conference on Experimental Safety Vehicles.
10. P. A. Weller, "Urethane Energy Absorbers for Automobile Bumpers." SAE Paper 730025.
11. P. A. Haines, "Urethane Flexible Body Parts," SAE Paper 730026.
12. H. Hintyen and R. A. Dunleavy. "An Energy Absorbing Elastomeric Bumper." SAE Paper 730027.
13. J. E. Gieck, D. A. Weitzenhof, E. J. Merz, "A High Energy Level Pneumatic Energy Absorbing Bumper." SAE Paper 730029
14. R. R. Golze and R. F. Kieule, "Bumper Design, Materials and Fabrication." SAE Paper 730030.
15. K. H. Carpenter, L. Kerr, "The 1973 G. M. Hydraulic Pneumatic Energy Absorber Bumper System."
16. U. Seiffert and P. Kirschner, "1974 VW Energy Absorbing Bumper System." SAE Paper 730033.

OCCUPANT PROTECTION DURING VEHICLE ROLLOVER

K. STONE
Ford Motor Company Limited

INTRODUCTION

Ford Motor Company Limited entered into a contract with the United Kingdom Government Transport and Road Research Laboratories (TRRL) in February 1973 to investigate:

- Vehicle and occupant kinematics during vehicle rollover.
- The effects of roof strength on roof intrusion and occupant injury.
- The use of interior padding to reduce occupant injury.

The Motor Industry Research Association (MIRA) would carry out the vehicle test work as required by the TRRL contract.

The program consists of three phases as shown in Figure 1. The initial phase is a general study of various sedan vehicles, the second phase is a study of one sedan model with variations in structural strength. The third phase consists of a padding materials study and their application in reducing occupant injury levels.

Earlier work on rollover test methods (1) concluded that:

- Injuries sustained by occupants were severest when ejection occurred.
- The most representative and repeatable rollover test method was the sled roll method. This method also facilitated the use of cameras and instrumented occupants to record injuries sustained.
- The roof crush test method used gives a static measure of roof strength and has a higher degree of repeatability.

With these points in mind both sled roll and static roof crush tests are incorporated in the program.

Rollover tests are carried out at the MIRA Crash Test Laboratory. To induce the broadside roll of the test vehicle, it is placed on a trolley which is accelerated to the required test velocity and then almost instantaneously stopped. The vehicle continues and rolls laterally off the trolley. This system is similar to the Mercedes-Benz arrangement in their early crash test investigations and the subsequent FMVSS procedure.

For the rollover tests dummies are fitted in the front seats only. This facilitates the dummy instrumentation and the filming of the dummy dynamics and is justified on the following basis:

- Occupancy of front seats is much greater than for rear seats.
- Ejection is much more likely for front seat occupants.
- Dummy dymamics, except for ejection, would be similar for rear seat occupants.

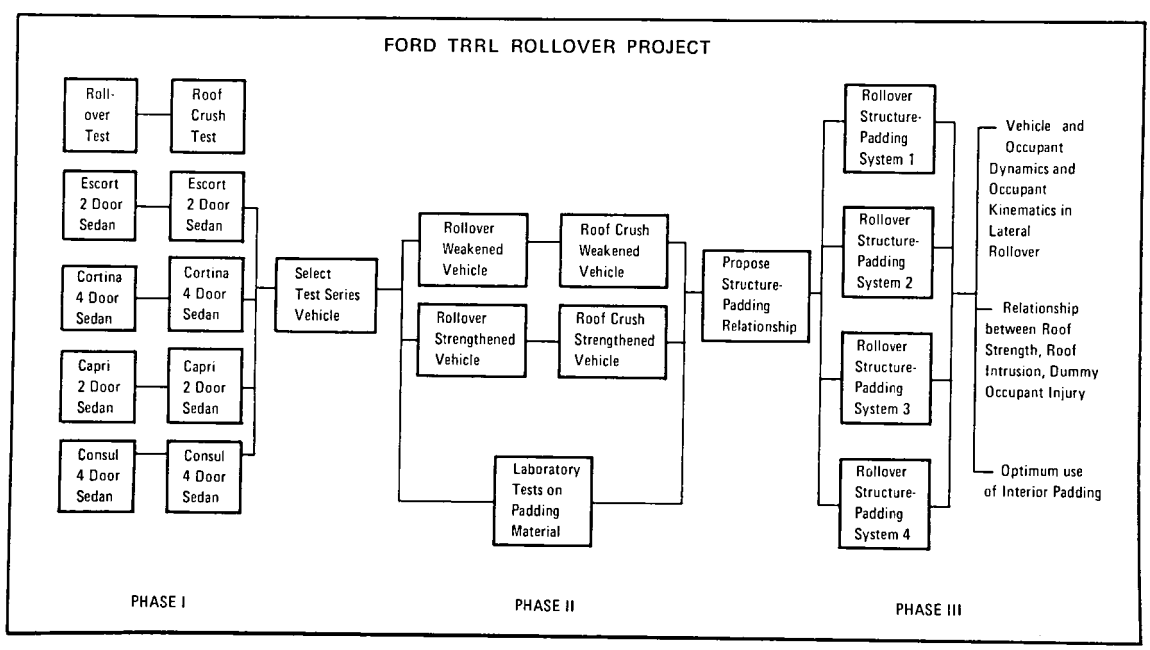


Figure 1

- Two lateral dummies would give more information than two dummies on the same side of the vehicle.

The roof crush tests are carried out in the MIRA Static Test Laboratory using a similar procedure to that set out in the FMVSS 208 requirements.

The purpose of these tests is to establish the strength of the roof structures for comparative purposes when studying the effects of structural modifications on vehicle deformation and occupant injury in the sled rollover tests.

In order to reduce occupant injury from impact with the vehicle's interior structure the program includes a study of the properties of various padding materials. The most suitable materials will be used as a crash padding in strategic areas of test vehicles. There will be rollover tests at MIRA to establish the effects on injury levels.

PROGRESS TO DATE

Phase One of the project has been completed and covered the testing of the following models:

- Escort 2 door sedan
- Mk III Cortina 4 door sedan
- Mk I Capri 2 door sedan
- Consul 4 door sedan

Having completed this phase, testing was confined to one model only. This removed the variable due to different vehicle packages enabling assessment to be concentrated on the specific effects of roof strength and interior padding. The model chosen was the 4 door Mk III Cortina sedan due to its mid range size and high volume of sales. Figure 2 shows its overall size and weight.

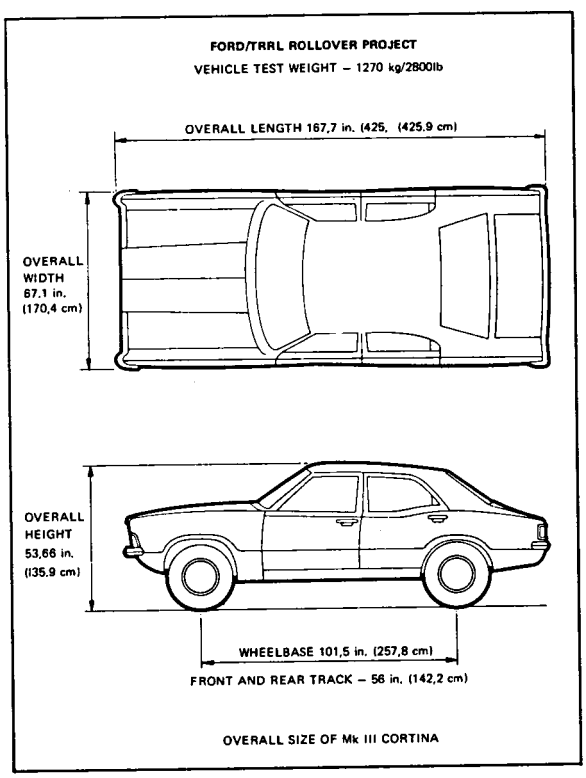


Figure 2

Phase One testing was carried out with unrestrained dummies. At the end of Phase One legislative trends to enforce wearing of seat belts became more evident. It is now a legal requirement in Australia, New Zealand and rural areas of France. It was decided that future tests would be conducted using dummies restrained with Ford production static seat belts. This was also expected to reduce the probability of dummy ejection through the windshield aperture.

Phase Two of the project has been largely completed and both weakened and strengthened vehicles have been tested.

Phase Three has commenced with tests on padding materials. The next rollover test vehicle will have additional padding in strategic areas.

TEST PROCEDURE

Rollover

Vehicle Preparation – The test vehicles are prepared at the Ford Motor Company Research Workshop. Preparation consists of modifying the vehicle structure as required, fitting laminated windshields, removing headlining, sun visors, interior mirrors, grab handles, fitting camera mounting brackets, draining gasoline tanks and refilling with low flammability test fluid, draining vehicle cooling and lubrication systems. The vehicles are then painted externally with a matt yellow paint to facilitate cine filming. The interior roof panel and center pillars are painted matt white to facilitate cine filming and show dummy head contact with the vehicle structure. Even markers and a deformation grid are strategically placed on the vehicle exterior. Figure 3 shows this marking configuration.

The hood top and roof contours, the windshield pillar marker locations are recorded before and after testing giving a record of the deformation occurring. During Phase One testing cinefilm analysis showed that the vehicle occupants could suffer injuries due to limb ejection through screen and glass apertures as shown in Figure 4. Static seat belts were fitted for Phase Two of the testing and Figure 5 shows the dummies in their seating position with seat belts fitted prior to testing.

Instrumentation – Instrumentation and fitting of the dummies is carried out at the MIRA

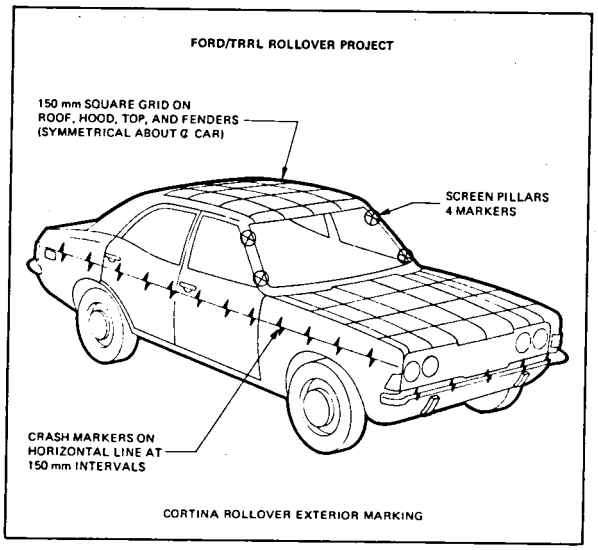


Figure 3

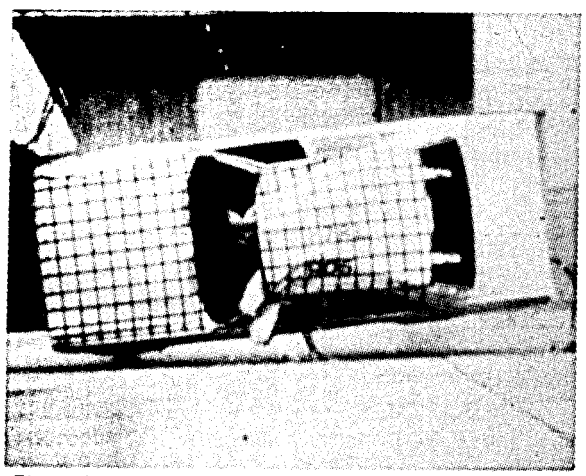


Figure 4

Laboratory. Alderson VIP 50A dummies, having rubber necks with wire restraints, are used.

Each dummy has duplicate accelerometers in the head and chest. Accelerometers are also fixed to the vehicle floor at the base of the center pillars on special mounts to avoid ringing due to panel oscillation. Accelerometer and event signals are fed back via an umbilical cable from the vehicle to the MIRA instrumentation recording unit. This consists of a duplicated 1-inch wide magnetic tape recording system and a UV trace recorder, the instrumentation being to SAE specification standard J211a.

The duplicated signals from the head and chest accelerometers and event marker signals are recorded on separate magnetic tape. One complete set of readings from the accelerometers and event markers is also recorded on the UV trace recorder.

Figure 5

Camera Coverage – High speed cine cameras incorporated in timing device are used inside the vehicle and around the test area perimeter.

Two cameras are mounted on the vehicle rear parcel shelf, both focussed onto the dummy heads, with a field of view extending from the sides to the center of the vehicle. Vehicle interior illumination is provided by sequentially fired photo-flood bulbs.

Five external cameras are used, a plan of their layout is shown in Figure 7. Two cameras are focused onto the front of the vehicle and positioned so that the first camera covers the first two thirds of the

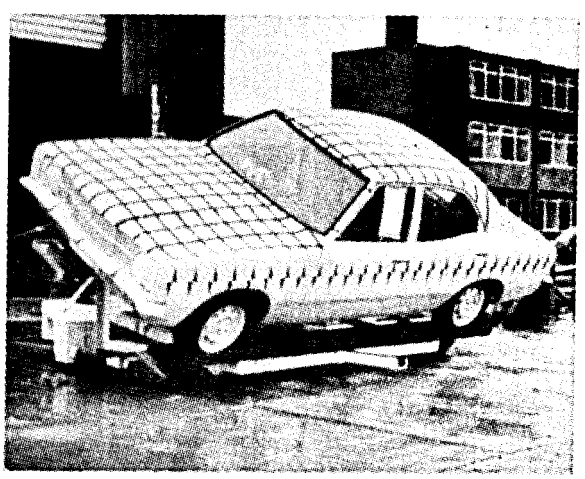


Figure 6

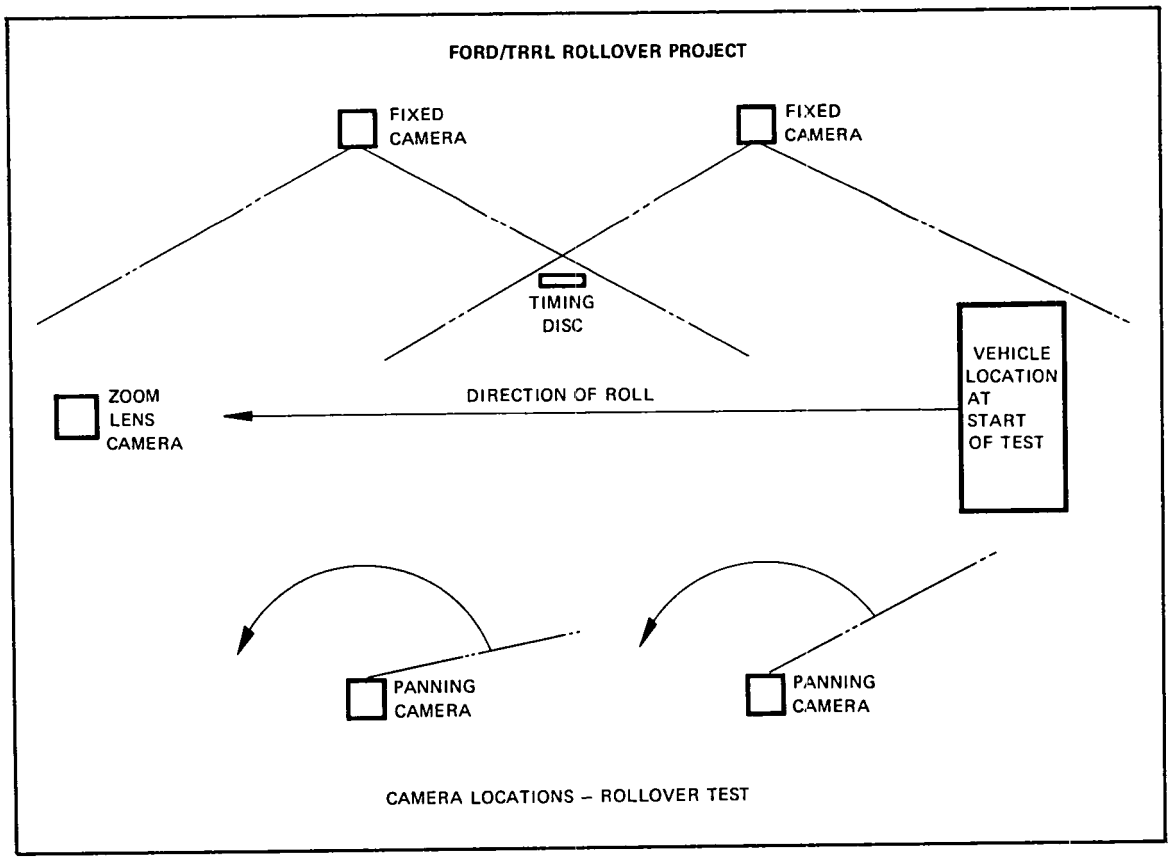


Figure 7

rollover and the second camera covers the last two-thirds of the rollover. In addition to the camera timing marks on the film these two cameras also have timing discs and electronic flash event markers (denoting start of test) within their field of view at all times.

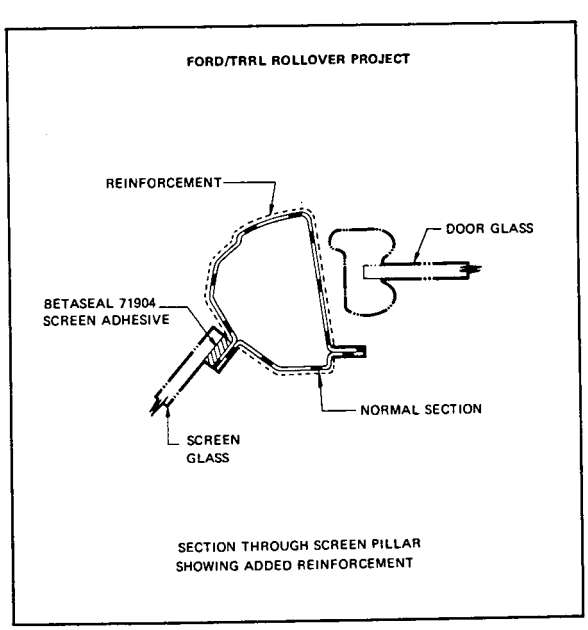


Figure 8

There are two cameras focussed on the rear of the vehicle, each camera panning on the vehicle for the duration of the test.

The fifth camera is focussed onto the vehicle side and by use of a zoom lens follows the progress of the vehicle during the test.

Sled Roll – The test vehicle is free standing on a trolley such that it is inclined 23° to the horizontal plane as shown in Figure 6. The dummies are placed into the vehicle with the front seats set in their mid position. When fitted, the static seat belts are production lap and diagonal type with the buckle attached by a semirigid stalk, to the lower inboard anchorage. All doors were locked and door glasses fully raised. The steering column lock was released allowing the front wheels and steering wheel to pivot in any direction.

The trolley and vehicle are accelerated to a velocity of 48 km/h/30 mph by a linear motor. At this speed the trolley is suddenly stopped causing the vehicle to roll off the trolley into a broadside rollover sequence.

Roof Crush

Production built 4-door body shells with complete door assemblies and front and rear screens fitted, are mounted onto rigidly fixed inclined mounting

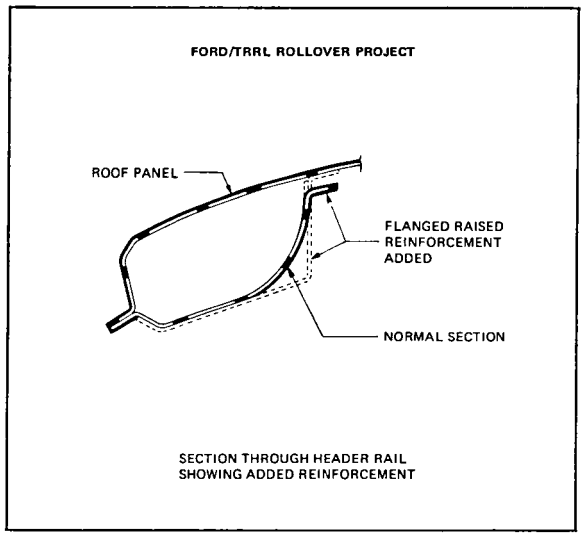


Figure 9

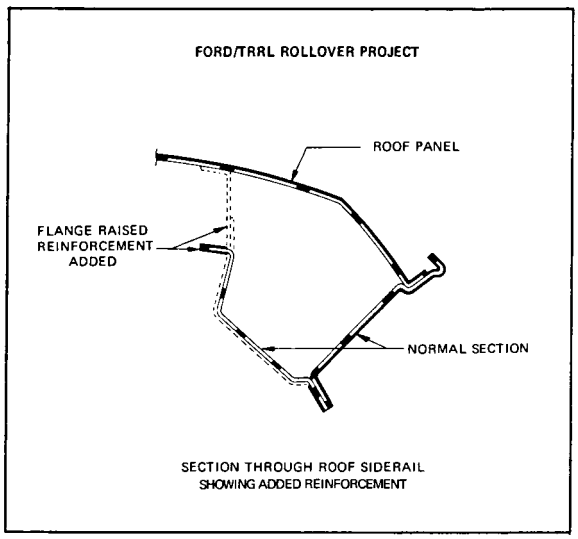


Figure 10

brackets. These brackets incline the body at 25° to the horizontal laterally and are fitted to the lower moveable platen of the static crush rig. This platen is then raised by hydraulic actuators forcing the roof of the body into a steel platen securely fixed to the stationary upper part of the crush rig and inclined at 5° to the horizontal in the longitudinal direction.

Movement of the rig is recorded on a UV trace recorder by means of a signal from a linear transducer. The roof crush load is recorded from load cells situation on the hydraulic actuators.

Vehicle deformation is recorded by two cine cameras positioned to give front and side views of the test vehicle. Figure 11 shows the crush rig prior to commencement of test.

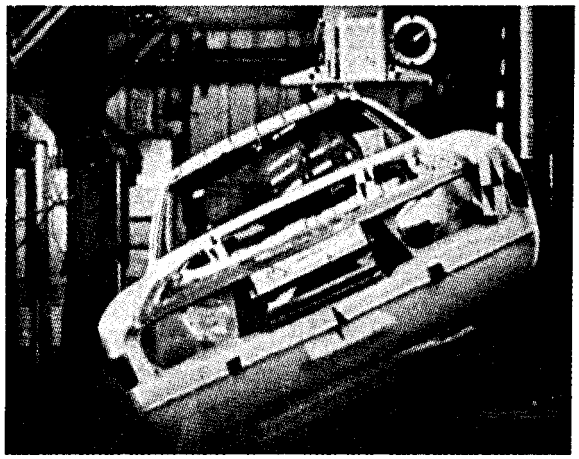


Figure 11

Padding Material Impact Test

These tests consist of impacting a sample of padding material with a standard head form at a set velocity using a conventional free fall pendulum rig. The impact velocity is set at 16 km/h/10 mph using a head form of 16.8 kg/15 lb effective mass.

The test samples are 300 mm/12 inches square and of 25, 50 and 74 mm/1, 2 and 3 inch thicknesses, retained in an open rectangular sheet metal carrier rigidly secured to a fixed beam.

A photoelectric timer gives the impact and rebound velocities. The acceleration level, duration of impact and rebound, and dynamic deflection are derived from accelerometers inside the headform. Static deflection is measured on the test rig. The energy absorbed is calculated for each sample.

From the results, specific padding materials will be chosen for use on rollover test vehicles. Figure 12 is a photograph of the test rig.

Result Analysis

Rollover – The head severity index (Gadd No.) and acceleration levels are evaluated, using a hybrid computer crash test analysis program, from the magnetic tape recordings. This program gives both a printout of the results and a trace of the acceleration levels to a time base.

Analysis of the cine films provides information about the vehicle and dummy movement during the rollover test. Impact of the dummy with the vehicle interior structure is observed from the interior cameras. This can be related to a time base and compared directly with the recorded accelerations.

Cine film analysis is carried out on a Vanguard analyzer with a digital data analyzer linked to a teletype terminal with a punched tape facility. This

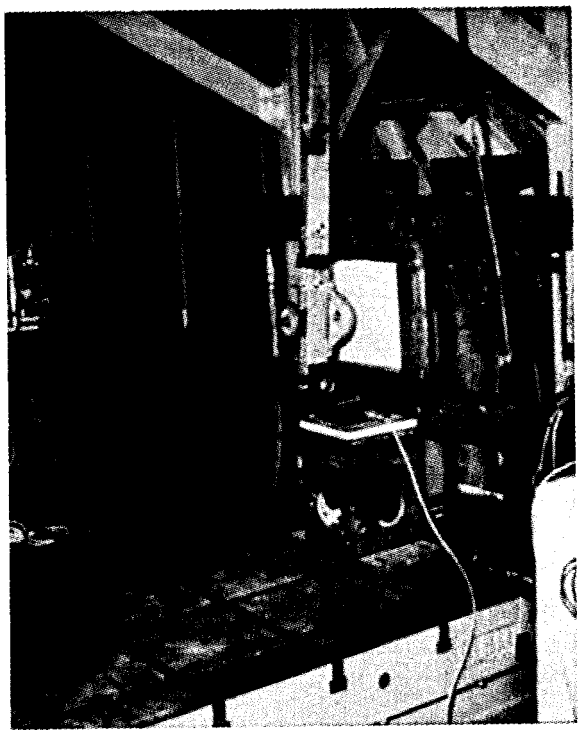


Figure 12

enables cine film results to be digitized and used in computer programs to ascertain deflection and velocity of the dummies and vehicle during the test.

Dummy-to-vehicle impact velocity can be calculated by two methods. The first by integration of the acceleration trace which gives change in velocity in a specific period of high acceleration level. The second method is by differentiation of the displacement measured on the cine film of the dummy's head relative to some fixed point in the vehicle interior.

Neither method is very accurate. Integration of the acceleration trace gives only change of velocity at the particular peak chosen. Due to its shape, the start and finish of the peak may be very indeterminate and can lead to inaccuracies in the velocity change calculated. Nor is it possible with this method to distinguish impact velocity from rebound velocity.

Evaluating velocity from the cine film accurately depends upon the camera remaining rigidly in position despite the deformation produced by the test. The major source of error is that the complex motions of the dummies result in their continually moving in and out of the focal plane of the close up cameras.

Similar problems apply to the evaluation of vehicle velocity from cine film because of variation in the focal distance of the reference point due to corkscrewing effect during rollover.

From the analysis of films and magnetic tape, graphs are compiled showing "vehicle" and

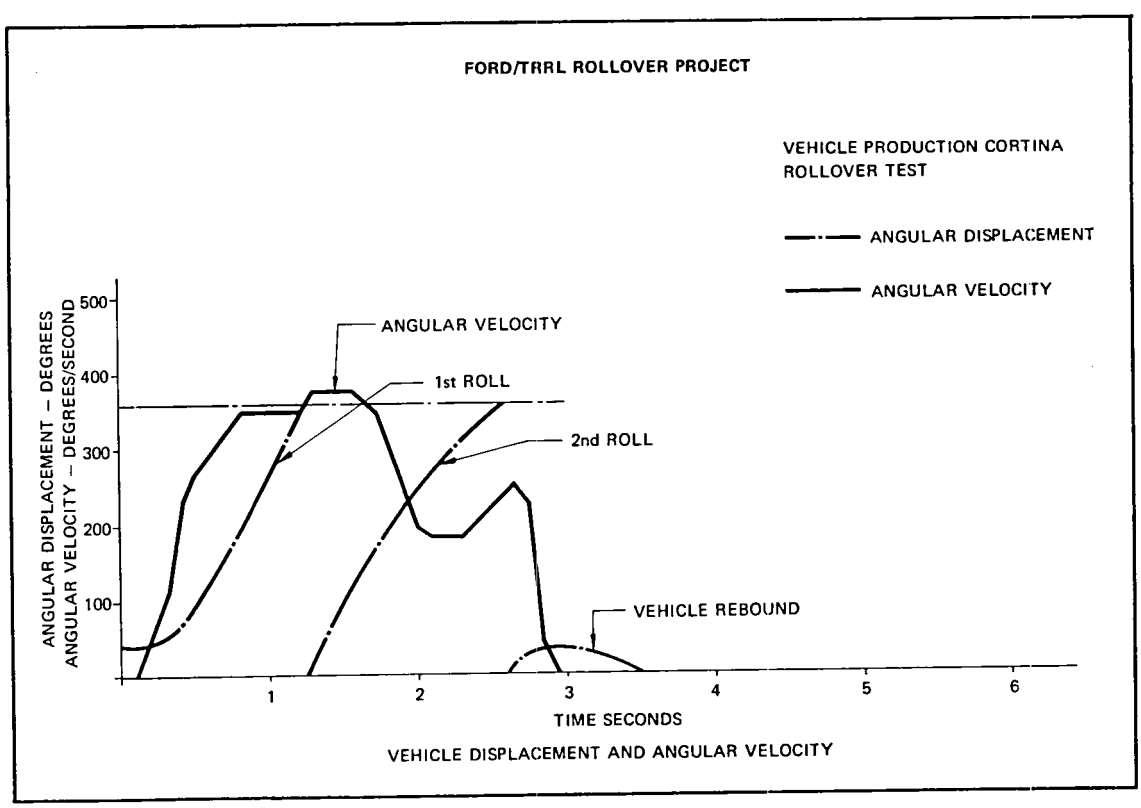


Figure 13

"occupant head" velocities and occupant accelerometer levels together with occupant-to-vehicle contact areas.

Roof, head top and fender contours are recorded both before and after testing as are the reference points on the windshield pillars. From these readings, deformation patterns are established for each vehicle tested.

Roof Crush – Force/deformation graphs are compiled, and crush energy evaluated, from the trace of rig movement and crush load. Displacement is also plotted against crush test time.

The cine films are analyzed to ascertain the deformation pattern of the vehicle structure. This can then be related to the force/deformation graph via the time base.

Deformation patterns of the roof panel and movement of the windshield pillars are established in the same way as for the follover test results.

Vehicle Modifications – Tests have been carried out on vehicles having modified structures to ascertain what effects these changes would have on occupant injury levels in rollovers.

The modifications consisted of:

- Weakening body above the waistline.
- Strengthening body above the waistline.
- Replacing conventional glazing system with direct glazing system.

The body was weakened by a 30% reduction of the spotwelds in the body structure from the waistline upwards. The roof side rail to the windshield pillar and the header panel joint was also weakened by removing part of the upper flange.

Strengthening of the vehicle structure was achieved by doubling the material thickness of the windshield pillars and extending the outer surface onto the roof and cowl top panels. In the roof structure, the roof side rails and header panel also had their material thickness doubled, and were welded to the roof panel instead of remaining as an open joint. Figures 8, 9, and 10 show cross sections of the reinforced windshield pillar, header panel and roof side rail.

Phase One tests showed the importance of retaining the front windshield within its aperture to reduce occupant limb injury (see Figure 4). A modification intended to improve retention was the use of a direct glazing system manufactured by Gurit-Essex Inc. (Switzerland) sold under the trade name Betaseal 71904. This is a one part paste system based on reactive polyurethanes. It cures by atmospheric moisture to an elastic adhesive seal.

Results

At this state in the program the following rollover and roof crush tests have been carried out.

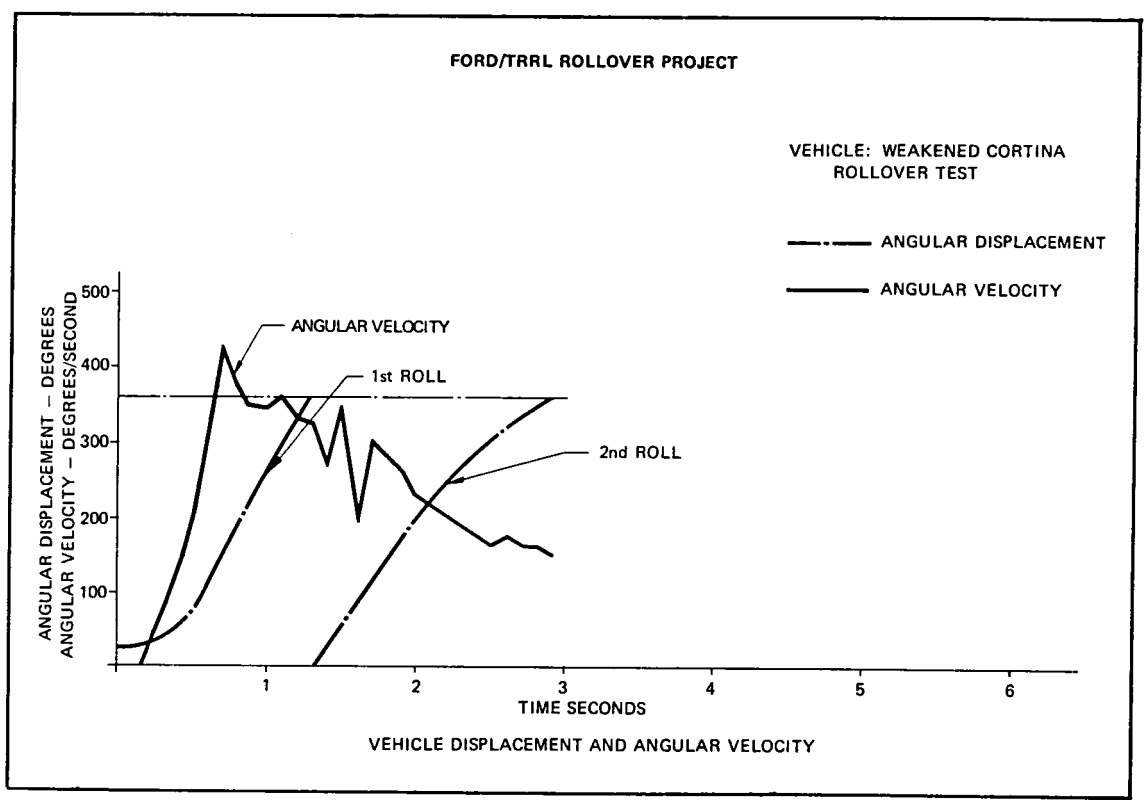


Figure 14

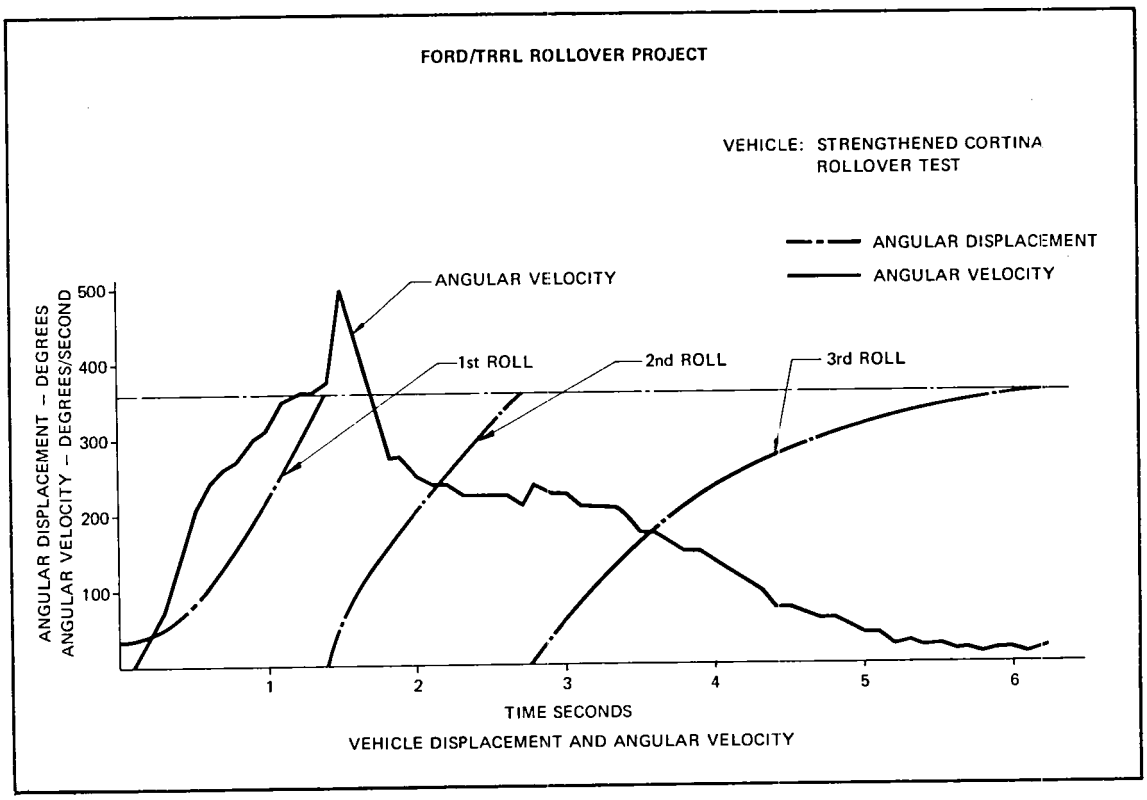


Figure 15

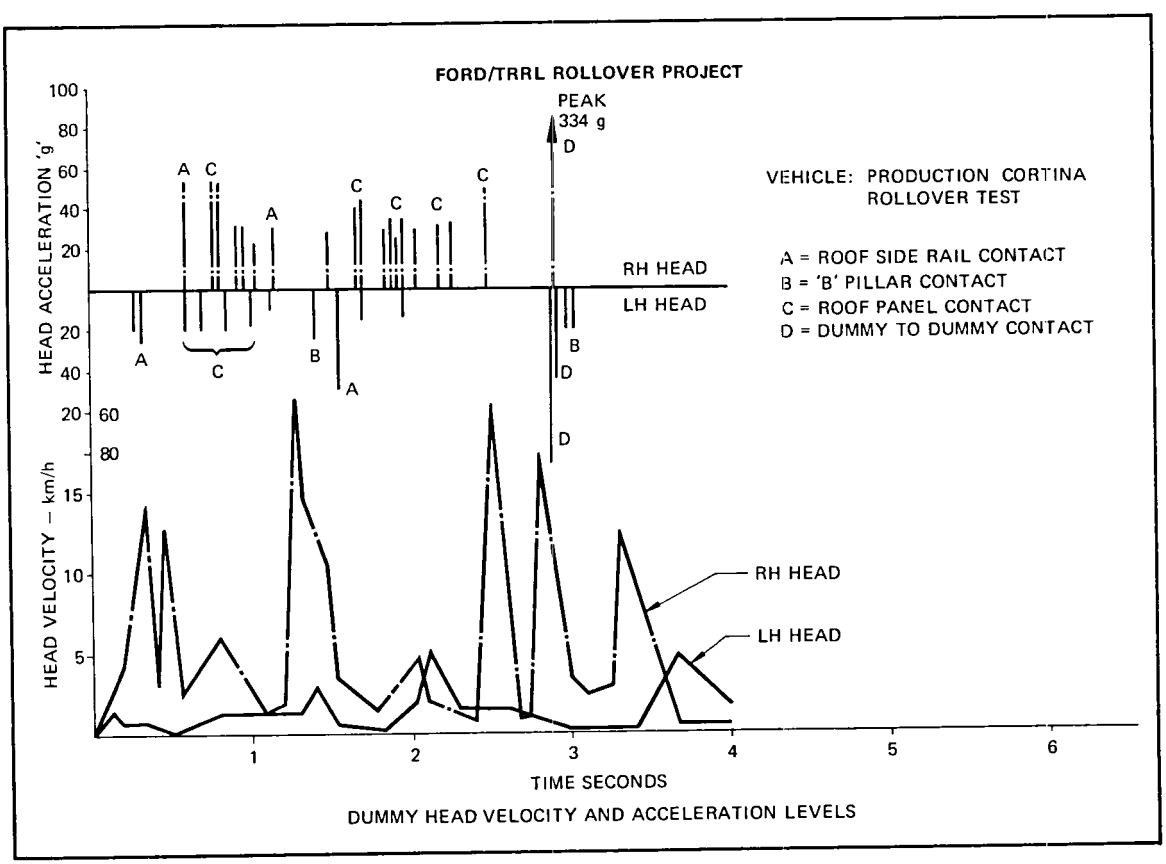


Figure 16

Rollover Tests

Escort	– 1 off
Mk. III Cortina	– 5 off
Capri	– 1 off
Consul	– 1 off

Roof Crush Tests

Escort	– 1 off
Mk. III Cortina	– 4 off
Capri	– 1 off
Consul	– 1 off

The most important results from these tests and the tests on energy absorbing padding are listed below.

Rollover Tests Phase 1 and 2

Tables 1 and 2 list significant events occuring in all tests conducted so far. Included in these tables are the maximum acceleration levels for the head and chest of the dummies with the time of occurrence and the structure impacted. Figure 25 shows a vehicle after test with generally typical deformation.

Figures 13, 14 and 15 show vehicle angular velocity and angular displacement for the production, weakened and strengthened Mk III Cortina vehicles respectively. Figures 16, 17 and 18 show dummy head velocity and acceleration together with the location of the vehicle structure impacted. Figures 19, 20 and 21 show the roof deformation for the same vehicles.

Dummy limb ejections that occured in Phase I tests (Figure 4) are described qualitatively in Tables 1 and 2. The test dummies had no facilities to record limb injuries due to ejection.

Roof Crush Tests Phase 1 and 2

The force deformation and crush energy graph for a Mk III Cortina production body having windshields retained with conventional rubber weatherstrip is shown in Figure 22.

Figures 23 and 24 show the force deformation and crush energy graphs for a production and strengthened Cortina body with the windshields adhered in location.

Figure 26 shows a body after test with generally typical deformation. A crush test on a weakened structure has not been carried out to date.

Energy Absorbing Padding Tests

To date about 70 padding samples have been tested and Table 3 lists the results from some of the samples which are considered worth testing further in the Phase III test program.

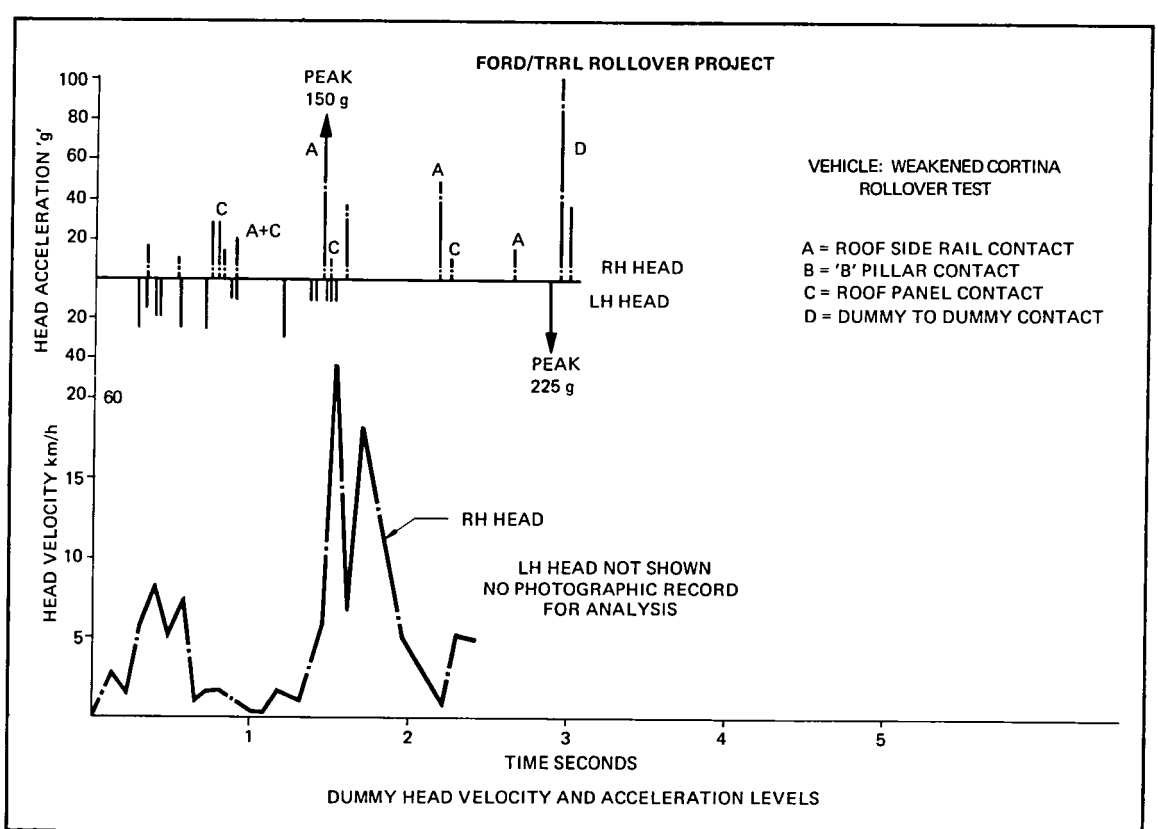


Figure 17

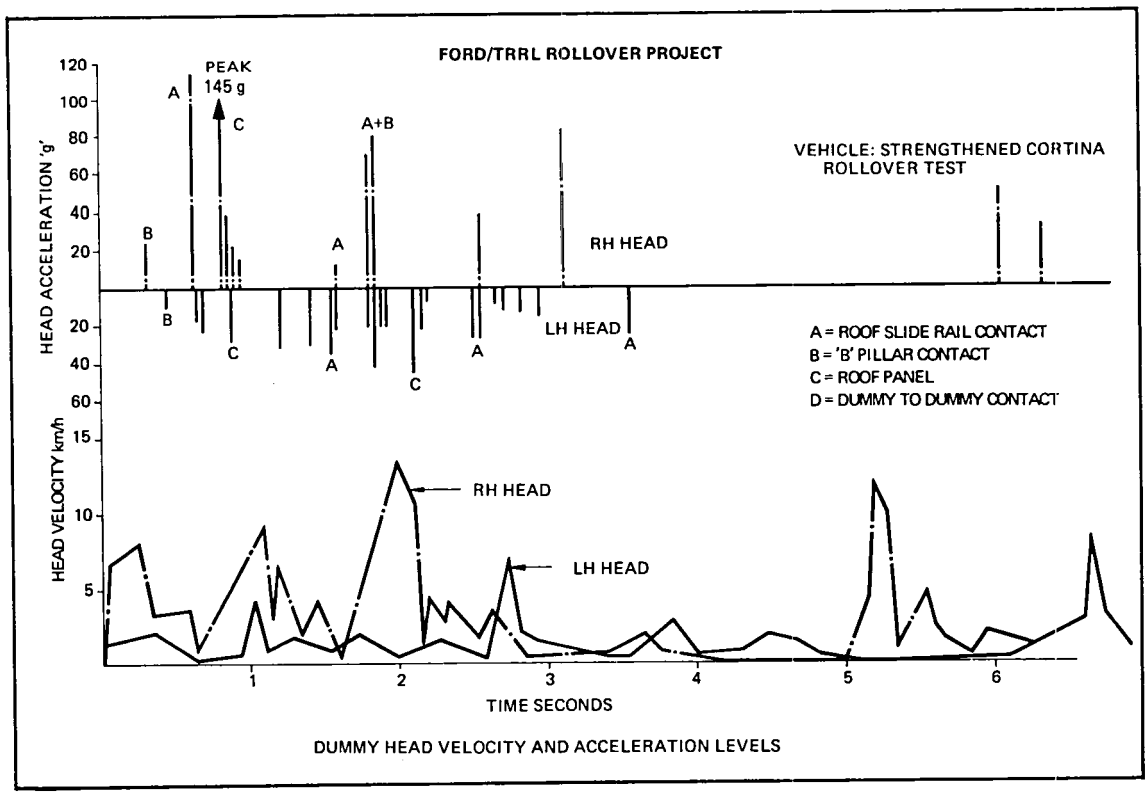


Figure 18

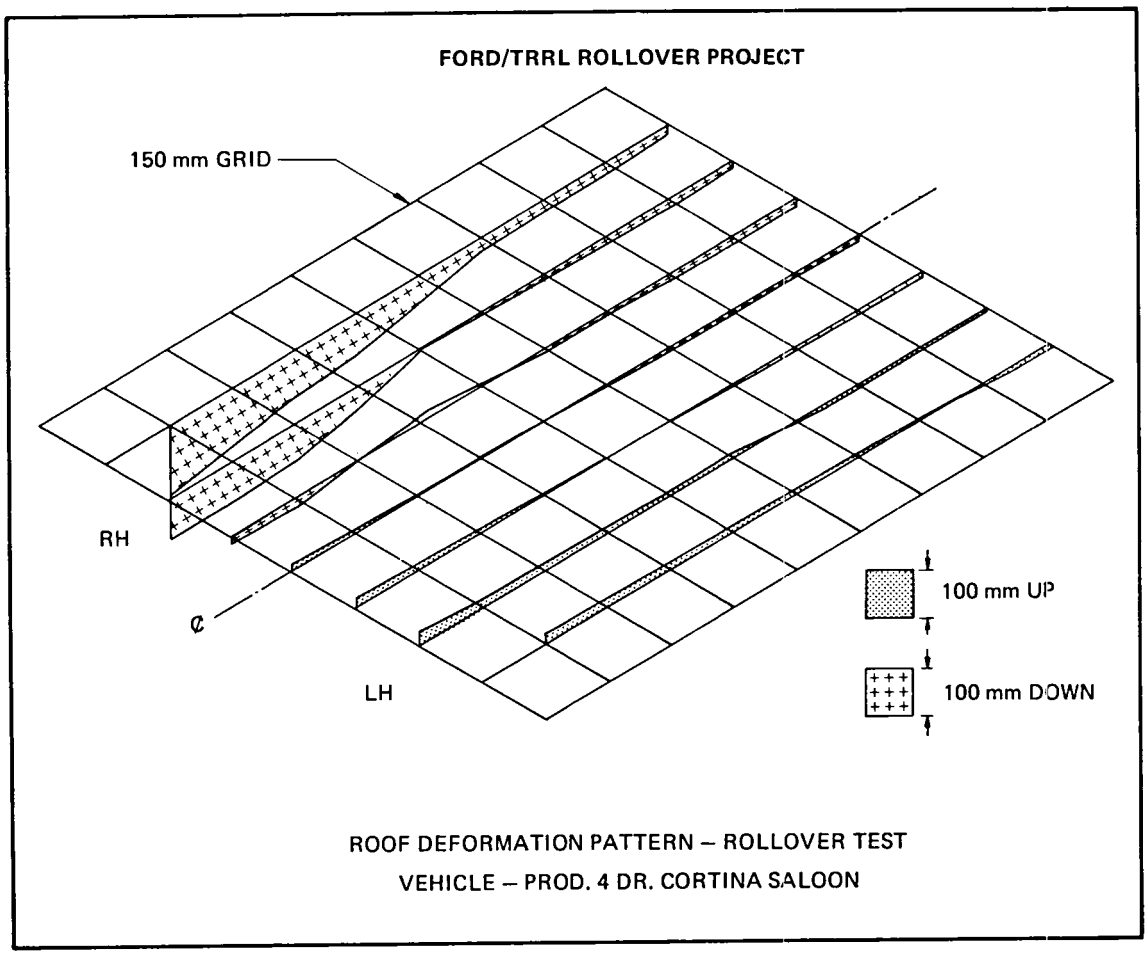


Figure 19

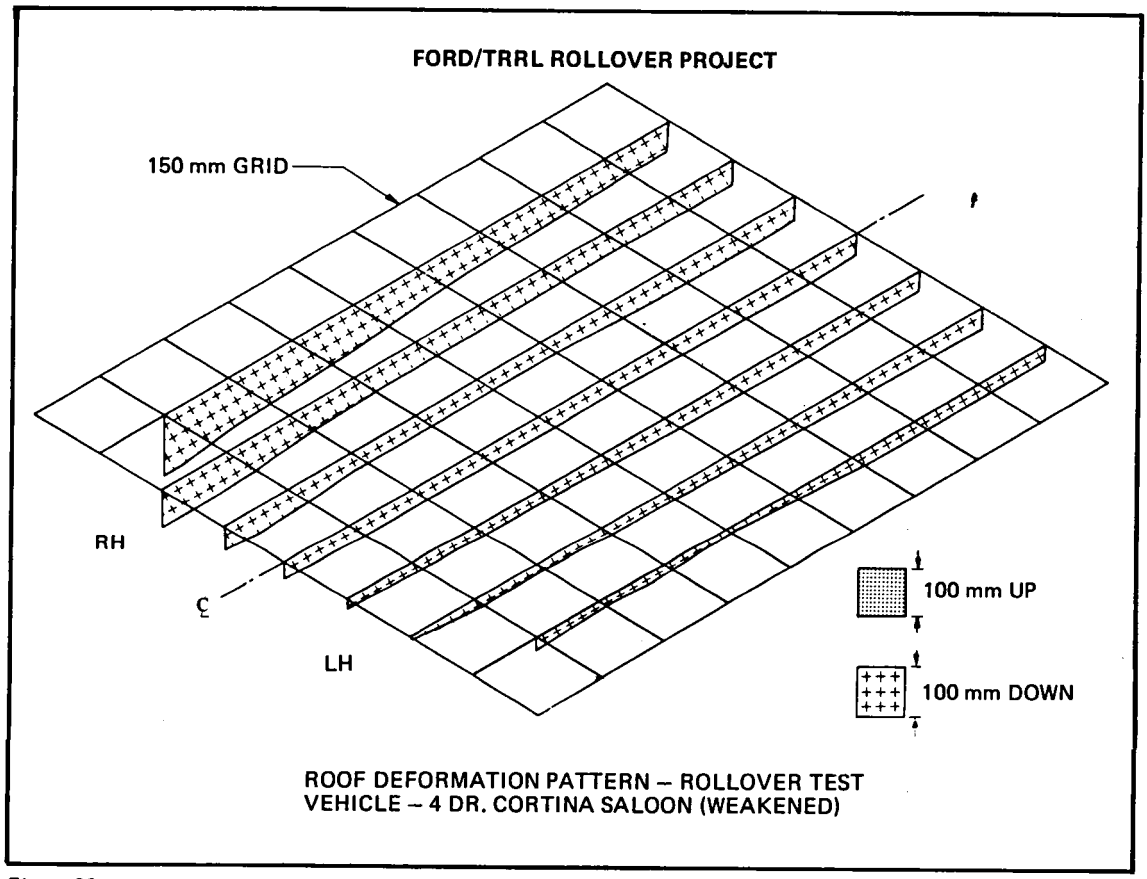


Figure 20

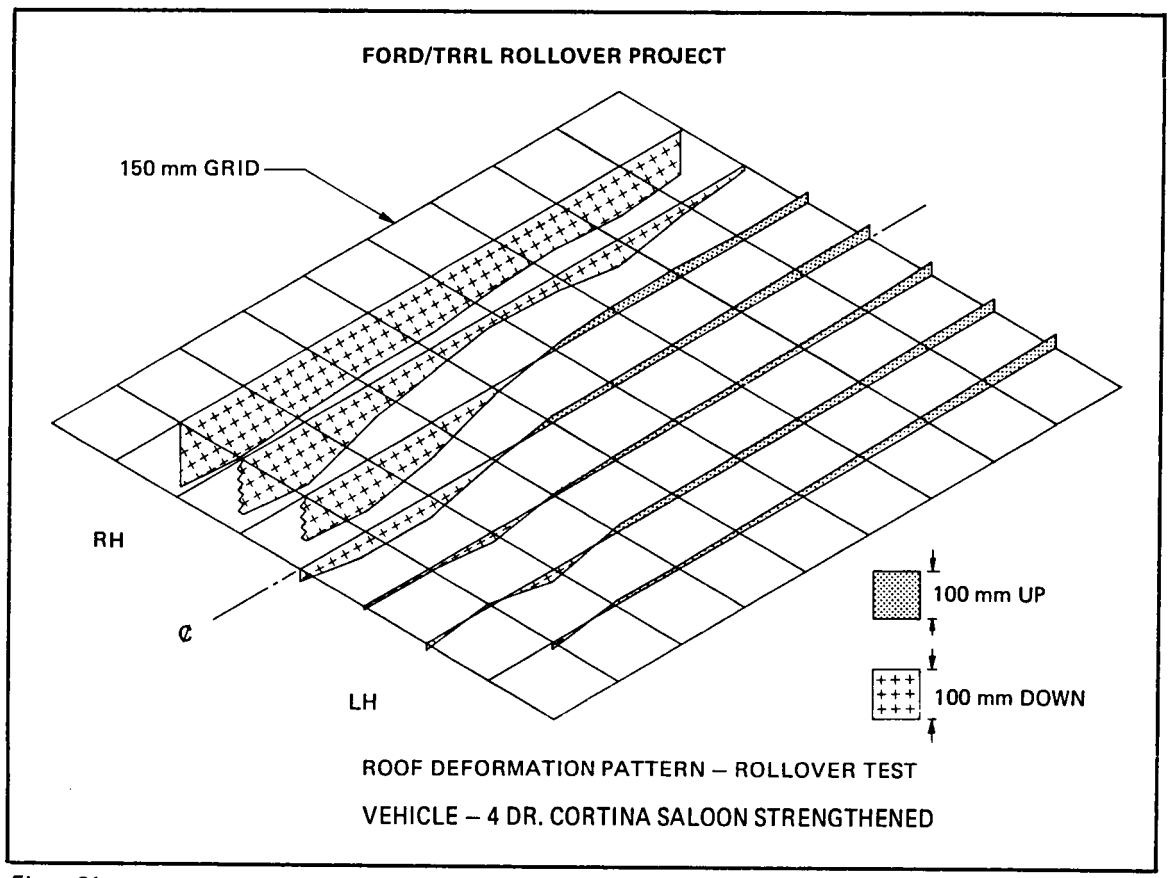


Figure 21

TABLE 1
SIGNIFICANT EVENTS DURING ROLLOVER TEST

FORD/TRRL ROLLOVER PROJECT

VEHICLE	ESCORT	CORTINA	CAPRI	CONSUL
Occupant HIC Gadd No./ (Time sec) RH	150/(2.34)	75/(0.75)	190/(2.60)	250/(0.46)
Occupant HIC Gadd No./ (Time sec) LH	40/(0.71)	375/(0.75)	150/(1.90)	240/(0.53)
Cause of Injury RH	Not visible on film	Head strikes roof interior	Head strikes RH door at waist	Head contacts roof
Cause of Injury LH		Head striks roof interior	Head strikes RH seat top	Head contacts roof
Dummy Ejection (time sec) * RH	No Ejection	Left hand through door glass aperture contact with ground. Right hand through windshield glass aperture (1.45)	No Ejection	Left hand & knee through door aperture (1.34 & 1.40). Right knee through windshield aperture (1.25)
Dummy Ejection (time sec) * LH	No ejection	Right hand, knee, thigh & posterior through windshield aperture. Contact with ground (1.34)	Left hand through door glass aperture (1.15). Head through door glass aperture	Right hand through windshield aperture (1.06). Right knee through windshield aperture
Front Windshield Ejection	No - Retained along lower edge	Yes Complete	Yes Complete	Yes Complete
Door Glass Breakage	Left hand door only	Right & Left hand front doors only	Right and Left hand doors	Right and Left hand front door RH rear door
Rear Windshield Breakage	No	No	No	Yes
Front seat structure failure	No	No	No	No
Roof Damage	Severe	Severe	Severe	Severe
Hood top condition	Open to safety catch	Open to safety catch	Open to safety catch	Open to safety catch
Fender Damage RH	Slight	Slight	Slight	Slight
Fender Damage LH	Severe	Severe	Severe	Severe
Luggage Compartment Lid	Closed	Closed	Closed	Closed
Doors**	Both doors could be opened	All doors could be opened	Both doors could be opened	LH rear jammed shut. All others openable.
Fuel Spillage	Slight leakage through vent tube	None	Slight leakage through vent tube	None
Chest – Max 'g' (time sec) RH	Instrument failure	17.5 (1.35)	15(2.06)	33 (1.34)
Chest – Max 'g' (time sec) LH	127 (1.32)	150 (0.71)	15 (1.86)	19 (1.33)
"B" Post Max "g" (time sec) RH	Instrument failure	Not recorded	Not recorded	24 (2.18)
"B" Post Max "g" (time sec) LH	28 (1.20)	Not recorded	Not recorded	16 (1.33)
No of rolls	2.5	2	2.25	1.75
Max. static test crush load (kN/lbf)	28/6300	22/5000	13/2900	27/6100

*Dummies were retained within passenger compartment at end of test in all cases

**No doors opened during test

TABLE 2
SIGNIFICANT EVENTS DURING ROLLOVER TEST

FORD/TRRL ROLLOVER PROJECT

VEHICLE		PRODUCTION CORTINA	WEAKENED CORTINA	STRENGTHENED CORTINA
Occupant HIC	RH	>2,000 (2.94)	275 (2.95)	1,000 (1.87)
Gadd No. (Time Sec)	LH	780 (2.88)	1,100 (2.89)	570 (2.10)
Cause of Injury	RH	Head Contact Elbow of LH Dummy	Head Contact LH Dummy Body	Head Contact 'B' Pillar & Roof Side Rail
	LH	Head Contact Roof Side Rail at 'B' Pillar	Head Contact LH 'B' Pillar	Head Contact Roof
Dummy Ejection (Time Secs) *	RH	Fingers of Left Hand through top of Windshield Aperture (1.50)	None	None
	LH	None	None	None
Front Windshield Ejection		Yes Complete	Yes Complete	No - Windshield adhered in location
Door Glass Breakage		RH Front Only	RH Front Only	RH Front & Rear Doors
Rear Windshield Breakage		No	No	Yes
Front Seat Failure		No	No	No
Roof Damage		Severe	Severe	Very Severe
Hood Top Condition		Open to Safety Catch	Opened after 2nd Inversion	Closed as normal
Fender Damage	RH	Severe	Severe	Slight
	LH	Slight	Slight	Slight
Luggage Compartment Lid		Closed	Closed	Closed
Doors **		All doors could be opened	All doors could be opened	All doors could be opened
Fuel Spillage		None	None	None
Chest – Max "g" (Time sec)	RH	21 (0.87)	13 (0.78)	34 (0.63)
	LH	61 (2.95)	63 (2.89)	13 (1.20)
"B" Post Max "g" (Time sec)	RH	19 (1.81)	23 (0.62)	19 (0.84)
	LH	168 (2.9)	100 (2.26)	16 (0.65)
		High "g level caused by rebound of car		
No. of Rolls		2.0	2.0	3.0
Max Static Test Crush Load (kN/lbf)		22/5,000	Not Tested	36/8,000

*Dummies were retained within passenger compartment at end of test in all cases
**No Doors opened during test

TABLE 3 FORD/TRRL ROLLOVER PROJECT									
Sample Results from Energy Absorbing Padding Tests Using Pendulum Head Form Impact Rig.									
Sample No.	Material	Thickness mm	Density kg/m^3	Head Form Deceleration 'G'	Energy Abs. (Joules)	Static Def. mm	Coeff. of Restitution	Impact Vel. km/h	Rebound Vel. km/h
15	Phenolic	76	37.8	38.0	70.1	39.70	0.24	16.88	3.98
16	Phenolic	78	23.5	24.0	72.3	54.03	0.18	16.89	3.09
17	Phenolic	75	42.3	39.0	59.8	39.45	0.26	16.17	4.18
21	Phenolic	76	31.6	26.5	70.5	53.89	0.21	16.88	3.33
32	Urethane	75	37.8	25.5	72.0	54.89	0.19	16.87	3.18
34	Urethane	75	34.6	29.5	71.3	55.49	0.21	16.88	3.65
38	Phenolic	76	23.9	36.0	62.7	57.50	0.33	16.36	5.35
40	Phenolic	52	27.6	26.0	64.4	44.02	0.29	16.36	4.72
46	Phenolic	52	41.4	28.0	64.1	43.04	0.30	16.36	4.84
47	Exp. Polyethylene	73	56.2	40.0	43.5	Nil	0.62	16.36	10.09

DISCUSSION OF RESULTS

Phase I Baseline Tests with Unrestrained Dummies

Vehicle Behavior – No vehicle in these tests has failed to roll although the number of rolls completed varied from 1.5 to 2.5. This suggests that the 48 km/h/30 mph) lateral rollover test being used is an extremely severe test since accident studies suggest that less than 15% of rollovers involve more than 360° of roll (2).

In no case did any of the doors open during the impact although in most cases door glass was broken and in only one case was the windshield glass retained to the end of the test.

The vehicles tended to roll onto their corners and sides (largest diameters) and were usually clear of the ground when right way up or totally inverted.

If there is any discernable trend associated with vehicle size it is that the larger vehicles tended to execute fewer rolls, but this was no guarantee of lower levels of occupant injury.

Dummy Occupant Behavior – In no case did complete dummy ejection occur but partial ejection of hands and knees occurred in four out of five tests carried out. This ejection took place through both front windshield and front side windows and was normally most pronounced for the dummy on the impact side.

The recorded accelerations were extremely low compared with accepted tolerance levels but the dummies had no facility for assessing the limb injuries that probably occurred.

Phase II Tests

For this part of the program the change was made to lap and diagonally belted dummies and work was concentrated on Mk III Cortina vehicles.

Dummy Occupant Behavior – The change to belted dummies almost completely eliminated ejection and partial ejection of dummy occupants. The kinematics of the belted dummies are quite different from unbelted ones. Although overall dummy motion is restricted, considerable lateral movement of head and upper torso is still possible and can result in severe head and shoulder impacts with the roof side rails and upper "B" pillar. These impacts are worsened in some instances by the dummy moving laterally out of the diagonal belt. This allows the lap portion of the belt to be slacken and the dummy is less effectively restrained. The most severe of these impacts occur at the end of the second roll when the Mk III Cortina vehicle comes to rest, and to a lesser extent at the end of the first roll due to the change in rolling motion that occurs when the vehicle's wheels are on the ground. One of the most severe impacts recorded to date coincided with a head impact on the elbow and shoulder of the other

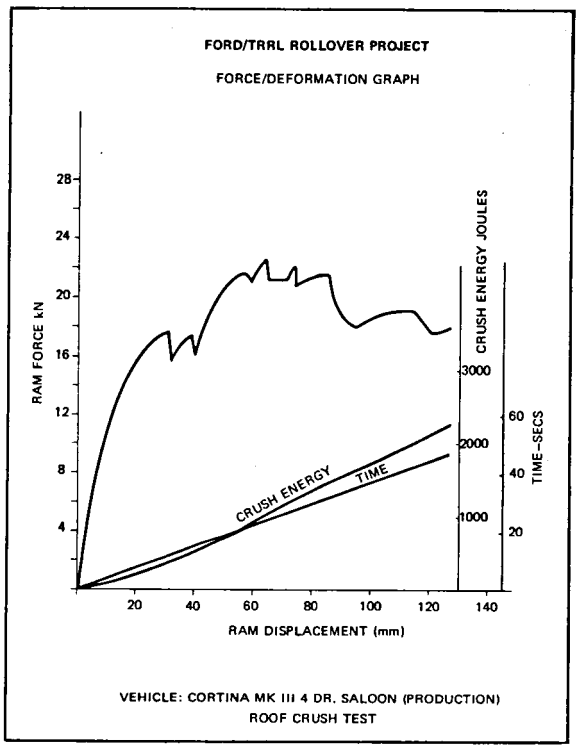


Figure 22

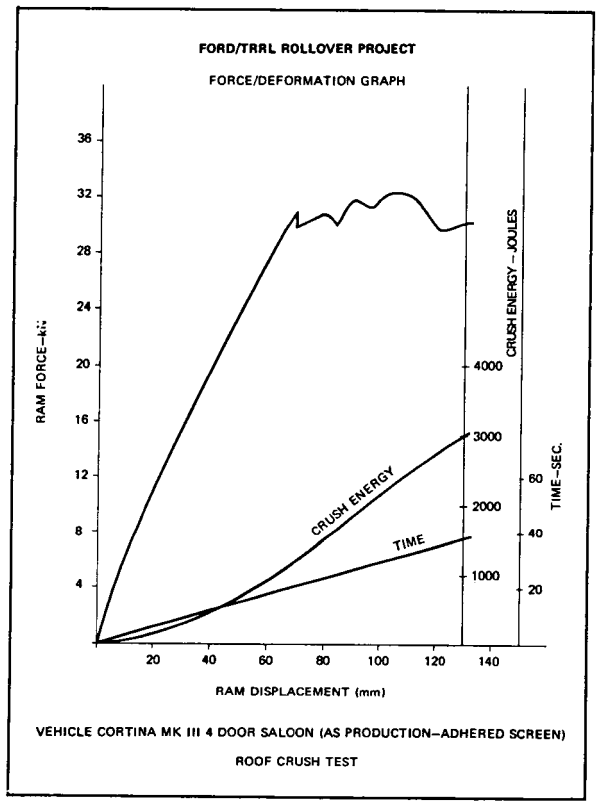


Figure 23

dummy. This result must clearly be considered suspect in the light of known limitations for crash test dummies.

Vehicle Behavior – The type of rollover test used is often criticized on the grounds of poor repeatability. During Phases I and II of this program five tests were carried out on the same model and the results only partially support this view. The overall vehicle dynamics and vehicle damage were fairly consistent and the test has undoubted value for development work. However highly specific data, e.g., the structural deformation at a particular point on the vehicle may vary widely from test to test and extreme care must be taken before making quantitative deduction from any one test.

Vehicle Strength Variations – The two modified vehicles were tested in this phase of the program. The stiffness change produced by a 30% reduction of spotwelds in all roof, pillar, and glasshouse areas did not produce a dramatic increase in rollover damage, or significant changes to vehicle and occupant dynamics.

The strengthened vehicle however did show different behavior although the reservations already expressed about repeatability must be borne in mind. The crush test showed an increase in roof strength of about 60% and in the rollover the vehicle completed 3 instead of the usual 2 rolls. Overall damage was not reduced and it appears that the increased stiffness

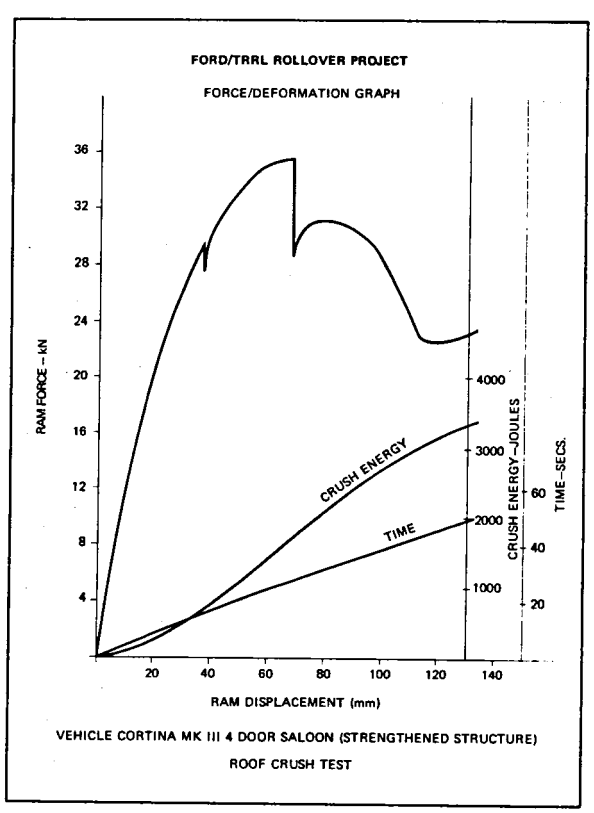


Figure 24

Figure 25

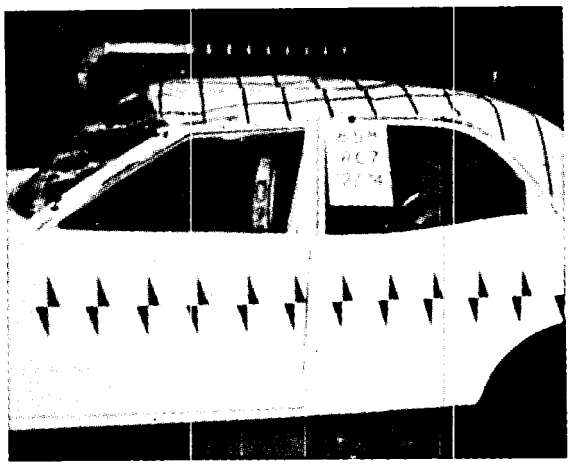


Figure 26

around the front windshield aperture resulted in a shift of damage towards the rear end of the roof and the center and rear pillars. The majority of the increased roof strength was due to the direct glazing system as shown in the roof crush results.

Energy Absorbing Padding Tests – Considerable analysis of occupant-to-vehicle velocities during rollover was done to establish a realistic speed for impact testing. It was found that impact speeds up to about 20 km/h (12 mph) occurred and tests were run at about 16 km/h (10 mph). The most promising padding from the samples tested to date is a low density phenolic foam.

CONCLUSIONS

In the first tests with unrestrained occupants, the measured injury levels are surprisingly low and occupant ejection did not prove to be the problem expected from studies of accident data. In no case did complete dummy ejection occur, probably because no doors opened during rollover. The fitting of seat belts greatly reduced the partial ejection that otherwise occurred and the retention of the windshield throughout rollover virtually eliminated it. However, using seat belts resulted in a change of injury pattern from partial ejection of limbs, to head impacts with the interior of the car. In assessing the significance of this trade-off in injury types two factors must be borne in mind. Firstly that the test being used represents an extremely severe test in the number of rolls it produces. Secondly, most rollovers in the real world will involve at their commencement, and probably at other instants as well, longitudinal vehicle decelerations that will influence dummy kinematics and increase the beneficial contribution of seat belts.

The investigations into the effects of roof strength have not been completed but it appears that moderate changes to the normal roof strength of the model tested, may have less significant effects on occupant dynamics than would changes in occupant restraint and interior padding. In particular the test on a strengthened vehicle suggests that increased roof strength is not necessarily beneficial. It can result in less absorbtion of vehicle kinetic energy through vehicle damage in the early stages of the rollover, and therefore increase the number of rolls and the distance covered. This is undesirable in practice as it increases the likelihood of occupant to vehicle interior impacts and it increases the chance of the rolling vehicle hitting another vehicle or obstacle.

FUTURE WORK

The program will continue in Phase III with some further investigation of roof strength effects but concentrating primarily on the problem of how to provide increased interior protection using energy absorbing padding with minimum intrusion into passenger compartment space.

ACKNOWLEDGEMENTS

The author would like to express his thanks to the staff of MIRA and TRRL, and colleagues of the Ford Motor Company Research and Engineering Center for their cooperation and assistance in this project.

REFERENCES

1. "ESV Rollover Test Methods," C. Ennos, 2nd Int. Tech. Conf. on ESVs (Germany 1971).
2. Field Studies of Rollover Performance," G.M. Mackay and I.D. Tampen, SAE 700417.

A PRACTICAL APPROACH TO THE PROTECTION OF MOTOR VEHICLES BY THE ABSORBTION OF IMPACT ENERGY

G. PERSICKE, F. I. PLANT,
and J. R. CHILD
Road Research Limited

INTRODUCTION

It would seem that today's motorist has never had it so good; his car has air conditioning, a powerful engine, disc brakes, power assisted steering, power operated windows and even a stereophonic radio, record player or tape recorder. He is able to drive in comfort and luxury for hundreds of miles, from one end of the country to another, without even the single interruption of a traffic light.

But there is a paradox here; for although he can drive in luxury he cannot drive in safety.

Motorization has exceeded all expectations, and has led to an incresae in traffic density which is reflected in high accident figures. The suffering, loss of life and high costs of injury arising from accidents on the roads of the world can no longer be tolerated. Ways must be found to reverse the trend and systems must be developed to reduce this morally and economically indefensible toll. We may classify such systems as active, passive, palliative and preventative. Active and preventative systems generally operate to reduce the number of accidents but their cost tends to be much higher than the passive and palliative types which aim to reduce the severity of the resultant damage. Examples of the latter include safety belts, roadsied safety barriers, soft interior padding and burst-proof doors. However, little attention has been paid to reducing the effects of those minor accidents occurring on impact between a vehicle and an obstruction which are rarely reported in statistics, but which cost much in time, money and inconvenience.

We must, therefore, give serious consideration to any device which reduces the initial severity of the impact between a vehicle and an obstruction, at either slow or fast speeds. This will have a dual effect by reducing material damage and lessening or preventing injury to the people involved. If really effective at low speeds, minor accidents immediately become negligible in their effects, thus alleviating a problem which is becoming more frequent both in the increasingly congested cities of wealthy countries, and among the rapidly growing traffic of the developing nations.

The first part of a vehicle to come into contact at the point of collision in most accidents is the front bumper. This is closely followed in terms of frequency of initial contact by the rear bumper. Hence, any system which uses the bumpers and their attachments to the car, to absorb part of the impact energy, offers a valuable route towards the reduction in the cost of accidents. In addition, any such system being both passive and palliatory, is also relatively cheap to use.

DESIGN CRITERIA

The essential requirements of energy absorbing bumpers are as follows:

1. High energy absorbtion.
2. Act without time-lag.
3. Fully effective at very low speeds.
4. Controlled retardation throughout their range.
5. Continued effectiveness at speeds well above the design range.
6. No rebound.
7. Low initial cost.
8. Re-useable after minor impact.
9. Self-resetting after minor impact.
10. Effective at all angles of impact.
11. Low repair cost.
12. Compactness.

(Not necessarily in order of importance.)

Criteria 1 to 7 can be met to varying degrees of efficiency by any means employing the permanent deformation of a suitable material, even such a simple solution as a stiffer version of the common bumper and dumb-iron design in use today. But, after the slightest impact, the effectiveness of such a system is seriously decreased and it is certain that few, if any, vehicle owners will bother with, much less pay for, repairs to such an apparently unimportant item. The major key to a practical system is, therefore, found in items 9 and 10, provided the other requirements are also adequately met.

EVOLUTION OF A PRACTICAL DESIGN

Road Research Limited accepted the challenge presented by the Transport and Road Research Laboratory in producing a specification for a highly advanced design of energy absorbing bumper system. In doing so, we recognized the well-known principle that a fluid flowing through an orifice can absorb great amounts of energy and when combined with a mechanism to restore the unit to its original state, will produce a system conforming to all the above criteria.

The requirement called for a system capable of being effective at speeds of up to 40 mph (64 km/h) and to prevent all damage at speeds of up to 5 mph (8 km/h). These specifications were fulfilled and subsequent tests confirmed the accuracy of our designs.

The resulting specification consists of a unit functioning in three stages, the design of which is dictated largely by considerations of size and economy. The first stage of action consists of a short stroke hydraulic buffer with a spring return, and this meets all the requirements of the energy absorbing bumpers at low speeds – up to 5 mph (8 km/h). At higher speeds – up to 20 mph (32 km/h) – the conflicting demands of criteria 4 and 12 dictated that the unit could not be self-resetting and that some part of the vehicle would suffer damage. As the vehicle would require repair it was thought permissible to envisage a simple garage repair that would recondition the bumper system in order that it may accept further service. Finally, in high speed impacts of up to 40 mph (64 km/h), where major damage would have been suffered by the car, the bumper system is designed to deform just enough to keep the forces applied to its attachment below the level at which the attachments fail, so absorbing a large amount of energy that would otherwise contribute to increasing the severity of other damage or personal injury. In doing so, the bumper system is irreparably damaged.

MATHEMATICAL TREATMENT

The design problems arose from the need to reconcile the conflicting requirements of the three functional stages and to achieve a smooth transition between them while maintaining a high efficiency at all times. The well-known theory of fluid shock absorbers using variable flow control orifices or valves is, of course, the basis of the design but the conventional solutions of shaped needles, multiple orifices, valves, etc., were all ruled out by the special considerations of the multi-stage action required. The answer was found in the patented combination of a specially designed liner to the cylinder, supported by an elastic member and supplemented by a yielding member which deforms to prevent the cylinder bursting at the highest rated speeds.

The design of the orifice is based on the retardation formula:

$$G = \frac{(V_1^2 - V_2^2)}{(L_1 - L_2) \times Kg} \qquad (1)$$

derived from $$\frac{M(V_1^2 - V_2^2)}{2g} = F(L_1 - L_2) \qquad (2)$$

The quantity of oil displaced by the piston movement from position L_1 to L_2 has to pass through the orifice and it is known that the hydraulic resistance of a rectangular orifice gives a flow rate per second of:

$$Q = \frac{P}{\eta}\left(\frac{t^3}{1}\right) \times K_2 \qquad (3)$$

So the displacement of oil is:

$$Q \times \frac{(L_1 - L_2)}{V}$$

units of volume, which (taking V to be the mean of V_1 and V_2) equals the volume of oil to be moved, i.e. $(L_1 - L_2) \times$ Piston area $= (L_1 - L_2)$ A; so $Q = AV$, which, inserted in formula 3 and rearranged, gives:

$$P = \frac{AV\eta}{K_2}\left(\frac{1}{t^3}\right)$$

But the retarding force 'G' is equal to Pressure P acting on the piston area A, so:

$$G = \frac{A^2V\eta}{K_2}\left(\frac{1}{t^3}\right) \qquad (4)$$

and combining with formula (1),

$$I = \frac{(V_1^2 - V_2^2)}{(V_1 + V_2)(L_1 - L_2)} \times \frac{t^3}{A^2\eta} \times \frac{2k_2}{K_1} \qquad (5)$$

where,

I	=	the variable width of our orifice
t	=	the fixed dimension of the orifice
A	=	the effective piston area
V_1 and V_2	=	the vehicle speeds at distances L_1 and L_2 along the stroke of the shock absorber
η	=	fluid viscosity
K_1 and K_2	=	constants depending on the units employed

So it is seen that the required width of orifice over any short lengths of stroke is proportional to the term:

$$\frac{V_1^2 - V_2^2}{V_1 + V_2} \qquad (6)$$

Having chosen suitable values for the terms that do not change during piston travel and a desired value for 'G' for a particular impact speed, 'V_2' can be calculated from formula (1), inserted in formula (5) to give an appropriate orifice width 'I' and the process repeated progressively, using the previous value of 'V_2' as the new value of 'V_1' in the next step, giving a profile of values of "I" which can be smoothed out as desired by adjusting the value adopted for "G". Normally, though, it is desired to keep "G" as near constant as possible for maximum energy absorption efficiency which results in values of "I" approximating to a parabolic curve. Special considerations, particularly those resulting from the inherent elasticity of the vehicle structure as well as the design of a unit to be of high efficiency at both low and high speeds of impact, suggest modifications to the basic profile and the effect of these can be calculated from the formula above.

For any chosen profile, however, the designed value of "G" will be exceeded in impacts of a higher initial speed. "G" is directly proportional to the internal pressure "P" so it is common to provide a pressure sensitive valve to "dump" excessive pressure. While such valves are valuable they do not provide an ideal solution due to a number of factors, of which inertia is a major item, and often allow over – or under – correction of the excess forces.

The orifice design adopted by Road Research Limited is constructed so that it is supported by the metal of the cylinder barrel and if excess pressure expands the barrel, dimension "I" increased by $\delta D \times \pi$. At the breaking stress of steel used, δD can be as much as 9 percent and the normal value of "I" in the order of $1/5 \times D$, therefore:

$$\frac{\delta I}{I} = \frac{.09 \times \pi D \times 5}{D} \qquad (7)$$

$$= 1.41 \text{ approximately}$$

So a 41 percent increase in orifice size can be obtained which provides an equally large drop in the excess "G" force without bursting the unit which, therefore, continues to absorb energy effectively. Even larger deformations can be obtained by interposing an elastic material between the cylinder barrel and the liner tube containing the orifice for special applications. Because the deformation of the barrel is generated directly by the internal pressure and because there are no rapidly moving components, inertia does not affect the response of the orifice which is rapid and precise.

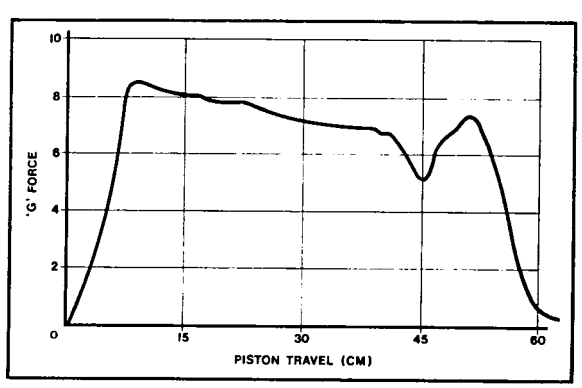
DEVELOPMENT, TESTING, RESULTS

Development of this system was undertaken under contract to the Transport and Road Research Laboratory, the necessary tests being carried out with the help of the Laboratory's impact test rig and recording equipment and with the extensive cooperation of the TRRL and its staff, to whom our thanks are due. Early problems were overcome, including a quite spectacular tendency of the fluid to catch fire under high-speed impact which was entirely eliminated by the rapid development of a suitable fluid by the Castrol Research Laboratory. The result of this work can be seen fitted to the Rover 2000 TC on display in the Exhibition. The results of the impact rig tests have shown that this unit has met all the requirements and these are summarized in Table 1.

Table 1.

Impact Speed mph	Impact Speed km/h	Cylinder Travel Inches	Cylinder Travel Cms.	Equivalent "G" Force for 1 ton vehicle
5.53	8.85	1.92	4.88	5.2

PERFORMANCE GRAPH OF SHOCK ABSORBER TEST OF TABLE 2



MEETING TODAY'S REQUIREMENTS

Private vehicles at present in production and under development do not, in many cases, have either the space or the strong points that will allow full use of the Road Research limited bumper system. For this reason, a lighter, lower rated and cheaper unit has been designed, using the same basic principle as the larger units but with a reduced specification that offers the same complete protection at impacts up to 5 mph (8km/h) and considerable energy absorbtion at

speeds up to 20 mph (32 km/h). Tests made at the TRRL demonstrated its effectiveness (Table 2).

Table 2.

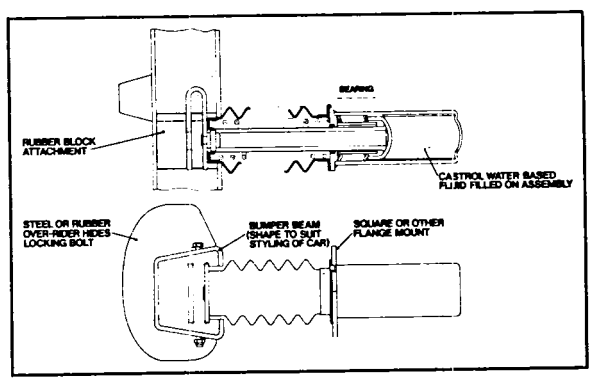
Impact Speed mph	Impact Speed km/h	Cylinder Travel Inches	Cylinder Travel Cms.	Equivalent "G" Force for 1 ton vehicles
6.72	10.75	2.5	6.35	8.08

FURTHER DEVELOPMENT

A special case of personal injury exists in the frequent incidents of small cars running under the overhanging rear of commercial vehicles. Where this occurs, even a light impact may be at the occupant's face level and therefore result in serious injury. Many heavy transport vehicles are fitted with rigid rear bumpers which can cause serious damage to cars hitting them, not only because they are too strong to absorb energy, but also because they are vulnerable to damage through incautious reversing or careless loading of the vehicle. Road Research Limited has therefore extended the use of its principle of energy absorbing pumpers to this application and is exhibiting a prototype of a heavy duty unit capable of stopping a one ton car travelling at a relative speed of 35 mph (56 km/h) with minimum damage to the occupants.

While we would not suggest for one moment that effort devoted to the prevention of accidents should be relaxed, we believe that the simple developments outlined in this paper offer an economical and effective means of reducing the loss and suffering caused by those accidents which cannot be prevented.

HYDRAULIC SHOCK ABSORBER AND BUMPER ATTACHMENT



DEVELOPMENT AND MANUFACTURE OF AUTOMOBILE BUMPERS

I. M. THOMSON
Enersorb Limited

This particular development stemmed from work being carried out by Cable Belt Limited, a company involved in the manufacture of heavy duty belt conveying systems. One of the principal components of their system is the belt. This belt must be transversely stiff so that it can span between the driving ropes and support the load, but longitudinally flexible so that it may easily traverse the drums at each end. This belt has these properties by having moulded within it steel or fibreglass straps of sufficient cross section and correct modulus so that it will trough naturally under load.

Obviously the manufacture of this belt is a time consuming operation with the hand laying of rubber and steel strips before final vulcanization of the belt. For many years Cable Belt has been lookint at a way of making rubber as stiff as if it were reinforced with steel, but still retain the high elastic elongation ability of rubber. Several preliminary designs had been tried but lacked all of the properties required. In 1972 I.C.I. developed a polyester fiber for other purposes which was sent to us for evaluation. This polysester fiber because of its modulus and construction has the unique property of when correctly combined with an elastomer acting as a "composite beam" construction. The fiber takes most of the tensile and compressive forces while the elastomer takes most of the shear when the component is bent, therefore the resistance of bending as in Figure 1 is considerably different from the plain elastomer.

The effect is quite remarkable. It will increase the modulus of rubber from the order of 1,000 psi to something like 100,000 psi with only a few relatively light layers of fabric being added. It also has other advantages in that with the correct weave of fabric the stiffness can be made uni-directional or vary in different directions. The flexibility of this type of design is extremely important, and the necessary development work is proceeding to change the belt manufacture to this type of construction.

One of the principal advantages of the Cable Belt conveyor is the relatively long life of the belting, lives of 10-15 years in field service under arduous mining conditions are not unusual. No satisfactory answer to this performance has ever been given but testing carried out would indicate that whereas the conventional belt has no bending stiffness, the Cable Belt belting, because of its internal stiffness, absorbs impact and shocks due to the springiness of its structure; therefore in the evolution of the polyester reinforced belt thorough testing was carried out in subjecting the construction to impact. In all these tests it showed itself to be much more efficient at absorbing impacts without damage than either spring steel or fiberglass reinforced Cable Belt belting.

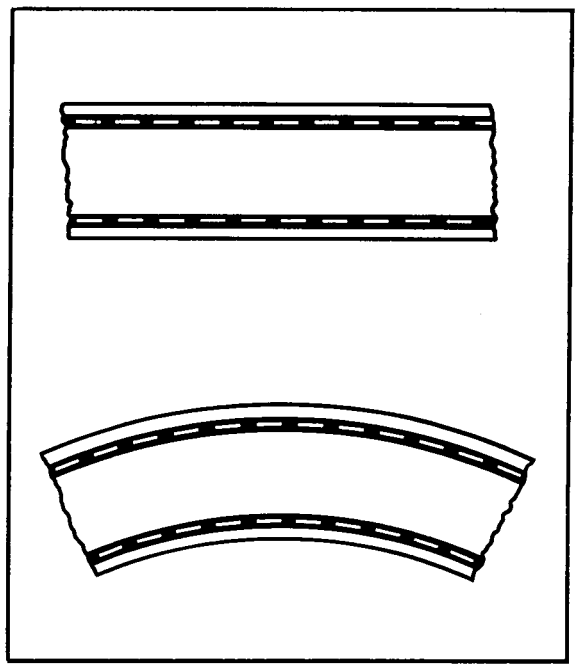


Figure 1

As the properties of the reinforcement imparted to the rubber seemed so unusual thought was given to possible other uses of such material. At that time the American federal government was introducing legislation which made it mandatory for vehicles to absorb low speed collisions without damage to safety systems. The work on conveyor belting that was being carried out seemed to lend itself naturally to use in such vehicle systems. It was obvious from reading available literature that the automobile engineer was really looking for a material which had the same modulus as steel but had an elongation comparable to that of rubber, and this new material which by then had been named "Enersorb" would appear to go some way to fulfilling these requirements.

Initial tests were carried out by reinforcing flat slabs to see whether they could be used to simulate steel panels and whilst it was quite obvious that we could make rubber rigid and still allow high elongations, during deformation the concept in itself

was of little use in vehicle design as the requirement was to transfer energy from the colliding mass into the vehicle body and controlling such transfer of energy so that the loads beyond vehicle body capability were not imposed.

Rubber is virtually incompressible, therefore for it to absorb energy and still restrict the loads passed to the rubber component it must deform. The usual method of allowing rubber to absorb energy is to place the rubber in shear, and there are many instances in vehicle and other designs where this principle is used. For example engine mounts, and rubber bushes have used this principle successfully for many years. We felt that the particular point about the Enersorb material was that it has great resistance in bending which is normally lacking from rubber. Therefore we decided to design the component such that this property was used almost exclusively. The initial design for such construction was a simple flat slab of Enersorb material which lay parallel to the load, therefore when loaded as a column the flat slab would buckle and thereby absorb the energy. This however is obviously an impractical design and as such did not prove particularly efficient. The next step was to move to a new type of construction with two parallel or nearly parallel legs connected by surface which could take impacts. This was the type of construction that was eventually adopted, as shown in Figure 2.

Contrary to expectations it was discovered that in bending the material had a non linear spring rate and the effect of this is that the bumper unit when deflected, very quickly rises to a high value and remains approximately at this level until finally the legs start to contact each other and the rubber goes into straight compression when of course very high loads are transmitted through the bumper unit.

Figures 3 and 4 illustrate the geometry of what happens during impact and the resultant force/deflection graph. The various stages of the force build

- Figure 3(a) is as the original part – Figure 4.
- Figure 3(b) is the point of commencement of bending and represents the stand of the plateau level on Figure 4.
- Figure 3(c) represent the plateau level as Figure 4.
- Figure 3(d) with the buckled legs now touching represents the rapid build up of load in Figure 4.

As in Figure 4 of the graph of force and deflection the area under the curve represents the energy. If the device were to be 100% efficient the load would remain constant over the total stroke.

Some hydraulic and mechanical devices come very close to achieving this condition, obtaining efficiencies between 90% and 95%. A normal metal spring will give an efficiency of approximately 50% and many rubber based devices acting in shear will give efficiencies between 60% and 70%. However we have with this particular device achieved efficiences between 80% and 85%. It is obvious that because of the geometry of the unit it is possible to achieve substantial strokes normally only available from hydraulic or mechanical devices.

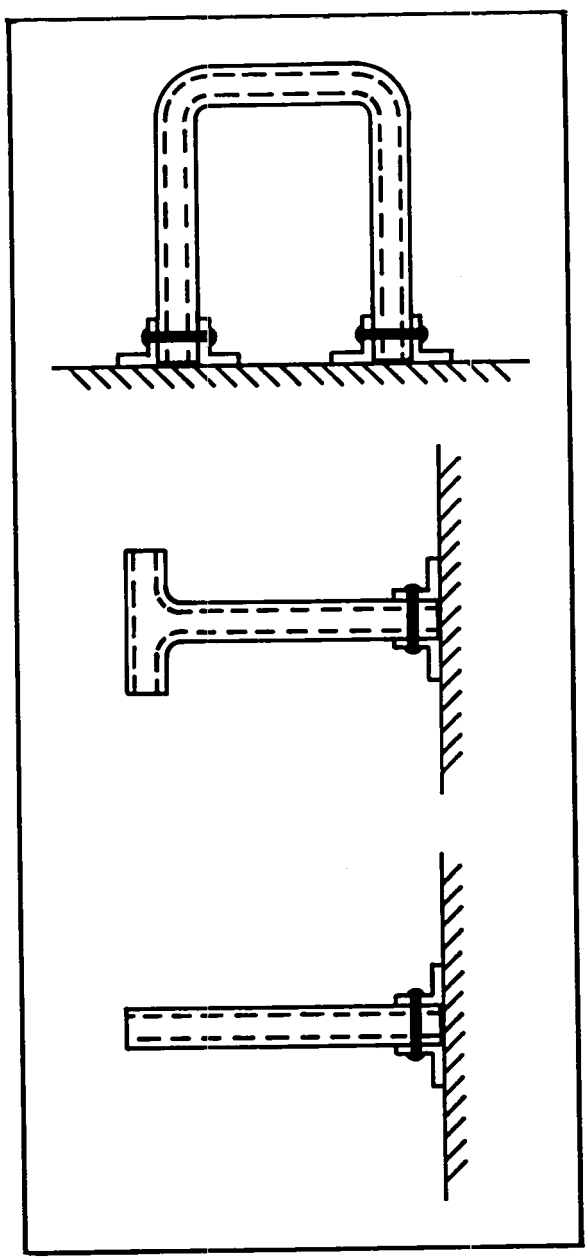


Figure 2

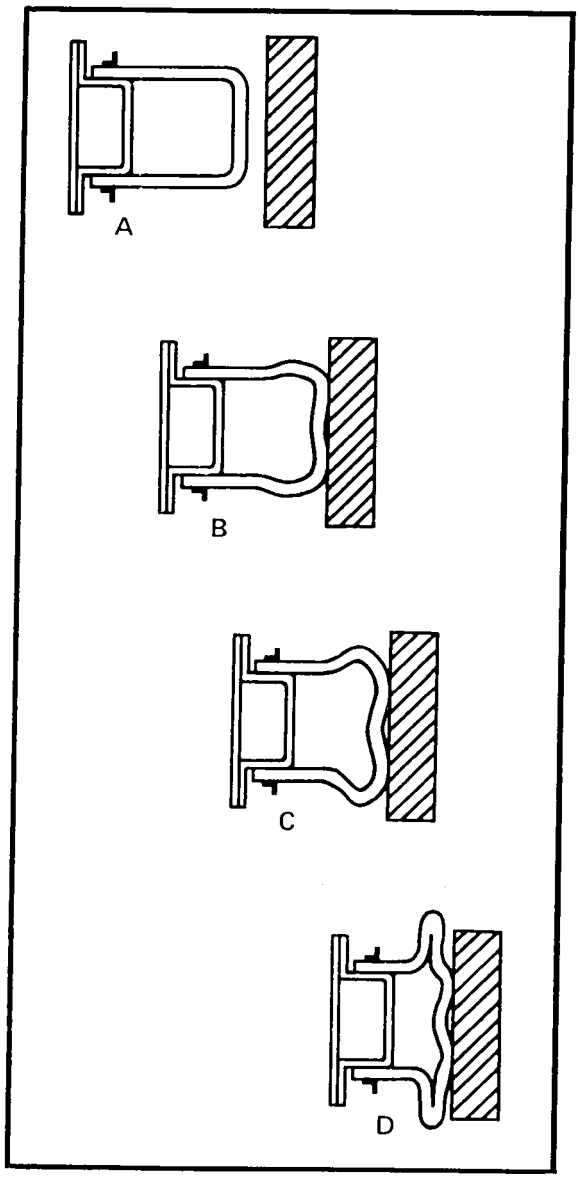


Figure 3

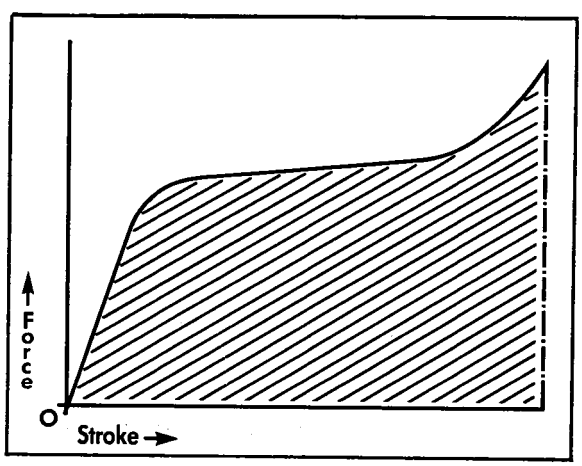


Figure 4

From the testing that has been carried out to date the following general design criteria must be taken into consideration.

A non linear impact surface such as normal pointed shape found in a car gives a much slower build up to the maximum load, but in meeting the U.S. Federal regulations such a non linear impact surface makes the load transmitted to the vehicle frame more nearly equal under the pendulum and barrier impact tests. The more nearly the legs are parallel the more quickly the rise to the maximum load. By adjusting the amount or spacing between the fabric of the legs and the top of the "U" significant differences can be achieved in the way that the unit bends and the resultant energy that it dissipates.

The unit is elastic, all the deformation is eventually recovered, but there is a significant difference between the first loading and the second and subsequent loadings, where the unit becomes considerably less stiff, something of the order of 70%-80%, than in the first test. This appears to be a conditioning process which it is possible to reduce the effect of by different pre-treatment of the fabric but this does not affect the dimensions or appearance of the unit.

The material is not velocity sensitive and tests have been carried out in the range between 1.5 feet per second to 12 feet per second with no measurable difference in its effective stiffness. There is of course a substantial increase in the dynamic stiffness over its static stiffness.

With regard to the mounting of the unit, it is important to rigidly clamp the rubber component and also that the metal beam supporting it should be rigidly attached to the vehicle body. As will be obvious if an impact occurs near the top of the rubber unit, such as in the pendulum tests, this will have a considerable turning moment on the armature.

Many of the properties of the unit are obtained from the elastomer used, including that of the surface finish, and final choice was influenced mainly by the following.

Various elastomers were considered but all the normal elastomers suffer from the same defect, which is that the rubber ages. It ages due to heat, sunlight, ozone, or any combination of these, but the ultimate effect is that it loses its elastic and other properties and becomes more brittle. For these reasons even although the bumper may be cheaper using conventional elastomers, like styrene butadiene rubber or polychoroprene, we decided to use one of the newer elastomers called EPDM, which has the important characteristics of not ageing at the same rate as conventional rubbers. It is almost impervious to ozone attack and in addition will retain its elastic

properties in strong sunlight probably 50-100 times more effectively. The good ageing properties make it suitable for placing near parts of the vehicle which may become hot, such as engine compartment or over the exhaust. One other consideration which greatly influenced the choice of this elastomer, is the effect of temperature.

The temperature range in the U.K. is quite low compared to that prevailing in the U.S. and large parts of Europe, but if an energy management system is to behave effectively and consistently over a substantial temperature range an elastomer which maintains its properties nearly constant is essential.

As can be seen from Figure 5 the unit when made from the DuPont elastomer Nordel remains reasonably constant.

There are many elastomers which at temperature of minus 10°C beome brittle, and as such lose most of their energy managing potential. Again at the higher temperature of 50°C to 60°C which can easily obtain in sunlight the decrease in modulus and the resultant lowering of the effective stiffness of the unit will result in a decrease in its energy management potential.

The final consideration which influenced our choice was that an effective painting system has been developed for EPDM rubbers.

We do not believe that the present practice of using black unpainted parts is satisfactory in the long run. It does inhibit the styling and the vehicle color range available, and we believe that flexible rubber parts to be fitted to vehicles will have to accept and give paint finishes as good as those obtained at present on steel. There has been a recently developed process in the U.S. by the DuPont Company allowing normal polyurethane based or flexible acrylic enamel paints to be applied to their EPDM rubber Nordel. The finish approximates closely to that obtained on steel and we believe that it is now possible to provide energy absorbing panels for automobiles which will not inhibit the stylists.

As this type of construction is dependent on a conventional textile reinforced elastomer type of process, obviously a substantial capital investment is required to manufacture in quantity. At present the manufacturing process is based on compression moulding techniques in large compression presses which manufacture 10 bumper units each cycle with a cycle time of 30 minutes, but it is anticipated that with advances in compound formulation production techniques and vulcanizing methods, that this cycle time will reduce substantially to allow greater production with a lesser capital investment.

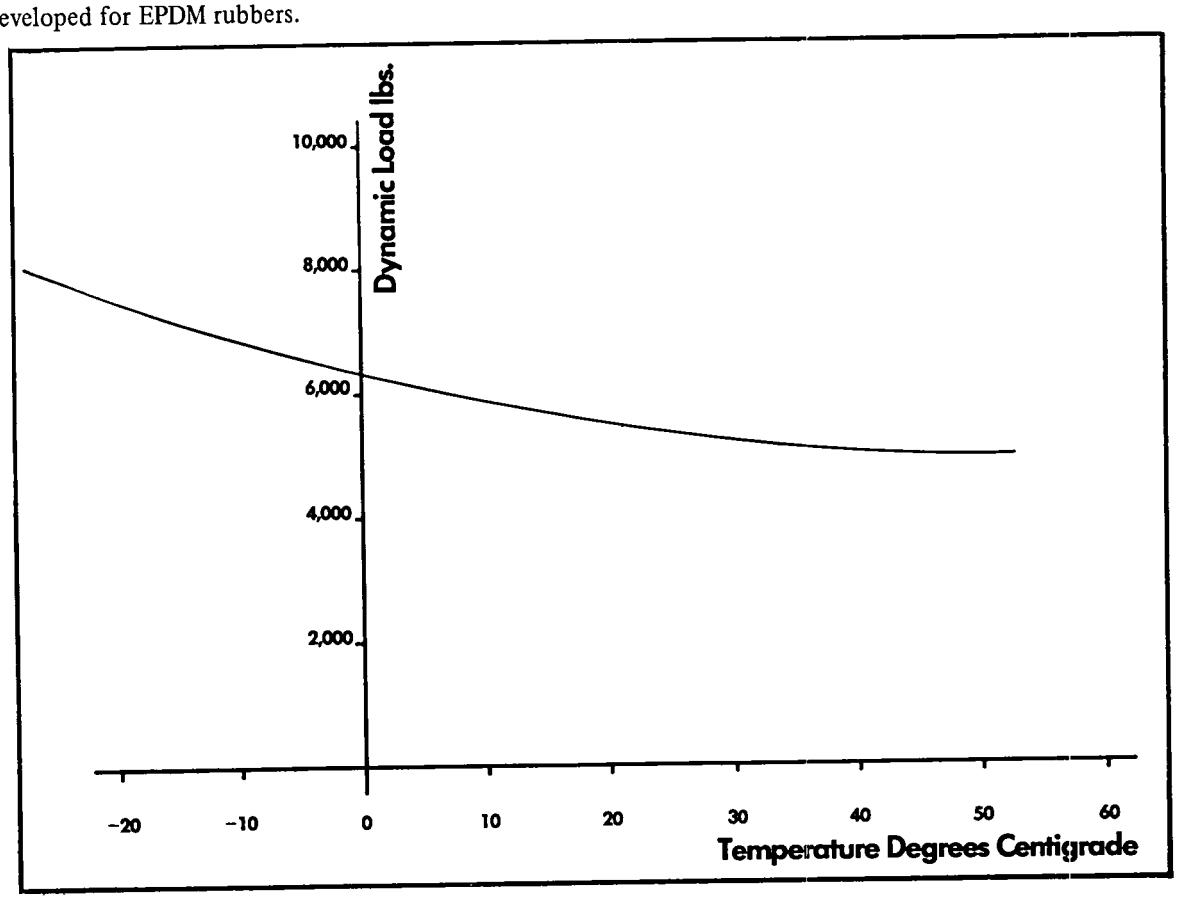


Figure 5

REVIEWS ON ACCURACIES AND RESULTS OF ESV COLLISION TESTS

MASAHIKO SAKIMURA
Sixth Research Dept.
Japan Automobile Research Institute, Inc. (J.A.R.I.)

SYNOPSIS

For the development of the safety vehicle, automotive engineers are forced to test the costly experimental vehicles in a very limited times and to get the accurate and reliable data, sometimes with a very few knowledge on test vehicles. Thus it is quite important and useful to review and discuss experimental technics and their resultant data to clarify to what extent they are reliable and how the methods should be revised. Here we introduce test measuring equipments and technics, discuss the data reduction methods and their feasibility and give a few comments on resultant data on Nissan and Toyota ESVs.

MEASURING EQUIPMENTS

The design features of electronic data acquisition system are shock-durable on-board instrumentation, multiplexed and wired telemetry by Pulse Code Modulation, multichannel instantaneous calibration method, and duplex-wiring for the redundancy of 1.5. Data reduction system, which consists of digital computing circuitry, contains digital filters specified in SEA J211 and Japanese ESV Test Method with other "optimum signal processing" softwares. (These are introduced in some detail by the slides, Figures 1, 9, 10, 11).

DUMMY HEAD ACCELERATION ANALYSIS

The triaxial acceleration sensors are mounted in each dummy head to ensure the complete redundancy for the head acceleration data; No. 1 sensor precisely at the gravitational center with No. 2 sensor on it. The dual data compared with one another and thus enabled us "data reliability cross checkings" (Figure 2). The coincidences in the dual data in general are fairly well except those of head impact to instrument panel and of rapid swing motions. We stated the accuracy of ± 4% in Test Specifications, but let us comment here it is "at peak amplitude." To ensure 3% accuracy for residual acceleration peaks naturally leads to ensure 1% accuracy for uniaxial acceleration data, and for this purpose cut-off frequency of 250 Hz is still higher than the optimum value we analyzed for each channel by a break-line path in process flow chart shown in Figure 2. We show one datum and an example of its analysis as frequency spectrum and filtering results. The datum (T7 DHA-1X: Toyota ESV Frontal Collision to Fixed Pole, X-axis

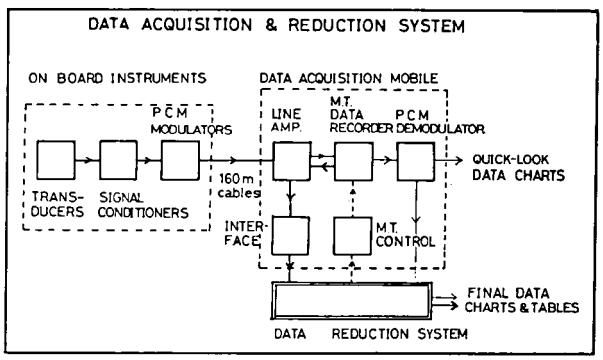


Figure 1

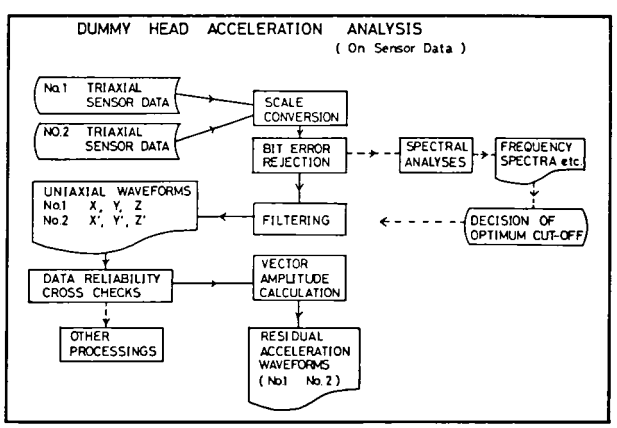


Figure 2

acceleration of driver dummy) is one of the best-S/N ratio data. It is seen, from the spectrum, the components above 150 Hz are random-phased and similar to white noise. But the announced data on Japanese ESV are *all* processed by the filter with 250 Hz cut-off frequency to follow the announced Test Specifications. (Other examples of the dummy head acceleration analysis are also discussed with the slides.)

VEHICLE COMPARTMENT ACCELERATION ANALYSES

The difficulty of reducing the precise waveforms of the vehicle compartment accelerations arises from the fact that no definite method of separating shock component from vibrational component is existent. We adopted the method of filtering and "space averaging" at the same time as shown in the flow chart (Figure 3), and get fairly good results. "Space averaging" means for averaging the data from several sensors distributed over a compartment floor. The

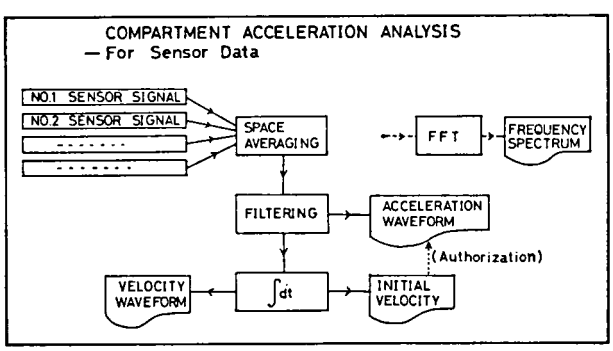


Figure 3

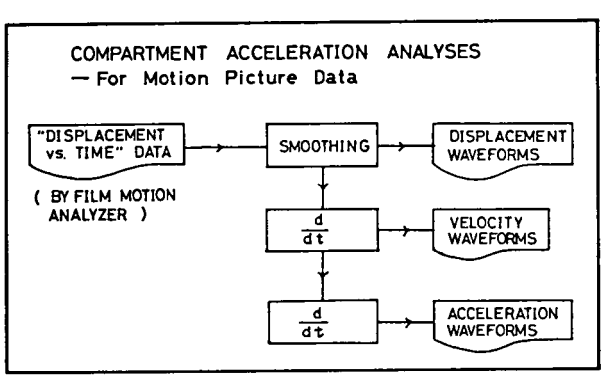


Figure 4

resultant data are authorized only if they passed the initial velocity checkings. Also we compare the sensor data with those from motion picture analysis in the forms of velocity and acceleration waveforms. (Some examples are shown by the slides.) We should attract attention to the fact that "smoothing" in the motion

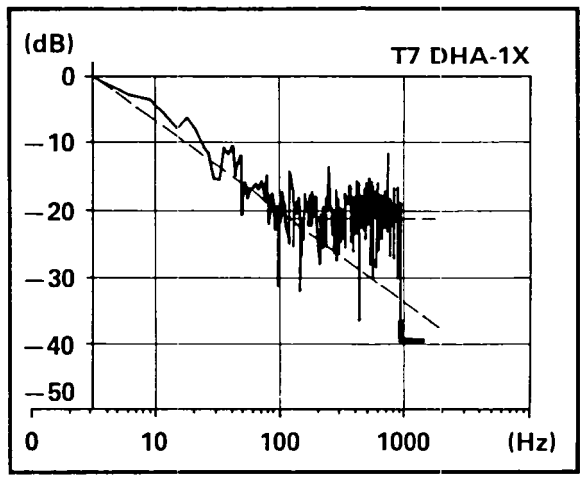


Figure 5

picture data processing decrease the signal bandwidth and thus can be compared with filtering.

OTHER DISCUSSIONS ON ESV TEST DATA

The discussions on transducer calibrations, mounting, experimental failure, data on restraints, and future prospects are introduced and discussed briefly by the slides.

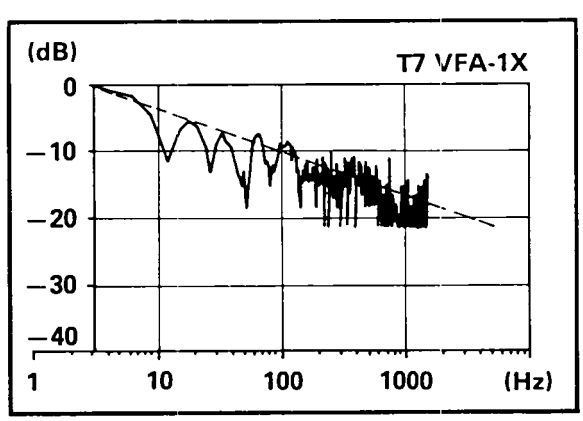


Figure 6

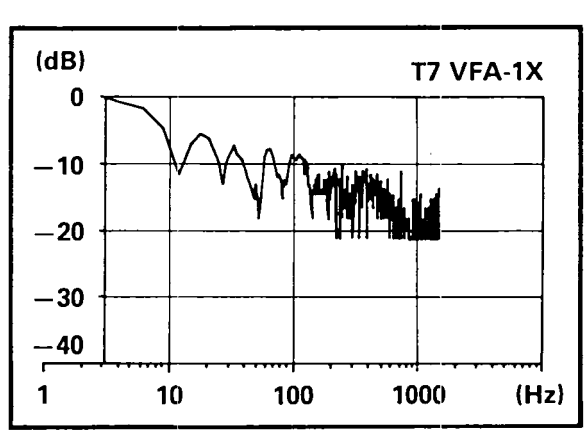


Figure 7

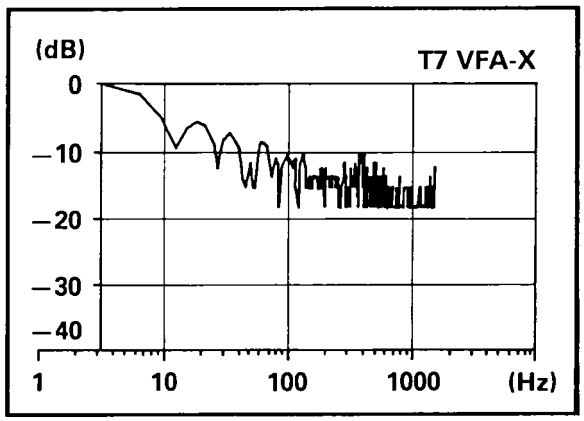


Figure 8

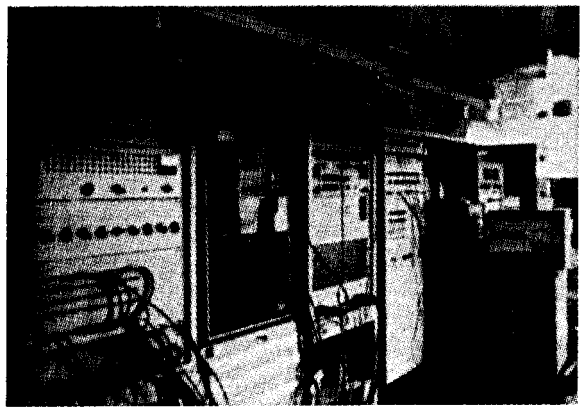


Figure 9. Inside the Data Acquisition Mobile shows (from left to right) PCM demodulator, Magnetic-Tape Data Recorder, Recorder Control and other charting instruments.

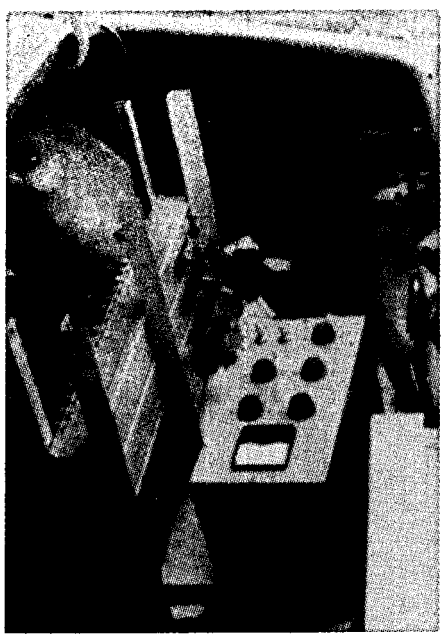


Figure 10. On-board instruments inside the vehicle rear trunk with calibration box is shown. Each of four signal-conditioner units contains 16 channels. Calibration box is removed before the vehicle launching. PCM modulators are below the calibration box.

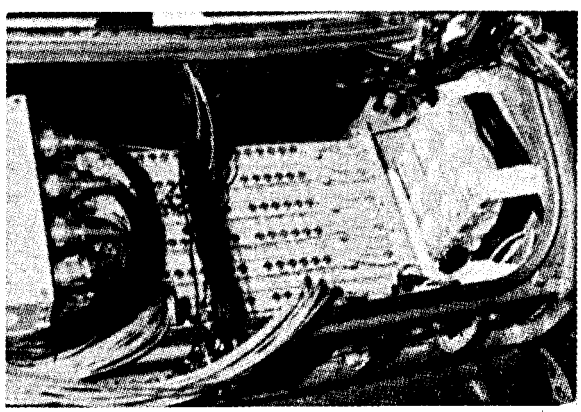


Figure 11. Another packaging of on-board instruments shows (from left to right) PCM modulator units, six signal conditioner units before wiring, and power units for modulators.

CHARACTERISTICS OF BODY ENERGY ABSORPTION AND RESTRAINT SYSTEM

KAZUO HIGUCHI
Honda Motor Co., Ltd.

INTRODUCTION

On account of its limited size and weight, the small car has real difficulties in the protection of its occupants in the event of collision.

In order to enhance the safety of a small car in the event of collision, the maximum utilization of its limited space must be achieved by not only improving efficiency of the restraint systems and crash characteristics of the vehicle, but by also making sure restraint systems and crash characteristics are well-matched with each other.

Factors having relation to the restraint system and crash characteristics are numerous, so that clarification of their effect cannot be made but with the help of computer simulations.

Shown in Figure 1 is a mathematical model for computer simulation used in Honda. It is a two-dimensional mathematical model based on the concept in which a vehicle occupant is considered to consist of seven masses: (1) head, (2) neck, (3) thorax, (4) buttocks, (5) upper leg, (6) lower leg, and (7) arm. This model is constructed with a special emphasis placed on the following points:

1. It should be of such construction that the characteristics of a dummy's movement are simulated closely as far as it will not make the whole system too complicated.
2. The results of tests conducted on various restraint systems can be used directly as they are as input data.
3. Considering its relation to the seat and the location of its anchorages, the belt system should be treated as part of a three-dimensional model.

This paper is on the results of studies conducted by the use of this model with respect to the following subjects:

1. Relation of various vehicle crash pulses having the same crash stroke to the occupant's injury.
2. Effect on vehicle occupants of the frequency and amplitude of high frequency components appearing in the vehicle crash pulse as a result of buckling of structural members of the vehicle, for instance.

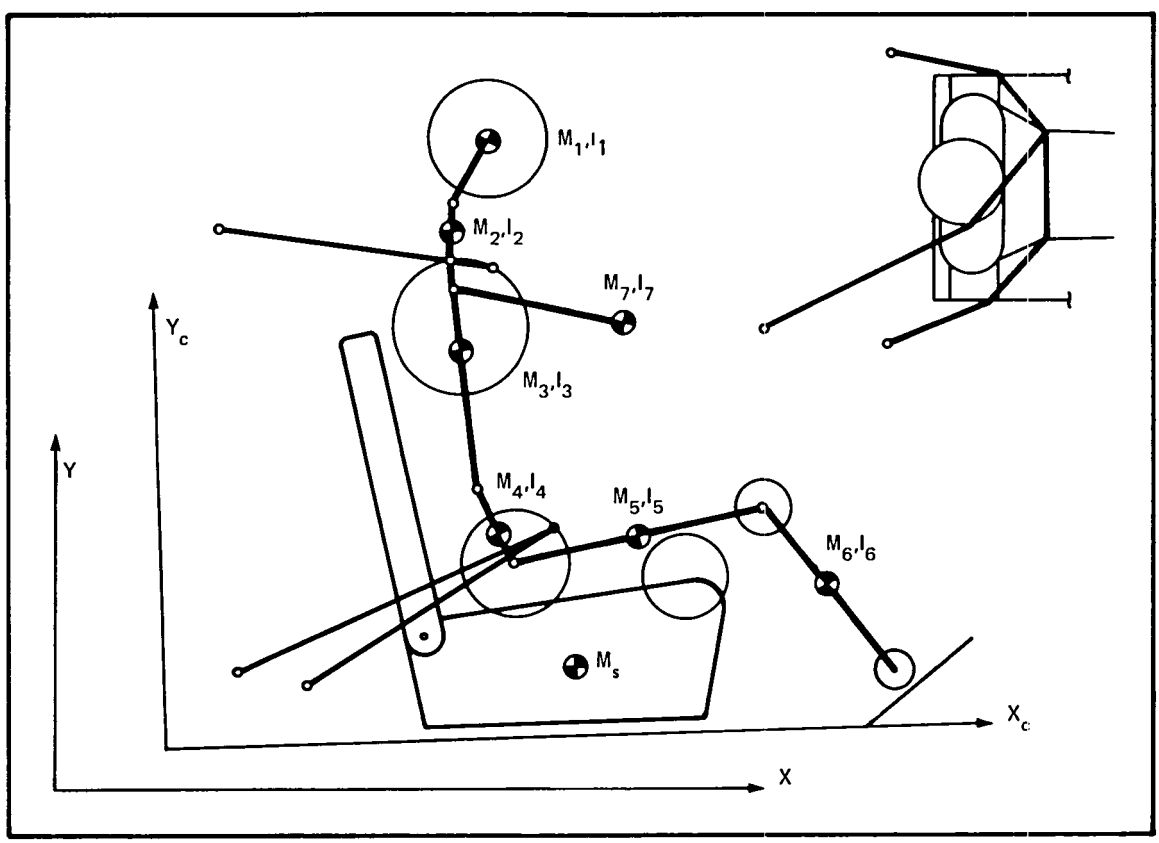


Figure 1. Mathematical Model

3. Effect on vehicle occupants of combinations of vehicle crash pulses and times required to operate an emergency locking retractor (ELR) used on the shoulder belt.

VEHICLE CRASH PULSE AND OCCUPANTS INJURY

Among various restrictions imposed on a small car, the one which is given in the first place is a crash stroke of the vehicle. For a given crash stroke, study must be made as to what crash pulse, if embodied, is how effective in mitigating occupant's injury.

Vehicle crash pulses are given by the following equations:

Deceleration:

$$\ddot{X}_c = -\left\{a_0 + a_1 \sin[(n\pi/t_1)t + \phi]\right\}$$

Velocity:

$$\dot{X}_c = -a_0 t + a_1 (t_1/n\pi) \cos[(n\pi/t_1)t + \phi] + c_1$$

Displacement:

$$X_c = -1/2 a_0 t^2 + a_1 (t_1/n\pi)^2 \sin[(n\pi/t_1)t + \phi] + c_1 t + c_2$$

when $t = 0$

$\dot{X}_c = V_0, X_c = 0$

when $t \geqslant t_1$

$\ddot{X}_c = 0, \dot{X}_c = 0, X_c = S_0 = \text{const.}$

where t = time after impact

t_1 = duration of impact

S_0 = vehicle crash stroke

Vehicle crash pulses are assumed to consist of a square wave component a_0 as a base with a sine wave having an amplitude of a_1 superposed thereon.

Now, the frequency and phase of a sine wave are given by n and ϕ respectively.

Shown in Figure 2 are those pulses which are used for computation. Assuming that a crash stroke is constant, the mean deceleration becomes lower and lower from those pulses which have the peak value at the end as Pulse A, to a square wave of Pulse C and to those having the peak value at the beginning as Pulse E (see Figure 3).

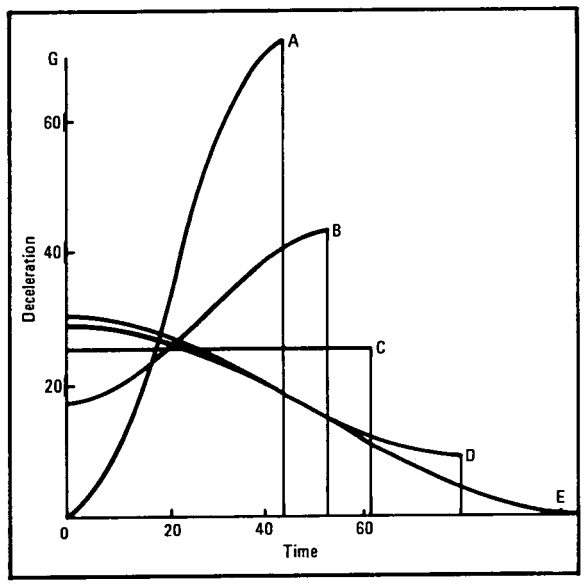


Figure 2. Crash Pulses.

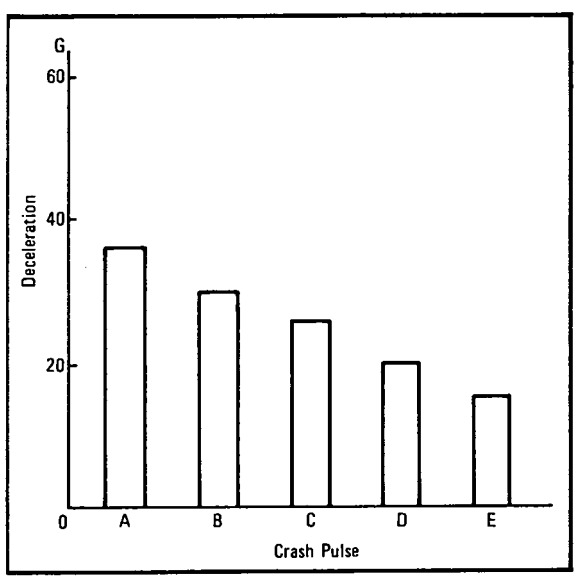


Figure 3. Mean Deceleration of Crash Pulses.

Shown in Figure 4 are head injury criteria (HIC) computed on these pulses. In the case of those pulses which, like Pulses A and B, are lower at the beginning as compared with Square Wave C and which have the peak value at the end, HIC increases considerably. This is considered to come from the fact that a low initial deceleration makes it unable to utilize the ride down effect sufficiently.

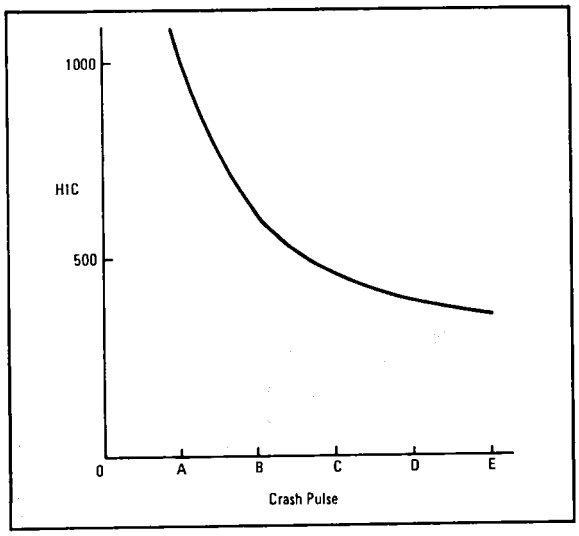


Figure 4. Crash Pulse vs. HIC.

On the other hand, those pulses which, like Pulses D and E, have the peak value at the beginning produce a better result as compared with the square wave; however, the difference between them are not much.

Those which are shown in Figure 5 are the crash pulses in Figure 2 expressed in the form of deceleration vs. crash stroke. You can see from this figure that there exists only a small difference in the deceleration between Square Wave C and those pulses which, like Pulses D and E, have the peak value at the beginning. Since it is extremely difficult from an engineering point of view to construct a vehicle of which the crash pulse is such that deceleration is high at the beginning and becomes low at the end, this fact, together with the consideration of occupant protection capability, makes such a crash pulse uncalled-for.

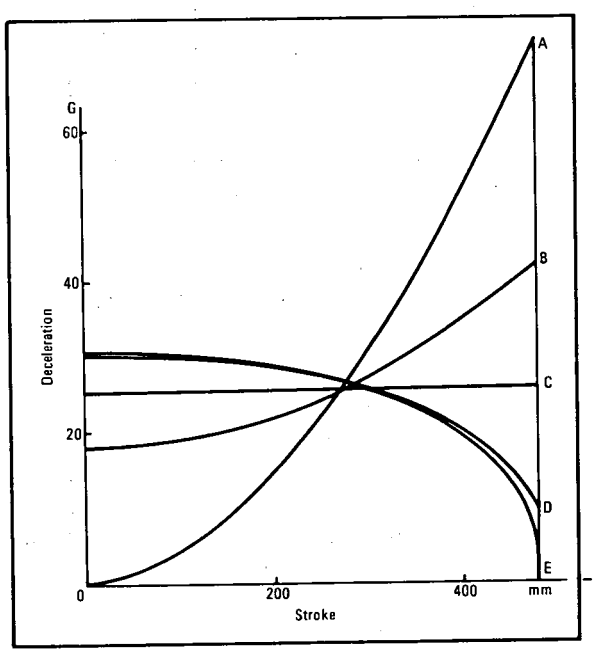


Figure 5. Deceleration vs. Stroke.

The foregoing result may lead to the following conclusion: Those crash pulses which are such that deceleration is low at the beginning and becomes high at the end must be avoided at any cost, for they produce a remarkably adverse effect on HIC. On the other hand those crash pulses which are such that deceleration is high at the beginning and becomes low at the end are far less effective as compared with its being technically difficult to be embodied. Therefore, as crash pulse for a small car, the square wave is the best.

In computation referred to in this paper, assumed ideal energy absorption belt characteristics are employed. If belt characteristics now available are used for computation, HIC becomes significantly worse; however, the tendency as a whole remains to be the same as with the case where assumed ideal characteristics are used.

EFFECT OF HIGH FREQUENCY COMPONENT IN CRASH PULSE

Even though an ideal pulse is aimed at, buckling of vehicle structural members, etc. makes it unavoidable to have high frequency components superposed; thus, it is necessary to conduct a study on their effect.

Shown in Figure 6 are crash pulses used for computation of effect of the frequencies of high frequency components. It consists of a square wave with cosine waves of 1 cycle (≐ 15 Hz) through 6 cycles (≐ 90 Hz) superposed thereon.

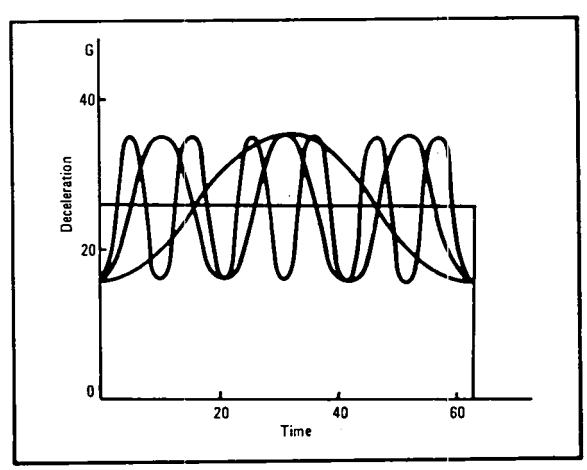


Figure 6. Crash pulses with high frequency components (parameter: frequency).

The computed result is shown in Figure 7, in which + indicates those pulses which are in the initial deceleration and – indicates those being low in the same.

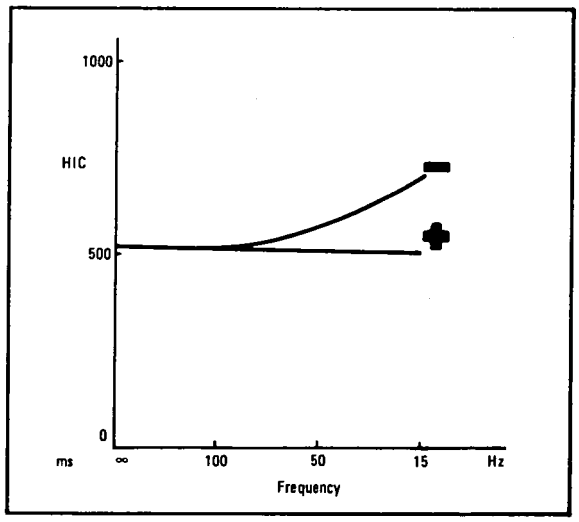


Figure 7. Frequency vs. HIC.

Those pulses which are in excess of 50 Hz in frequency have little effect on HIC, while those indicated with – show a bad result in the neighborhood of 15 Hz, which, however, is considered to have resulted, judging by frequencies in question, not from the influence of high frequency components, but from a worsening of the ride down effect as a result of low initial deceleration.

Study was made on effect of the amplitude up to approximately ± 20 g when cosine waves of approximately 15 Hz are superposed (see Figure 8). The result is shown in Figure 9. So far as the amplitude stays within the range of approximately –5 g to + 10 g, its effect is not noticeable, but it is different when the amplitude moves out of this range.

From the foregoing results, the effect of high frequency components are summarized as follows: Components which are in excess of 50 Hz in frequency have little effect on the result, while those of approximately 15 Hz which are likely to change the pulse entirely affect the result, particularly effect of the amplitude on the result is greater with those pulses which are low in deceleration at the beginning than those which are high in the same; therefore, special attention must be paid to the initial rise of deceleration.

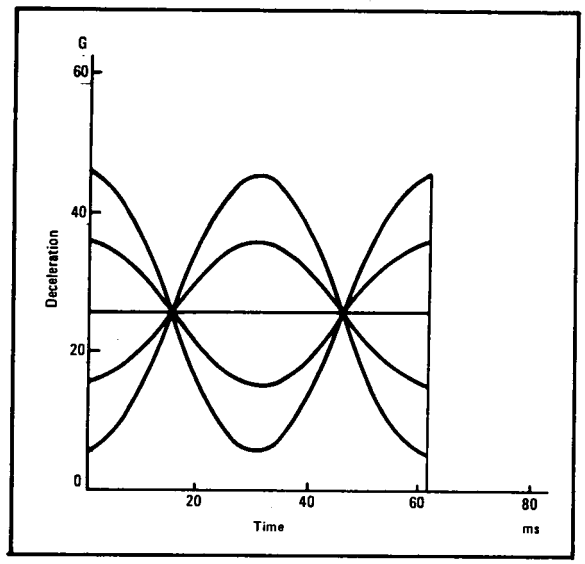


Figure 8. Crash pulses with high frequency components (parameter: amplitude).

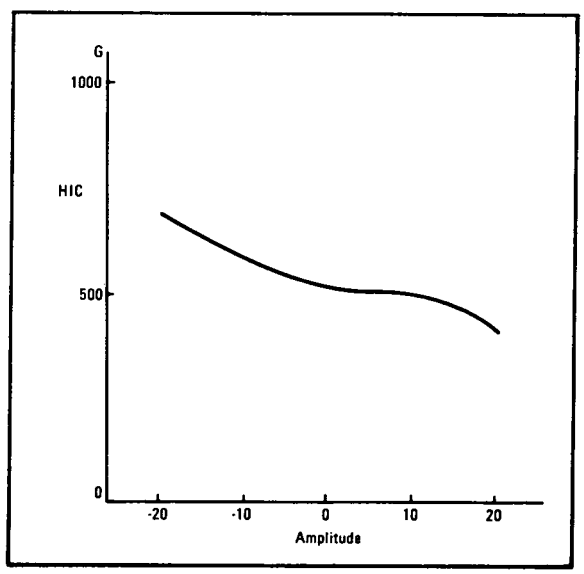


Figure 9. Amplitude vs. HIC.

LOCKING TIME OF RETRACTOR AND CRASH PULSE

If an ELR used on the belt, time required to lock it cannot be reduced to zero. Study was made on relation between the locking time of an ELR and floor G pulses. Shown in Figure 10 are the results of study on combinations of locking time of an ELR of the shoulder belt which varies within the range of 0 ms to 25 ms and vehicle crash pulses assumed to be Pulses A, C, and E in Figure 2.

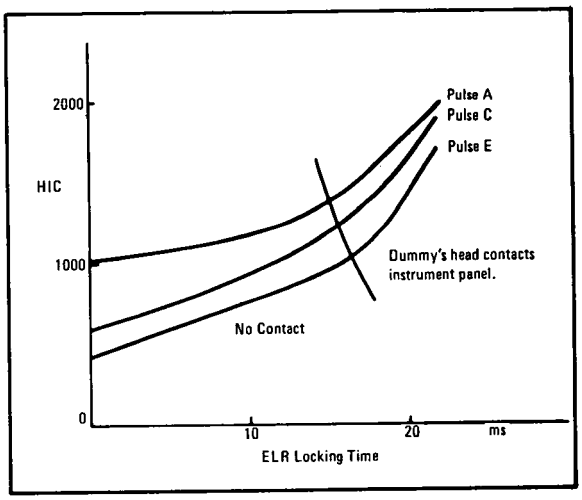


Figure 10. ELR locking time vs. HIC.

As the ELR locking time becomes slower, HIC grows naturally to be worse. However, in the case of Pulse A, effect of the ELR locking time on HIC is slightly smaller. This is presumed to be attributable to the following fact: since pulse A is originally a pulse with only a small ride down effect, influence of the worsening thereof as a result of a delay in the ELR locking time is smaller on it than on those pulse which utilize the ride down effect as pulses C and E.

In view of the foregoing, it can be said that the higher the ride down effect of a vehicle crash pulse is, the more adverse effect on HIC will be as a result of a delay in the ELR locking time.

In Figure 10, when the ELR locking time is over 15 ms to 17 ms, the dummy's head contacts the instrument panel to worsen HIC remarkably; such a locking time and HIC value are subject to change in accordance with location of the instrument panel and the crash characteristics of a vehicle to be studied.

CONCLUSION

By studying with the help of computer simulation effect of the vehicle crash characteristics and ELR locking time on HIC of occupants, the following conclusion is obtained for the report:

1. Among various crash pulses, a square wave is suited the most to small vehicles.
2. High frequency components in excess of 50 Hz frequency to be superposed on a crush pulse have little effect on HIC.
3. It is important for a crash pulse to have a good rise in the beginning for utilization of the ride down effect.
4. With respect to the ELR locking time, "the quicker, the better" is true; however, the better the initial rise a crash pulse has, quicker the locking time is required to be.

Factors other than those discussed in this paper, particularly seatbelt webbing characteristics, have also a large effect on vehicle occupants.

With small cars in particular, the characteristics of these factors must all be well matched with one another.

In computations referred to above, values of the VIP50 type dummy were used. However, if a dummy of different characteristics, particularly that with different neck characteristics is used, the result may be different to some extent.

The early perfection of a dummy with more humanlike characteristics is being looked for.

SECTION 3

TECHNICAL SEMINARS

PART FOUR – ACCIDENT AVOIDANCE

Chairman: DR. G. POCCI
Ministry of Transport
Italy

Themes: Estimation of Effectiveness of Safety Measures;
Engineering Systems;
Interaction of Driver and Vehicle

Introduction by the Chairman:

The reasons why active safety has a less important role in the ESV Program is that most countries already have governmental provisions on such active safety features as braking, lighting, and tires. This does not mean that these are minor problems. I must say that for active safety we all have a certain amount of tradition. Part of this derives from the progress which was made before the war. The increased production of sport cars made a major contribution to the solution of these problems. Things have changed a bit since that heroic pioneering era when the deaths which took place contributed to today's safety. But we can't have that happening again. We want to manufacture vehicles which will avoid accidents, and we want our colleagues to make sure that as accidents do occur the severity will be as low as possible.

The speakers we have this afternoon are in three groups. Group 1 will be from the United States and will be represented by Mr. Hayes, who will speak of road holding and the driver limits, and by Mr. Kline, who will present a research program to measure the behavior of the driver under varying road conditions.

The second group will be from Japan. Mr. Sakai will talk about the tires which were chosen after extensive tests for ESV vehicles. Then, Mr. Sano, from Honda, will speak about the characteristics of controlling vehicles with special safety tires.

The third group will represent the European Experimental Vehicle Committee.

Dr. Enke, from the Federal Republic of Germany, will discuss the relationship between road holding and the maneuverability of the vehicle and the possibility of avoiding accidents. Then, Dr. Weber and Dr. Manger will discuss transient slip angles of tires.

From Italy we will have Mr. Surace who will discuss progress in the mathematical method of simulating vehicle behavior and the research made in the relationship between the vehicle and the driver. He will also speak about the application of the mathematical methods to braking and cornering. Then, we will have Mr. Soardo, also from Italy, who will speak on the research on brake lights that he is carrying out in association with the Italian Transport Ministry. He is also going to discuss a system having to do with the intensity of the brake lights being linked to the speed of the vehicle.

From the United Kingdom, Dr. French, from Dunlap, will speak on new safety tires. Mr. Jones, from British Leyland, will discuss loss of control; Mr. Printice, from the Transport Road Research Laboratory, will speak on mistakes on driver limitations; and, finally, Mr. Jacobson will represent the consumers and will speak on the various possibilities for safety tires.

UNITED STATES OVERVIEW

FRANCIS A. DiLORENZO
National Highway Traffic Safety Administration

The decisions we make on what must be done to reduce the probability of a driver and his vehicle becoming involved in a crash present a multisided problem for which priorities are difficult to establish. We must consider not only a machine that has proven difficult to simulate, but human drivers whose simulation is even more elusive, and who individually possess extreme characteristic variations. Thus, we progress in our understanding of this complex system in a manner that could appear highly subjective and perhaps at times lacking sound logic. As an example, for a given dollar expenditure, would it be more cost beneficial to improve the visibility or braking performance of the vehicle? With our present knowledge of accident causation, the answer cannot be firm, since sound data to substantiate either choice is not available. We, therefore, continue our research at the NHTSA in the many areas of crash avoidance compatible within our manpower, time, and budget restrictions.

To this end, the NHTSA has many on-going research programs that address categories of the vehicle, the driver-vehicle interface, and the driver. In the few minutes I have, I can only outline what I believe to be the major programs where we are concerned and give you some idea of the research efforts.

Addressing the research concerned with the vehicle, we have four programs in existence to study its dynamic properties. The first program has to do with truck and bus handling. The objectives are:

1. To develop handling test procedures for truck and large buses.
2. To develop mathematical simulations of these procedures.
3. To verify the simulations using full-scale tests of the developed procedures.
4. To obtain the limit performance data on a large intercity bus and a large tank truck.

The initial data has been gathered for this program and we are awaiting the final report.

The second program has to do with bettering our understanding of the effects that tires have on passenger cars. The title of the program is "Tire Property Effects on Vehicle Handling," and there are two major objectives to the work statement:

1. To identify the properties of tires that affect vehicle dynamic response and describe those effects in quantitative terms.
2. To evaluate the degree to which the various tire parameters affect vehicle dynamic response and to assess their relative importance.

This program is also near completion with a final report due this summer.

The third program also has to do with tire effects on vehicle handling. However, it is concerned with vehicles-in-use. We are all aware that a vehicle is used only a short portion of its total life with original equipment in proper operating condition. For the majority of its useful life, the individual owner, applying his own judgements and experience, governs the maintenance level of his vehicle. To assess these variations, as caused by the tires alone, this program will address the following factors:

1. Vehicle loading variations.
2. Tire pressure variations.
3. Tread wear effects.
4. Different tire type mixes.

The fourth program is a study to determine the methodology of defining and measuring brake efficiency. We realize that there are many schools of thought in this area and that finding a solution amenable to everyone will be most difficult. There is strong argument that the highest braking efficiency, in the classical sense, is not always the best because trade-offs with stability are involved. Then, perhaps more quantitative methods of measuring and defining a braking stability criterion is needed along with the requirements for specifying brake efficiency. Certainly establishment of this need should not

preclude the necessity of research to define more meaningful methods of measuring brake efficiency, but rather identifies another potential aspect of braking performance that needs research. For the present, it may be acknowledged that the vehicle with the highest brake efficiency for a given set of conditions – and as defined in the classic manner – does not necessarily mean the safest vehicle for all other conditions of road, loadings, and driver abilities.

The current NHTSA brake efficiency program addresses only straight-line braking. Considerations for a combination brake and steer maneuver may be made in the future. In concept, this program will utilize a hypothetical vehicle that has been sized from average dimensions, based on our total domestic vehicle population.

The performance of this hypothetical vehicle, when simulated mathematically using frictional data for a referenced standard tire, will be compared to the test vehicle to establish a number we currently term brake efficiency. Brake efficiency will then be defined as the ratio of the hypothetical or reference vehicle stopping distance to the test vehicle stopping distance. The final report of this research activity is due this summer.

Next, we turn to areas of the driver-vehicle interface that present first order effects on the driver's ability or capability to perform. Such areas include: forward and rearward lighting, visibility, controls and displays, and driver vehicle dynamics. No one study has ever been made on any of these topics that can be considered all inclusive; and, hence, they remain as on-going work for the NHTSA.

With forward lighting we are concerned with sufficient illumination without glare. To achieve this goal we are in the planning stages of a field demonstration to determine the merits of systems that employ polarized lighting. For rearward lighting we continue to study how best to transmit clear, unambigious information to other drivers. What colors are best? Which locations, what intensities, etc., are optimum? Concerning visibility and controls, we continue to study how best to configure the inside of the vehicle. How wide can the roof pillars be before vision is compromised? By sampling the behavior of drivers in various driving environments to determine how often and where he obtains his visual information, we hope to get the answers to these and other questions. The subject of controls also affects the driver-vehicle interface. Is there an optimum configuration? What are the ergometric requirements that accommodate the entire driver population?

The last – and perhaps the most emphasized aspect of the driver-vehicle interface – is driver-vehicle dynamics or handling. This year, the NHTSA has initiated two separately funded contracts to study this problem. It marks the first in a hopefully continuing effort to resolve this problem. We have scheduled two U.S. papers this afternoon that will address the objectives and status of these programs.

The last topic I would like to touch on concerns the NHTSA research on the driver as an entity. Here, we must work with a variable that possesses great variations in individual abilities and behavior, while remaining fundamentally unchangeable as a group. We can educate drivers, but when and what should they be taught? The NHTSA is studying this problem and the methods of evaluating the effectiveness of the educators. Nearly all governments mandate driver licensing, yet requirements vary greatly. For the driving test, what should we test for and how do we test for it? What are the realistic medical standards for eyesight? What minimum levels of driving performance should an applicant demonstrate? Assuming that we have adequately educated and licensed a driver, what can we do to protect him from himself and others when societal pressures, personal problems, etc., causes him to turn to excessive alcohol consumption for escape? NHTSA is evaluating such devices as ignition interlocks and detectors as a possible solution to this problem. Perhaps there are easier or more economical ways to achieve the same goals.

To these and other questions, NHTSA continues to seek answers – however difficult. Only when the last life on the highway has been lost may we cease our efforts.

DETERMINATION OF MOTOR VEHICLE CHARACTERISTICS AFFECTING DRIVER HANDLING PERFORMANCE

GORDON G. HAYES
Texas Transportation Institute
Texas A&M University

INTRODUCTION

Contemporary passenger vehicles encompass a wide range of dynamic response characteristics which may influence a driver's success in performing safety related maneuvers. The need to objectively measure vehicle response characteristics led to the development of open-loop handling test procedures under a research program* sponsored by the National Highway Traffic Safety Administration (NHTSA). A subsequent study** compared the response of a spectrum of contemporary vehicles using the tests which included maneuvers requiring braking, turning and combinations of braking and turning. In addition, one test consisted of a turn maneuver with suspension excitation ("rough road") and another was designed to produce a large roll moment. Based on the selected response parameters, many vehicles did exhibit differences when subjected to equivalent control inputs. Many of these differences were expected due to known variation in weight distribution and suspension dynamics, and the comparison parameters were chosen to relate to vehicle stability in limit maneuvers. However, in order to say that one vehicle is "safer" in an emergency maneuvering situation based on parameters determined in open-loop tests, a knowledge of the interaction between driver and vehicle is necessary. That is, differences in vehicle response may be masked or overcome by the closed-loop response of a typical driver. Therefore, an NHTSA study now in progress is attacking this problem. This paper summarizes the experimental approach of the study which is being conducted by the Texas Transportation Institute under contract to NHTSA.

*H. Dugoff, R.D. Ervin, and L. Segel, *Vehicle Handling Test Procedures*, Final Report, NHSB Contract FH-11-7297, Highway Safety Research Institute, The University of Michigan, Ann Arbor, November 1970.

**R.D. Ervin, P. Grote, P.S. Fancher, C.C. MacAdam, and L. Segel, *Vehicle Handling Performance*, Final Report, Contract DOT-HS-031-1-159, National Highway Traffic Safety Administration, The University of Michigan, Ann Arbor, November 1972.

OBJECTIVES

The objectives of this study are:

1. Identify the more significant vehicle handling and stability performance parameters in maneuvering situations requiring extreme vehicle dynamic performance up to and including limit of performance in situations such as those which would result in vehicle spin-out or plow-out.
2. Determine and assess the extent to which a representative sampling of vehicle drivers utilize the full capacity of the vehicle with respect to the significant vehicle response and feedback characteristics identified in 1 above.
3. Rank those parameters of vehicle handling and stability relative to their importance in the vehicle-driver combinations in accident avoidance maneuvers, with consideration to both the vehicle performance required and the projected likelihood of the vehicle's full performance capability being utilized.

EXPERIMENTAL APPROACH

Vehicles which have been compared in previous open-loop limit performance tests are being driven in simulated "emergency" maneuvering situations by a cross section of participants from the general public. The input and response parameters are being recorded in an effort to identify those parameters which are significantly related to driver handling ability in "emergency" maneuvers and to estimate the extent to which drivers typically are willing to utilize inherent vehicle maneuverability.

The following subsections describe the various aspects of this study.

Vehicle Selection

Four vehicles from the previous open-loop comparison were selected to represent the range of response types and weight distributions included in

that study. These can be described as:

1. *Subcompact* – Represents the small, lightweight vehicles with rearward weight bias and high center-of-mass to track-width ratio.
2. *Standard Sedan* – Represents the midsize sedans with "average" weight distribution, power, and suspension characteristics.
3. *"Personal" Car* – Represents powerful sport vehicles with forward weight bias, low center-of-mass to track-width ratio, firm suspension, and low steering ratio.
4. *Semiluxury* – Represents heavy vehicles with large yaw moment of inertia. This vehicle has front wheel drive.

Driver Selection

From a large pool of potential volunteer drivers a sample of 100 was drawn to parallel as nearly as possible the general population based on age, sex, and yearly mileage. These volunteers participated in preliminary handling tests to determine the approximate individual driving skill. Each driver received the same familiarization and handling skill assessment in a vehicle different from those to be used in the main study. The assessment was "approximate" because this testing was limited in order to minimize learning effects before the main test sequence. From the handling scores and demographic data, a final group of 32 drivers was selected to participate. Eight drivers were assigned to each vehicle, with the four groups as closely matched as possible.

Instrumentation

The vehicles, which are being tested sequentially, are instrumented to measure the following parameters:

1. Speed (fifth wheel)
2. Lateral and longitudinal acceleration in the horizontal plane (stable platform)
3. Steering angle (potentiometer)
4. Roll, pitch, and yaw rates (gyros)
5. Wheel rotations (reed switches)
6. Throttle position (linear potentiometer)
7. Brake line pressure (pressure transducer)

From the rate gyros, roll, pitch, and yaw (heading) angles can be obtained by integration. Other parameters such as slip angle and resultant velocity are computed from measured quantities. The analog transducer outputs are recorded on-board on 14 channel magnetic tape for subsequent digitization.

Maneuvers

The handling tests are divided into two groups called "replication" and "surprise." The replication tests simulate the previous open-loop handling tests with the exception of straight-line braking which was omitted. In all of these, the driver receives precise instructions on audio tape. These test maneuvers are:

1. *"Trapezoidal Steer"* – The open-loop maneuver consisted of succeeding runs with increasing steer inputs until a limit lateral acceleration was reached. The closed-loop maneuver requires that the driver, on succeeding runs, attempt to remain in delineated lanes of decreasing radius of curvature.
2. *"Braking in a Turn"* – The open-loop maneuver consisted of a fixed steer input and incremented brake line pressure to increment deceleration on succeeding runs. It was decided that drivers would be unable to maintain sufficiently constant levels of braking, so the closed-loop test employs a brake limiter to increment deceleration. However, the drivers have steering control and attempt to maintain a constant radius curve while braking.
3. *"Road Holding in a Turn"* – The open-loop tests employed three curved paths with rubber bumps spaced to give excitations of 9, 11, and 14 Hertz at test speed. The closed-loop tests are essentially the same, with the driver attempting to maintain the constant radius curves over the bumps.
4. *"Sinusoidal Steer"* – The open-loop tests employed programmed sinusoidal steering inputs, whereas in the closed-loop tests the drivers are instructed to avoid increasingly wide barriers and remain in an adjacent lane. This forces increasing reverse-steer inputs but, of course, they are not in general symmetrical as in the open-loop tests.
5. *"Drastic Brake and Steer"* – The open-loop test consisted of establishing a turn, applying full braking, and then releasing the brake at a predetermined instant to produce a sudden roll moment due to resumption of cornering friction. These tests were designed to produce large roll angles. The drivers attempt to duplicate these conditions by following a delineated curve, applying the brakes as hard as possible and then releasing the brakes when they feel that wheel lockup has occured.

The surprise tests present the drivers with unexpected obstacles or instructions, and they are free to maneuver as they see fit. However, they are instructed to attempt to obey instructions and not

collide with obstructions. Any number of obstructions or signs may present themselves on any given run, or nothing at all may happen. This minimizes driver anticipation of the required maneuver. The surprise conditions on a given run, in addition to no event at all, are:

1. *"Sudden Obstacle"* – Any one of five pylons may suddenly appear in the initial path. (The approach is striped to represent a two-lane highway.) In addition, the distance from the vehicle to the pylon at the time of actuation can be varied. This maneuver simulates an obstacle (stopped vehicle, pedestrian, etc.) that for any reason unexpectedly appears in the traffic lane.

2. *"Late Decision at Intersection"* – A fold-down sign at a marked right-angle intersection may pop up indicating detour right or left at the intersection. Again, the distance in which this maneuver is required can be varied from a control station. The sign may also be erected with no instructions (blank). This maneuver requires braking and then making a 90 degree turn, usually while still decelerating, and could be related to a driver making a late decision or being forced to turn when entering a 90 degree corner or "T" intersection at unsafe speeds.

3. *"Late Decision at Freeway Exit"* – This is brought about by a sign (similar to the intersection) which may suddenly appear indicating an exit left or right onto a constant radius curve. The time delay can be set to achieve limit maneuver requirements. Turning and then braking is usually required, and this condition could be the result of a late decision to exit or entering a curve at excessive speed.

4. *"Blind Corner"* – In this maneuver the driver is instructed by a sign to exit as before, and an obstacle may (or may not) appear which requires an avoidance maneuver while in the turn. The similarity to possible real-world situations is evident.

5. *"Sudden Lane Change"* – At one point the approach lane is flanked by adjacent lanes, and over each lane is a traffic signal indicating amber. As the vehicle approaches, one light may turn green and the other two red, indicating that the vehicle should take the lane marked green before reaching a given "gate." Of course, any light may be set green, they may all turn red or green, or nothing may happen. This elicits, on certain runs, a very sudden lane change that could have been caused by an obstacle or a forced return from a passing maneuver in real life.

Test Matrix

Four of the eight drivers for a vehicle do "replication" maneuvers and four do "surprises." Each driver is used in one session lasting approximately five hours. The order of the tests for each driver is different and is based on a counterbalanced design to minimize learning effects. If, for example, all drivers followed the same test order, they would on the average respond in the last maneuver differently (due to "learning") than they would have had it been the first one performed. This would bias the comparison between maneuvers. Also, in the surprise maneuvers, the runs in which nothing happens or a blank sign appears are inserted in the sequence at random to further minimize anticipation of the required maneuver.

Safety Considerations

The safety and protection of the volunteers take precedence over all other considerations. In order to insure this, some of the steps taken are:

1. The driver and on-board observer wear helmets and fire-resistant clothing, and they are secured by adequate lap and shoulder belts.

2. The test vehicles are maintained in top condition and are equipped with fire extinguishers, extrication tools, roll bars, antirollover outriggers, a racing-type fuel cell, and an outside emergency switch to isolate the electrical system.

3. In addition to fully equipped special teams, all test personnel are trained in first aid, fire, and rescue operations.

4. An ambulance and medical doctor are on standby near the test area during all tests.

5. Approval of each driver's personal physician is obtained for his participation. (No volunteer is accepted who has had a medical condition that could be aggravated by physical or mental stress.)

6. Extraneous personnel and vehicles are kept clear of the test area.

7. All "surprise" obstacles are made from lightweight materials.

Although some of the precautions (such as helmets and fire suits) undoubtedly affect driver performance to some extent, the drivers have invariably commented that they are oblivious to all external safety equipment or paraphernalia during the actual test runs. (A questionnaire eliciting subjective impressions is administered after the testing.)

It is felt that the drivers' performance, if anything, is enhanced by the conscious or subconscious perception of the safety precautions. All vehicle instrumentation is placed to rearward of the front seat of the vehicle so that it is not distracting.

Data Reduction

The analog input-response data on magnetic tape are processed on an analog to digital converter, and the digital data are simultaneously recorded on another magnetic tape. In addition, the analog data are displayed on a strip chart. The digitized data are fed into an IBM 360-65 computer which converts the transducer outputs to engineering units, computes sideslip angle and other quantities of interest, and prints the data at 0.05 second intervals during the maneuver period.

The data from the replication maneuvers are plotted on a Gerber programmed plotter in a format compatible to the previous comparative study.

Methods are now being devised to statistically analyze the data from the surprise maneuvers to determine which response parameters best characterize the driver-vehicle performance level.

PROJECT STATUS

As of May, 1974, twenty-four drivers have participated, and three of the four vehicles have been utilized. The contract with NHTSA is being modified to select and test an additional 32 drivers in order to provide a better statistical base. It is anticipated that all testing and analysis will be completed before the end of the 1974 calendar year.

PRELIMINARY RESULTS

The first comparison being made is between the data from the replication runs and those from the similar open-loop maneuvers of the previous study. The data from the open-loop tests were presented as a series of parameter plots. This comparison is intended to answer the following questions:

1. For similar open- and closed-loop control inputs, are the response outputs similar? (That is, do we have the same vehicle input-response characteristics and the same vehicle control limits that existed in the previous study?)
2. Can drivers achieve similar control input rates and magnitudes as those in the open-loop tests? (Are the open-loop tests realistic from a driver standpoint?)
3. Will typical drivers even in controlled situations achieve limit conditions for all the vehicles? (Are they willing to "push" one vehicle farther than another, and if so, can this be predicted based on open-loop data?)
4. Are open-loop response parameters related to qualitative driver-vehicle performance? (For example, suppose that in the open-loop tests one vehicle exhibited a higher limit lateral acceleration than another. Is this indicative that the drivers are willing and able to control that vehicle in a more severe turn than the other one?)

At this time it appears that the typical driver is hesitant to achieve limit maneuvering conditions and, based on sublimit experience, makes a conservative estimate of vehicle stability. However, except for the "drastic brake and steer" maneuver, most drivers are *capable* of achieving control inputs similar to or exceeding those of the open-loop tests. The "drastic brake and steer" maneuver requires precise timing to obtain maximum roll angle, and thus far no driver has been able to achieve the roll angles observed in the open-loop tests.

It does appear that the drivers in general will achieve different absolute levels of vehicle response in different vehicles. The analysis to look for indicators of this in the open-loop test data is incomplete.

The development of methodology to analyze the surprise data is incomplete. In these maneuvers we will be looking for relationships between closed-loop handling data and vehicle-driver performance in instinctive maneuvers in which the drivers do not have time to plan their actions. Also, the extent to which drivers will "push" the vehicles toward limit response will be determined. Again, preliminary qualitative evidence indicates that drivers perform better in some types of vehicles than in others.

The data to date seem to indicate that some response parameters are very poor indicators (due to nonreproducibility) of driver-vehicle capability in real-world situations. The results will be studied to determine if parameters that were not analyzed in the open-loop tests may be more meaningful.

Subjective driver opinion indicates that the participants "like" some cars more than others, and this seems to have no relationship to the type of vehicle they normally drive. Their performance in the tests has not been compared with their subjective evaluation of the vehicles at this time.

CONCLUSION

It is hoped that the results of this study will form the basis for refining open-loop test and data analysis methodology to permit closely controlled and reproducible tests to be conducted and the results objectively analyzed to give measures of vehicle handling performance that are meaningful in

real-world driving situations. This is not to say that closed-loop testing and subjective evaluation by experienced test drivers are not useful. Rather, the two approaches are complementary in providing vehicles for the driving public which may give that small margin of performance separating safety from tragedy for those unfortunate enough to be confronted with an emergency maneuvering situation.

ACKNOWLEDGMENTS

The research described herein is being conducted by the Texas Transportation Institute (Texas A&M University) under contract with the National Highway Traffic Safety Administration, U.S. Department of Transportation. Mr. Lynn Bradford, Mr. Svein Larsen, and Mr. Craig Keifer of the Office of Operating Systems Research, NHTSA, have provided close technical liaison with the sponsor; and their cooperation has been instrumental in the performance of this study.

The contents of this report reflect the views of the author, who is responsible for the accuracy of the statements presented herein. The contents do not necessarily reflect the official views or policies of the National Highway Traffic Safety Administration.

RICHARD H. KLEIN
Systems Technology, Inc.

INTRODUCTION

This paper presents objectives and approach to a U.S. Department of Transportation, National Highway Traffic Safety Administration, sponsored contract involving driver/vehicle response research.

The objective of the first phase of this program is to develop hypotheses on driver/vehicle interactions and time effects. This will be done by measuring driver response and performance properties for a range of driving task variables, including: vehicle directional response configurations; normal and emergency maneuvers; random and discrete disturbances; and task durations. Additional overall objectives are to:

- Identify maneuvers and performance measures which quantify and discriminate optimum driver/vehicle systems.
- Correlate subjective driver evaluations and measured system response and performance quantities.
- Determine best driver performance and optimum vehicle configurations for a range of maneuvers.
- Determine sensitivity of vehicle characteristics from the optimum regions with respect to safety-related performance.
- Resolve the effects of time span during a six hour driving period, and whether an optimum vehicle configuration remains the same (for a given subject) in the presence of said time effects.

The initial phase involves simulator and full-scale driving experiments with test drivers in vehicles with a range of directional (steering) handling properties. Subsequent phases of the program will be aimed at validating the hypotheses developed on the basis of the first phase data. This validation testing will use 16 nominal driver subjects performing the maneuvers found to be most selective by the Phase 1 test drivers. Differences in response and performance between drivers will also be determined. The final validation testing will develop connections between the results obtained for the basic test vehicles which are similar in appearance but different in response characteristics and several production vehicles from different manufacturers.

The balance of this paper discusses the vehicle handling configurations being used, the maneuver complex and test scenarios, and the processing and analysis of the response and performance data.

VEHICLE HANDLING CONFIGURATIONS

In this study, the vehicle configurations are defined in terms of their directional response characteristics for steering inputs. An elemental directional response variable of primary interest is yaw velocity (r). A convenient way to describe the yaw velocity response to nominal steering inputs (δ_{sw}) is via the system transfer function, r/δ_{sw}. This operational form allows the frequency and time response of the vehicle to be examined in various ways. For example, the two-degree-of-motion (suppressed roll mode) yaw velocity response to steering wheel input transfer function can be written in Laplace notation form as:

$$\frac{r}{\delta_{sw}} = G_s \frac{A_r(s + 1/T_r)}{[s^2 + 2\zeta_1\omega_1 s + \omega_1^2]}$$

where

G_s = Steering system gain
A_r = Yawing acceleration steering coefficient
$1/T_r$ = Inverse lead time constant of yaw velocity response to steer input
ζ_1 = Damping ratio of directional mode
ω_1 = Undamped natural frequency of directional mode
s = Laplace operator

The frequency and damping ratio of the denominator roots can be plotted on the complex plane in polar coordinates (ρ, θ) where $\rho = \omega_1$ and $\theta = \cos^{-1}\zeta$ or in rectangular coordinates (x, y) where $x = \zeta_1\omega_1$ and $y = \omega_1\sqrt{1 - \zeta_1^2}$. Figure 1 is an example of the complex plane representation of four different yaw velocities to steering wheel transfer functions. Normalized yaw velocity time responses to a step wheel input are also shown alongside the respective transfer function poles (X). Note that the higher frequency configurations respond more rapidly, and the amount of overshoot is proportional to the damping ratio and the value of the numerator yaw time constant T_r. Lower frequency configurations rise more slowly and may be thought of as having a compressed time scale. A third dimension, not shown in Figure 1, is the magnitude of the response which is scaled by the steering system gain. These three variables of the directional response, i.e., directional damping (ζ_1), directional frequency (ω_1), and steering ratio (G_s) are the vehicle variables being studied in this test program.

Several methods have been used to identify the vehicle dynamic characteristics experimentally. The

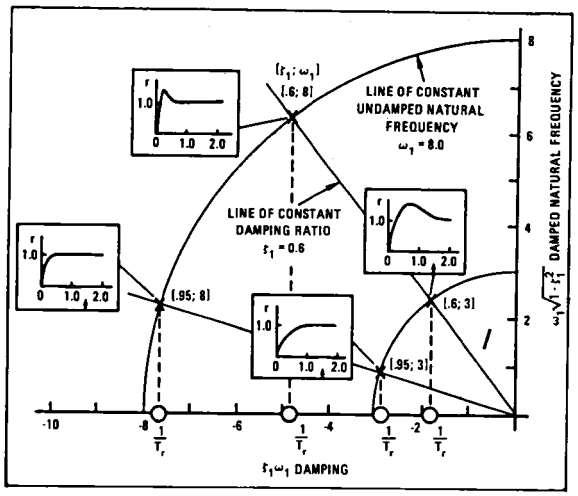


Figure 1. Complex Plane Representation of Vehicle Configurations

most successful and least time consuming method has been the time domain response matching method. Step steering wheel inputs are obtained with a hydraulic actuator in series with the steering wheel or by skillful, hand generated inputs. The resulting yaw velocity response to steering input is then fitted to the two-degree-of-freedom representation. Key parameters in the fitted response are the time of first steady-state value crossing (t_1), time to peak (t_p), time of second steady-state value crossing (t_2), the percent overshoot, and the steady-state value. These parameters are shown in Figure 2.

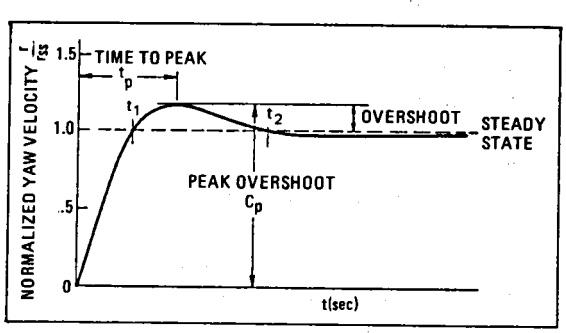


Figure 2. Yaw Velocity Indicial Response (Step Steer Input)

The test vehicles in the initial test phase are 1974 Chevrolet "Novas," of similar external appearance. They have been stiffened in roll, so that the main response to steer inputs is the lateral directional mode described above. The various vehicle dynamic configurations are obtained on the Novas by tire and weight distribution changes, and in one vehicle by feedbacks of various motion and control variables to an electrohydraulic servo system installed in series with the steering linkage. A functional illustration of this variable steering servo is shown in Figure 3. With this system, steering ratio can be varied experimentally from 9:1 to less than 50:1 and damping ratios can be varied from 0.5 or less to greater than unity. With tire and weight changes, the Nova's directional response natural frequency can be changed from a low of 2.25 rad/sec to a high of 7 rad/sec.

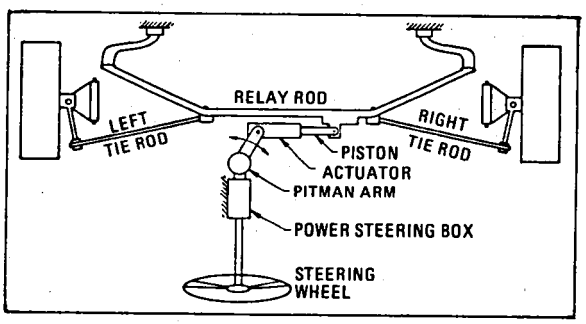


Figure 3. Variable Steering Servo Mechanics

MANEUVER COMPLEX

The experimental maneuver complex has been selected to ensure that all classes of steering functions are represented, that all modes of driver response are evoked, and to provide appropriate tie-in with other handling tests in common use.

Steering functions are those driver tasks related to:

1. Routine lane following, such as in normal highway driving.
2. Precision course control, such as required in narrow causeways, tunnels, etc., where a high degree of concentration is necessary.
3. Command changes, such as in a normal lane change maneuver or any maneuver in which the vehicle is commanded to change its lateral path in a particular way decided upon by the driver.
4. Regulation tasks, such as trying to maintain a desired lane position in the presence of wind gusts or an uneven roadway surface.
5. Unexpected tasks, such as a blown tire, an unexpected obstacle blocking the roadway, or an unexpected emergency lane change.

Each steering function has associated with it a particular mode of driver control. The three most common modes of driver control can be classified as:

- Compensatory
- Precognitive
- Dual mode

Compensatory behavior involves regulation against a disturbance, and the driver is applying a control action proportional to perceived error signal(s). As an

example, for vehicles with a constant level of yaw velocity to steering wheel response, i.e., $r/\delta_{sw} \doteq K$, the driver's steering wheel response can be proportional to the vehicle's heading error, plus a time delay. However, with degraded vehicle dynamics and the same perceptual situation, the driver must anticipate the heading error and hence make corrections proportional to the rate of heading change (i.e., the yaw velocity), as well.

Precognitive behavior involves programmed, open-loop control actions. For example, it is that exhibited when a driver becomes so skilled in a task that he could do it, in effect, with his eyes closed. A combination of compensatory and precognitive behavior has been called "dual mode." This type of behavior is most representative. For example, in a lane change maneuver the driver makes patterned control actions to change lanes; but small residual errors will be present and the driver must add compensatory control actions to place the vehicle in the desired final position. Hence, the precognitive and compensatory behaviors in dual mode control are exerted sequentially in time.

Table 1 lists the series of test maneuvers making up the nominal or nonemergency driving scenario used in the test program. Listed alongside each maneuver is the associated steering function and most likely driver control mode being evoked.

Table 1
NORMAL DRIVING MANEUVER COMPLEX

NAME OF MANEUVER	STEERING FUNCTION	DRIVER CONTROL MODE
Emergency Lane Change	Evasive command change	Precognitive (Dual Mode)
Highway Lane Regulation	Routine lane following	Compensatory
Random Input Regulation	Steady-state regulation (external input)	Compensatory
Step Input	Transient regulation (external input)	Compensatory
Double Lane Change	Normal command change	Precognitive (Dual Mode)
Precision Line Tracking	Precision course control	Compensatory
Unexpected Obstacle	Unexpected command change	Precognitive (Dual Mode)

A second set of maneuvers is used to represent a high workload level or emergency-like situations. These maneuvers include a high speed slalom and high speed double lane change such as used by driving schools, manufacturers, and researchers. A low speed random slalom also included is a variation of the conventional low speed slalom used in magazine road tests. The random slalom consists of randomly spaced pylons at 60, 120, 180, and 240 feet intervals. When run at a constant speed this maneuver exercises the vehicle's directional response capability at various lateral g levels.

DATA ANALYSIS

Measures of driver/vehicle response and performance are obtained in two basic ways. First is the direct measurement of closed-loop driver control parameters associated with the steering task. This takes the form of driver describing functions. Associated parameters are crossover and closed-loop natural frequencies, damping, phase margin, and effective time delay. Performance measures are also obtained, as noted below. In conjunction with direct driver measurements is the driver's subjective rating of the vehicle and difficulty of performing specific maneuvers.

The performance measures are those derived from the driver's control output and the vehicle response parameters. The variables measured and recorded on FM tape include the following:

- Steering wheel angle
- Steering wheel rate
- Brake pressure
- Throttle position
- Driver heart rate
- Yaw velocity
- Lateral acceleration
- Forward speed
- Roll angle
- Lateral placement (every 2 seconds for regulation task)

A data flow diagram depicting the on-line driver describing function measurement system and the tape recorded driver/vehicle response variables is shown in Figure 4.

After recording, the FM tapes are processed, off-line, to derive key closed- and open-loop performance measures such as rms values, peak values, exceedances of key variables from specified values, etc., for individual maneuvers. These can be correlated with the closed-loop driver control measurements in the data analysis stage. For instance, by bringing together these two forms of driver/vehicle response data, changes in driver behavior that compensate for changes in the vehicle dynamic configuration can be determined.

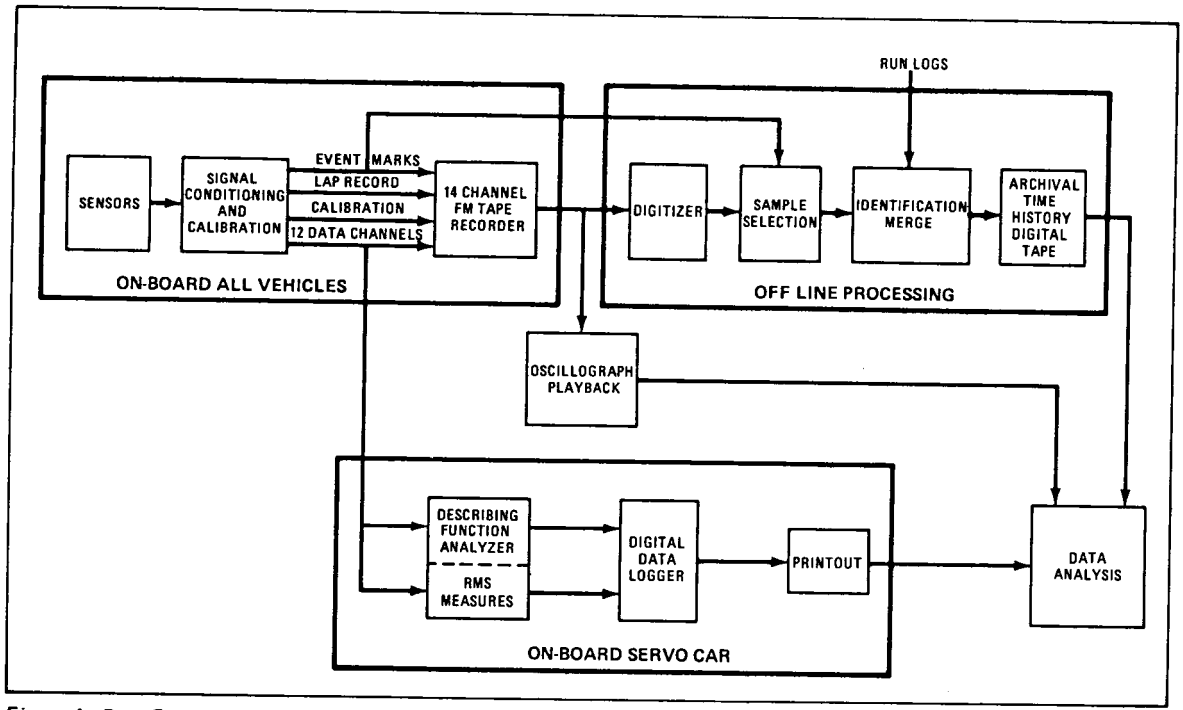


Figure 4. Data Flow Diagram

For these performance measures and data analysis procedures we hope to be able to determine optimum vehicle configurations that are:

- Within the "good" range of ratings for commandability, vehicle dynamics, and workload concentration in the ζ_1, ω_1, G_s space.
- Within the acceptable rating range of maximal "excess control capacity."

And that have:

- Adequate disturbance regulation performance, in terms of key performance measures being below a "knee" of the performance versus effective vehicle response time plane.
- Adequate discrete maneuver performance, in terms of minimal overshoot, sufficient rise (or settling) times, etc.
- Adequate maneuver performance margins with respect to maximum available control and side forces.

The results of the program are expected to be available later this year.

THE RELATION BETWEEN VEHICLE HANDLING AND ACCIDENT AVOIDANCE

KURT ENKE
Daimler-Benz AG

INTRODUCTION

In our research work for improving active safety, today we are facing a serious problem: the lack of knowledge on the relation between vehicle handling and accident avoidance.

The solution of this problem is of high importance for automobile manufacturers and governmental authorities. We should know which measures result in the best safety benefits and how to avoid negative safety effects resulting from regulations on handling (such as proposed in some countries). Therefore, an exhaustive discussion of this topic is necessary and many efforts have to be made, if we wish to make appreciable advances.

DESCRIPTION OF THE PROBLEM

Accident avoidance by vehicle handling is of a dual nature: vehicle handling as a scientific and technical problem is superimposed by the statistical problem of utilization of handling in actual emergency situations.

Vehicle handling, the performance of the closed-loop system: driver – vehicle – environment, is a very complex network of individual components with many interdependences.

We have to consider the quality level of the individual handling disciplines and the optimum compromise or balance of these levels. For our question, the balance is the more important subject.

The balance of the handling disciplines of modern cars is based on two factors: on technical reasons – for instance, one part of the suspension is of influence on several disciplines – and on the decisions of design and test engineers by subjective assessment of the order of importance and of the graduation of the individual disciplines.

The yardstick of these engineers is their experience from handling tests and from driving in actual traffic. Handling tests are being carried out on proving grounds covering the whole range of possible maneuvers up to and beyond the physical limits established by the friction between the tires and the road.

As a result of such testing, vehicles today perform equally well in the hands of an experienced driver and of the average user, under normal and even under extreme conditions.

There is only one point which could not and cannot be considered: all test runs are carried out on free roads or under normal traffic conditions and we assume that the results obtained will also be significant in real emergency situations. We are quite sure that this assumption is correct in a few cases regarding, for instance, directional stability during braking.

But in most cases we are not sure if our assumption is right or wrong. It might be that the reaction of skilled and unskilled drivers – in the stress situation shortly before a possible real accident – differs considerably from their reactions in normal situations. Therefore, it might be that the handling performance and the balance of the individual disciplines do not represent an optimum for drivers in real emergency situations.

Since accident avoidance, above all, relates to driver and vehicle behavior in emergency situations, we must admit that our knowledge of this item is rather poor. We need to take into account drivers' abilities in the emergency case during the development phase of handling qualities and handling balance, but the methods of accident investigation used today do not provide the necessary data.

Particularly, we have absolutely no information about:

1. The reaction of the driver in the extraordinary and unprecedented stress situation of foreseeing a possible, imminent accident.
2. Probability data of steering motions and pedal actuations before real accidents.

Our knowlege is, furthermore, incomplete on:

3. Probability data on types of maneuvers which are carried out before real accidents.

4. Probability data on weather and operating conditions during accidents.

Presently, available methods cover (4) completely; (3) only qualitatively; and (1) and (2) not at all.

After an accident, there is only a useless fraction of these data stored in the memory of the driver. Asking him, therefore, is a vain effort. Traces on the road, if ever visible, are of little additional value.

IMPORTANCE OF THE NEEDED DATA

The manufacturer needs this data in order to learn the particular handling disciplines which are most important for accident avoidance and where most benefit can be expected from improvement, maintaining the overall balance of the disciplines. The legislator needs this data in order to be able to direct particular attention to all conditions and matters connected with those types of accidents occurring most frequently.

The manufacturer must have the above data:

- In order to know the utilization ranking of the different handling disciplines in emergency situations, if he wishes to change the balance – in other words to improve individual handling disciplines to the disadvantage of others;
- In order to know which reactions of the driver have to be superimposed in handling tests;
- In order to know which information the driver utilizes to control his vehicle in emergency situations;
- In order to be able to adapt the control behavior of the vehicle to the ability of the driver in emergency situations;
- In order to find out which degree of yaw or sideslip motion can no longer be kept under control by drivers in emergency situations;
- In order to know the variations of driver reactions and the handling utilization in the emergency case in cars handling differently.

The legislator must have the missing data:

- In order to find the range of steering inputs for such tests, not exceeding the inputs generally used by humans;
- In order to come to a reasonable and effective selection of test procedures for rulemaking;
- In order to find the limits for the acceptable range in each test.

There is no doubt that we must avoid a disadvantageous change of the balance of handling disciplines by too high demands on individual disciplines which are not utilized by drivers to an adequate degree. Under all circumstances, we must avoid such a change of balance that a discipline, utilized never or rarely, has to be improved to the detriment of a frequently utilized discipline.

We may well say that in our work on active safety we have reached an obstacle, hard to overcome.

PROPOSAL FOR OBTAINING THE LACKING DATA

We have discussed the possibilities of simulators; studies of the reaction of drivers with different experience on proving grounds; and tests with artificial accident scenerios and recognized that they can only deliver a small part of the data wanted.

The only method to obtain this significant data, as we see it today, requires the installation of a great number of data recorders into a representative lot of vehicles. Perhaps, we may call them "drive recorders" in analogy to flight recorders in commercial airplanes.

These recorders would be able to tell us what happened in the seconds shortly before and during an accident.

Such a program of investigations, however, is so comprehensive and entails so many problems that it cannot be carried out by the automotive industry alone. This is partly due to the cost, while on the other side, problems of an organizational nature have to be solved. This includes, among others, the problems of:

- The inspection of the installation of the instrumentation;
- The particularly careful description of accidents in cases of vehicles with built-in recorders;
- The transportation of the recorders; and
- The central processing and evaluation of the gathered data.

In addition to the above, there still remain some legal problems to be solved. The only change of success to solve these problems is for all interested groups to work together and have governmental authorities assume a dominant role.

Although such recorders have already been presented, we have developed an independent idea on the scope of instrumentation which will require that the data to be gathered and recorded should include: steering wheel angle; brake fluid pressure; vehicle speed; longitudinal acceleration; lateral acceleration at the far front of the vehicle; lateral acceleration at the far rear of the vehicle; and throttle and gear lever positions.

Perhaps it might be useful to collect additional data with the same recorder which could further the insight into passive safety engineering. From the recordings the following data may be calculated: steering wheel velocity; brake pedal force; yaw acceleration; yaw velocity; and yaw angle.

It would be sufficient to limit the storage time to 30 seconds. The recorder should be switched off when both longitudinal and one of the lateral accelerations are zero. The optimization of the recorder requires a specialized engineering effort. Our approximate cost estimate is: £200 to £300 if 10,000 units are built.

Regardless of the type of recorder used, there is no doubt that the data obtained is of extremely high value for progress in safety and nothing else could serve as a substitute.

Regardless of the possibilities to put such a drive recorder program into immediate effect, we should start immediately to optimize methods and equipment for such an investigation.

MEASUREMENT OF TRANSIENT SLIP ANGLES OF TIRES -- A CONTRIBUTION OF EVALUATING SAFETY-RELEVANT DRIVING CONDITIONS

DR. RÜDIGER WEBER
Leiter der Reifenentwicklung
Porsche AG

INTRODUCTION

If driving maneuvers are to be examined from theoretical standpoints or if test track results are to be interpreted, then a knowledge of tire properties is very important. The dependence of lateral force on the sideslip angle is especially significant. The more exact and comprehensive the data which are available, the more precise will be the predictions about conditions relevant to driving safety as well as measures which can be taken to realize an improvement in the active safety.

Critical driving situations, during which quick steering maneuvers, yaw angel and sideslip angle changes occur, take place within a short time span. As a result of this fact, various velocities of slip angle changing occur. Therefore, the question arises: Can the curves of lateral force as a function of slip angle, which are almost always used and obtained by means of a tire test stand with constant or only gradually changing parameters, be applied under these conditions?

If one plots the harmonic excitation and the response of a single-degree-of-freedom system with damping, there is a phase loop that is influenced by the phase angle. It is known that a phase lag exists between the lateral force S and the slip angle α in the case of a transient measurement [2, 4, 6, 8, 9, 10]. Consequently, the function S(α) must have a phase lag loop when excited by a sine wave, for example.

Figure 1 contains two experimentally determined plots in which the lateral force as a function of slip angle is recorded. The slip angle was varied by a hydraulic pulsing machine programmed for a sawtooth function. A constant slip angle velocity $d\alpha/dt$ was achieved. The results are surprising: the solid curve represents an extraordinary small excitation frequency of 0.01 cps. In fact, it approaches that of the steady-state case. However, even at low frequencies of about 0.1 cps, discernible phase lag loops were recorded. The dotted curve in Figure 1 shows this result. Even though the change in the slip angle took place slowly at 1.6°/sec, there is no longer only a single function S(α). The phase lag of this particular loop is about 10°. It should be noted that this quantity can easily be one order of magnitude greater.

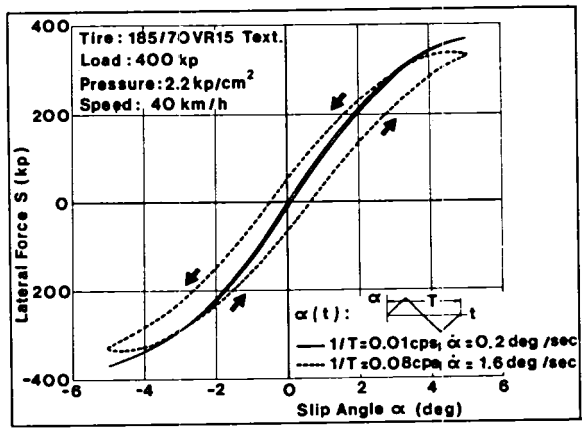


Figure 1

Figure 1 provides convincing evidence of the fact that for an exact consideration of a dynamic driving situation, a steady-state tire characteristic curve is not adequate. The dependence of the lateral force is very strongly influenced by the manner in which the slip angle is changed. Therefore, a single curve S(α) does not exist in actual driving situations.

In order to make more precise predictions of dynamic driving conditions possible, research must continue to examine the problem of transient slip angles. It is also necessary to examine a region with frequencies below 1 cps, because it is important in many driving maneuvers [11]. The previous considerations about the well-defined case of harmonic excitation of the slip angle around an average value of $\alpha_m = 0°$ should make a contribution in this area of research. Repeated lane changing or slalom driving are actual driving maneuvers, which are modelled by the test presented here.

TEST FACILITY

At the Porsche development center, there is a large tire testing machine, which has an inside drum with a diameter of 4.5 meters. The maximum speed is 250 km/h and the maximum wheel load is 1,000 kp. Figure 2 gives an impression of the complete test apparatus. All types of steady-state tire diagrams obtained on dry surface can be produced. Increased capability for wet surface testing is projected. In addition, measurements of transient slip angles are possible. The "Hydropuls" machine, which is visible in Figure 3, allows the use of various time dependent functions of the slip angle. The rolling wheel may be pivoted about its vertical axis for 0° camber. The slip angle vibration is thereby not only achieved by a

lateral displacement of the rolling wheel [8], but the rest position, about which the slip angle is varied, can be selected at any angle between $a_m = 0°$ to $a_m = 20°$. The maximum allowable amplitude of the pulse machine is $a_0 = 5°$ at 2 cps and falls off hyperbolically to $a_0 = 0.1°$ at 20 cps. Forces and moments on the tire are measured by a 5-component transducer and recorded by various types of equipment. All of the results shown in this paper have been obtained with the described apparatus [3, 12].

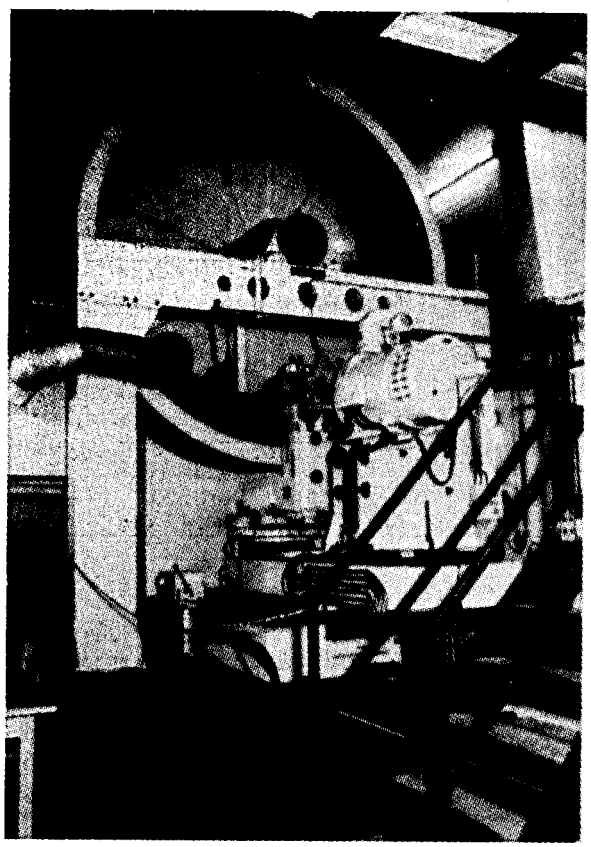


Figure 2

A plot of transient slip angle measurement is shown in Figure 4. The driving function $a = a_0 \sin \omega t$ is shown, as well as the rest positioned lateral force displaced by the phase angle $S = S^* + S_0 \sin(\omega t + \zeta)$. All the necessary values are shown. The lateral force constant S*, which is superimposed on the time dependent function, is in principle similar to – but not exactly of the same magnitude as – the lateral force which can be measured on a rolling wheel with no slip angle.

The lateral force vibration is not strictly harmonic. Therefore, the phase lag angle must be calculated from an average value of the lag times $t\zeta_1$ to $t\zeta_4$ at the zero and maximum values.

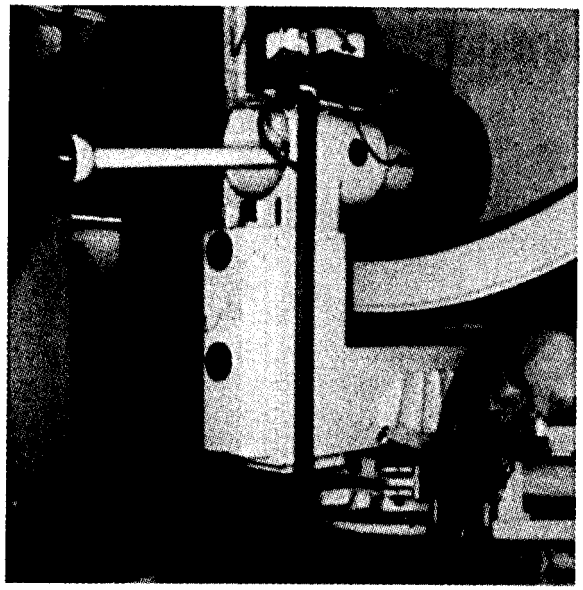


Figure 3

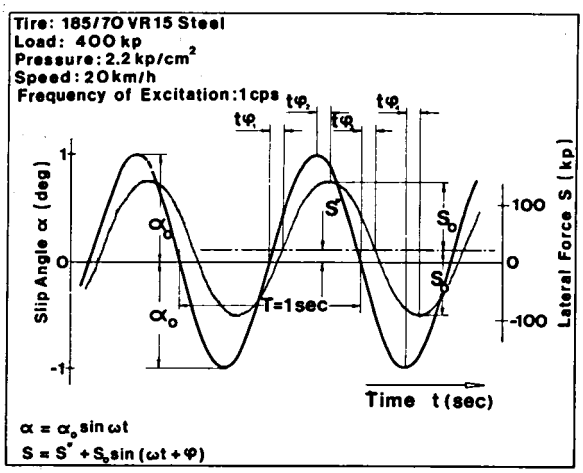


Figure 4

TEST RESULTS

As described, the slip angle of a rolling wheel was varied according to a time dependent sine function. The pivot axis was the vertical axis of the wheel; the camber was set to zero. The slip angle was varied symmetrically about the rest position of $a_m = 0°$. The slip angle amplitude a_0 and the excitation frequency f were varied. Figure 4 presents the results of a typical test with given parameters. The determination of the lateral force amplitude S_0 and the phase lag angle ζ were arrived at in the described manner.

The region which was possible to examine was limited in two ways: the slip angle amplitudes, which are permissible on the test machine, are reduced as the frequency is increased; on the other hand, at very slow tire velocities the lateral force vibration is so

small that, especially at higher frequencies and small amplitudes a_0, it is not possible to evaluate the data. The dotted lines in Figures 5 and 8 fall in this poorly measurable area, where extrapolation must be employed.

The precision of the measurements is enchanced when the tires, which are used, exhibit a small lateral force value S* and show good uniformity.

Lateral Force Amplitude S_0

The amplitude S_0 is a measure of the available lateral force for a given change in slip angle. Figures 5 and 6 show the basic relationships. Figure 5 depicts the dependence of S_0 on the frequency f for various speeds. At a frequency f = 0, the steady-state values are shown; slip angle amplitude a_0 and slip angle a in a conventional plot S(a) are the same in this case. The solid curve in Figure 1, in which there is hardly any phase lag loop, serves as an illustration. When f = 0, S_0 increases as the speed increases. The explanation for this fact lies in the manner in which the side force increases as a function of the slip angle for the static case. At small slip angles the lateral force, as well as the cornering stiffness, increases as the velocity increases; whereas at large slip angles, the lateral force decreases as the velocity increases. The latest measurements on the "Calspan" test machine [1] also show this tendency.

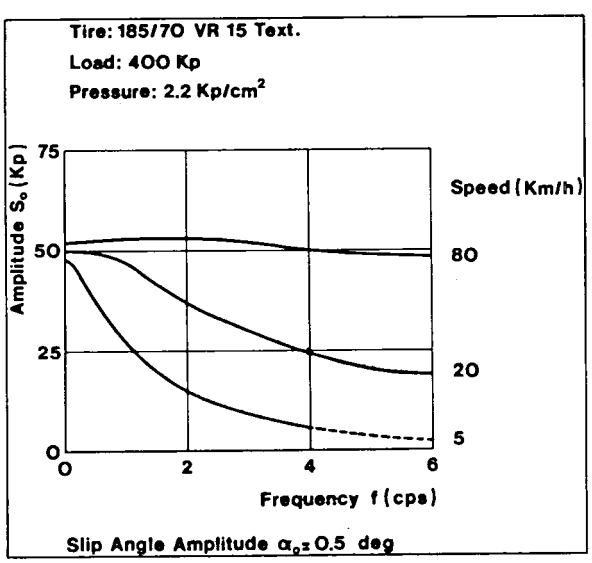


Figure 5

At low velocities the influence of the frequency is the greatest. The amplitude S_0 decreases rapidly as the frequency increases. As the velocity increases, the amplitude is less influenced by the frequency. At very high speeds this tendency reverses, and the amplitude increases slightly as the frequency increases.

In Figure 6 there is an example with a different tire size and excitation amplitude of two driving frequencies, which shows how the amplitude varies as a function of velocity. The previous observations regarding Figure 5 are further qualified in Figure 6. Moreover, it is clear that small speeds influence the lateral force amplitude most. Very small velocities have an especially large effect. However, from a safety standpoint they are probably not of great importance.

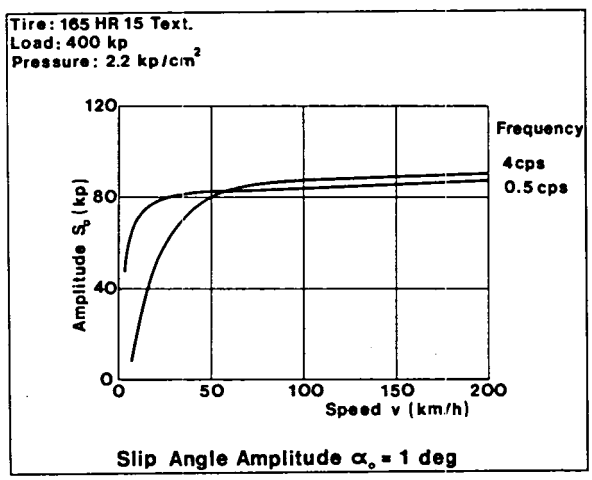


Figure 6

A three-dimensional presentation allows the amplitude to be shown simultaneously as a function of the excitation frequency and the tire speed. Figure 7 depicts such a case. The amplitude S_0 is normalized by dividing by the value of the lateral force S_{st} obtained by steady-state measurement at a_0. The

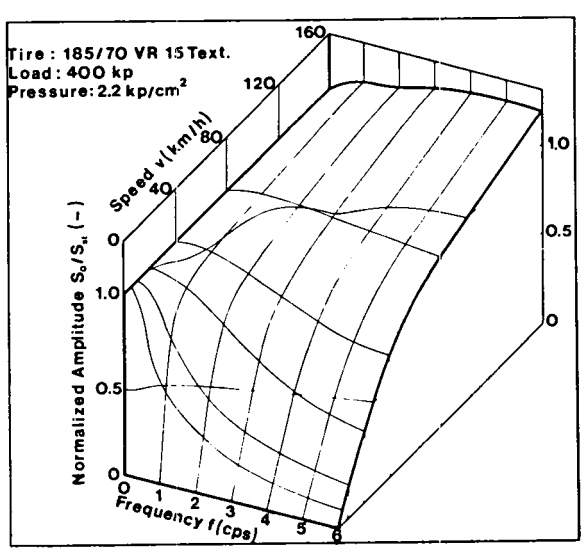


Figure 7

range of test values of a_o varied between $a = 0.5°$ to $a_o = 5°$. The magnitude of a_o influences S_o as a function of frequency and speed. The experiments proved that in spite of this influence a uniform diagram for the examined a_o-range is possible, if the ratio S_o/S_{st} – the magnification factor – instead of S_o is used as the ordinate. The surface of the three-dimensional diagram in Figure 7 contains all of the magnification factors which occurred with the shown parameters and with variation of speed, frequency, and excitation amplitude.

Phase Angle

In Figure 8 the average phase angle ζ is shown as a function of the frequency for three different velocities. The value of ζ decreases more radically at low speeds when the frequency decreases. Even at low frequencies large phase angles were recorded. As previously demonstrated from Figure 1, the function $S(a)$ is not single-valued if phase lag exists.

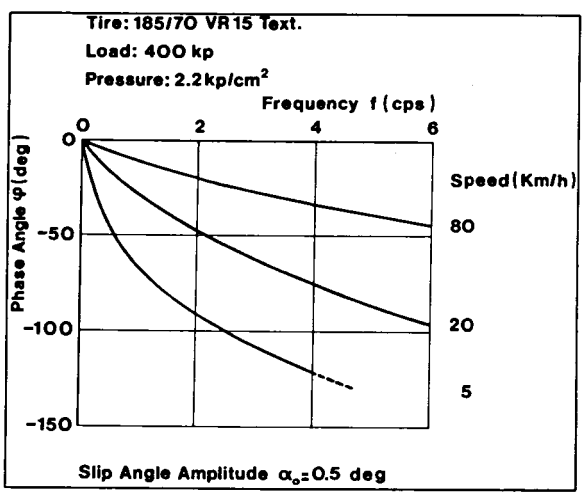


Figure 8

Figure 9 shows the phase angle as a function of velocity for two frequencies. The tire size and slip angle amplitude are not identical with those tested in Figure 8. It is clear that low speeds have a special influence on the phase angle. It has not been possible to determine the approximate phase angle at zero velocity; however, this question is of more interest from a theory of friction standpoint [5, 7] than from an actual dynamic driving state.

In another three-dimensional plot, the effect of frequency and velocity on the phase angle is shown in Figure 10. This diagram also includes slip angle amplitude variation up to $a_o = 5°$ with adequate accuracy; ζ is almost independent from a_o in the examined range.

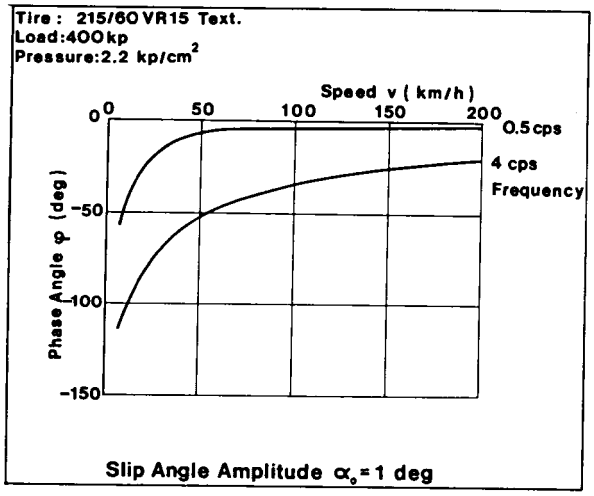


Figure 9

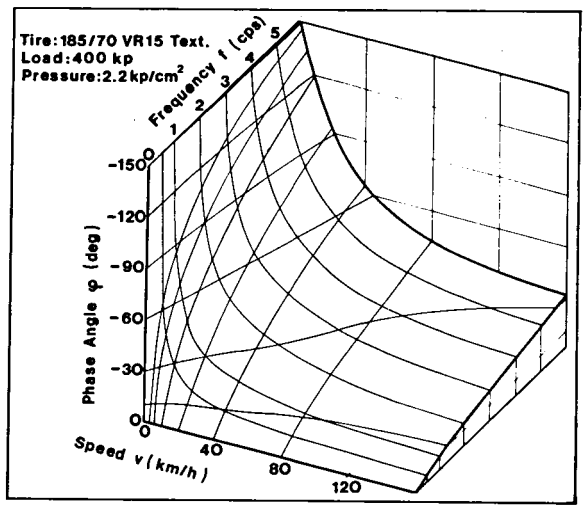


Figure 10

The results presented here were compared with the theoretical and experimental results of Pacejka [6]. The agreement was good.

PROSPECTS FOR ADDITIONAL WORK

The experiments which have been presented are a first portion of a larger series which shall be continued at the Porsche Co. One of the areas to be examined more closely will be the frequency range below 1 cps. Previous literature suggest that this range is important for many driving conditions and maneuvers [11]. In addition, it is worthwhile to choose a slip angle vibration $a(t) = a_m + a_o \sin \omega t$ with a middle value $a_m \neq 0$. Thereby, it will be possible to examine the nonlinear range of the lateral tire characteristic.

Steering inputs are in general not strictly harmonic vibrations. Hence, different time-varied functions $a(t)$

must also be examined. Of special interest is a step function.

Finally, it is necessary to find the correlation between measurements obtained on a tire testing machine and from test driving. A comprehensive experimental program in this direction has already been initiated at the Porsche Co.

ACKNOWLEDGMENTS

The author would like to express his thanks to Mr. H.G. Persch and Mr. K. Köfler for their help in finishing measurements and Mr. J. Stockley for his work in translating this paper.

REFERENCES

1. Calspan Co., TIRF, Tire Research Facility.
2. Clark, S.K., "Mechanics of Pneumatic Tires," NBS Monograph 122, 1971.
3. Bröder, K.; Haardt, H.; and Ulrich, P., "Reifenprüfstand mit innerer und äusserer Fahrbahn," Automobiltechnische Zeitung (ATZ) 75 (1973) 2, S. 39.
4. Fritz, G., and Fritz, W., "Das Verhalten von PKW-Reifen bei schnellen Schräglaufänderungen," Automobil-Industrie (AI) 4 (1973), S. 43.
5. Geyer, W., "Beitrag zur Gummireibung auf trockenen und insbesondere nassen Oberflächen," Dissertation, München, 1971.
6. Pacejka, H.B., "The Wheel Shimmy Phenomenon," Proefschrift ter verkrijing van de graad van doctor, Delft, 1966.
7. Rieger, H., "Experimentelle und theoretische Untersuchungen zur Gummireibung in einem grossen Geschwindigkeits- und Temperaturbereich unter Berücksichtigung der Reibungswärme," Dissertation, München, 1968.
8. Schubert, K., "Seitenkräfte an rollenden Luftreifen bei periodischer Felgenquerbewegung," Dissertation, Hannover, 1972.
9. Strackerjahn, B., "Fahrversuche und Berechnungen zur Kurshaltung von Personenkraftwagen," Dissertation, Braunschweig, 1973.
10. Tambrun, J., "Presentation du banc d'essais de pneumatiques Renault-Peugeot," Fisita 1974 (Paris), C-1-14.
11. Waldmann, D., "Korrelation Fahrer – Lenkung – Fahrzeug (Theorie und Versuche)," Fisita 1974 (Paris), C-1-16.
12. Wolff, B., and Stehle, H., "Das Porsche Entwicklungszentrum," Automobiltechnische Zeitung (ATZ) 75 (1973) 2, S. 33.

DR. HANSJÖRG MANGER
Robert Bosch GMBH

INTRODUCTION

Further development of motor vehicles occurs today with the following objectives: safety, emission control, and economy.

Safety in traffic means trouble-free function in common with driver, vehicle, and environment. It can be increased by the following measures:

- Improvement of driver and traffic education.
- Safer vehicles, i.e., vehicles with higher active safety for reducing the risk of accidents and higher passive safety for reducing the consequences of accidents.
- Improvement of road conditions and better information for the driver on traffic and environmental conditions.

In the following text some examples will be shown of the possibilities which exist to increase active and passive safety of vehicles by means of devices and systems developed by the motor vehicle equipment industry. In this, considerable improvement can frequently be made with relatively low expediture.

DEVICES TO INCREASE ACTIVE SAFETY

"See and Be Seen"

By far the most important information in driving a vehicle reaches the driver visually. Any avoidable interference to vision should therefore be removed.

While it is sufficient during the day to make sure that the windshield and rear window are clean, further measures must be taken for night driving. The general introduction of H4 headlamps is an essential step towards better illumination of the road. Headlamps can however only yield their full light when the headlamp lenses are free from dirt. Even a small amount of contamination reduces by half the light emitted by a headlamp. Heavy contamination, e.g., by snow slush, can reduce the light emitted by 90 percent. Good illumination of the road, independently of the weather conditions, is guaranteed by the use of a headlamp cleaner, which is already legally required in Sweden.

In many accidents the people involved give "dazzling by an oncoming vehicle" as the cause of the accident. Even with correctly adjusted headlamps, dazzling of the oncoming traffic often cannot be avoided because the light-dark-border of the dipped light varies with the loading condition. The principle of a hydraulic headlamp levelling system which maintains the light-dark-border constant within narrow tolerance limits, independently of the loading condition, is shown in Figure 1.

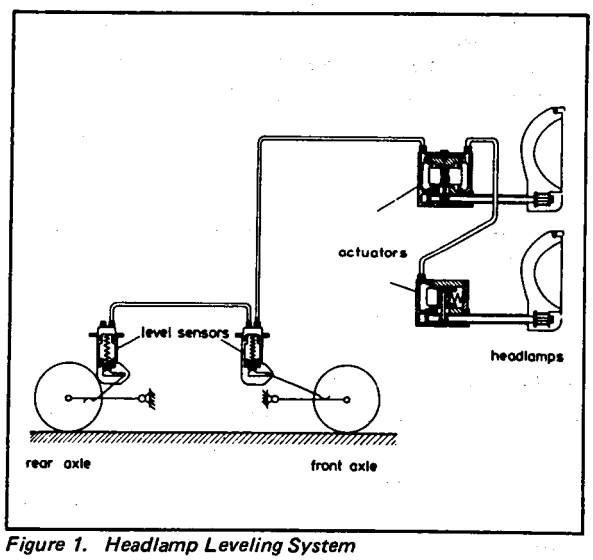


Figure 1. Headlamp Leveling System

Level sensors on the front and rear axles sense the position of the vehicle body relative to the road. Headlamp adjustment is provided by actuators on the headlamps. Level sensors and adjusting elements are connected by hydraulic lines. With uniform loading of both axles there is no adjustment; but with nonuniform loading the inclination of the vehicle is compensated by adjustment of the headlamps. Besides the hydraulic system shown, pneumatic and electronic systems are also possible.

Complete avoidance of dazzling would be possible by the general introduction of headlamps with polarized light. The solutions suggested up to now with polarization foils have an efficiency of only about 30 percent, i.e., only about a third of the light emitted from the headlamp is available to illuminate the road.

In order to achieve the same visual conditions as with normal headlamps and to compensate for the attenuation in the analyzer, the light output, and the concomitant electrical power, requirement must be increased approximately 5 fold.

With the solution developed by Bosch, whose principle is shown in Figure 2, an efficiency of over 60 percent could be achieved by using polarization on the reflecting layers as well as in the foils. Thus, the prohibitive head dissipation of the other systems can also be avoided.

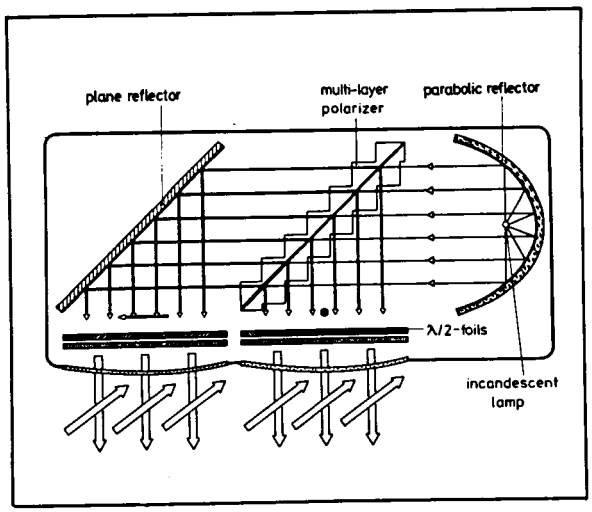


Figure 2. Polarized Headlamp – Principle

The light emitted by the incandescent lamp is first resolved into two linearly polarized beams by means of the multilayer polarizer. The beam passing through the multilayer polarizer is deflected by a plane mirror in the same direction as the beam reflected from the multilayer polarizer.

The two parallel beams are polarized at right angles to one another. To achieve uniform polarization suitably oriented birebringent plastic foils must be used to rotate the direction of polarization by 45° in each case.

Figure 3 shows a more advanced design on this principle for use in motor vehicles. It has the

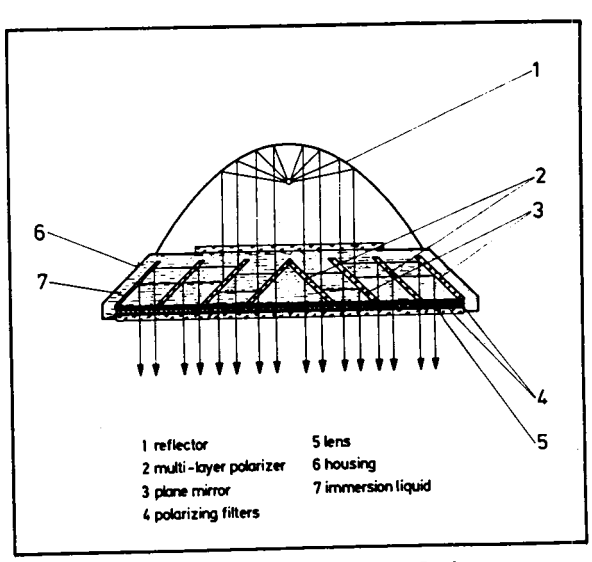


Figure 3. Polarized Headlamp – Compact Design

particular advantage of being compact. The multilayer polarizer and the deflecting mirror are each divided into four individual elements and positioned symmetrically with respect to the reflector axis. These elements are placed in an immersion liquid in order to achieve suitable optical conditions.

Figure 4 shows the meeting of two oncoming vehicles to illustrate the advantages of polarized light in comparison with normal H4 light. One can see clearly that the lighting conditions with polarized light are considerably better. In particular it is possible to recognize obstructions such as parked cars, pedestrians, or cyclists on the right hand edge of the road at a distance at which they would still be invisible with conventional light. With the introduction of polarized light the traffic safety could be considerably increased.

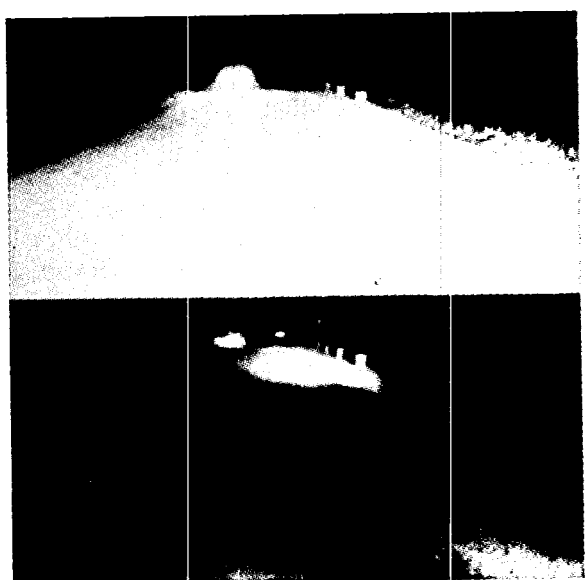


Figure 4. Comparison of Headlamp Systems – (a) H4-H4; (b) Pola-Pola.

In connection with "see," it must be pointed out that for road traffic safety to "be seen" is just as important. The motor vehicle equipment industry supplies for this purpose a variety of tail lamps, flasher lamps, stop lamps, fog tail lamps and warning flasher lamps.

Brakes

The braking system is of central importance for the active safety of a motor vehicle. The development of brakes in the past has led to the present, technically highly developed braking systems. These are characterized by high braking forces, uniform distribution of braking force, low fading, high reliability, and low pedal forces.

However, despite these advantages, the braking process always depends principally on the driver and therefore on his skill and experience. Emergency braking under panic conditions leads (as a rule) to

locked wheels, which result in loss of stability, steerability, and generally a longer braking distance. This can be avoided by automatic antiskid systems.

Criteria for evaluating the quality of an automatic antiskid system are:

Stability during straight-on braking with homogeneous coefficient of friction.	Caused by lateral guidance of the rear axle.
Stability during straight-on braking with inhomogeneous or asymmetric coefficient of friction.	Caused by lateral guidance of the rear axle.
Stability and steerability during braking in cornering.	Caused by lateral guidance of the rear axle and by lateral guidance of the front axle (steerability).
Reduction of braking distance.	Caused by optimal utilization of the coefficient of friction road surface/tire present at each wheel.

The stability must be placed first in this evaluation, since even with straight-on braking, loss in stability leads to a driving condition which can no longer be controlled by the driver, even with medium speeds.

Steerability gives the driver the possibility of carrying out a swerving maneuver or maintaining the original lane curve when braking while cornering.

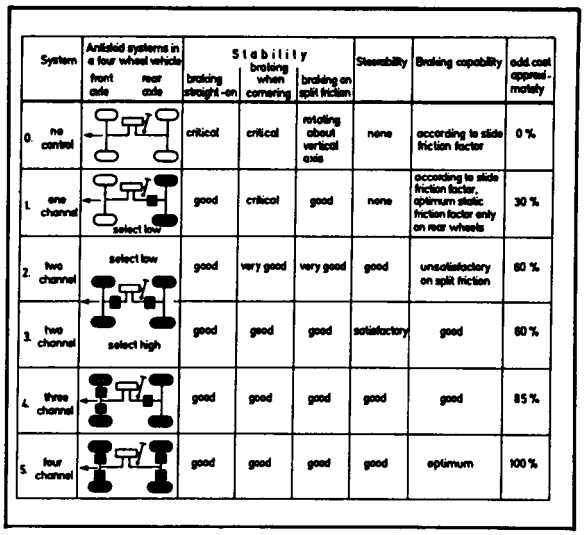
The length of braking distance, and the possibility of shortening it, depend on the road conditions and the antiskid system used.

A shortening of the braking distance is advantageous, but in comparison to maintaining stability and steerability, is not of prime importance.

In Figure 5 different antiskid systems for 2-axle vehicles are listed and are roughly evaluated according to their characteristics and the required cost.

As system 0, a vehicle without an antiskid system is given in comparison. System 1 shows a one-channel control in which both wheels of the rear axle are controlled together and the front wheels are not controlled.

In systems 2 and 3, the front wheels are additionally controlled, not independently, but together. Systems 2 and 3 differ in system design – in one system the braking pressure on each axle is given by the wheel with the lower coefficient of friction (select low) and in the other system by the wheel



System	Antiskid systems in a four wheel vehicle: front axle / rear axle	Stability: braking straight-on	Stability: braking when cornering	Stability: braking on split friction	Steerability	Braking capability	add. cost approximately
0. no control		critical	critical	rotating about vertical axis	none	according to slide friction factor	0 %
1. one channel	select low	good	critical	good	none	according to slide friction factor, optimum static friction factor only on rear wheels	30 %
2. two channel	select low	good	very good	very good	good	unsatisfactory on split friction	60 %
3. two channel	select high	good	good	good	satisfactory	good	60 %
4. three channel		good	good	good	good	good	85 %
5. four channel		good	good	good	good	optimum	100 %

Figure 5. Comparison of Antiskid Systems

with the higher coefficient of friction (select high). System 4 is a 3-channel system in which the rear wheels are controlled together and the front wheels individually. System 5 is a 4-channel system where all wheels are controlled individually.

The stability during straight-on braking, stability during braking in cornering, stability with asymmetric coefficient of friction, steerability, braking capability, and approximate additional cost for these systems were chosen as evaluation criteria. The additional costs are all given relative to system 5, the 4-channel system.

For the vehicle without an antiskid system, stability during straight-on braking and braking in cornering is critical and depends very strongly on the road conditions and the skill of the driver. The vehicle rotates about its vertical axis. The vehicle is also no longer steerable with full brake application.

If Figure 5 is read from top to bottom, it is realized that, in general, the characteristics improve and reach their optimal values with system 5, with 4-wheel individual control. However, it is also realized that for systems of medium cost (e.g., system 3), a considerable improvement in stability and satisfactory steerability have already been achieved.

Four-channel automatic antiskid systems with individual wheel control can therefore make a large contribution to traffic safety, but simpler designs can also contribute considerably to safety.

Relieving the Driver

A very high percentage of all traffic accidents with injuries to people are caused by the behavior of the vehicle driver. It is well known that fatigue plays an important role in this. The motor vehicle equipment industry can relieve the driver by a number of devices

and thus counteract premature fatigue. These devices, among others are:

- Power steering
- Power brakes
- Automatic transmission
- Speed control for maintaining constant speed
- Air-conditioning systems to achieve a pleasant climate in the vehicle
- Adjustable seating.

A further contribution to relieving the driver would be sidewind compensation with which the behavior of the vehicle in a sidewind can be improved by electronic means. In this, the lateral acceleration of the vehicle is measured by an accelerometer and the unwanted side deviation is reduced by actuating a steering motor. Such a device not only relieves the driver but also improves the handling, because the electronic control systems acts faster than the driver can.

Figure 6 shows the side deviation of a vehicle with and without sidewind compensation for a constant steering wheel position. The lateral deviation is reduced to about a third by the sidewind compensation.

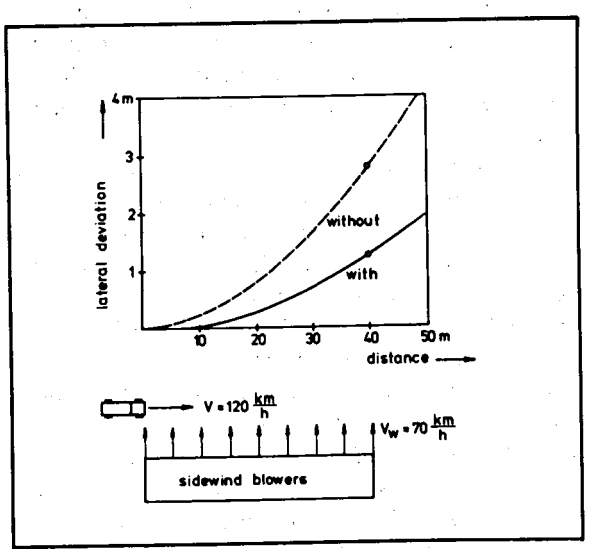


Figure 6. Sidewind Compensation

A further device to increase traffic safety is the distance warning system on which research work is being done at different places. Figure 7 shows the basic principle of such a collision avoidance system. By measuring the differential speed and the distance from an object in front, the critical distance is calculated and the driver is warned by an acoustical or visual signal. In a later development stage this signal could be used to trigger an automatic braking process.

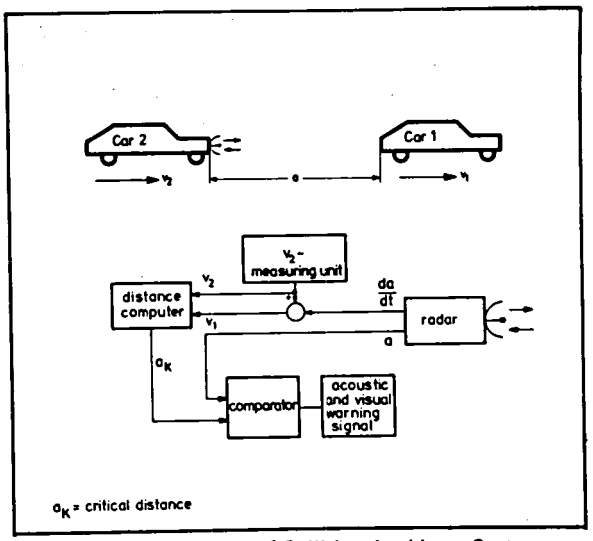


Figure 7. Basic Principle of Collision Avoidance System

A number of problems remain unsolved with these systems, in particular the distinction of relevant and irrelevant echoes. If these problems could be solved, a decisive contribution to increasing traffic safety could be made.

Since a very large percentage of all accidents can be traced back to human error, it would be practical, for reasons of safety, to carry out as many as possible of the driver's operations by mechanical or electronic means. The last consequence would be automatic vehicle guidance (e.g., along guidance cables) in which all functions of the driver would be performed by mechanical or electronic equipment.

Monitoring of Operational Safety of the Vehicle

The technical condition of the vehicle plays an important role in traffic safety. In practice, however, one frequently meets vehicles which show deficiencies.

While some deficiencies (of the lighting system) are still relatively simple to recognize, this is not the case for other assemblies (brakes). Here the motor vehicle equipment industry can add a contribution to safety by a variety of monitoring devices. With these devices it is possible to recognize deficiencies both before starting and during a journey. Examples are: oil level; cooling water level; brake fluid level; flushing fluid level; engine temperature; oil pressure; brake pressure; tire inflation pressure; brake lining thickness; battery charging condition; lamps; and door and hood locks.

Figure 8 shows, as an example, a sensor for monitoring the brake lining thickness on disc brakes.

A ceramic substrate with six thick film resistances connected in parallel is positioned in a clamp between the brake linings. Due to lining wear, a stepped shear

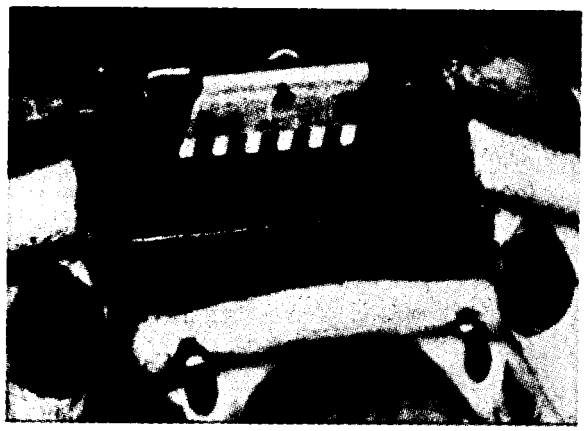


Figure 8. Brake Lining Monitoring Device

sheet breaks off the teeth one by one. The total resistance thereby increases stepwise and is a measure of the brake lining thickness.

Figure 9 shows the principle of a light monitoring circuit as a further example. The light lead is led through a magnetic core. In addition, two more wires, the primary and secondary leads, are led through.

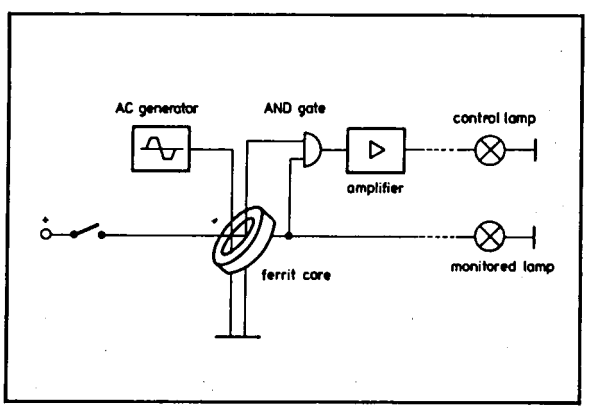


Figure 9. Lamp Monitoring Circuit

The light current saturates the core so that the coupling of the test pulses from the primary to the secondary lead is interrupted. When there is no current in the lamp, an alternating voltage is induced in the read wire. In order to be able to distinguish whether the lamp is defective or the lamp switch is off, the read voltage and supply voltage of the lamp are connected in an AND-gate. An alternating voltage exists at the output of the AND-gate only when the lamp switch is on and a lamp is defective.

Since distraction of the driver affects traffic safety, suitable design of the display instruments must ensure that the driver only receives information which is important to him. This can be provided by signal processing which only gives signals in the case that specified limits of important quantities are exceeded. Quantities which have no dangerous consequences can be displayed at the start of a journey or when the vehicle is halted.

Driver Information

An important means of averting damage to vehicle and passengers is the avoidance of accidents by decentralizing the traffic and informing the driver at the right time of the traffic situation in front of him and on traffic obstacles caused by the weather. Today's road traffic participants are only informed on the traffic situation and the road conditions immediately in front of them.

As a first remedy, a traffic warning broadcast is introduced in some states, giving the vehicle driver a rough picture of the traffic situation.

Figure 10 shows a block circuit diagram of such a unit which is designed as an additional device for an ordinary car radio. The traffic warning broadcast decoder receives the traffic broadcasts from special transmitters and automatically switches on the car radio or interrupts another current program. A disadvantage is the fact that the broadcast can only apply to large areas and only take place when the traffic obstructions have already occurred.

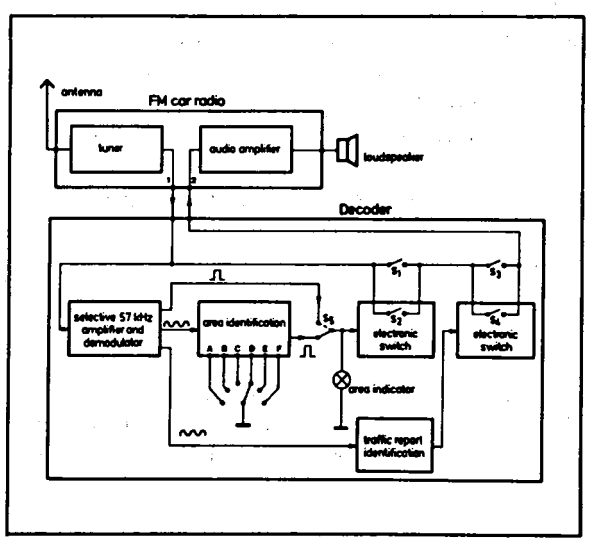


Figure 10. Traffic Broadcast Receiver

These disadvantages would not occur with a traffic guidance system in which the most favorable route for each vehicle is calculated according to the actual traffic situation by means of electronic devices partly in the vehicle, on the road, and in a central control station. It could be transmitted to the drivers individually, together with other important information. By giving a destination address to the central guidance station, potential traffic jams could be recognized in advance and could be averted by

corresponding diversion measures. An increase in traffic safety would then be simultaneously linked with better utilization of the road network.

Figure 11 schematically shows the necessary equipment in the road network.

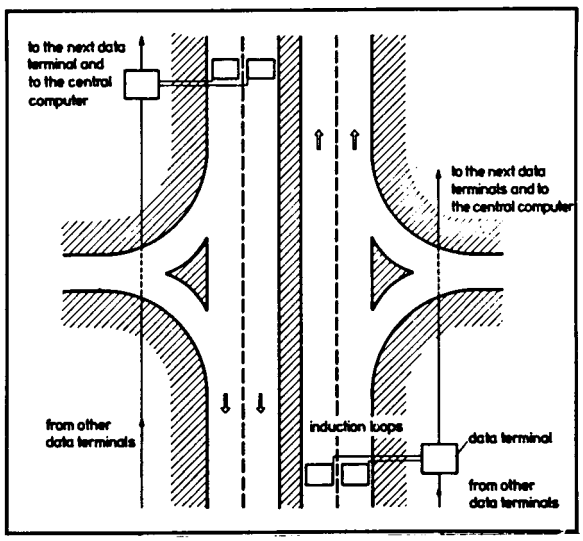


Figure 11. Traffic Guidance System

Figure 12 shows an actual display model in the vehicle. The driver specifies the required destination by means of a keyboard and receives advice on the most favorable route for him at each junction. Furthermore, the driver can receive other information – ice, traffic jams, fog, maximum speed, etc. The necessary expense is low and comparable with that of a car radio. However, to introduce such systems, the problem of financing the road installation has to be solved.

DEVICES FOR PASSIVE SAFETY

While the devices mentioned up to now are designed to avoid accidents and to increase active safety, the motor vehicle equipment industry also

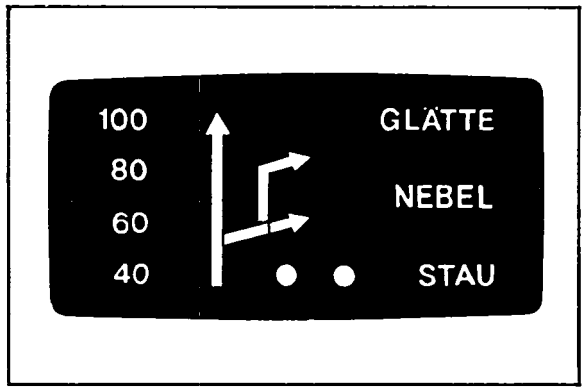


Figure 12. Traffic Guidance System Vehicle Display

supplies devices for passive safety. Among others, there are safety belt systems, airbag systems, systems for switching off fuel and power supply in emergencies, and many measures for making the vehicle interior safe.

DEVICES FOR SUPERVISING THE DRIVER

The safest and most technically sound vehicle is no longer safe if treated carelessly by the driver or if the installed safety measures are ignored. The motor vehicle equipment industry also supplies corresponding devices for this task, such as the safety belt interlock system to compel the wearing of seat belts and an ability tester to prevent drivers from using the vehicle when they are unfit because of alcohol, drugs, or fatigue.

CONCLUSION

The above mentioned devices and systems show that, besides the motor vehicle manufacturer, the motor vehicle equipment industry can also make considerable contributions to the active and passive safety of the vehicle. It is essential that, by means of many of these devices, a considerable improvement in safety can be achieved with relatively low expenditure.

IMPROVING DIRECTIONAL STABILITY UNDER BRAKING

DETLEF BANHOLZER
Audi NSU Auto-Union AG

INTRODUCTION

There is no doubt that if improvements can be effected in the area of braking stability, this would represent a very worthwhile contribution to the "active safety" of the vehicle.

Just as landing and take-off are the most dangerous maneuvers for an aircraft, so with an automobile braking maneuvers still involve a high degree of risk and thus represent the most frequent cause of accidents.

The extent to which the vehicle maintains the desired course when braking often determines whether or not an accident can be avoided. Brakes which pull to one side are a frequent cause of serious accidents.

Extended brake testing under many different conditions – also with anti-lock systems – has shown up the weaknesses of conventional brake systems and led to a requirement for improved braking stability:

1. The vehicle should be able to brake from either straight ahead motion or cornering motion without pulling to one side, i.e. without deviating from the desired course, even if the coefficient of friction of the road surface varies from one side to the other and even if there are differing friction coefficients in the brakes themselves: this should be possible both with the brake system intact and when one circuit is out of action.
2. The understeer/oversteer behavior of the vehicle should remain unaltered, or change only slightly, during braking even if one brake circuit is out of action.
3. If at all possible, the vehicle should still retain steering control when one brake circuit is cut off and the braked wheels lock.

Two factors threaten to reduce the directional stability of the vehicle during braking: unequal braking forces on one side or the other, and locked wheels or axles (both front wheels or both rear wheels).

Nowadays it is possible to go a long way towards preventing the wheels from locking by the use of expensive anti-lock systems. But uneven braking forces will always occur even when anti-lock systems are used, since it is to be expected that the front wheels will in all cases be controlled independently of each other.

One of the most urgent requirements for increasing active safety must therefore be to reduce the effect of unequal braking forces on the directional stability of the vehicle to a minimum.

CAUSE AND EFFECT OF ASYMMETRIC BRAKING FORCES

Uneven distribution of braking forces can be caused by many factors:

- Varying friction characteristics of brake pads
- Friction between brake pad and brake disc reduced on one side by the presence of oil, water or ice
- Tire adhesion on the road surface varying from one side to the other, e.g. one wheel on a dry road and the other on water, ice, oil or grass

Asymmetric braking forces of this kind may have very different effects on the directional stability of the vehicle, depending on the front suspension geometry and the location of the front wheel brakes.

The factor which determines whether the vehicle pulls to one side or retains good braking stability is the length, or more accurately, the direction of the lever arm of the braking force on the front wheels.

INBOARD SCRUB RADIUS

With a conventional front suspension design with outboard brakes this distance is determined by the inboard scrub radius (king pin offset at the ground, Figure 1). In the case of inboard brakes (Figure 2) the braking force acts through the center of the wheel with a positive lever arm a (king pin offset at the

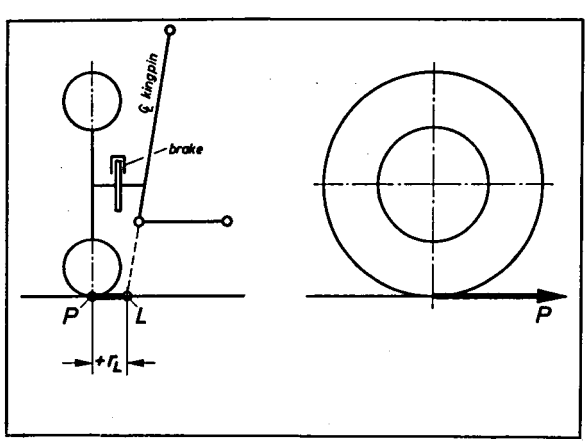


Figure 1

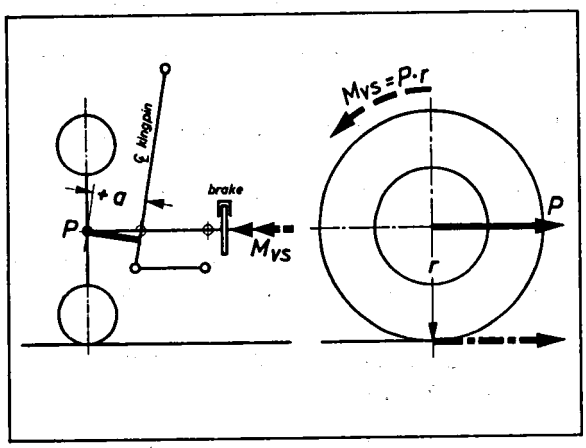


Figure 2

wheel center), because the moment $M_{vs} = P \cdot r$ does not react on the front suspension but is transferred to react on the mountings of the engine and transmission assembly.

The poor directional stability of vehicles of this design when braking forces are applied on one side only is due to the following factors (Figure 3):

1. Braking force P_1, acting on one side, turns the vehicle about its center of gravity towards the side which is being braked.
2. In addition, braking force P_1, acting on one side, turns the front wheels towards the side which is being braked as a result of the positive lever arm of the braking force.
3. The resulting slip angles α_1 and α_2 generate the lateral forces S_1 and S_2, whose direction is such as to intensify the effect of the braking force which tends to make the vehicle yaw to one side. In addition to this, the lateral force S_1 increases the lever arm y_{g1} of the braking force by the amount of lateral tire deformation so that the deflecting torque on the steering is exaggerated.
4. The lateral forces S_1 to S_4, generated by the increasing sideslip angle of the vehicle, are all directed to the side which is being braked and tend to displace the vehicle to this side.

The result is therefore a chain of reactions which all contribute to the same effect, i.e. the vehicle yaws and drifts.

Keeping the vehicle to the desired course under these conditions demands steering forces of such magnitude and a reaction time which is so short that the driver, particularly in emergency situations, cannot retain control of the vehicle.

ZERO SCRUB RADIUS

If the inboard scrub radius is reduced to zero, the tendency to pull to one side is also reduced, but not fully compensated even with zero scrub radius (Figure 4).

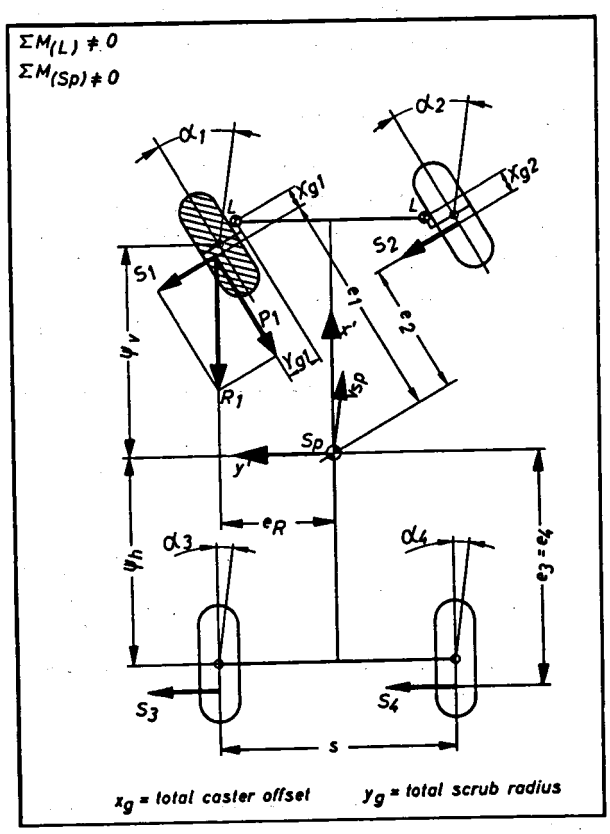


Figure 3

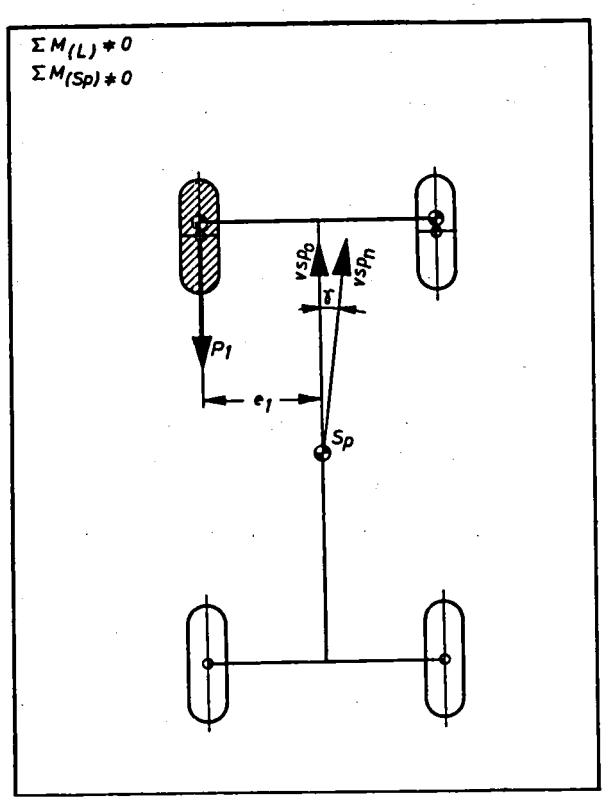


Figure 4

Braking force P_1, acting on one side, admittedly does not turn the front wheel immediately because there is no lever arm from the king pin center line during the initial braking phase, but the disproportionate braking force on one side turns the whole vehicle about its vertical axis (S_p) towards the side on which the braking force is greater (Figure 5).

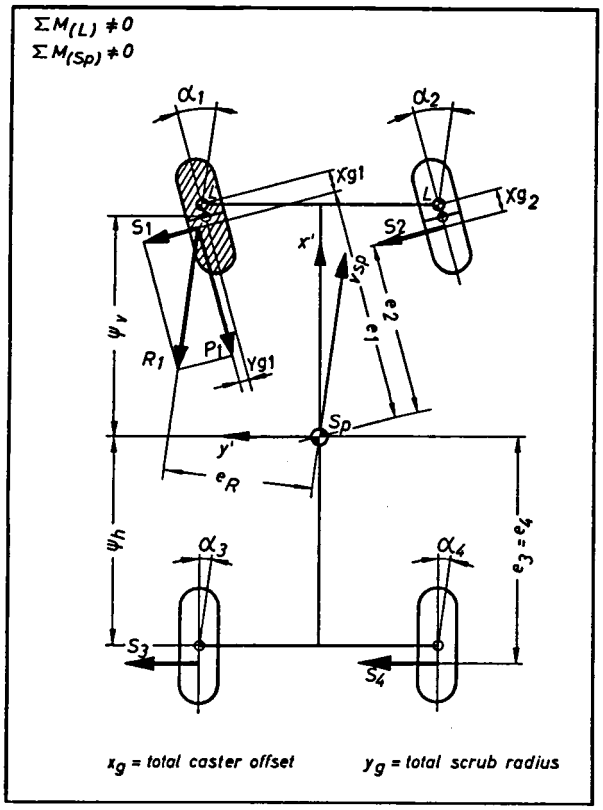


Figure 5

As soon as this turning motion is imparted to the vehicle, slip angles and therefore lateral forces are created at the wheels which then exaggerate the tendency of the vehicle to turn to the side on which the braking forces are greater.

In other words, when disproportionate braking forces are acting on one side, a vehicle with zero scrub radius exhibits similar characteristics to those of a vehicle with inboard scrub radius, except that the steering torque is not so great.

A self-stabilizing effect cannot be obtained from the steering when the scrub radius is zero. A reaction at the steering wheel and on the whole vehicle will still be present when uneven braking forces are applied.

OUTBOARD SCRUB RADIUS - STABILIZING EFFECT UNDER BRAKING

The yaw effect of uneven braking forces can only be counteracted by the use of outboard scrub radius (negative lever arm of braking force).

Since it is extremely difficult to design a system with inboard brakes where dimension "a" (king pin offset at the wheel center, Figure 2) is negative, it is necessary to incorporate outboard brakes together with outboard scrub radius to achieve a negative braking force lever arm (Figure 6).

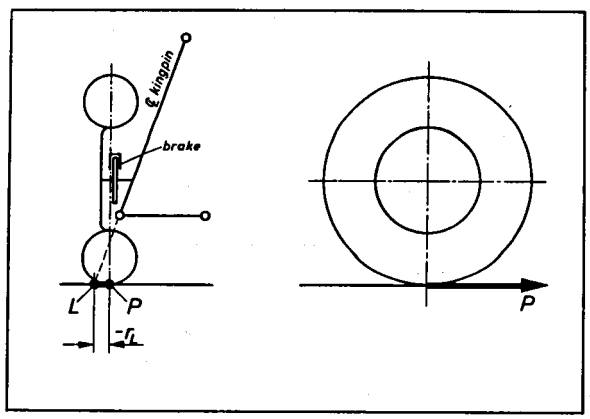


Figure 6

The good directional stability of such vehicles, even when disproportionate braking forces are applied on one side, is due to the following factors (Figure 7).

Braking force P_1, acting on one side, tries to turn the vehicle towards the side which is being braked in this case as well, but since the inboard scrub radius has now been converted to outboard scrub radius the front wheels are now turned towards the side which is not being braked.

As a result of the lateral elastic deformation of the tire caused by lateral force S_1 acting on front wheel 1 (braked), the outboard scrub radius incorporated in the design is increased by the amount of lateral dynamic tire deformation, thus also increasing the stabilizing moment $P_1 \cdot y_{g1}$.

Lateral forces S_1 and S_2 thus generated at the front wheels are now also directed towards the side which is not being braked and so compensate the deflecting moment $P_1 \cdot e_p$ of the braking forces about the center of gravity S_p, so that the sum of the moments of all the forces acting at the tire contact areas about the center of gravity is reduced to zero or a minimal value.

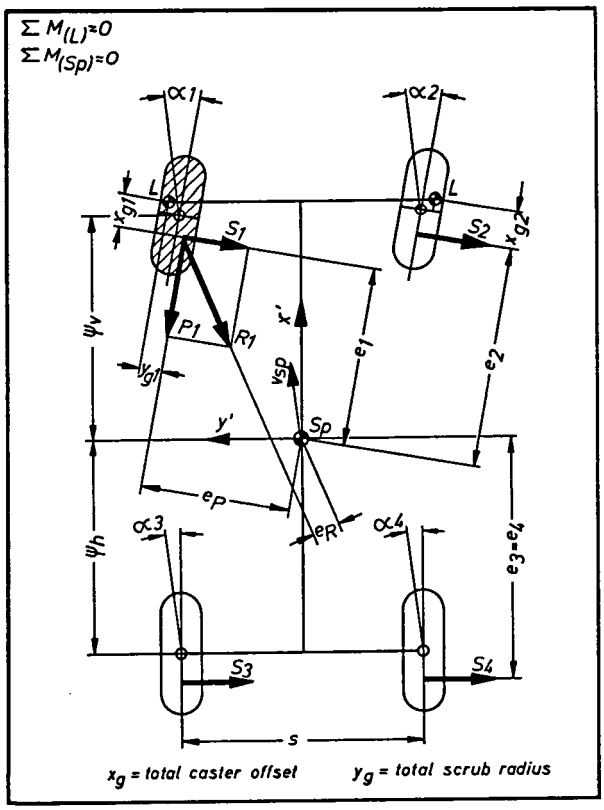


Figure 7

$$M_{(Sp)} = R_1 \cdot e_R + (S_3 + S_4) \cdot e_3' - S_2 \cdot e_2' = 0$$

Under these conditions, the vehicle no longer yaws (turns).

This yaw compensation also involves a self-stabilizing steering effect: unlike conventional front suspension designs, where the road wheels are turned to full lock if the steering wheel is released (free control) and a braking force is applied on one side, with a front suspension with outboard scrub radius the wheels will, under the same conditions, take up a *stable* position, applying steering correction. When braking forces are applied on one side, the front wheels turn only until the forces acting at the tire contact areas about the king pin center lines (points L) are in balance: $P_1 \cdot y_{g1} = S_1 \cdot x_{g1}' + S_2 \cdot x_{g2}'$ (Figure 7). There is now no torque on the steering and the front wheels remain stable in the corrective position. The torques required on the steering to keep the vehicle exactly on the desired course are almost zero.

Yaw can always be effectively compensated if at least one front wheel is not locked and if there is sufficient elasticity in the steering system. Tests have shown that full braking compensation can be achieved even with very inelastic steering designs (rack and pinion) because the greatest elasticity in the complete system is always provided by the driver's arms.

EXPERIMENTAL RESULTS

The tests were carried out with a conventional front suspension with an inboard scrub radius of 62 mm and with a McPherson front suspension with an outboard scrub radius of 18 mm and floating caliper brakes (Figure 8).

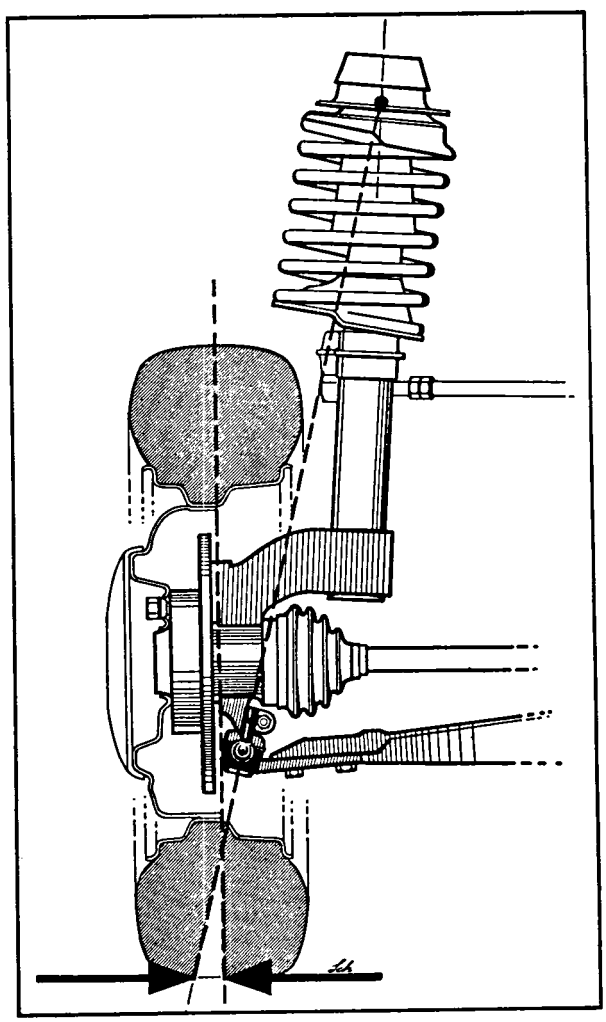


Figure 8

Figure 9 shows the front wheel angle of two vehicles on which the front left wheel was braked with the steering wheel released. In the case of

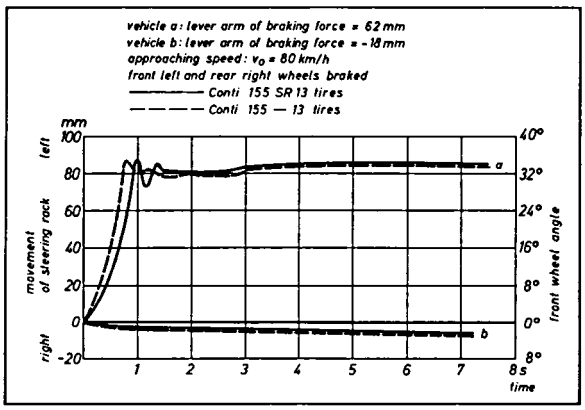


Figure 9

Vehicle A with a conventional front suspension, the wheel turns to full lock inside 2 sec. But with Vehicle B with outboard scrub radius the braked front wheel turns approximately 1° to 2° in the opposite direction and remains stable in this position (self-stabilization).

Figure 10 shows how the same Vehicles A and B turn about their vertical axes (skid) during braking. After 3.5 sec. the conventional vehicle has already spun through 90° and after 6 seconds has almost completed a 180° spin. Vehicle B with outboard scrub radius turns hardly at all. It maintains a stable course.

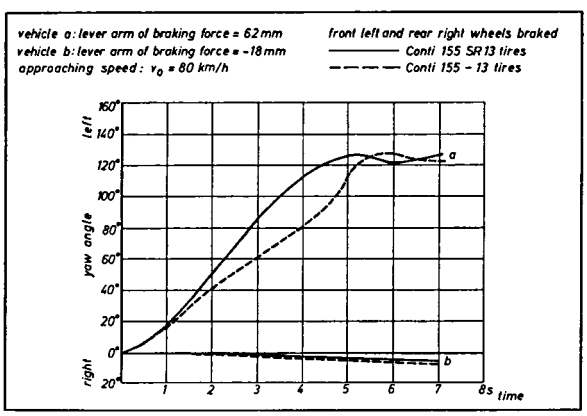


Figure 10

Figure 11 shows the steering torques required to maintain course, i.e. the forces which the driver must exert on the steering wheel if he is to hold the vehicle precisely on the desired course when unequal braking forces are applied. The sharply rising (0.5 seconds) steering torques required for conventional front suspensions with inboard scrub radius are immediately obvious, and similarly the increasing steering torques for Vehicle C with zero scrub radius. Only in the case of Vehicle B with outboard scrub radius does the steering torque remain practically zero, i.e. the steering is self-stabilizing.

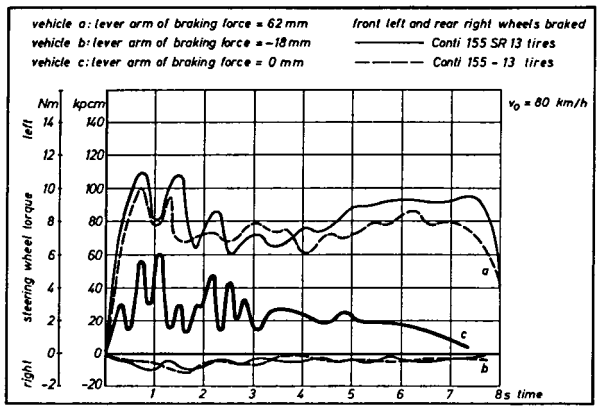


Figure 11

This braking compensation is influenced very little by the type of tire construction (Figures 9, 10 and 11).

To sum up: a vehicle with outboard scrub radius can be kept exactly on the desired course with very little steering effort, even when excessive braking force is applied on one side, as a result of its self-stabilizing characteristics.

The effect of this feature on directional stability can be observed not only when a braking force is applied on one side, but also when the vehicle is braked after a front tire is suddenly deflated (Figure 12). The lateral deformation resulting from the braking force on the deflated tire means that the outboard scrub radius changes, so that the braking force has a short positive lever arm.

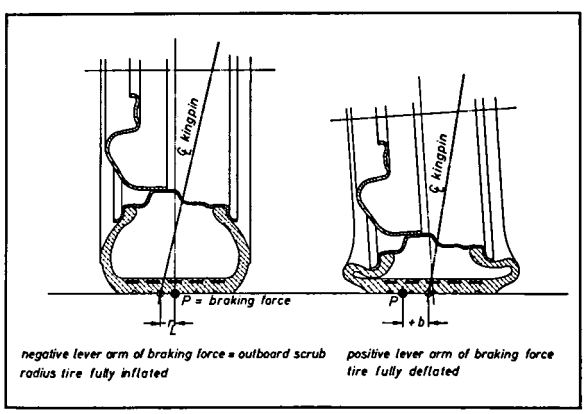


Figure 12

However, in the case of inboard scrub radius with normal king pin inclinations, the lever arm of the braking force is increased considerably (Figure 13).

The steering torques required to keep the vehicle on the desired course under braking with a deflated front tire vary accordingly (Figure 14). The correcting forces needed at the steering wheel in the case of outboard scrub radius are considerably reduced. They amount to only about a fifth of the

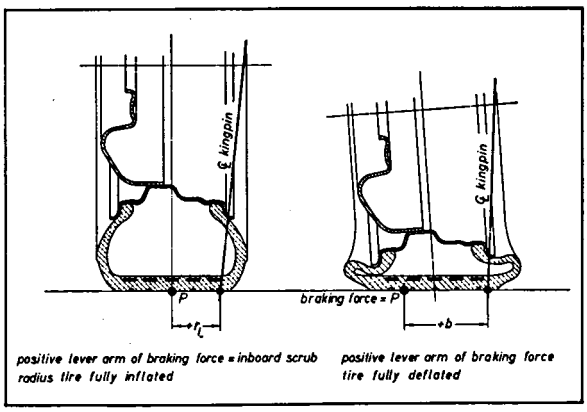


Figure 13

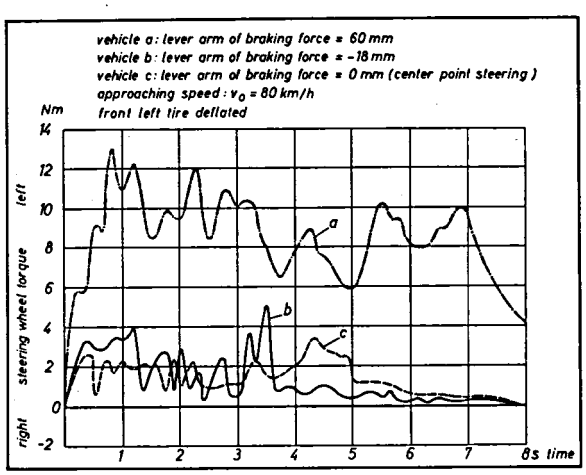


Figure 14

steering torques required in the case of inboard scrub radius.

DIAGONALLY SPLIT BRAKE CIRCUITS WITH OUTBOARD SCRUB RADIUS ON FRONT WHEELS

A diagonal split of the brake circuits takes on a completely new significance when considered with regard to the compensation of asymmetric braking forces, since it enables braking stability to be greatly improved when one circuit fails.

It is well known that if the brake circuits are split so that it is possible for both wheels on one axle to lock at the same time when one circuit is out of action, this has an adverse effect on directional stability. If both front wheels lock, steering control is lost; and if both rear wheels lock, the vehicle becomes unstable. If this happens during cornering motion, the understeer/oversteer characteristics of the vehicle change to such an extent that the average driver can hardly retain control of the vehicle.

In contrast, diagonally split dual brake circuits together with outboard scrub radius at the front wheels offer important advantages over all other methods of splitting the brake circuits which have previously been put into practice:

1. Vehicle behavior under braking remains constant when one circuit fails, because it is still the braked front wheel which will lock first.
2. The vehicle does not pull to one side even when two diagonally opposite wheels are braked because yaw is compensated by the outboard scrub radius.
3. Neither braking stability nor steering control is lost when one circuit fails because two wheels always generate lateral forces even when the two braked wheels have locked.

Figure 15 shows the stabilizing effect of lateral forces S_1 and S_2 on the front wheels when two diagonally opposite wheels only are braked. Even when both wheels on a diagonal circuit lock (P_1 sliding, P_4 sliding), braking stability is still provided by lateral forces S_2 and S_3 on the wheels which are not braked.

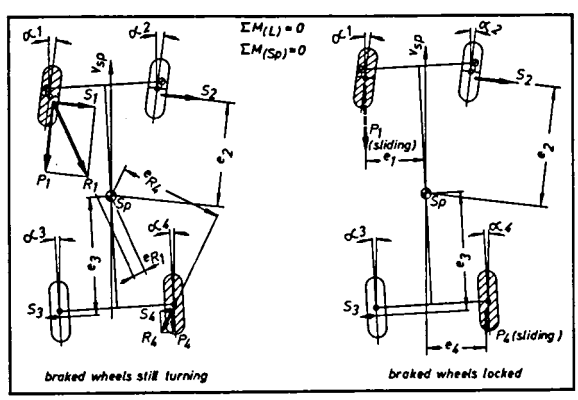


Figure 15

The oversteer/understeer behavior of a vehicle incorporating outboard scrub radius changes only slightly; this applies not only to straight ahead motion but even to cornering motion when one diagonal brake circuit fails. The vehicle still handles well and control is retained.

When the outer front wheel and the inner rear wheel are braked (Figure 16) an understeering moment is generated about the vertical axis by the braking forces and this moment is increased in the case of inboard scrub radius, but is partly compensated by the elastic oversteering correction imparted by both front wheels in the case of outboard scrub radius. This applies irrespective of whether the wheels lock or not.

When the other diagonal circuit is braked (Figure 17) the oversteering moment generated by the braking forces is again exaggerated in the case of inboard scrub radius, but is partly compensated by

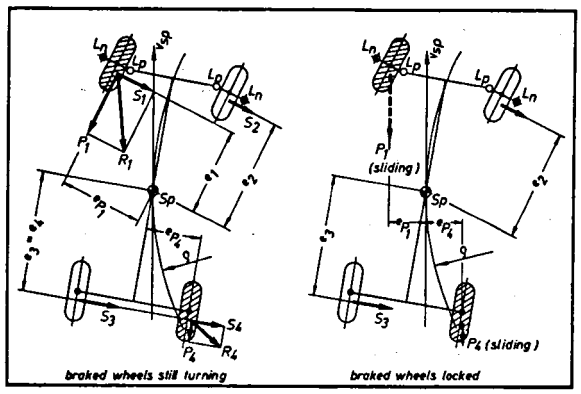


Figure 16

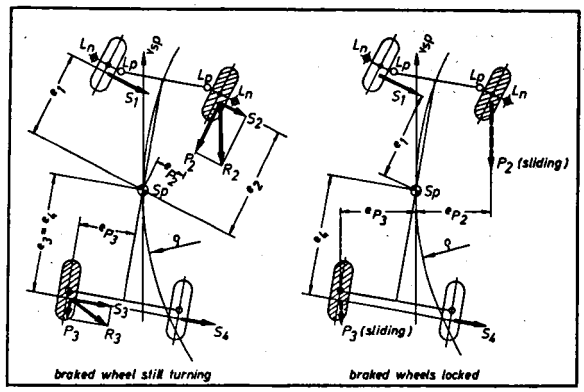


Figure 17

the elastic understeering correction imparted by the front wheels in the case of outboard scrub radius.

Tests have shown that a medium-sized passenger car with diagonally split brake circuits and an outboard scrub radius on the front wheels of 10-20 mm possesses exceptional braking stability and controllability in all braking maneuvers. In addition, this combination of design elements requires a minimum of driver reaction to shortcomings and defects both in the brake system and the road surface.

This not only contributes to the active safety of the vehicle but also minimizes driver fatigue.

FURTHER RESEARCH ON THE DRIVER/VEHICLE SYSTEM STUDY OF BRAKING IN A TURN

F. SURACE
Alfa Romeo

At the Kyoto conference we have shown the last results obtained by means of our mathematical model of the driver/vehicle system and the successful correlation between the theoretical data and the experimental results, in the case of an overtaking maneuver. This type of maneuver was chosen for its complexity.

At the time, the comparison with the experimental tests was good. It was yet impossible to use the mathematical model as a way to forecast the behavior of the vehicle. This because the time of extrapolation of the path was a parameter to be evaluated through the comparison between the theoretical and the experimental results. We therefore stated that further studies were underway to correlate the time of extrapolation to the reaction time of both the vehicle and the driver. This correlation is essential for the application of the mathematical model as a way to forecast the behavior of the driver/vehicle system.

We will now describe this correlation; in the second part of the paper we will show some applications of the mathematical model to the case of breaking on a curve.

RELATIONSHIP BETWEEN THE EXTRAPOLATION TIME OF THE TRAJECTORY AND THE REACTION TIME OF THE DRIVER AND THE VEHICLE

We briefly remind you that according to our mathematical model the driver's actions on the steering wheel are functions of the configuration of the extrapolated path of the vehicle with reference to external data: as external data we consider the trajectory that the driver wants to keep, the safe distance from the curbs and other obstacles.

Actually the driver must forstall the maneuvers in time to counterbalance the delay of the response of the car to the control, and therefore he must choose a well timed distance of extrapolation of the vehicle trajectory.

Thereafter the time of extrapolation must be linked to tune constant of reaction time of the driver and the vehicle.

The experiences gained in different situations have shown to us that it is necessary to consider the variability of the time constant of the vehicle during the maneuvers. In fact the time constants vary according to the degree of transversal acceleration of the vehicle.

Therefore in our mathematical model the time constants of the vehicle are evaluated at every instant of calculation.

More precisely we assume to apply to the vehicle, in the conditions of transversal instantaneous engagement relative to the considered maneuver, a step of steering infinitesimal such as it does not alter the degree of transversal engagement.

As an example, we show the Figure 1 the behavior of a car travelling in a curve. The lower line shows the angle of the steering and the upper line the lateral acceleration. At a certain time we apply an infinitesimal step of steering and see that the acceleration at the beginning becomes lower; then it takes an asymptotical trend.

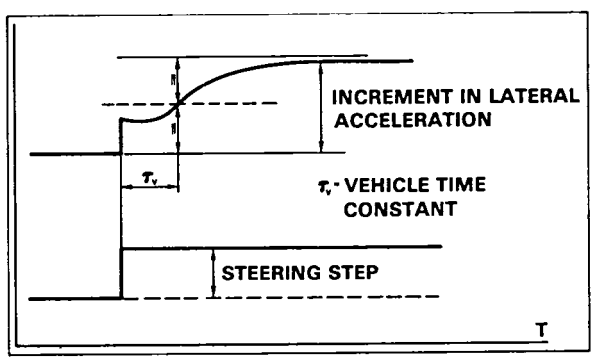


Figure 1

As "time constant" of the car we consider the time necessary to obtain a transversal acceleration equal to ½ of the asymptotic increment relative to the infinitesimal step of the steering.

With regard to the driver the equation of the delay in the steering maneuver is of linear type and of first order and it permits a classical definition of "time constant."

However, following the definition used for the vehicle behavior, we assume a time constant equal to the time necessary to reach ½ of the asymptotic value of the steering angle which is obtained by a step input. (Figure 2).

Defined in such a way the "time constant" of the driver and of the vehicle, we have attempted to fix an equation for "the extrapolation time":

$$T = K(\tau_v + \tau_p)$$

where K is a non dimensional constant to be defined experimentally and τ_v and τ_d are the time constant of the vehicle and of the driver, respectively.

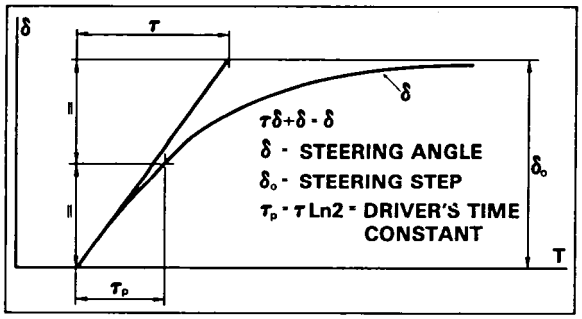


Figure 2

The expression for "extrapolation time" becomes more valid as "K" diminishes in its relativity to the various maneuvers.

For the maneuvers studied up to now we have found:

$$1. \leqslant K \leqslant 1.2.$$

It follows that the expression proposed for the "time of extrapolation" is valid assuming for K a mean value of 1.1. This conclusion permits us to utilize the mathematical model as a means to foresee the behavior of the system driver/vehicle in various types of maneuvers.

APPLICATION OF THE MATHEMATICAL MODEL TO THE BRAKING IN A TURN

The degree of refinement of our mathematical model does not allow us to consider the driver action on the brake and/or on the accelerator depending on the travelling conditions or on the external events. In other words the feedback from the environment can only affect the steering angle. However we can use the mathematical model introducing a predetermined shape of the longitudinal acceleration.

We have carried out several tests to compare theoretical and experimental values in relation to maneuvers of braking in a curve and utilizing for the calculation the experimental laws obtained in the tests.

Because of the satisfactory results obtained, we think it is interesting to show some examples of application of the mathematical model, where, for comparison with known test procedures, we took a step deceleration law. The examples shown indicate the effects of braking distribution between front and rear on the vehicle stability and therefore on the degree of engagement of the driver.

Tests Description

The braking tests are simulated starting from a speed of 120 Km/h (~74.5 mph) with a transversal acceleration of 0.3 g corresponding to a curvature radius of 380 m (1,250 ft). The deceleration steps considered are:

$$0.3\ g; 0.5\ g; 0.7\ g.$$

The road has the dimensions and configuration shown in Figure 3 corresponding to those of a motorway. We consider a dry pavement.

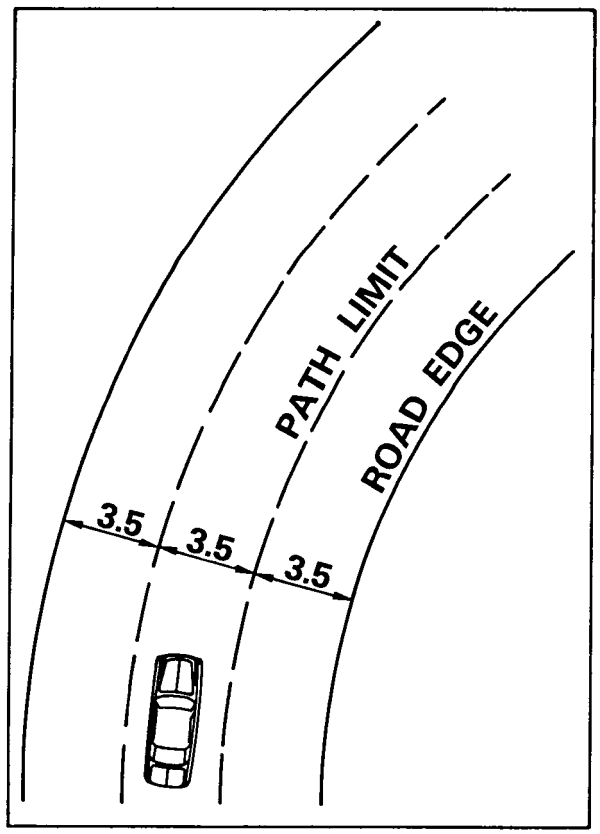


Figure 3

Vehicle Characteristics

The vehicle is a medium size sedan with equal performances to that shown in our paper presented to the Fourth ESV Conference. We now add the diagram of the braking distribution (Figure 4). Starting from that distribution and for the examined deceleration we have considered distributions symmetrically modified by ± 5 points as shown in the diagram.

Driver Characteristics

To show more clearly the effects of alteration in the braking distribution, we have considered a medium skilled driver.

The driver, as shown in the Fourth ESV Conference, is distinguished by two parameters:

- Time response linked to the delay of operation maneuver

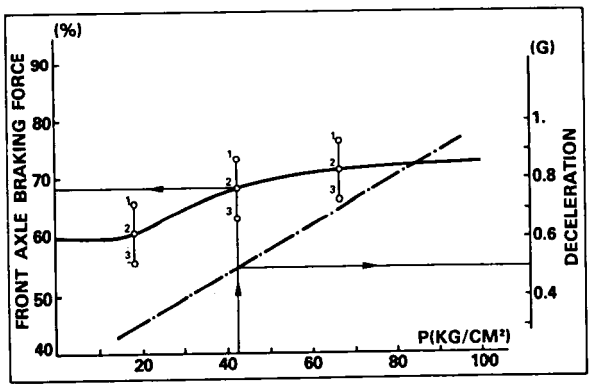


Figure 4

- Distance characteristics linked to the precision on evaluation of the extrapolated trajectory.

According to our experience a medium skilled driver has:

- Time response equal to 1.1 sec.
- Distance characteristic 130 m = 425 ft.

Analysis of Calculations

We will first of all show the results obtained by the mathematical model of the vehicle with driver (see Figures 5, 6 and 7).

Further we will show the results obtained with the steering wheel blocked (open loop, see Figures 8, 9 and 10). With reference to Figure 4, in Figure 5 to Figure 11 the numbers on the diagrams have the following meanings:

- Braking distribution with prevalence of the front part to the standard situation
- Braking distribution in the standard situation
- Braking distribution with prevalence of the rear part to the standard situation.

Vehicle with Driver (Closed Loop)

In Figures 5, 6 and 7 we have shown the following parameters with reference to the time:

δ = steering angle measured on the steering wheel

ω = angular velocity of the vehicle around the yawing axis

β = vehicle angle with reference to the trajectory.

We have not shown the trajectory because the deviations from the ideal one are always less than one meter. The diagrams are limited to a time inferior to the stopping time of vehicle. In Figure 5 we show the results relative to the case of braking deceleration equal to 0.3 g.

Due to the low level of deceleration we do not notice great differences among the various braking distributions. With the exception of some swinging of light intensity the steering angle diminishes with the time. Actually with the reduction of the speed and of

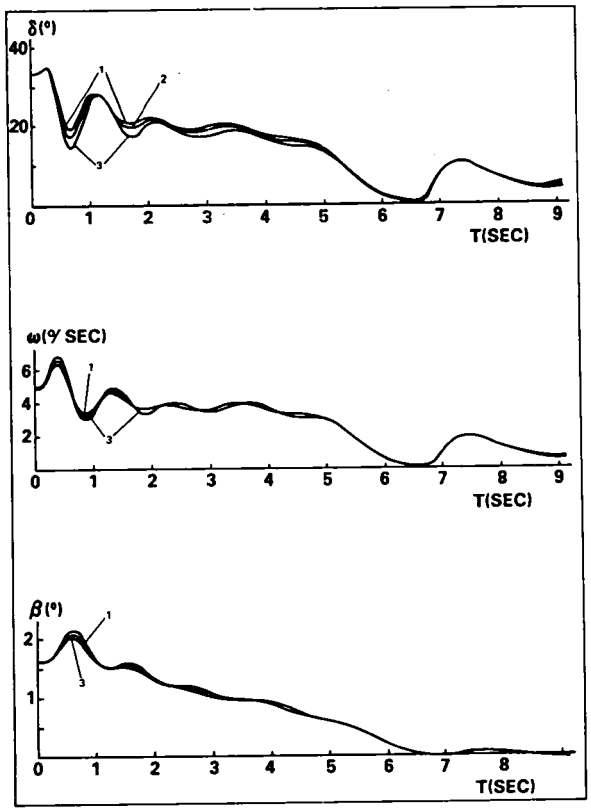


Figure 5

the angular acceleration, the vehicle would travel on a path with a greater curvature due to its characteristic of understeer. The driver easily counterbalances that trend reducing the steering angle.

The parameters "ω" and "β" diminish fairly progressively with the time showing light swinging. In Figure 6 we have shown the results of braking deceleration of 0.6 g. Due to the greater difficulty of the maneuver, the effect of the three different braking distributions is clearly shown.

Due to the understeer characteristics of the vehicle it remains a trend to reduce the angle of the steering wheel through swinging. The swinging amplitude depends on the braking distribution. With the increase of the braking action on the rear axis the vehicle becomes unstable and the amplitude of the swinging increases.

Nevertheless, even in the unstable condition the swinging of the steering angle is restricted. The same comment can be made for the parameters "ω" and "β". In the last case we can see that the parameters "ω" and "β" increase sharply at the beginning of the application of the brake, before the driver can react; the reason for this is the tire action.

Figure 7 shows the braking with 0.7 g deceleration. As you can see from it, the vehicle with a greater braking load on the front axle shows a trend

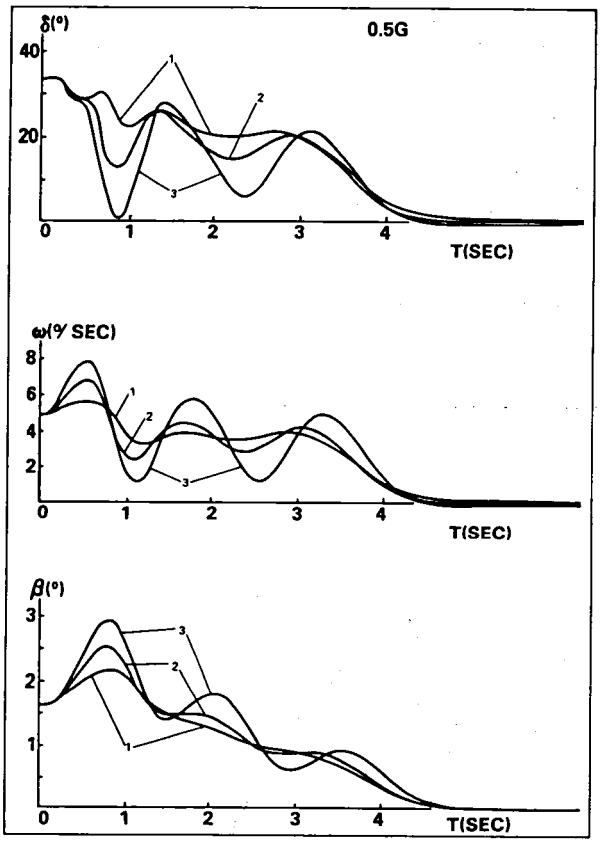


Figure 6

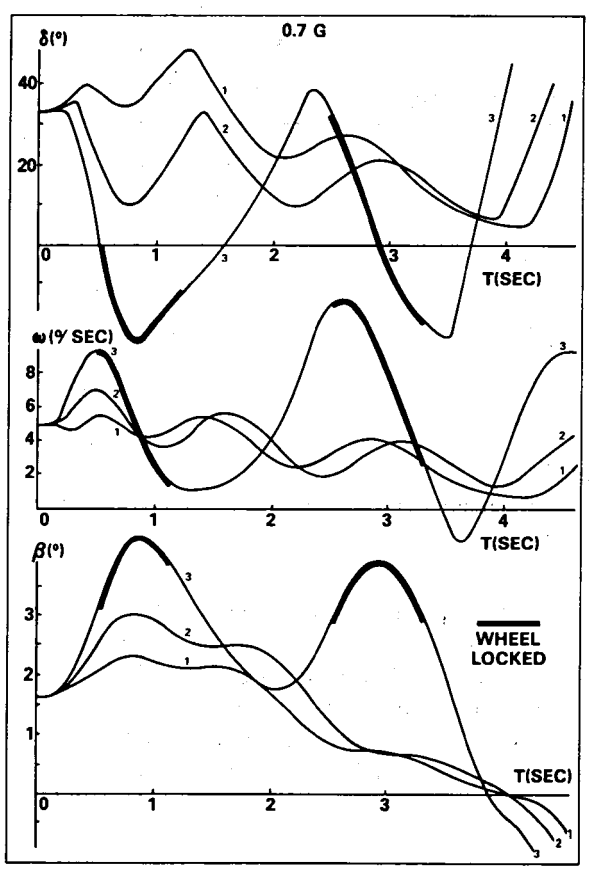


Figure 7

to travel on a smaller curvature; the effect is emphasized by the high "g" level.

The first driver's action is to increase the steering angle.

The unstable effect created by the greater braking force to the rear is emphasized in the diagram number 3; in fact you can see several countersteering actions by the driver.

This is the only case in which a wheel has been blocked – the rear inside one.

The shape of the diagrams of "ω" and "β" shows the differences of vehicle behavior in the three different braking distributions.

Vehicle Without the Driver (Open Loop)

Figures 8, 9 and 10 show the comparison between the results with blocked steering wheel for lateral acceleration equal to 0.3 g, 0.5 g and 0.7 g.

Instead of the law of variation of the steering angle, we have shown the vehicle shifting from the ideal path. In Figure 11 we summarize the shifting from the ideal path calculated after 2 seconds from the beginning of the braking in function of the longitudinal deceleration.

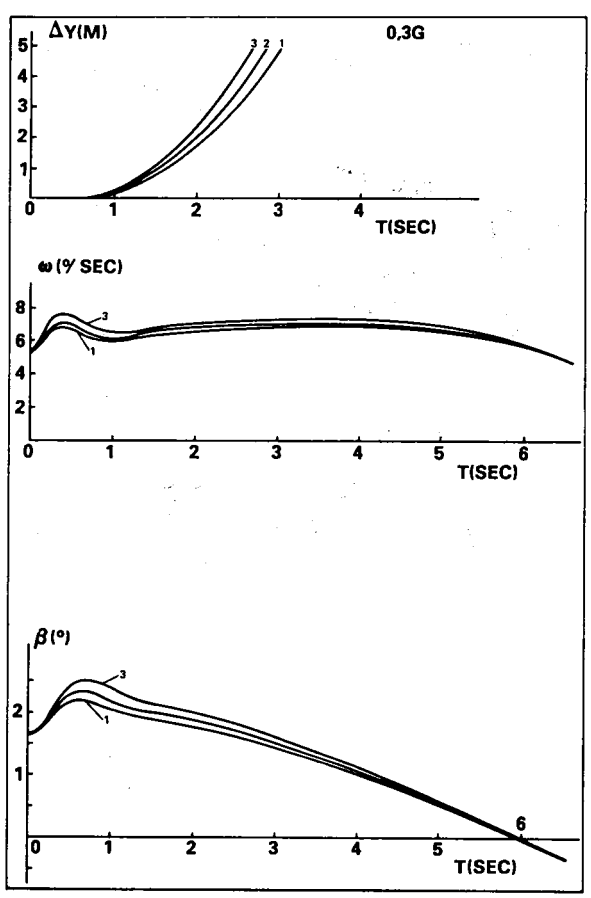


Figure 8

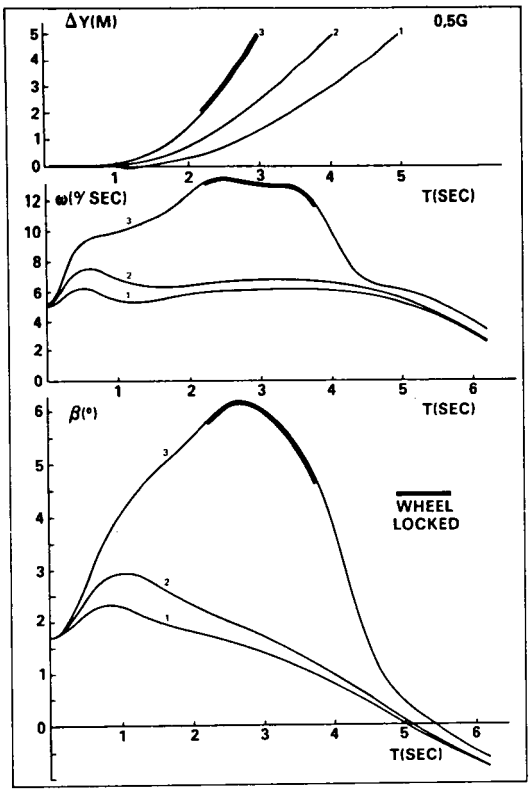


Figure 9

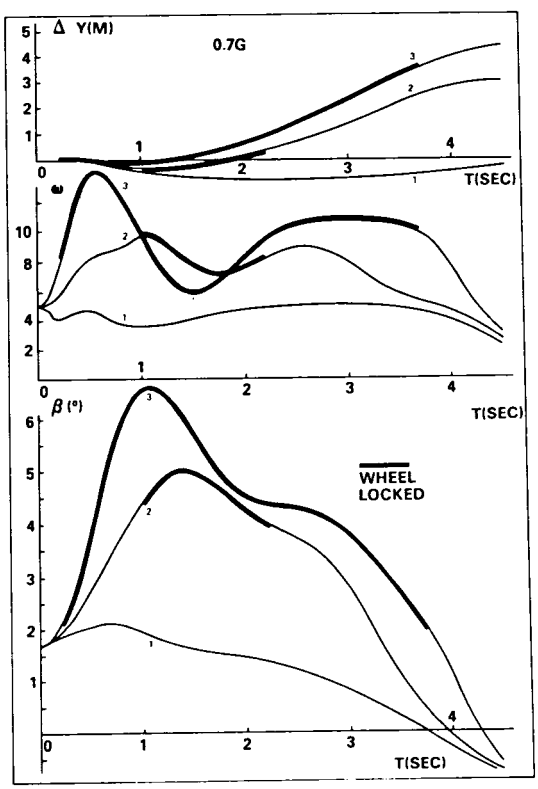


Figure 10

As it was foreseeable, the non-intervention of the driver with its corrective actions lowers the oscillation components of "ω" and "β" while it increases the transversal engagement of the vehicle.

In fact we can notice the skidding of the rear inside wheel for the braking distribution type 3 at the 0.5 g deceleration and for the braking distribution type 2 and 3 at the 0.7 g deceleration.

It is difficult to correlate the result of simulation of braking (closed loop) with that of (open loop) on the basis of the few cases shown.

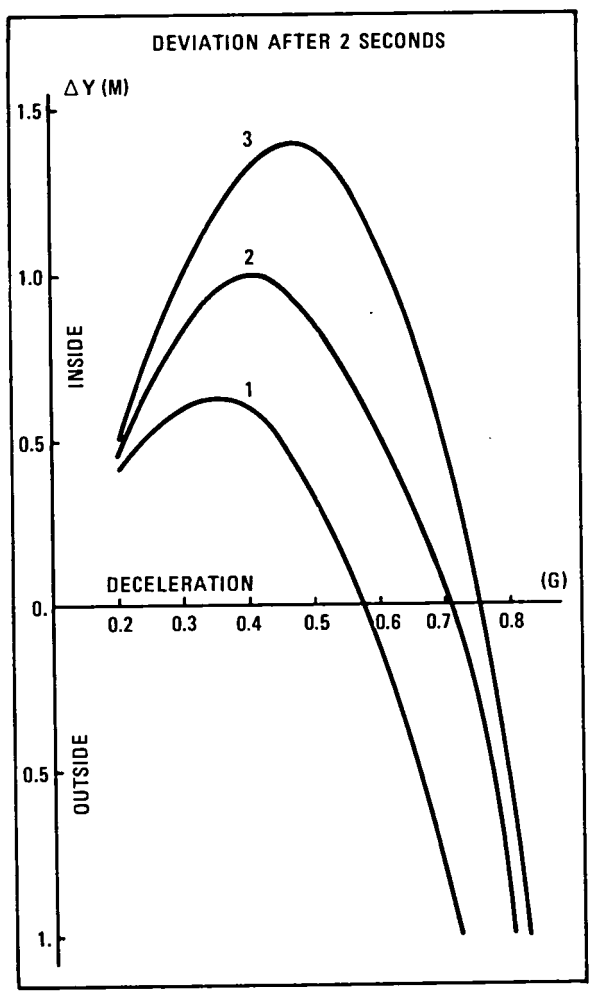


Figure 11

However quantitatively we believe that there is a good agreement between the two types of maneuvers. In fact the alteration of the degree of stability of the vehicle is noticeable in the same sense in both cases.

This agreement seems more noticeable for the angular velocity and for the vehicle stability.

CONCLUSIONS

On the basis of our actual experience, the proposed mathematical model is satisfactory for the maneuver studied.

It is our conviction that the use of mathematical models of driver/vehicle systems is an essential support of the experimental tests.

Moreover the mathematical models can give a contribution to the very confused situation of the active safety.

Unfortunately it seems that up to now not enough attention has been given to these models and the possibility that they can offer a correct interpretation of the experimental results has been underestimated.

They also can be useful to situations difficult to achieve in practice or dangerous for the driver.

We believe that only with the help of these instruments is it possible to obtain a deep knowledge of the dynamic behavior of the vehicle in the real travel condition and of the limitation of the principal dynamic parameters for safety.

BIBLIOGRAPHY

W. Bergman, effect of traction on cornering force, SAE Transaction (1961).

L. Segel, On the lateral stability and control of the automobile as influenced by the dynamics of the steering system, ASME Transactions (1965).

Alfa Romeo S.p.A., Notes on vehicle stability parameters with respect to active safety, Report on the Second ESV Conference, Stuttgart, October 1971.

Alfa Romeo S.p.A., Active safety: a contribution to the study of the vehicle/driver system, Report on the Third ESV Conference, Washington, June 1972.

L. Chidini, Analysis of the dynamics of the driver-car system as a contribution to safety, XIV Fisita Congress, London 1972.

Alfa Romeo S.p.A., Further research on the driver/vehicle system, Report on the Fourth ESV Conference, Kyoto, March 1973.

THE STOP LAMP: SOME OPTIMUM VISIBILITY CONSIDERATIONS

PAOLO SOARDO
Instituto Elettrotecnico
Nazilonale Galileo Fennanis (IEN)

SUMMARY

The luminous intensity distribution for a stop lamp can be calculated with special consideration to glare limitation. If luminous intensities are chosen so as to give the maximum acceptable glare for any position of the observer, maximum visibility by night is reached. In these conditions a good day visibility is also to be expected.

The drawbacks of this lamp are its sensitivity to mounting height and aim, but on the other hand it overcomes the disadvantages of the double level system.

INTRODUCTION

The luminous intensity distribution of a rear lamp can be optimized with special reference to the reduction of glare and to the improvement of visibility: these are to a certain extent contrary aims; but a compromise solution can be looked for.

Glare is important by night, at very low adaptation luminances, and it calls for a limitation of maximum luminous intensity and luminance for the lighting device. On the other hand, through a device designed to produce the maximum acceptable glare for any position of the observer it seems possible to reach a high night visibility with a low discomfort for the driver. A good day visibility could also be expected in these conditions.

A better identification of the different rear lamps (direction, parking, stop) could be helpful for the drivers. Besides a change of the color code, one could also think of a stop lamp showing to the rear a variable luminous area depending on the deceleration of the vehicle; this indication could be very helpful for accident limitation.

In this paper the rear lamp visibility optimization is carried out with particular reference to the stop lamp.

OPTIMIZATION CONDITIONS

A glare limitation of the lighting device should take into account:

a. luminous area of the lamp;
b. difference between the height of the observer eye (the driver of the car which follows) and the mounting height of the lamp;
c. inclination of the vehicle for different load conditions.

A study on the influence of the luminous area on the visibility of the lamp was carried out by T.C. E.3.3.5. of the Commission Internationale de l'Eclairage (References 1 and 2). Lamps having different luminous areas were compared with a reference lamp with an angular aperture of 8 minutes and the illuminances on the observer eye giving the same visibility were measured. Tests were carried out with background luminances of 0.1, 30 and 10,000 cd/m^2. Figure 1 shows the experimental results in the first case (night conditions).

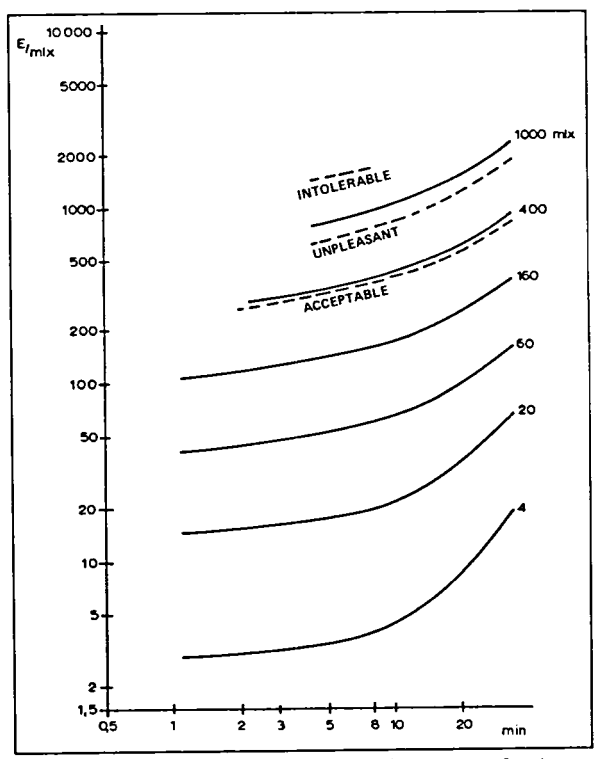


Figure 1. Equal visibility and equal glare curves for lamps with different angular aperture.

Each curve gives the illuminance versus the aperture angle for a lamp having the same visibility as the reference lamp with the illuminance indicated as parameter of the curve. Some glare tests were also carried out; the results are shown in the dashed curves representing three glare levels: acceptable, unpleasant and untolerable.

Concerning point (b), the present regulations allow a great variety of mounting heights for the stop lamp, and it seems quite clear that no considerable improvement could be reached without some limitation in the lamp position. A theoretical design is

carried out in this paper with 0.75 m difference between the heights of the observer eye and of the lamp. About point (c) a ± 1° (about ± 2%) inclination of the vehicle will be allowed, which is quite normal in most European cars.

GLARE LIMITATION

The glare is commonly described in two different forms, discomfort and disability; this paper is mainly concerned with the former one.

The evaluation of discomfort glare leads to a "glare index" which is related with the observer percentage who deemed non glaring a luminous source (References 3, 4, 5, 6, and 7). Some experimental formulas are given which allow us to calculate the glare index with reference to the adaptation luminance, the luminous intensity of the source and its angular aperture.

The experimental conditions of CIE document (Reference 1) can be reproduced through the formulas given in References 6 and 7. Figure 2 shows the comparison between the CIE glare curves and the calculated ones with a percentage of 50% and 70% of

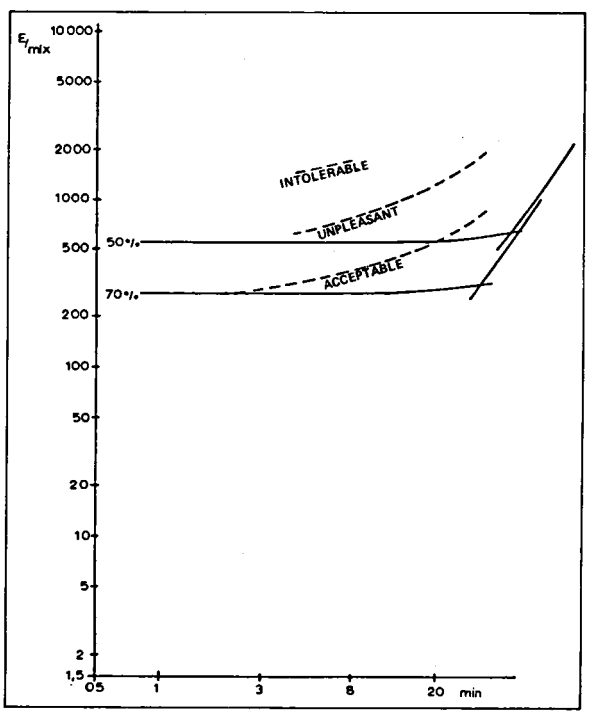


Figure 2. Glare calculation for a lamp: the percentages refer to the "non glaring" assessments.

non glaring assessments. The two curve branches (the formula changes with the angular aperture) are almost the asymptotes of the acceptable and unpleasant CIE curves.

The good agreement of the two types of curves could allow the use of the formulas for traffic simulation. The design of the luminous intensity distribution of the stop lamp will be based on the CIE "acceptable" glare curve which shows a high percentage of positive assesments.

DESIGN IN NIGHT DRIVING CONDITIONS

The glare must be limited for any load of the vehicle; the worst condition is a front tilt of 1°. With a 0.75 m difference in height between the lamp and the observer eye, the luminous intensity along the lamp axis is seen at a distance of about 43 m. Below this distance, and then for directions above the lamp optical axis, a calculation of the lamp luminous intensity distribution limited to the CIE acceptable glare level (Reference 1) is easily carried on. For distance above 43 m this calculation will lead to lower glare (and lower visibility, too), since it seems useless to increase the luminous intensity below the lamp optical axis, which will contribute only to "road illumination" for different load conditions of the vehicle.

Figure 3 shows the calculated vertical luminous intensity distribution of a round lamp with a diameter of 100 mm, while in Figure 4 a comparison is shown, for three typical load conditions of the vehicle, between the CIE glare level curves and the illuminance given by the said lamp on the observer eye versus its angular aperture and the observation distance.

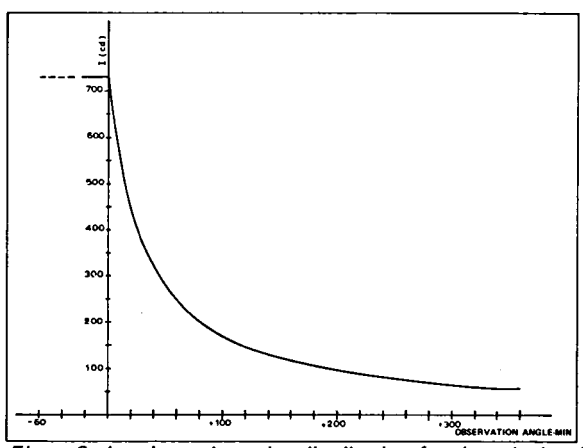


Figure 3. Luminous intensity distribution for the calculated round lamp with a diameter of 100 mm.

In order to compare the characteristics of this theoretical stop lamp with a double level one, the minimum and maximum illuminances allowed by the present regulations (Reference 8) for the night and day levels are superimposed in Figure 5 on the calculated illuminances of Figure 4 and the CIE glare levels. It goes without saying that a comparison between manufacturing tolerances and theoretical characteristics could raise some doubts, but it seems

anyway that according to Figure 5 the single level design described in this paper could bear comparison with the double level lamp.

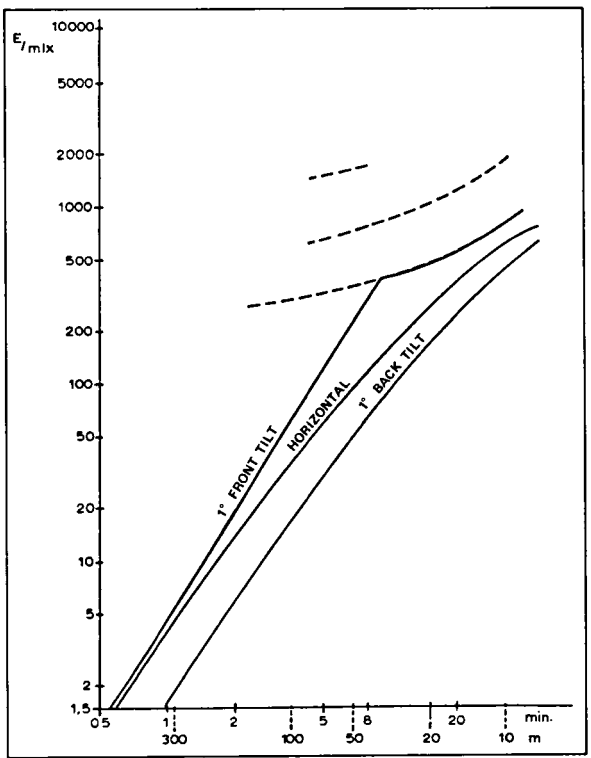


Figure 4. Illuminance on the observer eye for the calculated round lamp with a diameter of 100 mm.

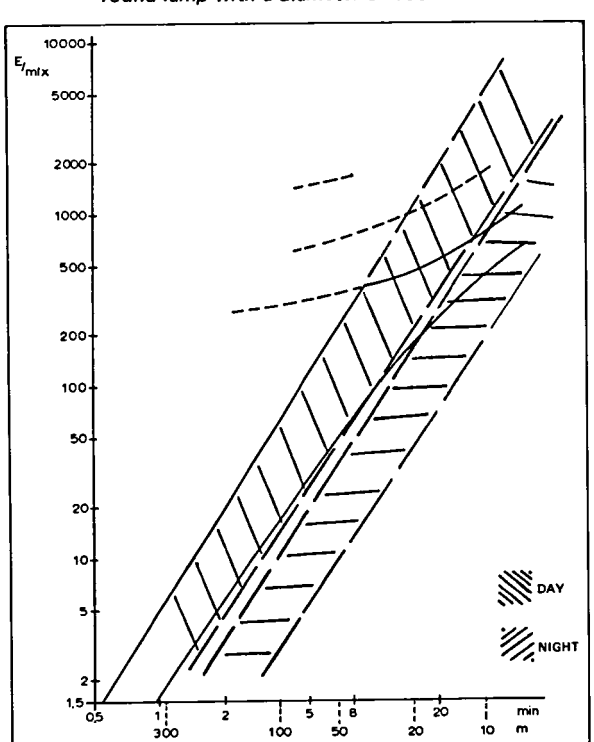


Figure 5. Comparison between the calculated round lamp (Ø 100 mm) and the tolerances allowed by the double level regulations.

This opinion is confirmed by Figure 6 where the same comparison is carried out for a calculated rectangular stop lamp 200 mm long and 100 mm high. Since no requirement is given in the international regulation for the luminous area, the calculated lamp appears even better.

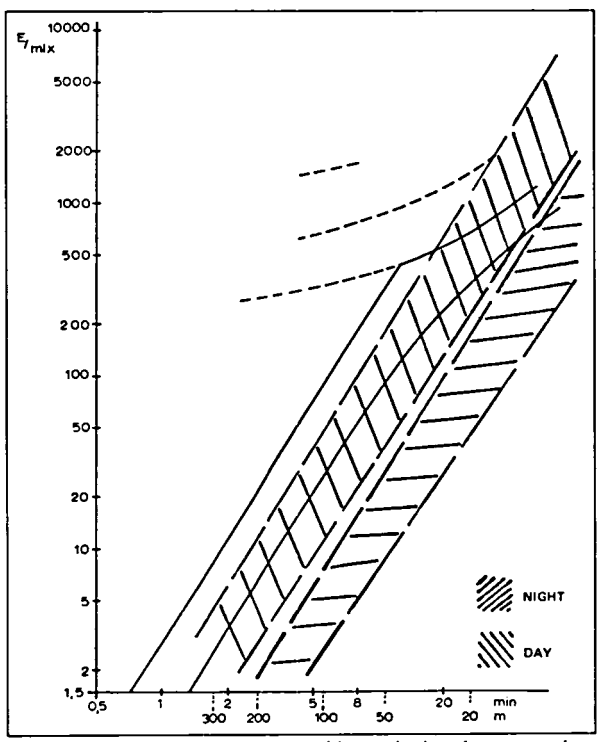


Figure 6. Comparison between the calculated rectangular lamp (100 x 200 mm^2) and the tolerances allowed by the double level regulations.

On the other hand the calculated lamp is sensitive to the height difference between the observer eye and the lamp. Figure 7 shows again the illuminances on the observer eye for the round lamp with horizontal vehicle and with three height differences: 0.5, 0.75 and 2.0 m.

The horizontal luminous intensity distribution leads to much lower visibility and glare variations and could be substantially flat.

DAY VISIBILITY

While the glare design in night driving conditions assured maximum visibility, day visibility can only be verified. Many authors (see e.g. References 9, 10, and 11) carried out laboratory tests to measure the threshold visibility, i.e. the luminance contrast between a luminous source and its background. This threshold depends on the background luminance and on the luminous source angular aperture. Figure 8 shows the minimum illuminances on the observer eye

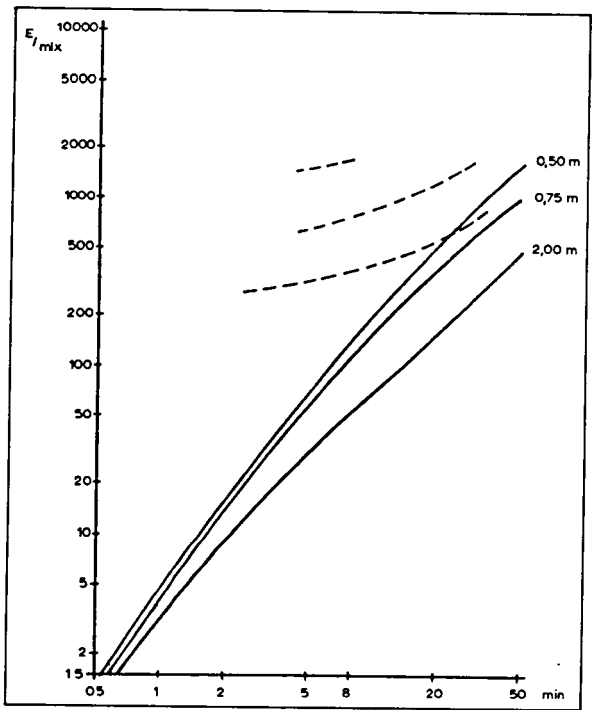


Figure 7. Illuminance on the observer eye for three height differences between the observer eye and the lamp (round Ø 100 mm lamp).

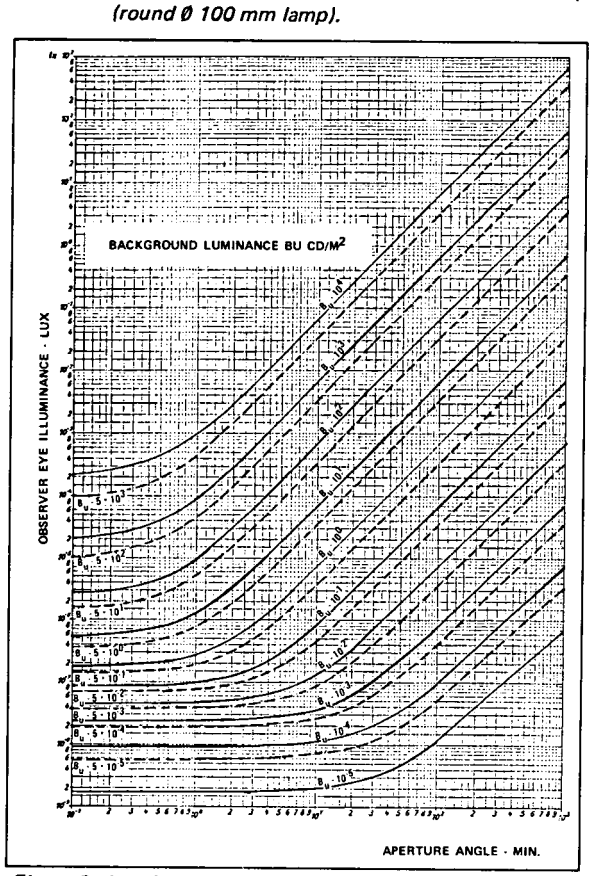


Figure 8. Luminance achromatic contrast thresholds.

necessary for the source visibility versus its angular aperture and the background luminance (Reference 11).

For a good visibility, an eye higher illuminance (supra-threshold visibility) is usually required. Unfortunately, the above mentioned experiences were carried out with white source and background (achromatic threshold) and fewer data are available on chromatic threshold (References 12, 13, and 14). Many authors agree that a chromatic contrast could allow good visibility with luminance contrasts equal to the achromatic threshold or even lower.

Table 1 compares the round lamp with the reference to the observation distances, the "safety coefficient," i.e. the times the eye illuminance given by the lamp is higher than the achromatic threshold. The background has a high day luminance, 10,000 cd/m^2; the visibility assured by the calculated lamp seems to be more than sufficient.

TABLE 1

	Safety coefficient	
Observation distance (m)	Round lamp Ø 100 mm	Rectangular lamp 100 x 200 mm^2
20	1.5	1.1
50	3.3	2.0
100	4.3	2.8
200	4.4	3.3
300	4.3	3.2
500	3.5	3.1

CONCLUSION

The stop lamp design just described seems to assure a night and day visibility comparable or even better than the present double level lamp with lower glare.

The drawbacks of this design are the sensitivity to the height difference between the driver eye and the stop lamp, which requires low mounting height for the lamp, and the necessity for a good aiming of the lamp. On the other hand, this design overcomes the disadvantages of the double level system.

As for the possibility of indicating the vehicle deceleration through a variable luminous area stop lamp, nothing prevents us from putting many lamps (typically three or four) near one another horizontally except probably a luminous intensity reduction. The higher the deceleration, the higher could also be the number of lamps alighted, starting of course from the outer ones. The number of on lamps could be controlled by the vehicle deceleration

following a brake application or by the pressure of the brake fluid.

ACKNOWLEDGMENTS

The authors are greatly indebted to Dr. Artom and C. Gentile for their great contribution to the planning of this research, and wish to thank Mr. P. Devaux, G. Danese and G. Pocci for the helpful discussion which allowed the stating of the aims of this design.

REFERENCES

1. CIE/E/3/3/5, "Etude de l'influence de la luminance sur la perceptibilité des feux de signalisation pour automobiles. Rapport commun franco-allemand" Document E.3.3.5/031 (January 1966).
2. CIE/E.3.3.5, "Feux de signalisation des automobiles. Rôle de la luminance. Relation entre les intensités lumineuses et les surfaces des plages éclairantes des feux de même perceptibilité" Document E.3.3.5/036 (October 1966).
3. S. K. Guth, J. K. Mc Nells, "Further data on discomfort glare from multiple sources" Illuminating Engineering, Vol. 56, No. 1, 1961, pp. 46-53.
4. R. G. Hopkinson, "Evaluation of glare," Illuminating Engineering, Vol. 52, No. 6, 1957, pp. 305-316.
5. R. C. Putnam, K. D. Bower, "Discomfort glare at low adaptation levels," Illuminating Engineering, Vol. 53, No. 4, 1958, pp. 174-184.
6. G. A. Fry, "The evaluation of discomfort glare," Illuminating Engineering, Vol. 51, No. 11, 1956, p. 722.
7. Report No. 6 of the Committee on Standards of Quantity and Quality for Interior Illumination of the Illuminating Engineering Society. "Evaluation of direct discomfort glare in lighting installations," Illuminating Engineering, Vol. 54, No. 7, 1959, pp. 463-468.
8. Ministero dei Transporti e Dell'aviazione Civile, Circolare No. 10, 1968.
9. H. R. Blackwell, "Contrast threshold of the human eye," Journal of the Optical Society of America, Vol. 36, No. 11, p. 624 (November 1946).
10. W. Adrian, "Die Unterschiedsempfindlichkeit des Auges und die Möglichkeit ihrer Berechnung" Lichttechnik, Vol. 21, No. 1, p. 2A (1969).
11. Norma Din 5037.
12. N. E. G. Hill, "The recognition of colored light signals which are near the limit of visibility" Proc. Phys. Soc. Vol. 59, p. 560 (1946).
13. N. E. G. Hill, "The measurement of the chromatic and achromatic thresholds of colored point sources against a white background," Proc. Phys. Soc. Vol. 59, p. 574 (1946).
14. W. Arndr, E. A. Voit, "Farberkennbarkeit und Blendwirkung von Signallichtern," Lichttechnik, Vol. 9, No. 4, 1957, pp. 186-190.

SAFETY ASPECTS OF DENOVO RUN FLAT TIRES

T. HOLMES and T. FRENCH
Tire Technical Division
Dunlop Limited

DESIGN OBJECTIVES

The pneumatic tire, so vital to the operation of automobiles, has in its normal or standard form, one overriding fundamental defect: when it is punctured by a penetrating object, the "pneumatic" content is lost and the vehicle is effectively rendered immobile and actually or potentially unsafe. It has to be accepted that there is no prospect of preventing sharp metallic-like objects from penetrating the rubber tread and casing of the tire, just as there is equally no prospect of guaranteeing sealing the full air pressure inside the tire after the puncturing object has entered it; the sheer size and geometric diversity of puncturing objects totally precludes this. On the average, a motorist has a puncture every 2 to 2½ years. In the U.K. 35% occur in darkness; more than 75% of women drivers cannot change a wheel. These and other related facts constitute the necessity from the user's standpoint of making a basic improvement in the behavior of pneumatic tires during and after a puncture. From the safety standpoint, clearly a sudden deflation (a blowout or burst) can be a hazardous event, and the more common slow leak type of puncture leading to progressive reduction in tire pressure, can be equally dangerous. Also the act of parking a vehicle on the roadside, often in a dangerous or unlit place, and attempting to change a wheel can be the cause of secondary accidents. The availability of a tire such as Denovo which manifestly contributes to safety in these fields and also guarantees continuation of vehicle mobility under defined conditions, gives much needed scope to the basic vehicle designer in terms of better space utilization in that the vehicle can be equipped with four instead of five tires and wheels. To enable this "breakthrough" position to be realistically attained, coupled with total elimination of roadside wheel changing, it is necessary to design a tire/wheel system which provides virtually a 100% assurance of safe mobility following a puncture. This was in fact the design objective for the Denovo; the objective has now been attained with massive independent corroborative evidence to support this contribution.

DENOVO TECHNOLOGY

Having accepted right from the start that tires would always be prone to puncture and thereby loss of air pressure, it followed that the collapsed (i.e. punctured) tire would need to remain a coherent engineering structure capable of exerting progressive control forces and of providing a reasonable degree of endurance under acceptable speed conditions. The following key factors were identified as being essential to attain this end and the production Denovo unit incorporates them all.

- Only a radial ply tubeless tire is capable of accepting the very high strains involved in deflated running; new design parameters have been evolved to minimize destructive heat generation.
- To enable the vehicle load to be transmitted essentially through the tread band, thus giving good rolling mobility as compared with the totally noncoherent shape of a flat normal tire, a particular shape has been evolved for the Denvo tire. (Figure 1). This leads to the Denovo being, in its normal inflated form, one step further along the already successful and well established development towards "low profile radial tires." The fact that this is coincidental with desired modern practice is highly advantageous and indeed in most cases, it has proved possible to install the correct size of Denovo tire within the spatial envelope available to the normal tire equivalent (Figure 2).

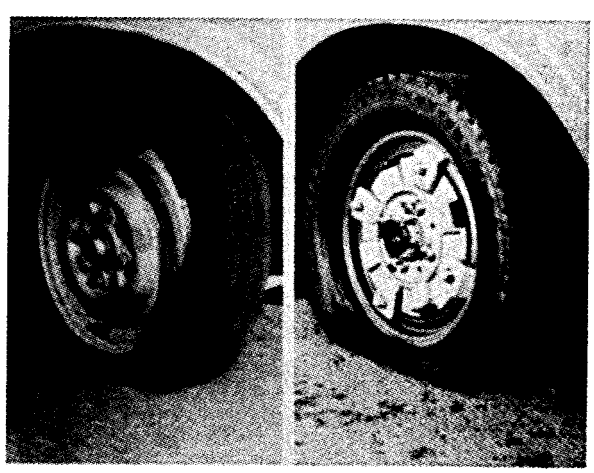


Figure 1

- When the tire collapses without air, the design must ensure that the load is carried from the rim to the ground through a carefully designed "column of material." Also in this condition, it is necessary to eliminate the destructive effects of internal friction when one rubber face slides over the other at velocities of 20-30 feet per second.

The friction is reduced to acceptably low levels by a lubrication system fitted within the tire and operated when the tire collapses (Figure 3).

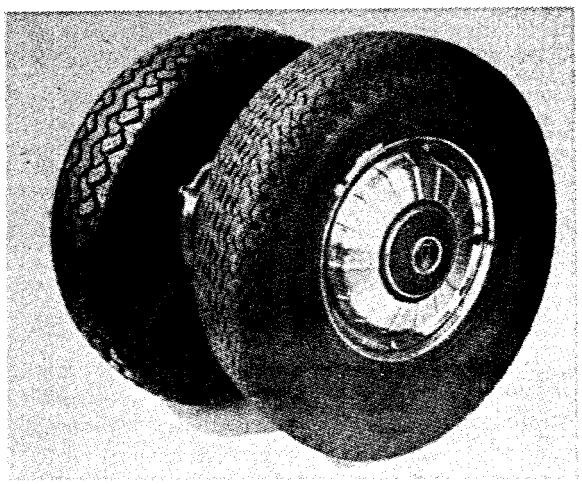


Figure 2

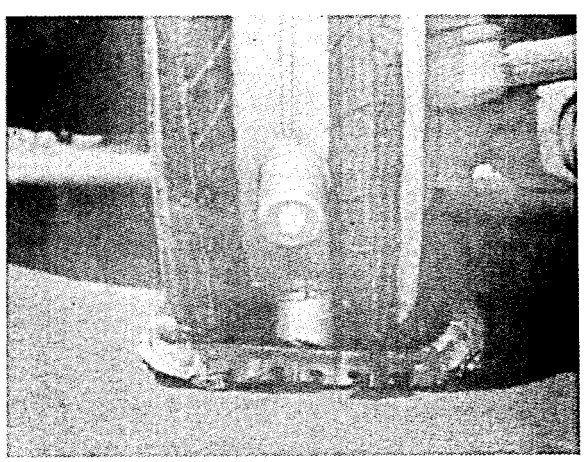


Figure 3

- As the punctured tire is driven along, the controlled degree of residual heat generated is used to vaporize constituents of the lubrication system to provide 4-6 p.s.i. reinflation pressure in the vast majority of puncturing incidents. Even if a perfect seal is not obtained, in many of the residual cases there is sufficient reserve of vaporizing components to provide for several hours' running. In the minority of cases where the puncturing object is large or irregularly shaped, the Denovo concept still offers a high degree of mobility and a minimum performance of 50 miles at 40 mph is then available.
- To enable all the foregoing factors to operate correctly, it is absolutely vital that the tire structure is held in its correct position in relation to the wheel rim. This is attained by a specially designed combination of tire and rim, so that under cornering or any other forces which tend to dislodge the tire from its correct position, the outer controlling bead is restrained from moving across the rim. This principle has proved extremely effective and no cases of dislodgement even under extreme motoring conditions when the tire is fully deflated, are known.

During the early design and development period of Denovo, a great deal of testing and information fact finding had to be carried out. This was because we were moving into an entirely new tire concept area. No longer could we just consider all the past evidence of inflated pneumatic tires, because it was exactly that word "pneumatic" which was going to be missing for some part of the new tire's life. We now had to introduce test procedures for flat running tires and also most importantly, to integrate the short period of flat running performance required with the very good inflated performance, which we and other tire manufacturers throughout the world have achieved over the last 80 years or so.

Some of the factors to be established during this early period were:

- How vehicle stability and handling was affected with a flat tire on one of its corners in both wet and dry conditions.
- How safe the Denovo tire/wheel unit was after a high speed blowout or a slow leak.
- Establish the facts about incidence of punctures and the state of spare wheels, etc.

These and many more facts on how this information was obtained are described in the following texts.

The flat tire performance was assessed first to establish how good a product we had.

RUN FLAT ENDURANCE

The minimum performance of 50 miles at 40 mph and with the tire in most cases initially punctured beyond the accepted repair limits, has been achieved many thousands of times. This minimum performance was set very early on, mainly by industry specifications. To ensure that tires were run completely flat, the valve core was removed. They were also driven under the most severe usage conditions on all types of roads, motorway and urban, and also on test tracks when the driving configurations were considered hazardous.

When the tire puncture seals, thus ensuring that partial vapor reinflation occurred, mileages have been considerably higher than the minimum performance.

The tools used to puncture tires were a 3/8 inch square pyramid or a 3/4 to 1 inch slitter type, and the puncture position was in the tread center or shoulder area. Some joint Rover/Dunlop test results have shown outstanding performance with the tire in this state, e.g. 347, 382, 347, 207, 382 and 406 miles at 70 mph.

Under no circumstance was the car in anything but a stable condition during these tests. The driver maintained full directional control at all times and no tire showed any sign of failure when taken off the car.

HANDLING AND STABILITY

A series of tests was carried out to establish the handling and stability levels of a number of cars with various tire fitments. These tire fitments included Denovo tires in various stages of inflation and fully deflated on each position of the cars, and at minimum and maximum load confitions. The tires with a non-run flat capability were tested at reduced inflation pressure and a pressure of 10 p.s.i. below the recommended pressure was used. Previous evidence had shown that the largest differences between inflated and under-inflated or deflated tires were obtained when the under-inflated or deflated tires were on the rear, and the vehicle subjected to transitional cornering maneuvers. These transitional maneuvers were of the type experienced by drivers performing fast lane changes on motorways or emergency avoidance maneuvers. For the purpose of these tests, therefore, it was decided to use the "Chicane" maneuver, which is well known and accepted in the car industry as a method of assessing vehicle stability under conditions of transitional cornering. This assessment is used by many leading vehicle manufacturers in the development of new vehicle and suspension designs.

The chicane course was set out as Figure 4 and for each test condition, three characteristic speeds were obtained. These are defined as follows:

1. Holding speed: the maximum speed at which the vehicle is able to negotiate the course without any awareness by the driver of any deviation of the vehicle from the prescribed course.
2. Trace speed: the minimum speed at which the driver becomes aware of deviation of vehicle from prescribed course, or change in vehicle characteristic.
3. Limit speed: the maximum speed at which the driver is able to negotiate the course without straying beyond its limits.

The driving was carried out by an experienced test driver to maintain consistency between each condition of vehicle fitment. In addition, the progression and the driver's observations during the progression were observed and noted. The progression is the speed range between the holding and limit speeds and is related to the warning time the driver has before impending vehicle instability. The results quoted in Figure 5 are for the car which is already in production on Denovo tires – the Rover 3500.

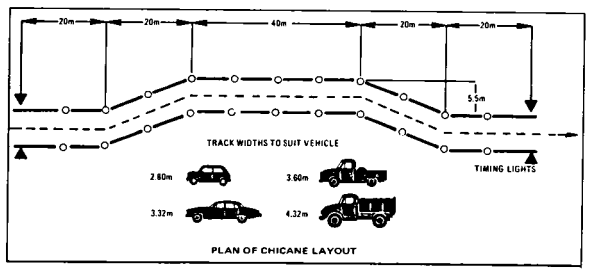


Figure 4

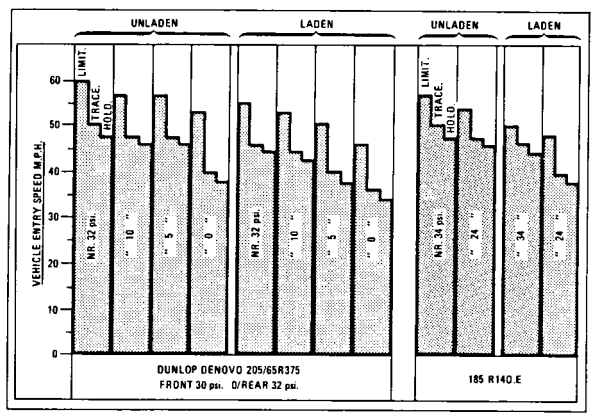


Figure 5

SNOW AND ICE TESTING

Winter testing was carried out in Norway using two Rover 3500s (one automatic, one manual gear box). The testing program was a three part one as follows:

1. Handling, traction and braking on an ice covered lake and driving up and down snow covered hills, covering both hard packed and loose snow.
2. Puncture sealing and vapor inflation assessment by running tires inflated and deflated in cold climatic conditions.
3. Endurance running deflated at low ambient temperatures.

Conclusions

1. Comparisons of Denovo and Rover O.E. snow tires were made with the Denovo tire performing very satisfactorily when tested against the Dunlop SP44 winter tire, which has for some years been widely used as a reference point.

2. No problems were encountered with the internal lubrication system operating at temperatures as low as -22°C. Vapor inflation and tire punctures sealed against the low reinflation pressure.
3. Some 30 tires were run successfully for 50 miles at 40 mph with the tire completely flat (valve out) and 100 miles at 50 mph in the vapor inflated condition.

FAILSAFE TESTING

To justify the failsafe claim for the Denovo tire, tests were carried out at high speeds up to 120 mph. The following failure circumstances were investigated.

1. A slow leak resulting in a low pressure tire failure.
2. An instantaneous tire burst (3/4 inch hole in the sidewall).
3. A partial tire break up at high speed when the driver either does not know, or chooses to ignore a flat tire condition.

The first step in proving the claim was the use of a drum type test with an anchored Rover 3500, front end running on a loaded drum at speeds up to the maximum speed of the vehicle. Having proved by observation and high speed cinephotography that when a tire failed at a very high speed (100 mph +) there was sufficient warning time, and the type of tire failure was not likely to cause vehicle instability, it was decided to extend tests to the next phase.

The next phase was carried out on a Rover 3500 at the MIRA high speed circuit at speeds up to 80 mph investigating the three tire failure modes as described above. In order successfully to monitor the correct impressions of both driver and observer during these tests, a tape recorder was used. Such comments as when vibrations started and differences in noise pitch, etc. were all accurately observed. Tires were put through these test procedures on different positions of the car and in every case although vibrations, noise and vapor smell became obvious in the car at advanced states of failure, the car was always capable of being brought satisfactorily to the side of the circuit, the driver having the car under directional control at all times.

Final proof testing to the claim "failsafe" was carried out on the Autostrada of Southern Italy. Again the three types of failure which could be encountered were tested over 40 times successfully up to the full speed capability of the vehicle which in this case was 120 mph.

In each case taped transcriptions were taken by the driver and observer, the tape being used to give them the necessary time to make as many comments as possible on the vibration and their characteristics, changes in noise levels and pitches and any other comments such as increased steering angle as the tire approached breakup and also, if the window was opened, the smell of the lubricant being vaporized off (Figure 6).

The procedures used to produce the three failure conditions as described previously were:

1. Slow leak: a steel valve cap was used with a small drilled hole, the tire pressure was increased before the test run commenced such that at 120 mph car speed, the tire pressure was 32 psi, the operating tire pressure of the car.
2. Instantaneous burst: a mechanical system using a 3/4 inch diameter rubber bung in the sidewall of the tire which was blown out at the speed and the position on the road required (Figure 7). Following the burst the car was brought to a satisfactory halt.
3. Instantaneous burst by system as used in (2) but the car was driven on until a complete tire failure occurred.

To establish a practical mobility figure for these high speed conditions, tires were burst at 100 mph and driven on until such warnings as extreme vibrations or noise levels occurred which could not be ignored by the driver. The car was then stopped, the tire visually inspected and then driven on at 40 mph. The distances achieved using this practical approach averaged 22 miles with a spread of 15 to 38 miles.

The main conclusions drawn from this testing program were that:

1. No hazards occurred following any burst at high speed and the driver had full directional control at all times.
2. There was some mobility following a high speed burst, particularly if the driver took notice of all the warnings that were available to him and he reduced his speed to 40 mph.

While in Southern Italy with ambient temperatures of +40°C, tire endurance testing was carried out at 50 miles at 40 mph completely flat, and 100 miles at 50 mph in the vapor inflated condition completely satisfactorily on 30 tires.

Field Test

A field test was started in December 1972 firstly on police patrol Ford Escort cars. This has led to more than 600 units being placed, involving many police forces throughout the U.K. No major problems have been experienced with these tires or with the divided type wheel. The divided wheel concept was deemed a marketing and servicing necessity, but has proved to be a very satisfactory solution to what at one time seemed an impossibility, i.e. a cheap, reliable, uniform and manufacturable wheel using present rolling and press tooling systems (Figure 8). A

few wheels have sustained impact damage, resulting in flange distortion, but the numbers involved with this sort of damage would have been expected from standard rims with this type of usage.

Of the Denovo tire/wheel units which have been fitted to police Ford Escorts size 185/60-13, 44 have been serviced and reissued with new tires fitted to the wheels. "Servicing" of the wheels has been confined to the normal "O" ring replacement and replacing the valve "O" ring where necessary.

Approximately 200 205/65-375 Rover units have been issued, of which to date, three have been serviced, retired and reissued.

TEST PROCEDURE

1. Tire fitted to the left front of the vehicle.
2. Tire inflated to 50 psi.
3. Stop at a convenient lay-by on the Autostrada.
4. Remove the valve core and fit a plastic valve cap with a 3/16 inch diameter hole drilled through the center.
5. Accelerate up to maximum speed.
6. Hold the speed until the driver, who had not been involved in any of the previous tests, thought that something was wrong.
7. Stop and the driver examine the tire and decide whether or not to continue at 40 mph.

Driver: J. Sanders (Rover Ltd)
Observers: R. A. Lyall (left front – Rover), J. M. Fitzgerald (left rear – Dunlop) M. C. Clatworthy (right rear – Dunlop)
Tire fitted to the left rear of the vehicle.

Distance – km	Comments
1.0	100 mph
1.3	105 mph
1.4	110 mph
4.9	115 mph slight vibration
6.1	Slight roughness
7.9	Stopped to inspect. Continue at 40 mph.
8.4	Stopped. Tire flat. Tire seems to be failed. Slight thumping. No characteristic flat running noise.
19.4	Rougher ride.
19.9	Slight thump under vehicle. Driver got out to look at tire. Looks OK externally. Decided to carry on. Yawing on acceleration up to 40 mph.
25.9	Appears to be getting rougher all the time.
29.9	Stop to examine the tire.
34.9	Increased vibration.
38.4	Pull increasing, having to steer 10° to left to keep the car in straight line. Gradually became more noticeable.
44.8	Slight increase in vibration. Driver stopping at first lay-by to look at tire.
45.1	Sidewall 170°, shoulder 180°C, bead 180°C. No outward appearance of failure.
50.0	Vibration increased. Yawing noticeable in the rear.
54.9	Vibration much worse.
56.0	Stopped to inspect tire. Tire appears the same externally as all previous stops. Smell observed. Carrying on.
57.2	Noticeably rougher than before.
58.1	Smell disappeared.
58.9	Smell returned. Vibration increased significantly and knocking.
62.3	Driver stopped to look at the tire because of change in noise level. Tire appears the same externally. Tire leaving mark on the road. Clicking noise still there and getting worse all the time.
64.3	Thumping increasing
64.6	Driver stopping.
70.0	Stop. Test car brought to a satisfactory halt at the side of the road under complete control.

Figure 6

More than 12 million miles have now been run on Denovo tires with a maximum individual divided wheel mileage of approximately 30,000 miles (excluding the tread wear test car wheel miles).

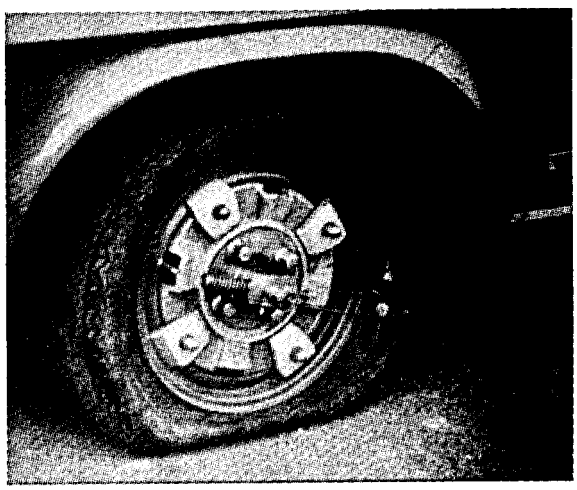


Figure 7

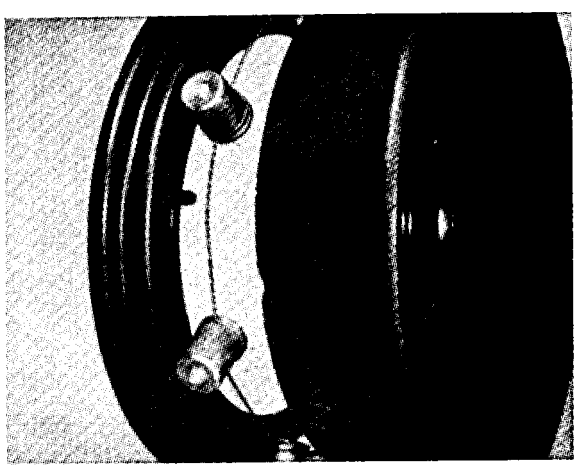


Figure 8

A number of police incidents have been filed both by individual police drivers to their superiors, and to the Dunlop coordination team, giving extremely encouraging results of Denovo tires being used in a general road usage as well as being used to their limit and beyond as far as the flat running speed and distance is concerned. These include more than 100 natural punctures, including three high speed blowouts and a puncture frequency of 20,000 miles plus. As further new Denovo sizes are developed, the field exercise will be expanded to cover them.

During the field tests carried out by the police forces all over the country, an independent test assessment was carried out by the A.A. This assessment included a considerable amount of both inflated running and partial deflation right down to the fully deflated state. The test procedures carried out and the results are included in a paper by the Chief Engineer of the Automobile Association at this Conference.

Two sets of Denovo units were fitted to a Ford Escort and a British Leyland Morris Marina in the RAC Rally of 1972. These Denovo tires had steel wheels and the forecast from experienced rally people was that steel wheels generally suffered considerably more damage than the normally used magnesium alloy wheels. This however, was proved not to be the case, as little or no damage at all occurred to the steel Denovo wheel. It is thought that a certain amount of protection was given to the wheel by the use of a wide low profile 65 series tire.

BURST DATA

In addition to the many bursts carried out to justify the failsafe claim, many thousands more have been carried out in all parts of the world, at both public and industry demonstrations. These bursts have been over the speed range 70-120 mph in straight line conditions. In addition, many hundreds of bursts have been carried out on 100 foot radius curves and simulated motorway lane changes. Finally, we are so confident of our product, we are now bursting two tires on the loaded side of the car in a curve and one on each side of a car at high speed in a straight line condition.

In addition to the 100+ natural punctures which have occurred during the field test, a number of others have been met both in inflated running tests on our test fleet plus some external outlets, and each one of these punctures has been logged – puncturing object, position on car, number of post puncture miles run, position of puncture in the tire and whether the tire was repaired and placed back in service for a continuation of its life. In every case so far reported, the car was driven on in the flat tire or vapor inflated state until the driver found it convenient to have the necessary repair carried out. One must remember that these are in addition to the many hundreds of induced punctures during test program where the sealant worked to give the resultant vapor inflation, or the tire ran in the completely flat condition for at least the minimum performance depending on which type of endurance test was being carried out.

STATISTICS

Over the past 2½ to 3 years the Dunlop Company has built up a massive bank of information relating to tire punctures; the results of this information have

helped us to determine what future development paths to pursue. The first survey carried out in early 1972 covered 2,528 motorists and the details are as follows:

1. Average man-hours lost per puncture before regaining mobility are 2.1. This surprisingly high figure involves driver and passenger time and includes time taken to seek and obtain assistance by those motorists who cannot change a wheel themselves. What is the direct cost of a delay of 2.1 hours? One could easily guess "£5" these days. 13% of motorists lost *more than 8 hours per puncture.*
2. Of high mileage motorists (15,000+ per year), 45% had had a puncture with their current car; frequency with these motorists is "one puncture each 17 months" – a very inconvenient figure.
3. Of all motorists interviewed, well over 50% said that a puncture is a "major nuisance." With women motorists the percentage is higher.
4. 40% of punctures were noticed when the car was in motion.
5. 35% of punctures occurred in the hours of darkness.
6. Over 75% of women drivers said they could not change a wheel following a puncture, not to mention spoiling the new dress, even if they tried.
7. Practically all punctures are caused by nails or similar objects.
8. In more than 25% of instances, the condition of the spare wheel is such that, if used, the journey could not be safely continued.

After a concentrated 2½ years working on the Denovo project, we have no reason to doubt these figures, but we have of course expanded our knowledge in other directions, for instance:

- Unlike seat belts, the Denovo tire is a passive safety feature, it calls for no effort on the part of the driver.
- If all car users drove on Denovo tires, we estimate that there would be an annual saving in U.K. of at least:

  44 deaths
  550 serious injuries
  1540 slight injuries

  and a not inconsiderable reduction in "damage only" accidents. The total annual saving to the community would be about £2M.

It must be admitted at once that these figures represent a very small proportion of the total road accidents. In fact they amount to about an average of 2 days. The saving would not seem to justify the effort (unless of course you happen to be one of the 2,000 or so dead or injured), but when we compare these figures with the results from other road safety measures, we find that in terms of cost effectiveness, the Denovo tire is probably one of the best safety features of all.

It can be established by analyzing our field service data relating to post puncture flat running, and using an analysis combining motorists' journey habits with the Denovo tire's mobility specifications, that there is a virtual 100% guarantee to the motorist of reaching his journey's end or a point at which he can have his tire reserviced. This can be done without the hazard of wheel changing in the dark or on a heavily used road, and that he can do this in full control of his vehicle at all times.

Deflation Warning Devices (DWD)

During our Denovo project development, the subject of DWD has been very much in our minds and in fact we have had a separate development program investigating a number of ideas. However, our analyses of a range of puncture happenings at various speeds has shown that while a DWD may be desirable from an economic point of view, it is not essential with the failsafe performance of the Denovo tire. We are continuing to work in this field as are many other organizations, and because of its even higher desirability for the present tire situation, it is hoped that some satisfactory solution will be found. However, we feel that it will not be in the very low cost area. One very important point that has to be established with any DWD is the fallibility of the device itself.

Inflated Performance

While the investigations were being carried out testing the Denovo tire in all its flat tire running modes, the inflated tire performance, which over the years we have all come to know as extremely reliable and predictable in all its usages, had to be checked and rechecked whenever any alteration was made to the tire construction, in order to enhance the run flat capability. Such testing as inflated endurance running and tread wear had to be checked; these features, even with the modifications required to give the run flat capability, had to be equal or superior to the O.E. equipment it was designed to replace. This was proven in all cases for the Denovo tire.

Because the Rover 3500 car has a high speed requirement of 125 mph we had to meet this, along with the wet grip requirements, using a low profile tire which was part of the concept of the Denovo run flat design. In the case of the Rover Denovo radial tire, the design concepts had to be such as to ensure that the cornering force generated was superior to the 185-14 O.E. tire it was replacing, although the self

aligning torque at low speed was somewhat higher. The rolling resistance of the Denovo construction of large sections is very much the same as the O.E. tire it is replacing, but in the smaller sections there is a distinct advantage to the Denovo tire.

Of course throughout the whole of this development phase, we have been closely involved with the car industry project engineers and their acceptability tests have to be met as is the case with any new tire, be it Denovo "failsafe" or any other.

SERVICING

In order to assess the fitting and stripping of the two-piece Denovo rim in the field, an exercise involving six depot visits throughout England and Scotland was carried out. This exercise consisted of presenting a fully assembled Denovo unit to an inexperienced fitter at each of the six depots and without any technical instruction from observers, they were asked to strip the two-piece Denovo wheel and remove the tire. The lubrication system which had been actuated was removed, the tire washed out with water as the lubrication solution is water soluble, and a puncture repair carried out in the normal radial tire manner. The tire was then reassembled, including the lubrication system and the whole operation was timed at between 25 and 35 minutes depending on what little skill the fitter had. Emphasis must be placed on the fact that the fitters chosen to carry out this exercise were very inexperienced indeed, and this was the first Denovo unit seen by any of them. A skilled fitter was timed at 10 to 15 minutes. This compares with some 10 minutes for a standard tire stripping, repairing and refitting exercise.

A major fact emerging from this exercise was that in no case was any damage caused to the tire, particularly in the bead area when stripping and fitting a Denovo unit. This is certainly not so with a standard tire fitment as all too frequently considerable damage is caused to the beads during the stripping and fitting, sometimes making the tire unuseable as a tubeless unit.

Tests have been carried out, in which the wheel flange nuts were deliberately loosened to check the air leakage results, bearing in mind the safety aspect. This simulated the condition of an operator failing to remove the valve core or inadevertently slackening the wheel flange nuts instead of the wheel fixing stud nuts. He would have great difficulty in doing this however, because the wheel flange nuts are on the rear side of the wheel when it is on the car. Under these tests, the air leaked immediately and the tire deflated but the wheel remained adequately secure to support the load.

Although servicing of the Dunlop Denovo unit is considered straightforward, and can be carried out by tire fitters with little experience, it was felt necessary to have a Denovo Service Network throughout the U.K. and Europe so that a first class service would be available for Denovo motorists. We therefore designated certain service and tire distributor centers as specialists for the servicing and sale of Dunlop Denovo tires. Centers so designated have been appointed Dunlop Denovo Dealers and are required by a charter to maintain certain stocks of Denovo tires, wheels and accessories for servicing the product.

The spread of Denovo dealers throughout the U.K. totals 600 plus, giving a very good coverage. There is a heavy concentration of service outlets in urban areas. In Greater London, there are over 50 appointed Denovo dealers.

Travelling distance from any puncture point to a service outlet in England will not be much in excess of 20 miles.

A training program for dealer tire fitters was arranged through the regions and one tire fitter from each Denovo dealer attended a one-day course at convenient venues. The training course of instruction included tire fitting and removing, permanent repair and temporary repairs. This enabled each fitter to have first hand experience of the Denovo unit and those who satisfactorily completed the one day's course received a certificate of training.

The U.K. service network will, in the main, only operate during normal working hours. However, there is a back up emergency temporary tire repair service given by the RAC and AA. They will carry out a blue plug temporary repair which will allow the Denovo motorist to drive on at normal speeds. Considerable testing of temporary plug systems have been carried out on the Denovo product and this, plus the "failsafe" end result if the plug does come out, allows the user to carry on at the Denovo specified speed. The temporary repair should be replaced by a permanent repair as soon as possible, within a maximum of 500 miles.

If any Denovo motorist is in doubt about how to use any of the Denovo services, there is a National Freefone available at any time. The Freefone number is published in the car brochure. Advice can be obtained on how the motorist may resolve a particular Denovo problem.

For Continental touring, a European Touring Service Network has been arranged and covers Western Europe through the Dunlop Organization. The European Denovo dealers will provide service at distances of 50-100 miles. Naturally, when Denovo is fitted to locally produced European vehicles, the

service support level will be made equivalent to the U.K. scheme.

As an additional back up service for motorists touring in Western Europe, the AA has included Denovo units in their 5 Star Travel Service.

TOTAL SUMMARY

It is now widely recognized that the Denovo concept has created a completely new situation in tire technology and usage. There are significant gains in overall safety and user convenience. Additionally, vehicles with more efficient space utilization can now be designed, without the need for a fifth wheel. A massive amount of company and independent testing has fully authenticated the claims made for the Denovo product, including those in the vital field of automobile safety.

FACTORS LEADING TO LOSS OF CONTROL – A GUIDE FROM ACCIDENT INVESTIGATIONS

G. JONES
Manager, Safety Engineering
British Leyland Motor Corporation Limited

INTRODUCTION

In making design changes to have a real benefit in accident avoidance it is of vital importance to have first the evidence to show what is wanted. Endless discussions and arbitrary tests which are not based on basic evidence can be useless, and accident information is the only source from which this evidence can be obtained. The writer, being a member of the Austin-Morris Division of British Leyland, has for many years been involved in passenger car development, and investigation of accidents has, during the past ten years, been an important part of his duties. These were in many cases initiated by an approach from the police or claims by experts that there was a defect which led to loss of control.

Many of the investigations have involved careful analysis to determine the true cause by examination of the vehicles involved, the site of the accident and various accident documents. In some cases it has been necessary to carry out extensive tests to establish the facts where this has been possible.

In the analysis of accidents involving loss of control there is a difficult interaction of human, environmental and vehicle factors.

It is hoped that the information contained in this paper will help to show where the emphasis is needed in passenger car developments.

THE OVERALL SITUATION

An on-the-spot survey carried out by the Transport and Road Research Laboratory has shown that in 996 accidents in which a car was involved there were 319 accidents involving car loss of control and 321 cars in which loss of control occurred. Loss of control therefore occurred in 32% of the accidents. This survey was carried out in an area having a bias towards rural roads. TRRL statistics also show (Reference 1) that in Great Britain the number of cars in use in 1971 (assuming that all registered cars are in use) was 12,056,000 and in the same year there were 256,950 cars involved in injury accidents.

From earlier statistics (Reference 2) it is estimated that the number of injury accidents is about 80% of the number of cars involved in injury accidents and so we can say that in 1971 there were approximately 205,000 injury accidents in which at least one car was involved.

Relating the survey figures to the overall figures we therefore note that annually the car-involved injury accidents are approximately 1.7% of the cars in use and loss of control accidents is therefore about 32% of these, i.e., about 0.5%. It is of interest to ask the question – why, of 12,056,000 cars travelling on our roadways should 0.5% of them be involved in a loss of control accident at, in most cases, one particular instant during the year of motoring?

VEHICLE DEFECTS

Referring again to TRRL on-the-spot data, leaflet LF 374 (Reference 3) quotes 130 cars with contributory defects in a total of 996 accidents involving cars. Assuming that there was only one car with a contributory defect (or contributory defects), in cases where two or more cars were involved, the ratio of accidents with contributory car defects to accidents involving cars is 130 in 996 or 13%. More than one defect was noted for some vehicles and the breakdown shows that tire defects account for the major proportion with foot brake defects next, followed by mechanical failure and steering defects.

The number of accidents involving cars with a contributory defect represents approximately 0.17% of the cars in use.

EXPERIENCE IN AUSTIN-MORRIS DIVISION OF BLMC

In the more serious accidents there occasionally occur failures which may appear evidence as being apparently responsible for, or an important factor in causing an accident. Some of the cases we have investigated have involved such litigation, others have come to our attention by an approach from the police in cases where the cause of a particular accident has been uncertain or where the conditions were thought to be of direct concern to the manufacturer.

The cases with which we have been concerned have therefore not resulted from obvious defects such as brake failure following lack of maintenance or from obvious tire defects, and so our sample may be biased because of this. The sample is also biased in terms of the particular car models concerned because impact failures occur on some and not on others.

However, an analysis of these accidents indicates attention which could be a benefit in reducing the incidence of loss of control.

Table 1 shows 60 loss of control accidents grouped according to the significant factors determined in

Table 1 CONTRIBUTORY FACTORS INDICATED IN 60 LOSS OF CONTROL ACCIDENTS	
Human and Environmental Factors only	18
Braking or Traction on Dry or Wet Road Surfaces (Including Cases of Brake Defect)	15
Ice on Road Surface	4
Swerve – or Avoidance Maneuver – without Evidence of Braking	8
Tire Defects	9
Maintenance Defect other than Brakes or Tires	3
Mechanical Failure	3

each case. In 18 cases there was no evidence to show that an improvement in stability would have been beneficial or that there was any contributory defect.

Human and environmental factors result in loss of control whatever vehicle characteristics are provided but in some accidents where excessive speed is indicated it is possible that feel characteristics could more adequately impress drivers of their own fallibility. Possibly for these reasons cars having good steady state cornering capabilities and good swervability in normal test schedules do not appear to have better accident records than some older cars which, on present day standards, are out of date but which tended to inhibit self-confidence. Although the degree of influence is uncertain vehicle factors which could have the most significant influence within the group examined are braking, response to swerve or avoidance maneuvers, and tire defects. Mechanical and maintenance defects do not figure prominently and though in many cases a defect was claimed to be responsible for loss of control the damage experienced was in most cases a consequence of the accident and not the cause.

THE INFLUENCE OF BRAKING ON TRACTION

In fifteen accidents with evidence of brake application in ten and an indication of braking in two, there were seven in which the car deviated to one side as occurs when the rear wheels lock before the fronts.

There was evidence of braking in six of these seven cases, four of which occurred on wet roads and two on dry roads. In all except one of the seven the cars were in a light load condition with either driver only or driver plus one passenger.

In the remaining case the car was fully laden.

In two further cases the evidence showed front wheel slide due to braking when approaching a bend on a wet down gradient. In another two cases on wet surfaces the evidence in one was not sufficient to determine whether the drift resulted from front wheel slide due to braking or traction, but in the other case the loss of control followed front wheel slide due to traction.

In these four cases of front wheel slide the driver only was in the car. In one case loss of control resulted from traction on rear wheels causing slideslip on a wet surface.

In the remaining three cases one was grossly overloaded and lost control as a result of inefficient braking, one had contaminated friction material on one of the front brakes and the other lost efficiency due to water.

In the above mentioned accidents the consequences of rear wheels locking before the fronts were far more serious than with front wheel slide mainly because of movement into the path of approaching traffic and because occupants in side impact have less protection than in frontal impact.

Figure 1 illustrates the motion resulting from early rear wheel lock when braking in a turn, this being a reproduction from a previous paper by the writer (Reference 4).

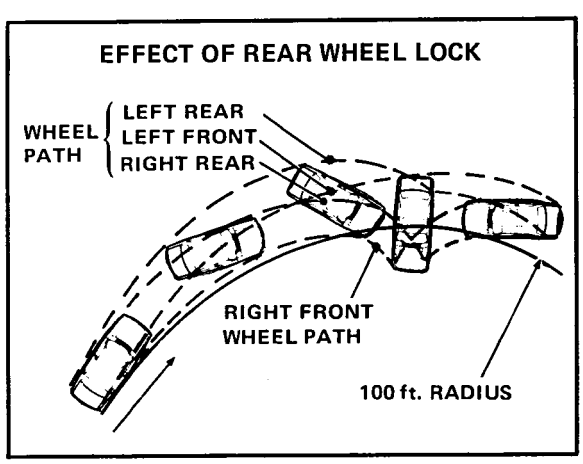


Figure 1

With present brake systems which in theory approach maximum use of the available adhesion, and which do not have an antilock, system, the variations which occur in practice are such as to make it extremely difficult to avoid a condition in which the rear wheels lock first in some situations.

The alternative with normal brake systems is less adhesion utilization and a greater incidence of front wheel lock resulting in accidents in which the car fails to negotiate a bend in the road. Figure 2 is an

example of the type of roadside obstacle sometimes struck in this type of accident.

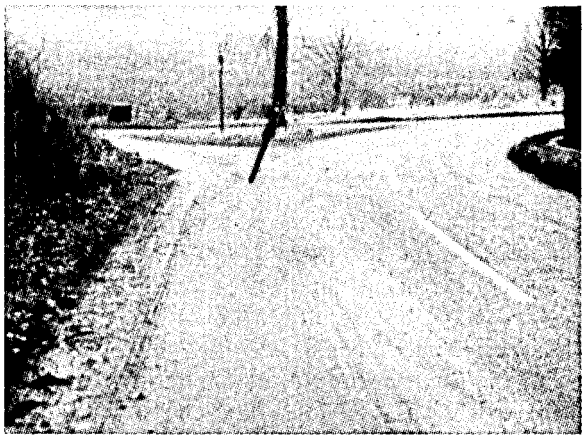


Figure 2

From the evidence of our investigations it would appear that a braking system which can ensure the avoidance of premature rear wheel lock and at the same time ensure maximum adhesion utilization has much to commend it providing there is no inducement to bolder driving habits.

LOSS OF CONTROL ON ICY SURFACES

There were four cases of loss of control on icy surfaces but with no evidence of braking. On such surfaces it is uncertain as to whether even a full antilock system would have been beneficial. In one case the consequences would most probably have been fatal if seat belts had not been worn. In the other three cases there was negligible injury.

COST-EFFECTIVENESS CONSIDERATIONS

In 1971 there were 3,000 car occupants killed and 39,602 seriously injured in Great Britain. During the same period, accidents involving car loss of control were about 32% of car involved injury accidents. On the basis of our observations in 60 loss of control accidents braking was involved and could be a contributory factor in 25% of these.

25% of 32% then gives us 8% as an approximate figure for loss of control accidents involving braking as a percentage of car involved injury accidents. If by changes in brake system performance this 8% of accidents could be avoided there is a possible saving of about 240 lives and 3,170 seriously injured per year.

Assuming that all cars in use would require such a change in brake system performance a total of 12 million cars would require the changes to achieve this saving in lives and injuries.

The range of injuries under the classification "Seriously Injured" is so wide that any attempt at cost benefit based on figures for the number of injured can have a very uncertain meaning. If however we make a broad assumption that the number of permanently disabled is equal to the number of fatalities we have a figure of 480 lives and permanent disabilities possibly saved per year in a car population of 12 million. This is a ratio of 1 to 25,000 and so for every 100,000 units of cost one could relate to a fatality or permanent disability the cost per car would be 4 cost units per year of car life. Such figures do not however indicate the amount which the average motorist can be expected, or is prepared, to pay in order to provide greater security against loss of control.

The provision of antiskid, even for rear wheels only, can be expected to add considerably to the cost of a braking system and it is questionable whether the average motorist would purchase it if given the option. European legislation is aiming to optimize front to rear balance on a theoretical basis but this will not ensure success in practice. The most suitable compromise in practice, short of fiting rear antilock, appears to be one of sacrificing adhesion utilization at the rear wheels at the expense of a possible increase in accidents involving front wheel lock.

LOSS OF CONTROL WITHOUT BRAKING

In seventeen accidents in which there was no evidence of braking it was found that there were nine in which tire defects were a contributory factor. Four were rear tire deflations, three front deflations, one with 0.5 mm of tread on one of the front tires, and one with radial front and cross-ply rear tires. Whether it is a front or rear tire deflation which occurs the consequences can be serious and there is no doubt of the benefit to be obtained from avoidance of complete deflation and its effects.

There is also a great benefit to be obtained by ensuring that tire pressure distribution is correctly maintained so that unexpected steering response is avoided. In a number of the accidents within this group the cars were in the more heavily laden condition, a condition in which additional load at the rear tends towards oversteer. In some cases in which there was evidence of a swerve or avoidance maneuver it is apparent that loss of control is associated with the phase lag between steering movement and response. This was illustrated by the writer in a paper presented at the 3rd ESV Conference in Washington (Reference 5), a drastic change in tire pressures towards oversteer having the effect of very much accentuating the phase lag. Figure 3 is a reproduction of results illustrating this effect. Note that the period

of yaw velocity response is in this case very similar to the steering input period, and response lags behind input by about 40 degrees. The more important

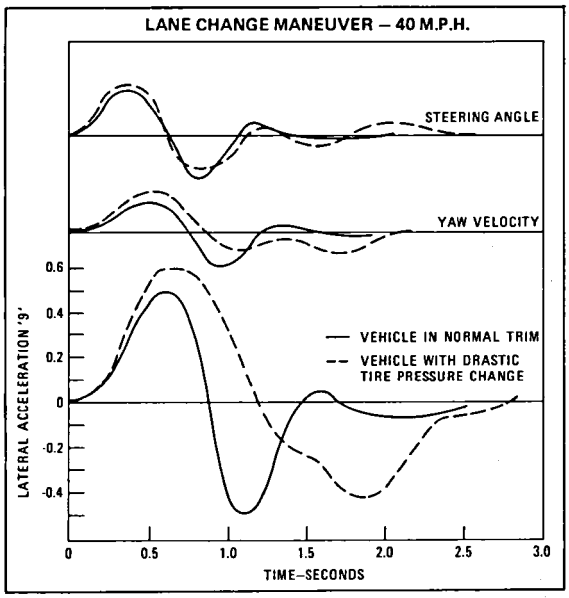


Figure 3

derivative which is yaw angle is 90 degrees out of phase with velocity and so the lag between displacement and steering input is about 130 degrees. It is desirable to have a bias towards understeer to reduce the lag, and in some cases it is necessary to have rear tire pressures higher than is necessary for comfort and load bearing so that there is a margin to allow for pressure neglect.

An illustration Figure 4 derived from Den Hartog's "Mechanical Vibrations" (Reference 6) also serves to show that one should in theory design for a yaw frequency outside the range which can be experienced in practice, and that yaw damping is important when, as normally occurs with sedan vehicles, this is impracticable. This applies particularly to the larger amplitudes of oscillation applicable to accident situations.

From research carried out at the Cornell Laboratory some years ago (Reference 7) we know that transient response damping increases as the conditions approach oversteer.

In discussion on their work one of the authors, Mr. Leonard Segel, also explained that when the human element is used to close the loop with an oversteering car the result can be a divergent oscillation. This latter condition is one which must obviously be avoided without having an alternative problem of a poorly damped transient response. Although the significance of transient response in relation to accident situations appears to be unknown there are instances of motion involving driver reaction.

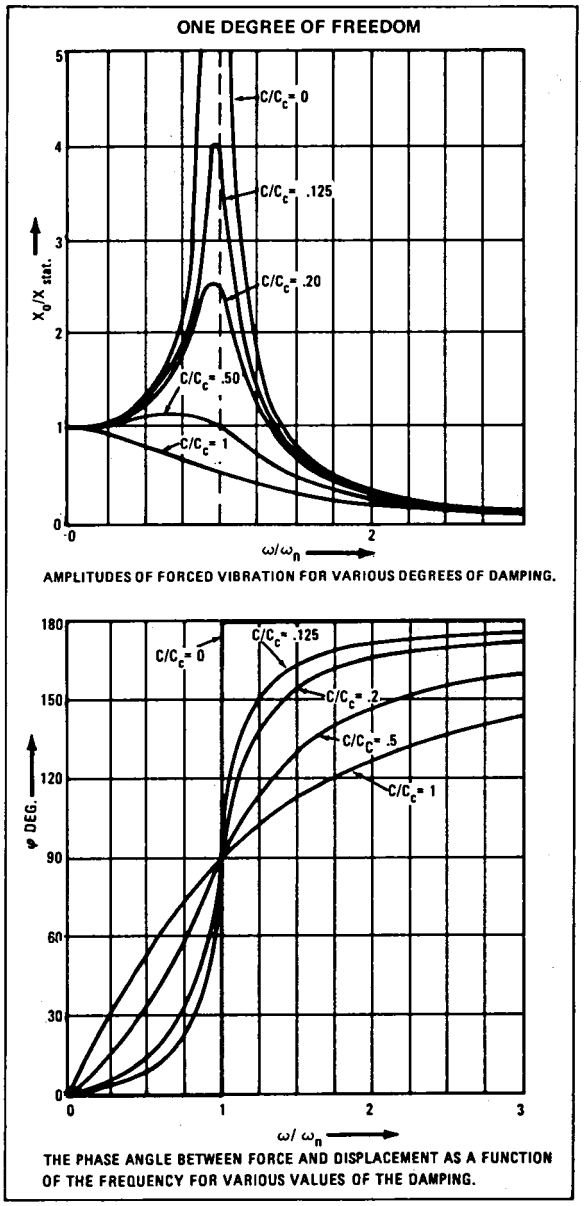


Figure 4

Figure 5 shows evidence of a vehicle having oscillated in yaw when coming out of a left hand bend, the base of the photograph being the direction in which it was travelling. Normal motion would follow a path on the right hand side of the photograph, that is driving on the left hand side of the road as applies in the United Kingdom. The bend is one which an experienced driver could normally negotiate at about 55 miles per hour, but would consider 60 miles per hour as being rather too fast in the circumstances. From measurements of the marking it is determined that the peak lateral

accelerations over the above mentioned speed range would be 0.8 g to 0.95 g with a yaw frequency of 0.4 to 0.45 cps.

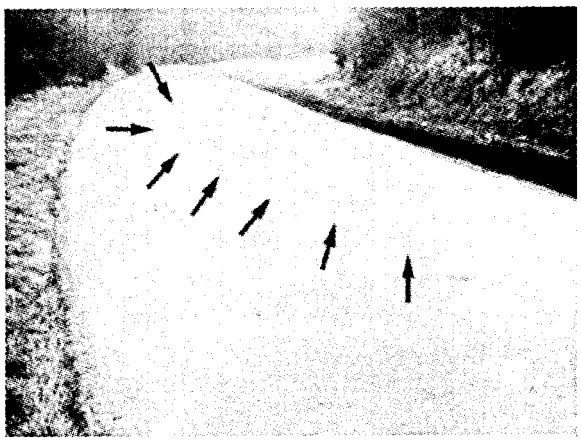


Figure 5

Figure 6 is an accident scene in which a similar form of yaw motion occurred at the exit from a bend, the vehicle striking the verge and overturning prior to coming to rest on its wheels. Figure 7 is a plan of the scene, the markings indicating a motion in which extreme lateral accelerations are involved, and in which phase lag could result in a loss of coordination between driver and machine.

ESV SPECIFICATION AND DEVELOPMENTS

The specification for U.K. Demonstration Safety Cars requires the cars to incorporate the best of current practice on a cost benefit basis. At the same time the test specifications are such that they cannot be satisfied with anything less than a full antilock braking system. Such a system is necessary to satisfy the requirements on split coefficient surfaces, braking in a curve on a wet surface of very low friction coefficient, and braking during a lane change on a wet surface of very low friction coefficient.

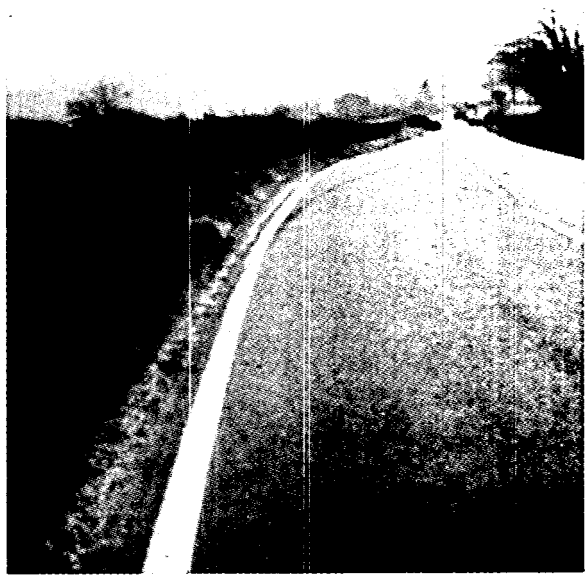


Figure 6

The indications from our investigations are that tests on the very low friction coefficient may not be representative of the majority of accidents involving braking, and tire/road adhesion coefficients of about 0.3 and upwards should be our primary concern in the United Kingdom. Rear antilock together with good adhesion utilization are desirable technical aims if reliability and cost/benefit levels become acceptable. Irrespective of cost benefit considerations full antilock systems have not, in our opinion,

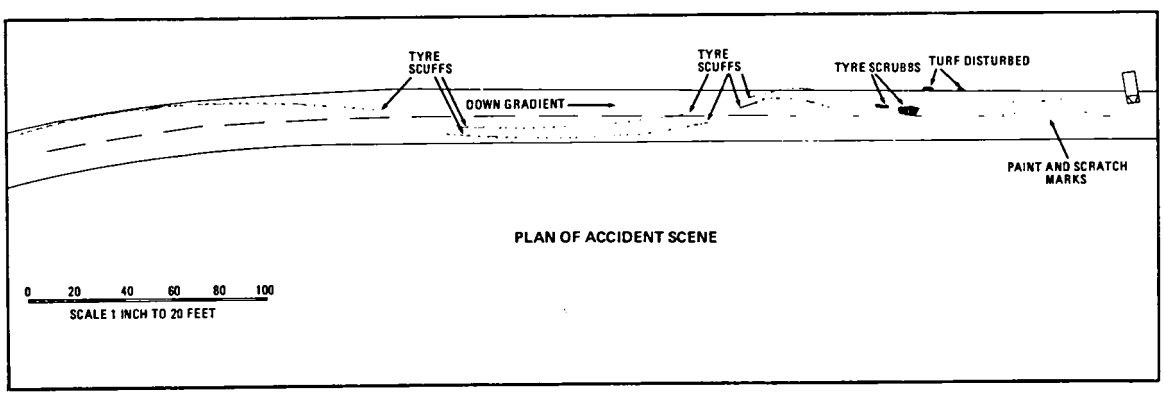


Figure 7

reached the stage of being suitable for practical application to small cars.

In the specification on handling the intention is to avoid a condition in which a driver has to cope with a sudden and unexpected change in behavior.

To this end a very important contribution is provided by wheel and tire assemblies which avoid major displacement of the tread relative to the wheel when sudden deflation occurs.

As illustrated in the text the effects of changes in tire pressure distribution are also of great importance.

REFERENCES

1. Road Accidents in Great Britain – 1971, Published by Her Majesty's Stationery Office.
2. Road Accidents in Great Britain – 1966, Published by Her Majesty's Stationery Office.
3. Transport and Road Research Laboratory Leaflet LF 374 – June 1973.
4. The Skidding Behavior of Motor Vehicles – Automobile Division Proceedings of The Institution of Mechanical Engineers 1962/63.
5. Report on Third International Technical Conference on Experimental Safety Vehicles – Washington, May 30 thru June 2, 1972.
6. Mechanical Vibrations, by Den Hartog. McGraw-Hill Book Company.
7. Research in Automobile Stability and Control and in Tire Performance – Automobile Division Proceedings of The Institution of Mechanical Engineers, 1956/57.

DRIVERS' LIMITATIONS

H. A. J. PRENTICE
Transport and Road Research Laboratory

INTRODUCTION

The ESV program to date has been mainly concerned with improving the vehicle structure to provide an envelope giving the maximum protection to its occupants in the event of an accident. Clearly the idea of protecting vehicle occupants as much as possible is sound common sense, nevertheless even with safer structures there will remain some risk of injury. Some disquiet has been expressed that the introduction of intrinsically safer road vehicles will lead to a reduction in the emphasis on accident avoidance. This might lead to an increase in damage only accidents, the material cost of which could be considerable; it might result in more accidents overall and thus nullify the benefit from vehicle structural improvements. Most progress is likely to be made therefore by reducing accident numbers as well as improving the safety of the vehicle itself.

Accident studies have concluded that the primary causative factor is driver misjudgment. However it is very difficult to arrive at the precise extent to which any particular aspect of driver performance is responsible for a particular accident. Such studies indicate various broad areas in which errors of judgment occur. These include excessive speed for the particular situation, incorrect overtaking and too close following.

CONTROL SYSTEM

The driver of a vehicle is an element in the man-machine system. He is the prime element as far as choice of action is concerned. He gathers information from his surroundings and from the performance of his machine. He then uses this to make decisions on the course of action he should take. For the greater proportion of the time the amount of information can be handled without any difficulty. However, when a driver is required to perform several different functions at the same time the load on him increases considerably, greatly increasing the probability of his making errors. There is a point at which his ability to assimilate and correctly analyze the information he is receiving breaks down.

In view of this we need to examine the abilities of drivers which are essential for them to reach the correct decisions in particular circumstances.

PERCEPTUAL JUDGMENTS

Accurate perceptual judgments are essential if correct maneuvering decisions are to be made in particular circumstances. Such judgments must be based on visual information present in the environment.

Various studies have indicated that the apparent cause of nearly 70% of right-turning accidents was due to misperception of the situation. In addition, perceptual misjudgments contributed to nearly half the overtaking accidents.

Three related perceptual judgments are continually being made by all road users:

1. The judgment of distance to vehicles or pedestrians on the roadway, or to specific features of the environment, for example, the distance across the road or the distance to a bend or junction.
2. The judgment of the speed of approach of oncoming vehicles or pedestrians or of the vehicles ahead, and the relationship between an individual's movement and his environment.
3. The comprehension of the time required to complete a maneuver. This particularly applies to pedestrians crossing in heavy traffic.

From the decisions that are made on distance and speed as in the above a driver presumably makes an estimate of the time available to carry out a particular maneuver which is then compared with his estimate of the time he thinks he needs to carry out that action based on his past experience and his vehicle's performance.

Clearly, judgments of speed and distance are fundamental to achieving correct decisions.

The experimental studies which have been conducted in this field of work have shown that road users lack the inbuilt ability to make the more critical of these decisions with the precision required.

If the overtaking role as a particular situation is considered then all of the factors connected with the road in front have to be assessed, the layout of the road, the vehicle type being overtaken and an estimate of its likely speed. Experiments suggest that the judgment of the speed of an oncoming vehicle is very strongly influenced by the speed of the vehicle which is being overtaken. In effect the driver expects a certain situation. However, if very fast or very slow moving vehicles are involved it is not until quite late on in the maneuver that the message from the visual perceptual system is sufficiently strong to overcome the driver's expectancy.

This suggests that aids in or on the vehicle or on the roadway to assist in the judgments of speed and distance would have a considerable potential to reduce accidents. Already fast moving public service vehicles, e.g. police vehicles, ambulances and fire engines are indicated as such by means of flashing lights. Also in some countries slow moving vehicles must bear special warning signs.

The problem would then appear to resolve itself into determining the limits of the abilities of people to accurately assess the varying parameters of the environment through which they are moving. Having understood these limitations the need is then to develop driving aids which supplement the deficiencies or if necessary even take away the freedom of choice of the individual to carry out certain maneuvers.

The decisions as to which equipment can be made available depend then to a large extent on the cost of the device or system as against its success in accident reduction.

The following sections give details of some devices currently under investigation in the U.K., both for use on vehicles and in the environment, and makes some suggestions as to areas in which research might be undertaken to give profitable results.

Radar and Related Systems

One approach to the problem of safe following of vehicles at high speeds and short headways is to use radar or a related system. These techniques are being investigated in many countries besides the United Kingdom. In the first instance it is likely that such a headway measuring system will be used as an aid to drivers, to warn of too close following or of brake application by a vehicle ahead, and perhaps to tighten seat belts before a collision. After adequate operational experience the system could be extended to automatic control of throttle and brakes.

The ideal headway measuring system should be: (a) self contained, i.e. not requiring "cooperation" from the vehicle ahead or the road, (b) able to operate in all weathers, (c) able to detect and respond quickly to sudden brake application by the vehicle ahead, (d) fail-safe, (e) able to detect road obstacles as well as vehicles, (f) cheap and reliable with few false alarms, (g) able to handle lane changing and merging as well as simple following. No system known at present can meet all these requirements, but an obvious starting point is forward facing primary radar. This has many attractions, being self-contained, potentially cheap and easily introduced. However the great accuracy of range of relative velocity measurement needed for quick detection of brake application by a vehicle ahead is very difficult to achieve under real road conditions, cluttered as these are with roadside furniture and multiple vehicle targets. A high frequency (e.g. 30 GHz) is needed to give a narrow enough beam from an acceptably small antenna, and this increases the difficulties of making a radar work reliably in the road vehicle environment which presents a most demanding combination of vibration, temperature changes, electrical noise, and exposure to the elements. Trials carried out at TRRL with 10 GHz Frequency Modulated Continuous Wave (FMCW) sets have not been very successful as they failed to measure ranges accurately on the road. A 34 GHz FMCW set using flat printed aerials is being developed, but there are some problems of reliability which remain to be investigated.

Research into ways of overcoming these difficulties includes studies of (a) radars using passive frequency changing transponders on the target vehicles; (b) an active cooperative telemetry system.

A passive transponder system using a frequency doubler has been demonstrated by Radio Corporation of America. It offers the great advantages of virtually 100% clutter rejection and alleviation of the primary radar's difficulties when following, for example the case of a very small car which is following a much larger vehicle. Its drawbacks seem to be (a) complexity and lower reliability, (b) need for spectrum space in two widely separated frequency bands, (c) need for all vehicles to be fitted with correctly working passive transponders. It also does not avoid the need for high range or velocity resolution to detect brake application by a vehicle ahead, i.e. it does not provide the equivalent of brake lights.

A telemetry system has potential benefits which may justify its extra complexities. Its principle is that each vehicle measures its own position with respect to the road and transmits this to all nearby vehicles. Each vehicle now knows where all other vehicles in the vicinity are and from this can derive a speed command voltage to control throttle and brakes to maintain its correct position. The advantages of this approach are that it can be used to deal with merging, crossing, and lane changing, which are beyond the capabilities of other systems. It should also allow following at very short headways without requiring very accurate range and relative velocity measurements.

Electronic Speed Control

It will clearly be some years before an electronic aid for headway control which would also allow control of vehicle speed could be in use on the roads

on all vehicles. Therefore, it is worth considering the possibility of preventing accidents in urban and other areas by limiting vehicle speeds by remote control. Doing this automatically has the merit of flexibility in that the speed imposed at any time could depend on traffic conditions, so that the introduction of such a system would probably be much more acceptable to drivers. Electronic systems to do this were proposed in the 1950s and are regularly reinvented. Although simple in principle they raise the questions of safety of operation and cost-effectiveness. The usual proposal is for a system of roadside units transmitting maximum speed commands to passing vehicles, for example, from short inductive cable loops or from cables laid continuously along the road. These signals are received by the vehicle and are compared with its actual speed. The throttle and brake are then controlled to prevent the commanded speed being exceeded. The need for cheap vehicle equipment immediately suggests "on-off" throttle actuation. However, if a tightly packed queue of vehicles all fitted with on-off throttle actuators is considered, each having different acceleration and deceleration rates and slightly different speed settings, the possibility of collisions is great. Proportional control of throttle position and perhaps braking therefore seem essential. Although this would make the vehicle equipment more expensive it has the advantage of allowing a "cruise control" or "speed hold" mode at no extra cost, increasing its value to the driver.

This area of electronic aids needing both roadside and vehicle equipment is one in which international cooperation during development is essential so that one system may be introduced on an international basis avoiding the difficulties that have arisen in the past on many technical items.

The kind of system that has been discussed above would enable automatic control of vehicles to be taken in the appropriate circumstances and yet still allow a very great degree of flexibility and freedom to the ordinary driver. The following items are devices which can only act as aids, or in other words can only give information to the driver as to what his actions should be.

Head-Up Display Speedometer

It is generally accepted that speed in excess of that suitable for the prevailing situation is one of the primary causes of accidents.

Experiments have shown that drivers used their speedometers very infrequently. It has also been found that drivers rely upon sensory information instead of their speedometer during maneuvers, or in traffic when making speed adjustment. In addition drivers are unable to make accurate visual estimates of their speed. A further factor is that time which is taken to read an ordinary speedometer. Experiments have shown that the average time required is 1.6 seconds and this is probably the reason why in heavy traffic or during difficult maneuvers the drivers do not look at their speedometers. If therefore it were possible to present vehicle speed to the driver without requiring him to look away from the view ahead then this information could be used much more readily. The information could then be obtained by a momentary change of view instead of requiring the driver to take his whole attention away from the road.

Prototype Head-Up Display (HUD) units have now been manufactured and a number of them have been undergoing extensive road trials.

The principle of operation of such a HUD system is relatively simple. Light from a rotating disc or electronic sequence of numbers is projected upwards onto the windshield through a lens system so that after reflection into the driver's eyes the images appear at a distance ahead of him. The image brightness may either be controlled automatically using photocells or manually by the driver according to his preference.

One problem that has become apparent during the trials is that the maximum brightness available is insufficient for viewing against snow, low sun, etc. The other problem which has arisen is that of height adjustment for individual drivers. The setting of the light reflected from the windshield is chosen in accordance with the standard eye position ellipse currently used in vehicle design. This apparently is not satisfactory and the range of positions will need to be extended.

The question very frequently raised is whether the speed information received in this manner is a distraction. Experience with this equipment shows that after a few days the novelty wears off and the information is taken in virtually unconsciously. An interesting point is that the driver is immediately aware of losing his speed display. This can occur when driving into the setting sun when he would of course experience great difficulty in reading a conventional instrument.

If HUD systems were adopted for use in large numbers of vehicles other advantages would follow. For example, it would no longer be necessary to have all instruments like oil pressure gauge, water level, engine temperature, etc., high up on the dashboard of the vehicle. As readings varied from preset limits a warning light would be projected through the display to alert the driver to consult his instruments.

Station Keeping Indicator

Observations of traffic on many roads have shown that vehicles tend to travel too close together for safety. One set of measurements showed that over half of all vehicles were travelling at less than 1 second interval from the vehicle in front. A number of factors are involved in such situations. One is the inability of drivers to judge correctly the following distance they require for safe travel for particular speeds and road conditions. It is also the case that the relative change in size of a vehicle to an observer must be above a certain minimum level before he is able to detect that the vehicle is closing or going away from him. Thus a driver tends to travel at a distance from the vehicle in front where its size is such that he is capable of quickly detecting changes in its distance from him. At high speeds the optimum distance for detecting the onset of lead vehicle braking may be less than that necessary to allow the following driver to react in time.

The problem then is to devise a simple instrument which enables the distance to be kept from the vehicle ahead to be shown as a function of the following vehicle's speed. This requires an indicating device which can change its gauging distance continuously as a function of the driver's own speed. In addition it is important for the driver to be able to see the display superimposed on his view of the road ahead and focussed in the same region as the vehicle being followed. Making use of the HUD principle two vertical lines are projected onto the windshield and by suitable focussing of the light the driver sees them in the road scene ahead. These lines then are separated by a distance which is controlled by the speed of the vehicle, i.e. the faster the speed of the vehicle the closer the two lines are together. In order to maintain a safe headway the driver has simply to adjust his distance so that the lines are roughly on the outside edges of the vehicle.

Having once placed this instrument in the vehicle it becomes a very simple matter to alter the following distances it indicates. For example in wet weather the switching on of the windshield wipers could automatically increase the separation distances and simple switches could be used to do the same for icy and other appropriate conditions. The instrument could be particularly valuable in poor visibility when the driver has lost a great many of his normal visual cues, e.g. in fog.

TIME

Many experiments have demonstrated that there is not a one-to-one correspondence between the time experienced by people and the passage of real time. A number of experiments carried out in the laboratory showed that in some cases, particularly in bad weather, that the subject's estimates of time vary considerably from that of real time. This is also clear from the decrease in gap width accepted by drivers when they have been waiting to move out into a passing traffic stream. In order to find out what a knowledge of real time meant to a driver an elapsed time device was installed on the hood of some vehicles. This operated by commencing to count seconds when a vehicle stopped and giving the total elapsed time in seconds. Using this device drivers allowed a much more consistent gap acceptance when waiting to cross a junction and appeared to be much more relaxed in these kinds of situations.

Automatic Headlight Dimming System

Some form of lighting is desirable on moving vehicles in all lighted streets so that road users are able to distinguish not only between moving and stationary vehicles but also to make better judgment as to the speed of approach. The degree of illumination required for this is clearly dictated by the level of streetlighting. In poorly lit streets the full level of dipped headlights is required. In well lit streets a lower level than this is desirable since full lighting can be a considerable source of annoyance to other road users.

Field trails had shown that the use of full dipped headlights had decreased pedestrian accidents but increased vehicle-to-vehicle accidents. The problem then was to devise a headlight system that would adjust itself to the existing streetlighting conditions.

A manually controlled dim dip system was devised and tested in 1966. However, it had a number of objections, so it was decided to look into the possibility of developing the following requirements:

a. It must not be sensitive to the headlights of other vehicles.
b. It must give a variable level of dimming ranging from no dimming at all where there is no streetlighting or poor streetlighting to full dimming in good streetlighting.
c. It must dim at a slow rate but must "brighten-up" at a fast rate – as is necessary when suddenly entering an unlit area.
d. It must not dim headlights when they are switched on in daylight fog.

A system has now been developed that satisfies these requirements in that it responds only to the frequency of fluctuation of the light source in streetlighting installations, i.e. 100 Hz. These fluctuations cannot be detected by the human eye and do not, of course, occur in natural daylight nor in

light from vehicle headlights. This device is being developed into a system suitable for mass production and a sufficient number will be produced to enable a fleet trial to be conducted to establish the quantitative effect on safety that might be expected if it were generally introduced.

Information Systems

In accident avoidance the supply of information to the driver is a part of the safety system, but whether information systems have any effect at all without their enforcement by the law is very difficult to ascertain. Enforcement of regulations is effective only to the extent that it is possible and the cost can be accepted. Very briefly then, information can be given by signs and signals on the road or by more recent developments in the vehicle. Radio broadcasts can give information on road conditions over wide areas allowing route planning or the more localized transmissions can give details to vehicles about conditions on the road immediately ahead.

Environmental Effects

The correct interpretation of the environment is crucial to safe usage of the road. If the environment is misleading then the interpretation is incorrect with the probable increase in accidents. Fortunately, the brain's memory of past events enables it to assume that, for example, the road continues over the hill or that red lights at night indicate the presence of a vehicle. Some visual impressions however exert such a powerful influence that the brain's capacity for making judgment is distorted. The process of speed adaption is such a phenomenon.

Speed Effects

Bar Lines. A well-known effect is that after driving continuously at high speed for some time, the speed appears to decrease and then when slowing down the speed appears to be slower still. Since this effect is caused by motion through the environment the question must be, can the environment be altered to produce similar or reverse effects. Experiments on a simulator and on roads suggested that various patterns could be used that altered the driver's visual impression of speed. A pattern was developed by which, using visual information only, a driver could reduce his speed to a value very close to what he required. One such pattern has been tried out on the roads for 3 years and continues to produce the desired speed changes. It has also given highly significant reductions in accidents. The existing pattern was produced for a particular set of road conditions. Research is now being directed towards producing patterns to alleviate conflicts and reduce speed in other situations. Fifty sites have been given the original line pattern and the reduction of accidents is under continuous review.

Rumble Strips. Rumble strips were developed to give drivers additional warning that they were approaching a hazard. This is done by generating a noticeable rumbling noise inside the vehicle. A surface to produce this noise level consists of 13-19 mm roadstones held in an epoxy binder. This increase in noise was judged to be very valuable by subjects who took part in the experiments. Three sites have been used for experimental purposes, but accident figures are only available for one site.

Lighting. So far we have been concerned with perceptual judgments of speed, relative motion, etc. But in nighttime driving, a more fundamental ability becomes important – the basic ability to see, whether this be seeing the run of the road, unlit obstacles or the various signs and markings which are patently visible in daylight.

It is useful to distinguish between lighting which is fundamentally for position and motion, and lighting which is for illumination. The former has already been mentioned in connection with the automatically dimmed headlamp; the latter includes roadway lighting and vehicle headlights.

Road Lighting. It is well established that adequate streetlighting makes a major contribution to the reduction of nighttime accidents. Analysis of accidents suggest that the savings in the cost of accidents on 70 mph trunk roads can be as much as 3 times the cost of the lighting.

The introduction of lighting columns into the environment however can itself cause accidents. Lighting columns are now available for use which break off at ground level when struck by a vehicle, the column falling to the ground at the rear of the vehicle after impact.

Vehicle Lighting. Despite considerable progress in headlamp technology, the basic headlight problem of lighting the road ahead without dazzling the oncoming driver still remains. In the short-term, self-levelling of headlamps will improve the situation. In the longer-term, novel headlamp systems, such as polarized light, will be needed to achieve significant improvements in a driver's ability to see.

Road Layout. The unambiguous interpretations of the road system is vital to the road user. Accident statistics indicate various danger points, such as junctions, hidden hollows and corners, but this does not provide the information as to how they should be changed.

Changing a junction for example is expensive and when completed may not give the desired reduction in accidents, equally a new junction may not be satisfactory from a driver's viewpoint.

Research is going on into the use of computer graphics to produce perspective eye views of road layouts from the coordinates defining the boundaries of the highway and all the items that would be found on or near the highway, i.e. street furniture, buildings. This is expected to lead to the portrayal in color of real time continuous runs through a junction over a range of speeds and viewing positions. This will enable alterations to be made to the layout, i.e. elevation, street furniture, angles of approach, and tried out on many subjects so that the correct perceptual conditions can be established before a new junction is designed or an old one altered. The same technique could clearly be applied to any road scene.

RESEARCH EMPHASIS

There are clearly many other parameters which can influence driving performance, for example training, education, motivation, etc. The fact however remains that as long as people have freedom of choice they will use the facilities they are given to the maximum; as road holding of vehicles improves, speed increases; as braking systems improve then braking is left to a later time. So the question that must be asked is if the vehicle occupant is so cocooned that he cannot be hurt below a certain speed, will he continue to drive at the same level of risk as before and therefore will there be no decrease in the number of accidents, but an increase?

If raw materials including oil and iron ore are going to be either in short supply or very costly, then there is a need to conserve their use and accident avoidance would be one way of doing this, provided the techniques necessary use less raw materials and conserve energy.

Where then should the research effort be directed? This is important as the effort available is limited. It has always been accepted that accident prevention is a worthwhile objective but increasing economic pressures, together with the limitations in human abilities against increased performance of vehicles, suggest it should become a major objective.

With the increasingly rapid development of low cost electronic systems with their ability to carry out calculations much more rapidly and accurately than people, it is inevitable that they will be increasingly used in vehicles. In accident avoidance it is suggested they can make a highly significant contribution by replacing driver's freedom of choice in areas of high accident risk.

Increased information on vehicle performance, such as warning of the onset of "rollover" condition or of braking system degradation, could be given to the driver. Electronic systems could be applied particularly for example to control vehicle speeds in urban, and other high risk areas. They could also solve the problems of station-keeping, lane changing, merging and overtaking, yet at the same time allowing the flexibility of use and personal convenience that makes the motor car such an attractive means of personal transport.

ACKNOWLEDGEMENTS

The work described in this paper forms part of the program of the Transport and Road Research Laboratory and the paper is published by permission of the Director.

THE PUNCTURED PNEUMATIC CAR TIRE FROM THE USER'S VIEWPOINT

M. A. JACOBSON
Chief Engineer
The Automobile Association

INTRODUCTION

For many years the Automobile Association has been concerned with the problem of vehicle and tire safety for two reasons above all:

1. to ensure that motorists are given the full benefit of technological improvements as they evolve
2. to be able to provide its over five million members with the type of breakdown assistance they may require.

How Significant Are the Problems of the Flat or the Seriously Underinflated Tire?

We have carried out systematic long-term evaluations of many different makes, types, and sizes of tires and wheels, independent of vehicle or tire manufacturers, to assess grip, wear, safety, and puncture resistance. Tire failures, including slow punctures, have diminished in significance over the years due to improvements in tire manufacture, the widespread use of tubeless tires, and the 1967 tire regulations.

Our findings suggest that many high-speed road cross-over accidents might be attributable to tire failures, the causes of which include neglect of tires. Tire failures – mostly punctures – still account for 4.3% of the calls for AA breakdown assistance, or some 80,000 incidents every year. Table 1 shows the number of AA breakdown call-outs from members due to tire failures while on the move. Added to this must be the number of cases of flat tires discovered while the AA member is still within ¼ mile of his home or where the member carries out his own wheel change or calls at a convenient garage for assistance. These are not attended to by AA patrols and are therefore not recorded.

How Would The Ordinary Motorist Benefit From Run Flat Tires?

There are two basic reasons why the ordinary motorist would benefit from having tires with good run flat characteristics as original fitment: convenience and safety. For, despite the great strides forward in car tire design and manufacture, punctures still do occur and the maintenance of correctly matched tire pressures is not as consistent as one might wish it to be. Repeated surveys carried out by the AA suggest that some 5-8% of cars in the UK or between ½ and 1 million cars are running around with significantly mismatched or incorrect tire pressures. Any tire/wheel combination which is more tolerant, in terms of car controllability and owner attention, is therefore of potential benefit to road users.

TABLE 1
BREAKDOWNS DUE TO TIRE FAILURE

YEAR	% OF TOTAL	ACTUAL NUMBERS
1966	12.1	86,759
1967	11.2	99,832
1968	8.3	92,212
1969	7.4	104,537
1970	7.1	115,821
1971	5.5	93,980
1972	4.8	84,974
1973	4.3	79,448

Ease and Safety of Roadside Wheel Change

In the early days of motoring, changing a wheel or replacing an inner tube was one of the accepted, tiresome tasks which motorists had to face up to as the cost of personal mobility, and it is no more pleasant, easy or convenient today than, say, 40 years ago. Half the motoring public – particularly women – find it a task physically beyond them. On a busy road changing a wheel can be far from safe in the dark or in bad weather. Yet it is a fact that more punctures occur when the roads are wet. Rain, snow, sleet, and slush all act as lubricants which assist sharp objects to work through the tread rubber and so puncture the carcass.

Conventional Tire Failures; Effect on Vehicle Handling and Safety

This paper attempts an analysis of the effect of certain tire failures of current conventional tires on vehicle controllability. It deals specifically with those aspects of tire behavior which can contribute to loss of vehicle control under unforeseen circumstances and be extremely hazardous. It compares conventional tires with a new range of safety tires and safety wheels under the following headings:

1. Significant tire pressure mismatch
2. Sudden deflation (blow-out)
3. Slow puncture
4. Curb damage to carcass
5. Total tire failure

It gives an analysis of the results of many tests carried out in the UK and overseas under extremes of condition rarely met in the UK, as well as in typical everyday situations, and relates them to real and simulated accident producing contingencies. These included snow, black ice, flooded roads, and surfaces varying from loose broken to highly polished ones, from slippery to bone dry, from new motorway to unmaintained roads.

How Real is the Danger?

Tire failures have been dreaded by motorists for many years – though often for the wrong reasons. The sudden blow-out of a front tire which sends a car careering out of control on a high-speed trunk road is a comparative rarity, and seldom has such dramatic effects as some Hollywood films vividly portray. Even so, tire failures – and particularly deflation of a rear tire – do cause injury accidents. Some 16% of Britain's motorway injury accidents are attributed to them. The driver is often not conscious of a slow puncture while he is driving at a steady, fast speed on a straight road. Only when attempting a rapid change of direction or speed maneuver or braking is called for does real danger arise. For instance when overtaking, braking suddenly, turning off a trunk road, or negotiating a roundabout is required, the handling response to normal application of such control inputs as brake, accelerator, and/or steering can be so exaggerated, unpredictable or frightening as to cause panic over-reaction. This produces the classic case of sudden loss of control in which bad or wet road surface conditions are an added safety hazard.

The AA Field Test Program

This paper summarizes the results obtained in 10,500 miles of field testing of a representative range of tire/wheel combinations at various stages of pressure mismatch on vehicles having differing suspension and power transmission characteristics. Road surface temperatures (below 0°C – 50°C) and coefficient of adhesion values from μ 0.2 to μ 0.95 were encountered. Observations on behavior on severely rutted roads covered in deep snow or mud, or on sheet ice are outside the scope of this paper.

We deliberately and progressively deflated tires on a score of cars possessing a wide variety of ride and handling characteristics in order to establish the threshold of controllability, i.e., the residual tire pressure level at which a moderately experienced and reasonably alert motorist tends to over-react to the point of loss of control of the vehicle. All road irregularities which are likely to be encountered in Europe and which could possibly excite significant side loading, vibration or transient response were investigated. These included adverse and rapid changes of camber in bends, tramlines, railway crossings, corrugations and undulations of varying pitch, manholes, raised white lines, "cats eyes," broken edges, road repair sections, joints in concrete surfaced roads, and a variety of μ values of road surfaces commonly encountered in Europe, as well as blind corners, hump-back bridges, and hills. Whenever possible both dry and wet adhesion were tested on the same vehicle with nominally identical tires to establish the influence of these factors on vehicle controllability. This allowed a relative merit rating of tires and wheels to be obtained.

Effect of Test Driver's Skill and Familiarity With the Circuit

Throughout the tests one factor was clearly established: the maximum safe speed or reaction time could vary from one test driver to another and could be improved upon by familiarization with the circuit. Yet these variations, in measurable quantities, were relatively small compared with, say, the difference in vehicle controllability with or without a severely deflated tire.

Method of Testing

The basic test procedures adopted were quite straightforward. On any one vehicle a tester took a car round a set course which could be a special test track free of normal traffic, a mixed set of public roads in the UK or abroad, an old aerodrome, a roundabout, a circular steering pad or skid pad, or, more usually, a combination of all these. His co-driver noted the handling characteristics, comfort, and noise level for at least two laps, one at a moderately sedate, and the other at a thrashing pace. Tire pressure was then lowered in steps of 2 or 3 psi from the standard setting until the driver lost control of the vehicle on two successive occasions, or until the prominent tire side wall markings were completely scuffed off due to continuous contact with the road, or the tire was pulled off its rim due to excessive flexing under side loading. Each series involved the progressive deflation of a front tire. This procedure was repeated on the rear. In some investigations, tires were deliberately rolled off the safety ledge of the wheel, and in others

tires were driven with explosive charges in the sidewalls to obtain a sudden blow-out. In others again, the sidewalls were deliberately weakened by cuts in such a manner that they were induced to fail by structural collapse of the carcass without giving much prior warning. Emergency brake application and swerve maneuvers were part of the test procedure. A spring-loaded marker on the rear bumper left a witness mark recording the actual movement of the vehicle at the point of imminent loss of constant or predictable steering response. Cine-film and still shots of the tire under stress were taken by observers either on the course or leaning out of the passenger seat. It was thus possible to build up a fairly comprehensive picture of loss-of-control threshold conditions and of the paths taken by various vehicles. Some of this work was carried out over a period of nearly three years before testing of tires or wheel with run flat capabilities commenced. The initial tests embraced both crossply and radial tires for most of the popular makes from the Mini to 1½ ton vans.

Do Commercial Vehicles and Cars React the Same Way?

It is important to remember that cars and commercial vehicles, with a 1½-2 ton van as the dividing line, do not behave in the same manner under conditions of tire deflation. On a commercial vehicle, loss of control is more pronounced with a fairly sudden front wheel deflation rather than with a punctured rear tire – obviously so where only one of the dual tire set is so affected.

How Lateral Forces Were Measured

Wherever possible "g" loadings were measured by an accelerometer mounted on or close to the front passenger footwell for ease of taking spot readings. Where this was not convenient, simple measurements of the road bends and the paths taken by the car on successive test runs were taken. The cars went into the control sections of the road circuit at substantially steady speeds which were measured by conventional means of calibrated speeds and/or 5th wheel. We are aware that this is not accurate enough for fundamental research work and fully accept that it may introduce an error of up to 6% since no roll correction was allowed for. Nonetheless, we suggest it does represent a meaningful, easily repeatable method of carrying out a fairly comprehensive field evaluation of tire and vehicle performance on a public highway or proving ground.

OBSERVATIONS ON CARS WITH CORRECTLY INFLATED TIRES ALL ROUND

Effects of Basic Vehicle Characteristics

We found that a standard tire could on a surface having a coefficient of adhesion of $\mu \simeq 0.8$ to $\mu \simeq 0.9$ give lateral "g" values from as low as 0.56 up to nearly 0.95 before controllability of the car was lost. Obviously this was a function of the basic car concept itself. The individual and cumulative effects of the suspension layout including dampers, roll center, roll stiffness back and front, weight distribution, width of track and length of wheelbase, position of center of gravity, and the location of the power unit relative to the steered wheels, were quite pronounced. Some cars were much inferior to the rest.

TABLE 2
MIRA STEERING PAD TESTS:
108 FT RADIUS CIRCLE OF FLAT UNCAMBERED SURFACE

LAYOUT	CHARACTERISTIC SPEED	LATERAL ACCELERATION "g"
Front engine – rear wheel drive	39-34 mph	0.95 g to 0.72 g
Rear engine – rear wheel drive	36-30 mph	0. 8 g to 0.56 g
Front engine front wheel drive	39-36 mph	0.95 g to 0. 8 g

Radial vs. Crossply Tires

The substitution of radial for crossply tires came next in order of importance within this context. It did improve road holding of practically all cars.

Bald Tire Effect

The state of the tire's tread was the next most important factor. The adverse effect of tires with little or no tread depth on the car's ability to hold a given course was found to be pronounced on the wet roads, particularly while cornering at speed.

Effect Of Different Makes Of Tire

On the other hand the differences between premium brands of tires in terms of carcass construction – steel versus textile, rayon versus nylon or polyester, rubber mix, wall stiffness, and tread

pattern – were all found to be of lesser significance, provided the roads were dry. Once the roads were wet there was, however, a marked difference between tires from the UK and Western Europe, which generally had good "wet grip" characteristics, and some of those from Eastern countries, including most East European and Japanese tires which lacked this feature.

What the Conventional Tire/Wheel Assembly Achieves

Once tire pressure had fallen to a setting of about 13-15 psi with the other remaining more or less at the recommended pressures, there was a strong tendency for all the cornering forces to be provided by the remaining three. A further drop in pressure to 8-12 psi often resulted in the tire venting to the extent that the tire rim seal was finally broken. At the limit the tubeless tire could be rolled off the wheel and rim to road contact took place. This drastically changed the car handling characteristics. In the case of a severely deflated rear tire it almost invariably produced a virtually uncontrollable snake, slide and spin when changing direction, although the tire itself was not always damaged beyond repair. In the case of the steered front wheel, the flopping tire was frequently cut by the rim. This produced a severe vibration and shudder right through the steering. The totally deflated tire could at best be slowly driven for about a quarter of a mile before it was ruined. After travelling a further 2 or 3 miles it was torn to ribbons and the wheel was then also made unusable.

SUMMARY OF DEFLATION TESTS ON STANDARD RADIAL OR CROSSPLY TIRES

Limiting "g" before slide, spin, or general loss of control occurred.

The degree of lateral holding force which a tire has to provide once a vehicle is changing direction from the straight ahead can be critical.

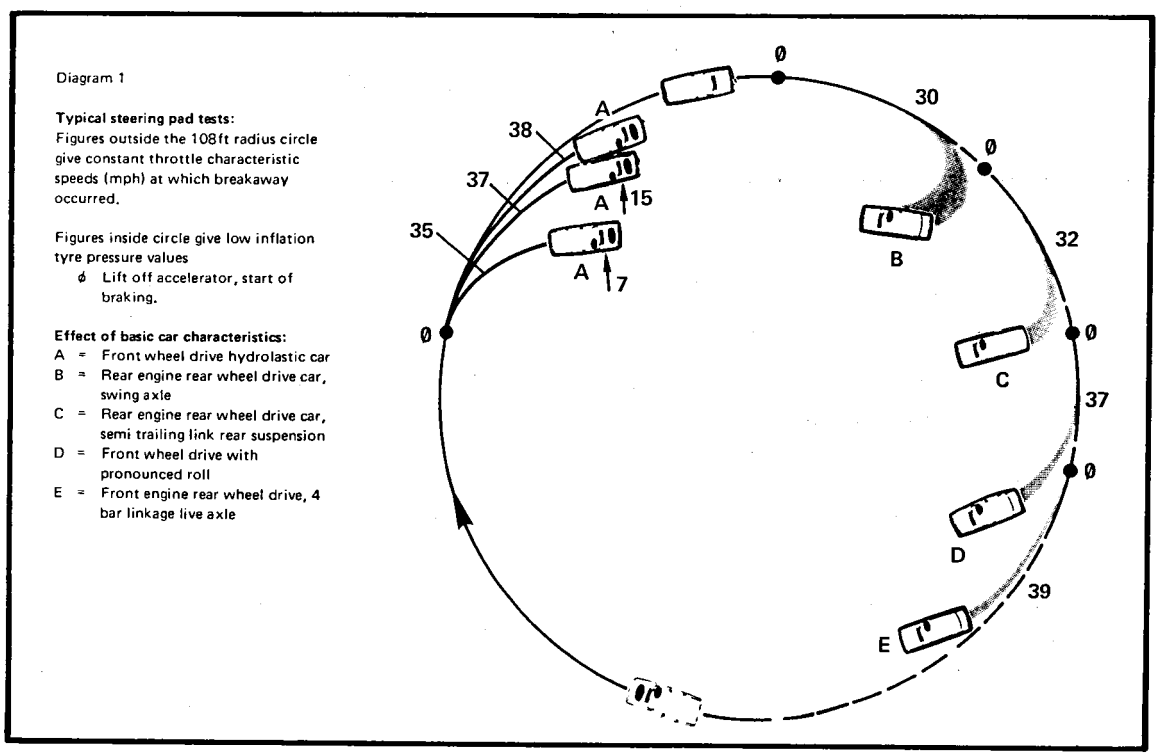
Tire deflation of both crossply and radial had a most marked effect resulting in "squelching" and tire scream.

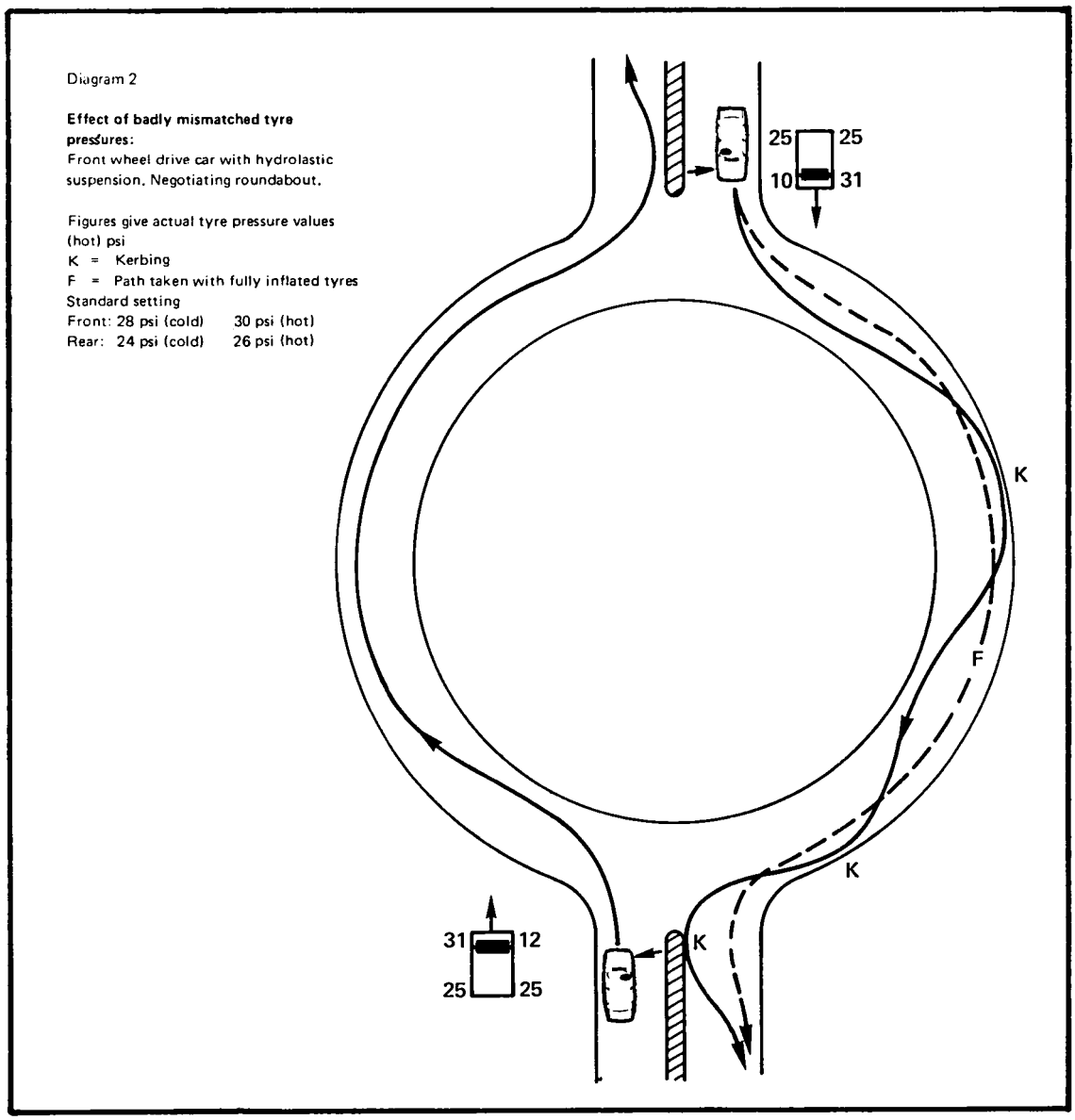
Typical values obtained when tire pressure was down to 10-12 psi were of the order of 0.23 g to 0.28 g for front tires and 0.20 g to 0.25 g for rear tires.

"g" values as low as .18 have been recorded just prior to the tire being rolled off. Cornering at speed could cause severe, abrasive road to tire contact halfway up the tire sidewall at about 10-13 psi. In most cases for a "soft tire" running at about 12-15 psi (cold) the limiting lateral "g" would generally be of the order of 0.35 g for a front tire and 0.30 g for a rear tire.

Typical Loss of Control Situations

Diagrams 1 to 6 show typical near or total loss of control conditions with severely deflated tires.





They are roughly to scale. When altering course on an overtaking maneuver, at speeds upwards of 50 mph the change in available cornering force of the tires as the weight transfer occurs from a standard pressure tire to the nearly deflated one and back again taxes most drivers beyond their capabilities and can be a potential killer. We had cars slide broadside to their intended course through a bend or chicane as a result of this change in handling characteristics. We were prepared for this, and it frequently resulted in the car rearing sideways onto two wheels, although we never actually rolled a car over.

Generally low roll stiffness cars were more prone to this and the subsequent heavy bounce as they rocked back on to all 4 wheels. We experienced four wheel sideways slides, spins, and running out of control off the track at anything from 25° to 150° deviation from the steered course. In most power-assisted steering systems we found a lack of feedback to the steering wheel. Little could be gleaned regarding tire to road forces, and there was particularly insufficient warning about the state of deflation of the tire.

Some of the heavier soft sprung cars were thus the most difficult to retain control over. The low center of gravity large wide tracked US car could readily have its low profile radial or crossply tires rolled off the rim and proceed in a straight line into the "rough" at a tangent to the course. On the other hand, the older high center of gravity, narrower track type of European car with power-assisted steering coupled to a pronounced degree of roll, tended to proceed with its tail leading after a slide and spin.

Diagram 3

Overtaking manoeuvre on flat road: motorway cross over simulation
3 lane motorway
with central reservation

Right hand front tyre deflating rapidly
Figures give overtaking car's speed (mph) at time of passing 1st car.
⇓ = direction of traffic
b = Brakes applied
.. = Dunlop Denovo fully deflated

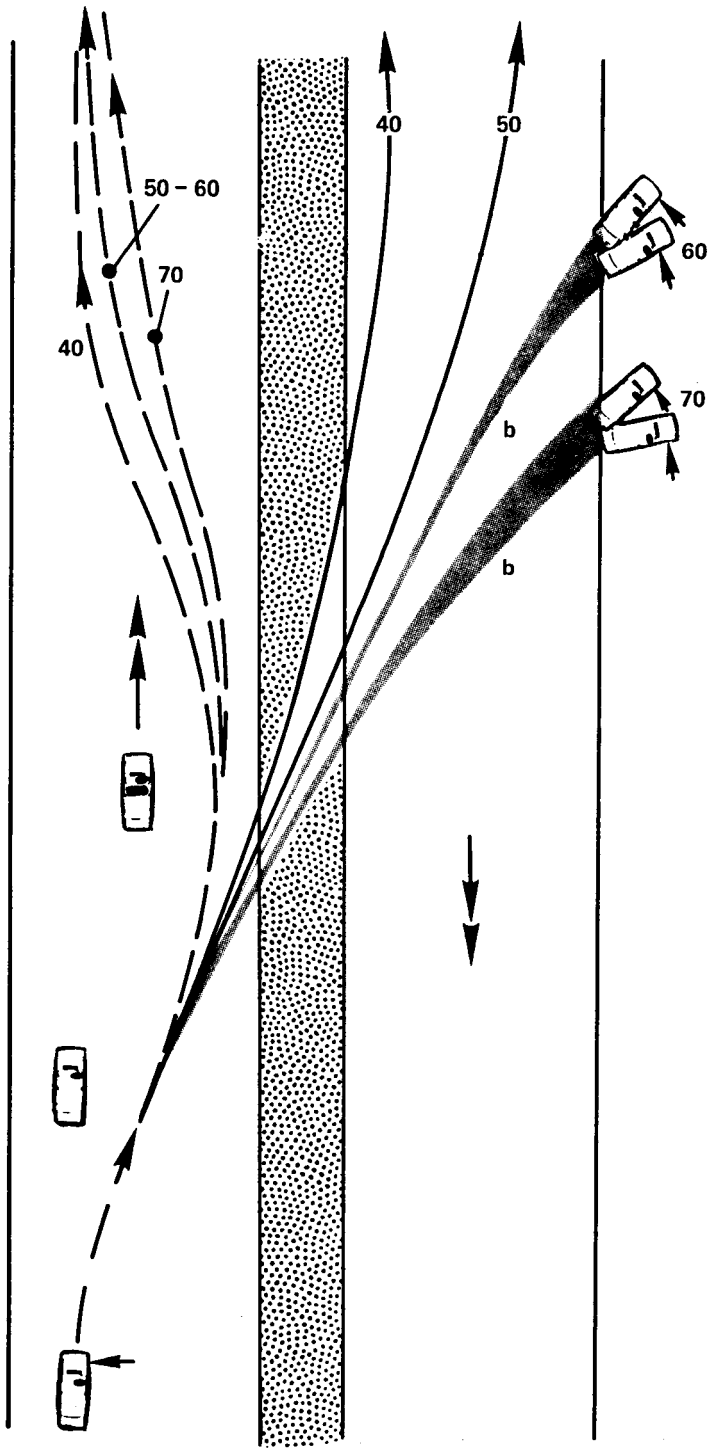
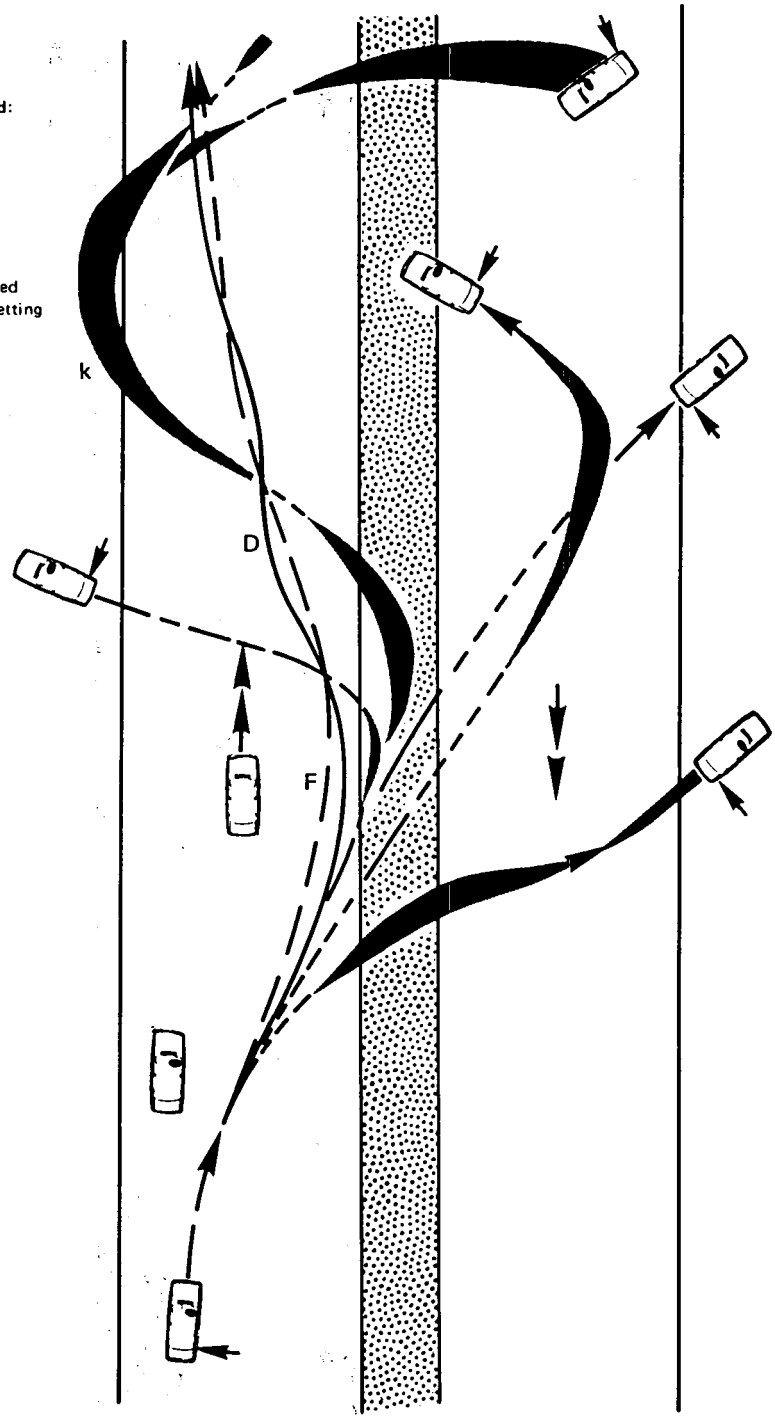


Diagram 3A

Overtaking manoeuvre on flat road: motorway cross over simulation
3 lane motorway
with central reservation

Rear wheel drive cars
RH rear tyre deflating rapidly
⇓ = Direction of traffic
D = Dunlop Denovo fully inflated
F = Tyre at standard pressure setting



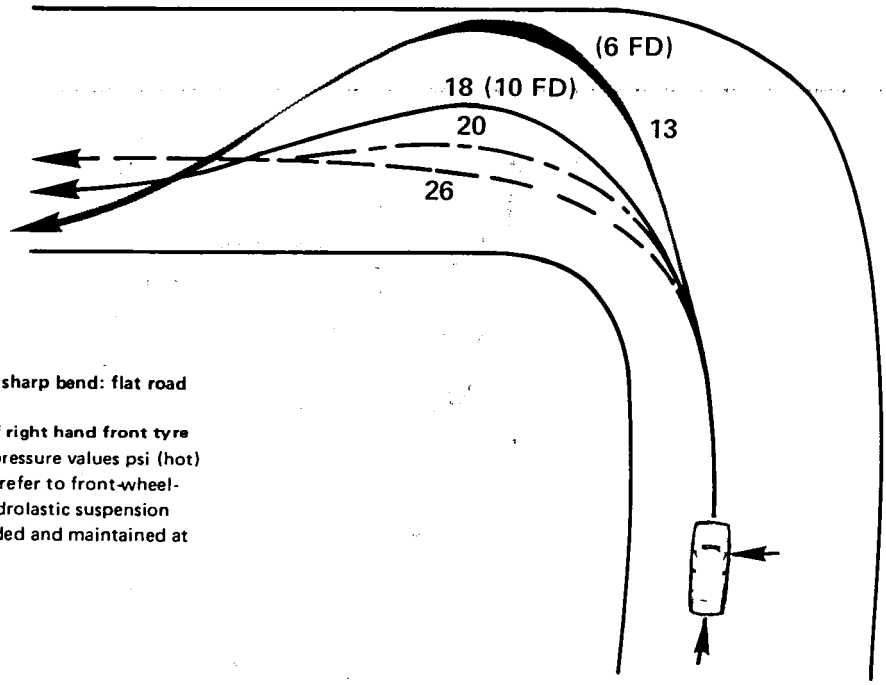


Diagram 4

Loss of control in sharp bend: flat road

Rapid deflation of right hand front tyre
Figures give tyre pressure values psi (hot)
Those in brackets refer to front-wheel-drive cars with hydrolastic suspension
26 is course intended and maintained at 26 psi (hot)

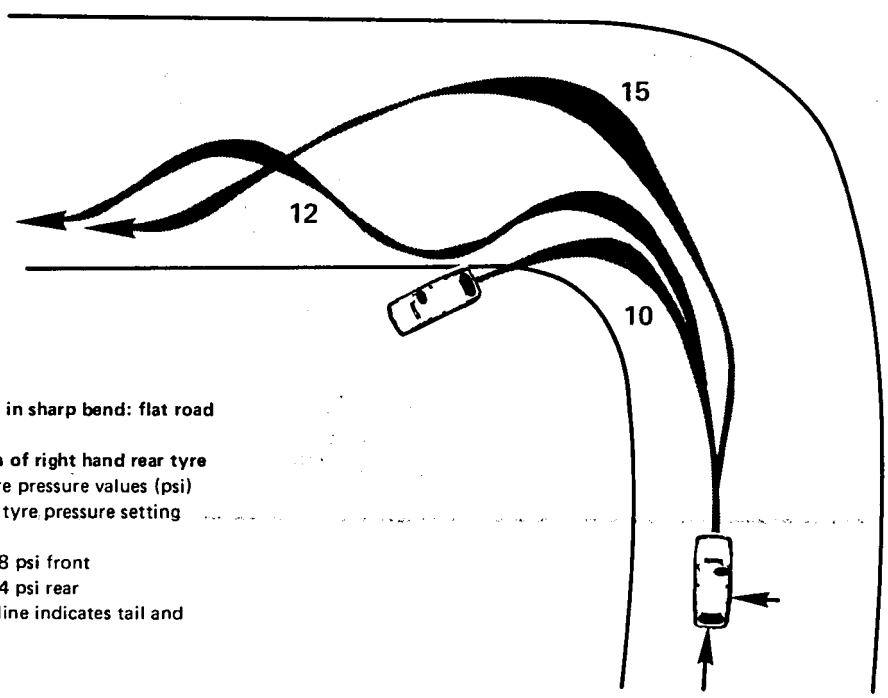


Diagram 4A

Loss of control in sharp bend: flat road

Rapid deflation of right hand rear tyre
Figures give tyre pressure values (psi)
(hot) Standard tyre pressure setting (cold)
28 psi front
24 psi rear
Broadening of line indicates tail and slide

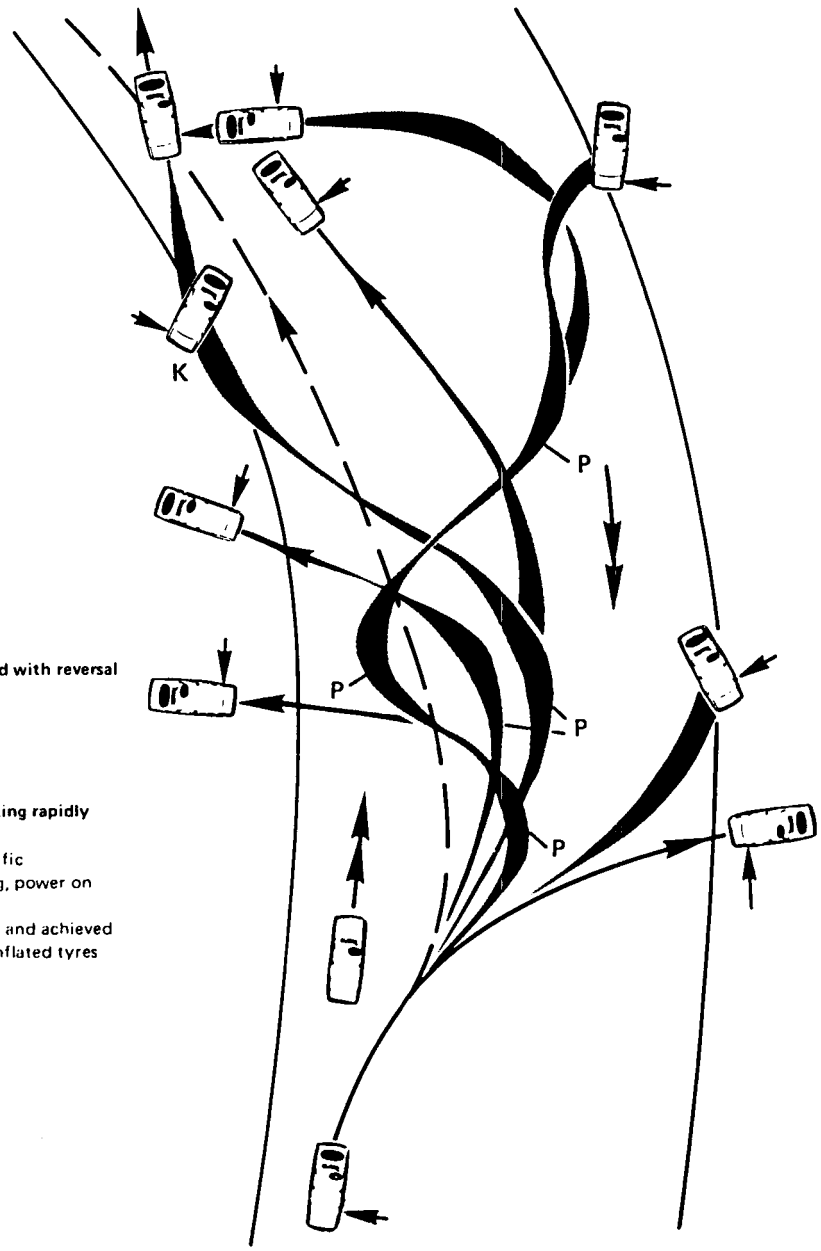


Diagram 5

Loss of control in fast bend with reversal of camber:

Overtaking manoeuvre:

Front-wheel-drive cars
Right hand rear tyre deflating rapidly

→	Direction of traffic
P	Throttle steering, power on
K	Kerbing
	Course intended and achieved with correctly inflated tyres

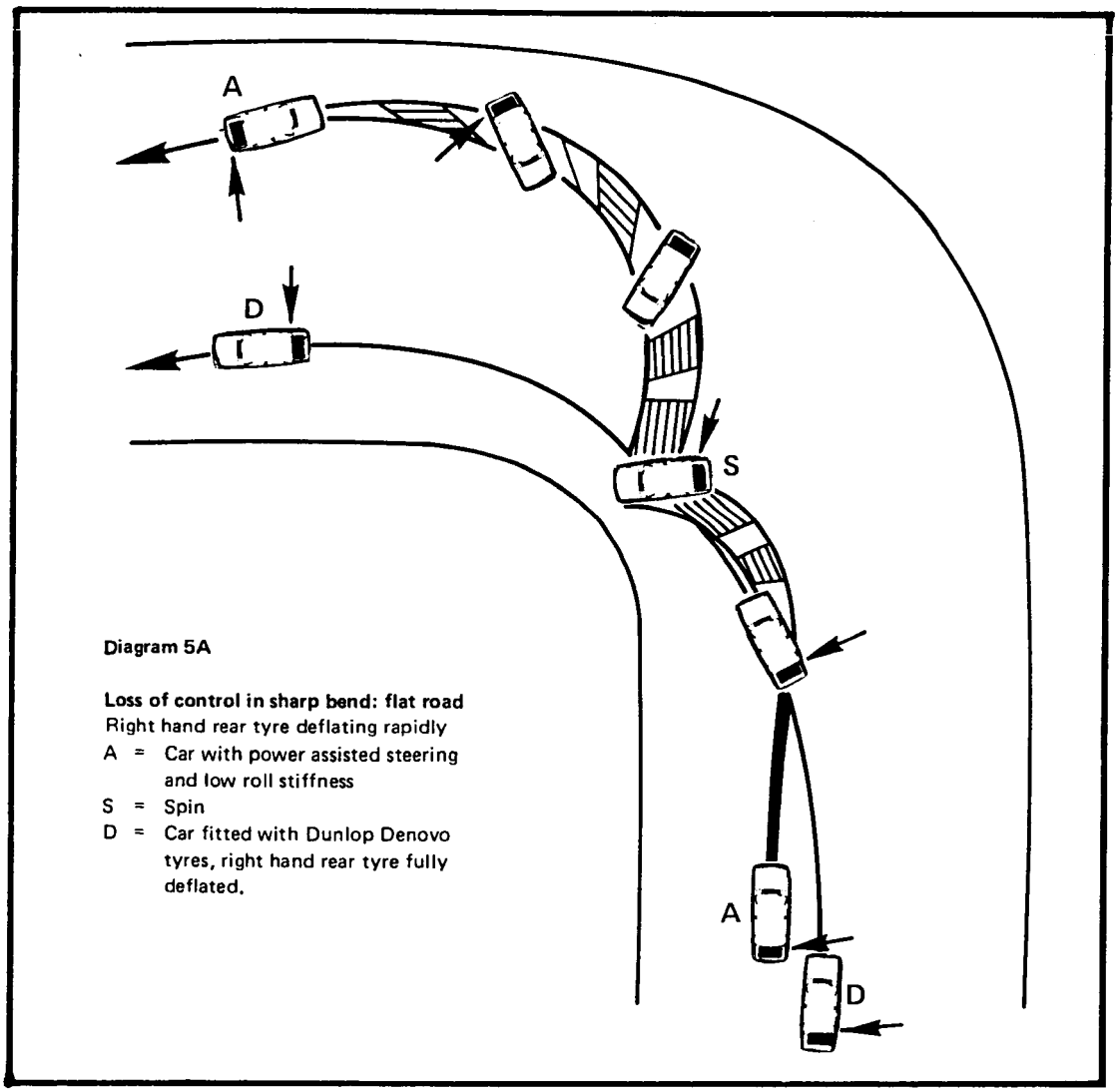


Diagram 5A

Loss of control in sharp bend: flat road
Right hand rear tyre deflating rapidly

A = Car with power assisted steering and low roll stiffness
S = Spin
D = Car fitted with Dunlop Denovo tyres, right hand rear tyre fully deflated.

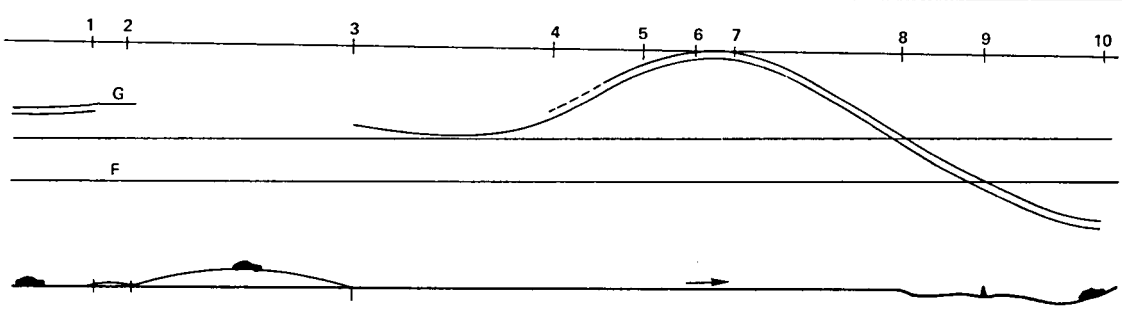


Diagram 6

High speed cross-over accident at (98 mph): multiple tyre bursts.

Type of car: Buick Skylark
1973 Model
Power Assisted Steering

G = 26 ft long 6" x 6" concrete beam
F = Fence
→ = Direction of travel

1. Front left hand tyre hits beam and collapses, car bounces up into air
2. Rear left hand tyre comes down heavily on beam and collapses
2-3 Car airborne
3. Car lands on right hand wheels only, only, canted over to right hand side about 10 to 15°
4. Left hand wheels contact road surface
5. Left hand wheels bite into road
6-7 Left hand wheels chew up shoulder of road
7. ¼ turn of steering wheel applied to overcome tendency of car to roll over, road regained
8. Car drives off highway into scrub land
9. Car smashes through fence
10. Car somes to rest in scrub land

No sudden application of brakes
Progressive braking after stage 7

Which is More Dangerous – A Rapid Deflation at the Front or at the Rear?

The tendency to spin off was significantly more pronounced once a rear tire was deflated. Whereas a front tire deflation due to total loss of air usually resulted in a more manageable and less dramatic loss of steering control, a rear tire at 10-13 psi usually produced a severe skid and spinning, once it was called upon to give some lateral guide force. Front-wheel-drive cars, particularly those with a "flat ride", i.e., little roll, tended to remain remarkably stable even with one severely deflated front tire. The rear tire could not stand anything like this deflation before loss of control became irretrievable.

THE PROBLEM

What a Safety Tire Has to Do

One of the essential requirements of a safety tire is that it should provide adequate, predictable, and consistently stable lateral guidance and cornering forces when maneuvering, taking a series of bends, or overtaking at speed, right to the point of total tire failure. Furthermore the driver must be able to accelerate and brake without fear of unpredictable swerving off the chosen course.

Some Solutions to the Punctured Pneumatic Tire Problem

Run flat tires are not new. Many attempts have been made and those which have had a partial success have all had severe limitations in their application.

Military Run Flat

The run flat tires for military application are basically of truck rather than car tire construction, with a safe continuous speed limit of around 60 mph. Since the weight is about twice that of a conventional tire, it is significantly dearer, and the increase in unsprung mass affects the suspension and steering elements. When punctured, speeds of up to 30 mph can be maintained for a strictly limited distance before the high heat generation in the stiffened side wall and shoulder buttress zones causes the tire to break up.

Self-Sealing Tires

A number of compounds inside the tubeless tire which are designed to seal small holes against loss of air pressure have been tested. Our experience is that they can be quite effective under certain conditions. A clean hole, such as will be left when running over a nail or even a row of nails fitted to a board, will be capable of being sealed provided the nail penetrates the tread at an angle of no less than 45° to the tire tread and pulls out cleanly; the nearer to vertical the tire penetration in the tread, the smaller is the hole to be sealed and the easier it is to form a pressure-tight internal skin over it. On the other hand a jagged hole, as left by pulling out a bent long woodscrew which has worked itself into the tire over a longish period, can be too much for a sealant, particularly those of the uncured latex type. In order to effect a kind of heat cure of the sealant to the tire carcass, it is essential to drive the car away very soon after the offending sharp object – preferably a straight nail – has been removed, and at speeds of at least 30 to 40 mph after reinflation to the normal tire pressure settings. This creates the essential frictional heat necessary for curing.

Leaving the car standing for an hour or more even with the tire at full pressure and the foreign object removed was found to substantially impair the sealing action. Likewise slow speed progress after inflation failed to ensure adequate sealing.

Where a form of sealant with a forcing agent to provide reinflation was employed, we found that in some cases there was a progressive increase of tire pressure above the correct one when the car was driven hard over 100 miles or more. The steady increase was indicative of the continuous "working" of the chemical compounds and required the bleeding off of internal air pressure. Neglect of this could lead to a very considerable tire pressure mismatch. We did not, however, experience a tire blow-out due to an uncontrolled rise in pressure, only a small adverse effect on ride and handling. We have not to date tested the Goodrich Golden Lifesaver which promises to be an improvement on the concept of uncured latex or similar liquid internal sealants.

Multi-Air-Chamber Designs

There are a number of multi-chamber and "tire within a tire" designs, all of which allow the car to continue on its course, after the tubeless tire has suffered a puncture. They will retain the tire on the rim, and permit progress, at a limited speed, to a repair depot, some miles away, provided the auxiliary air chambers are pumped up to the required pressure and retain the air. All the multi-chamber and internal sealant systems that we know of assume that punctures will occur in the tread area and that sidewall penetration, weakening of the carcass due to curbing, striking a sharp rock or brick, can safely be ignored. Our statistics and those of the tire retreaders do, however, suggest that a large number of tires are seriously damaged in the sidewall zones usually due

to driver carelessness, and that solving the problem of sharp objects piercing the tread zone is not enough. The effectiveness of these concepts in retaining safe vehicle handling when a sidewall puncturing or failure, regardless of its cause, has occurred is, in our opinion, not proven.

THE SAFETY WHEEL CONCEPT

There are two designs of safety wheels commercially available. Both rely on filling the well in the wheel, which is required to allow the tire to be fitted and removed. By preventing the bead of a deflated tire from sinking into the well while cornering, the tire is retained and a measure of steerability is assured. This does not, however, save the tire, which will be ruined in a matter of some 2-4 miles of slow motoring. Our testing confirmed that the safety wheel lived up to its claim of giving a measure of steering control in the case of front tire failure but we found very little residual controllability in the case of a drastic rear tire deflation or blow-out. The vehicle control when the weight was transferred onto the deflating tire during sharp cornering, was much better than with standard wheels. However it was, in my opinion, at least, barely adequate for an average driver. In some installations it resulted in the inner sidewall making severe rubbing contact with steering and also suspension elements. It generally prevented wheel rim to road surface contact, thus saving the wheel. Unlike the conventional tire and wheel assembly where the tire is being mangled by the wheel rim before the car comes to rest, there was no "kangaroo jumping" while braking.

Dunlop Denovo

The Dunlop Denovo system also has a well-baseless construction, but the assembly of the tire and its ring of canisters containing the sealant/lubricant is carried out from one side. The assembly is made airtight by fitting the second half of the metal wheel rim, and bolting the two halves together with an "O" ring trapped between them. The step by step reduction in tire pressure in the conventional tire on a conventional or well-baseless rim, and even the self-sealing tire (with a residual air leak) did in every case lead to loss of control long before zero inflation pressure was reached. The Dunlop Denovo, however, allowed fast safe maneuvering for considerable distances, even after the tire had been irreparably damaged. Typical stages of gradual deterioration of precise vehicle control are recorded in Appendices 1-4. These indicate that, even with Dunlop Denovo tires fitted, the basic suspension and handling characteristics of a car play a significant role. However, the most important factor is the additional overall safety and controllability which endure until the tire disintegrates 100 miles or so after self inflation pressure failure. The appendices give some quantifiable values of limiting safe cornering loads at various stages of tire failure. There are also comments on when a puncture manifests itself to the driver or passengers.

Is a Low Tire-Pressure Warning Device Necessary?

One of the new set of problems posed to the ordinary motorist is to determine when his Denovo tire is running partially or totally deflated.

If there is no risk of unpredictable handling and braking response then the further complication of fitting a low tire pressure warning device – while it may form part of the Japanese Experimental Safety Vehicle concept – commends itself only on the basis of economic advantages to an owner, if he can confidently expect to recoup its initial and maintenance costs by a saving on avoidable tire replacement costs. Our tests to date suggest that as far as the Dunlop Denovo concept is concerned it is doubtful whether the cost of a warning device can be justified for a standard family car, for the system always fails safe, even though the tire may be ruined.

There are clearly recognizable signs of imminent tire break-up, and this still allows sufficient mileage to be covered for a motorist to reach his destination in safety. This of course means that before long, car makers can dispense with the need for carrying a spare wheel and tire assembly, relying instead on strategically placed service support.

Dunlop Denovo Tire/Wheel Combination Sensitivity to Imbalance Forces

Since, to most ordinary motorists the first indication of a deflating Dunlop Denovo tire are ill-defined, we carried out an investigation in order to establish whether dynamic and/or static wheel imbalance could give the same signal to a motorist that he ought to stop and investigate the slightly "odd" behavior of his car. All tires were run at the recommended tire pressure setting. The fitting of two magnets of 1½ oz. each at approximately 6½" from the wheel center line on either side of the valve of the left hand front wheel produced no noticeable effect. The two magnets were then transferred to the already imbalanced left hand rear wheel, and above 55 mph the driver noticed an effect which was almost indistinguishable from the symptoms of a deflating Dunlop Denovo tire. At about 72 mph one of the magnets was flung off and a slight improvement in

ride noticed. The fitting of a total of 5½ oz. of stronger magnets to the left hand rear produced a signal to the driver which he mistook for a rapidly deflating tire almost at the point of zero inflation.

The magnets were removed from the left hand rear and two 2 oz. magnets were then fitted at the left hand front wheel disc, 6¼" from the center line either side of the valve. The imbalance, which was noticeable at 40-45 mph but ran out at 60 mph, could easily be confused with an almost flat front tire effect. This contrasts with the observation on the rear wheel imbalance signal which was at its peak at about 40 mph, ran out at 45 mph, and became of nuisance value at above 55-60 mph.

Reducing the imbalance weights from 4 to 3 oz. at 6¼" center line weakened the symptoms to the point where most motorists would tend to overlook them. A further reduction to 2 oz. had the effect of swallowing up all excitation throughout the speed range, 0-75 mph.

What Happens If More Than One Tire Fails at the Same Time?

There is little information available on the effect of more than one tire failing at the same time. The statistical likelihood, if one ignores deliberately laid man-made obstacles, booby traps, or detonating devices, is very low.

The author has only twice experienced unexpected and sudden deflation of more than one tire at a time. In the first instance he careered off at speed at an angle of approximately 25° to his intended course, finishing up some 150 ft. in the sage bush scrub off a US Federal Highway and, thanks largely to the sturdiness of the US car, lived to tell the tale; for although he lost both front and rear offside tires by rolling them off after striking an obstruction on the road, the car neither spun nor rolled over. In the second instance, both Dunlop Denovo front tires on a modest Escort 1100 were punctured at the same time – quite unexpectedly – as the car approached a busy roundabout at speed. There was no room to swerve in the fast moving and converging traffic. All that the passenger noticed was that the headlight beam appeared to have gone out. The car continued to travel safely in a distinctly nose-down position for about five miles before they stopped to investigate. It required considerably more effort to steer. Since they had no spares they carried on for another 23 miles without in fact damaging the tires.

The author suggests that in the not-too-distant future it will be reasonable to do away with spare wheel and jack. After all, motorists no longer carry a spare battery – so why a spare wheel?

APPENDIX 1
DUNLOP DENOVO TIRE TESTING ON FORD ESCORT 1100

Extensive tests were carried out on:

1. motorways and class A trunk roads carrying their fair share of mixed traffic
2. well-surfaced B class roads, narrow but relatively free of heavy traffic
3. metalled and/or rough surfaced unclassified cross country routes in North and mid Wales
4. in London's busiest traffic zones – Trafalgar Square, Leicester Square, Marble Arch, Hyde Park Corner, Victoria Monument, Elephant and Castle.

SUMMARY OF RESULTS

Running Fully Inflated

The first impression of the Dunlop Denovo when fitted to the Escort 1100 was that it transformed a car with average handling into a car of quite remarkable road holding in both wet and dry road conditions on most road surfaces. Pot holes had a negligible disturbing influence. Running over "cat's eyes" could be heard and felt but it was not objectionable. There was a measure of raised white line "nibble", but this could be eliminated by modification to the tire shoulder tread pattern. The tire did affect the basic noise transmission particularly when being driven on high-speed trunk roads and motorways. It shares this characteristic with a number of US and European makes of low-profile tires. This feature is, however, not obtrusive nor was the firmer and more positively guided ride objectionable. Many of the good features such as better dry and wet road grip in high speed cruising, braking, acceleration, fast lane and direction changes, and cornering at speed would, of course, be the same regardless of whether the tire had a built-in long distance run flat capability or not. In its road manners as a fully inflated tire it represents a significant step forward, inevitably leading to enhanced vehicle controllability. The true merit of the tire should, however, be assessed on its run flat characteristics.

In the case of current conventional tires, both of radial and crossply construction, deflation of a rear tire to half the standard setting could make a car virtually uncontrollable, in an emergency overtaking maneuver or on tight bends. The Dunlop Denovo gave controllable, though at times, far from precise handling right down to zero pressure.

The driver and his passenger first became aware of the uneven pressure and a consequent change in vehicle behavior when the tire pressure had dropped

from 27.5 psi to about 9 psi (hot). This could, however, be easily misunderstood as driving on rough, or loose surfaces and be ignored by the average motorist. Only when the tire pressure had dropped to 5 psi (hot) was "snaking" after coming out of tight bends noticed. At 3 psi, i.e., about 1/10th of recommended tire pressure setting, steering response was distinctly "woolly" – but the car remained controllable even in fast cornering slides by snapping the throttle shut or applying moderate braking.

Self Inflation Pressure

Starting off with zero tire pressure produced automatic partial re-inflation of the order of 2-5 psi and a marked improvement in the "rolling gait" of the car.

Running Without Air in the Tire

In order to check the claim that the car was completely controllable at speeds up to 50 mph with one tire totally deflated, and could be run in this condition for distances up to 100 miles, one of the Dunlop Denovo tires supplied had its valve core removed to obtain zero pressure; this valve core was then replaced. Speeds of up to 55 mph were obtained on the short straights, and the car was subjected to considerable side forces when negotiating very tight bends at relatively high cornering speeds. While the car tended to pull to one side and the steering was distinctly heavy, it could be maneuvered perfectly safely, responding to both steering, accelerating and braking efforts of the driver in a predictable and repeatable manner. In fact it was nearly as precise as on its original D75 crossply tires – fully inflated.

The more severe condition of a fully deflated tire on the rear was also tried. There was less of a pull to one side, and although the handling was much more "woolly", severe braking still produced a straight line pull up.

A run of 105 miles without air in the tire at speeds of up to 65 mph was completed over a variety of road surfaces, both wet and dry. The tire/wheel assembly was then thoroughly inspected and it was confirmed that the carcass had not suffered unduly despite the severe deflations and heat generation involved. The special sealant/lubricant had allowed the internal relative sliding velocities of one part of the carcass over the other to take place without reaching a destructive temperature rise. Relative internal sliding velocities reached nearly 20 mph. Then after fitting a new set of special fluid canisters testing continued for several thousand miles.

Running With a Tire With Sidewall Damage

In our experience the most severe punctures are caused by the penetration of a sharp object through the side wall or concussion damage. A deliberate cutting of all the cords with a sharp knife produced the desired severity. Two diagonally opposed 1″ wide slits kept open by two flints were made on the left hand side rear tire and the car was driven in this condition across some of the rougher parts of mid Wales with perfect safety and controllability. This was followed by some reasonably fast motoring on "A" class roads with speeds occasionally up to 62 mph. A total distance of 39.3 miles was covered before the driver became aware of any unusual car or tire behavior. Imbalance vibration and a cloud of blue-black smoke (displaced special fluid escaping) were the only indication. The car was brought to a halt 1.7 miles further on. This was done so as not to destroy tire and wheel completely. Bits of rubber were beginning to fly off.

Cold Cure Plug Repair

An unpremeditated puncture by a no. 8 woodscrew which remained embedded in the tread for some 130 miles, did not cause a rapid loss of air pressure until the screw was discovered and removed by a forecourt attendant. This then caused the tire to deflate rapidly. A distance of some 64 miles was covered, mostly on "A" roads and the M4 motorway, at speeds up to 55 mph before the tire was removed for the fitting of an approved cold cure plug. Despite severe rain, which created a continuous water film on the road, controllability was very good throughout a run of nearly 200 miles.

THE MID WALES TESTS

Introduction

A variety of road conditions, including some very tight bends, narrow bridges, reverse camber sections, pot holes, loose gravel on tarmac, and long stretches of high tarmac dressed with granite chippings, were encountered. Coefficients of adhesion were of the order $\mu \simeq 0.60$ to $\simeq 0.90$ dry and $\mu \simeq 0.40$ to $\mu \simeq 0.70$ wet.

Air and road surface temperatures were close to freezing in the shaded sections and up to about 4°C where the sun was shining. Roads were affected by frost damage and the presence of black ice.

Front seat passenger as well as driver reaction to change in comfort and car handling predictability was sought under normal motoring, rather than extreme conditions.

TYPICAL TEST RESULTS

Left Hand Side Rear Deflation Tests

Run 1 — Fully inflated. Tire pressure psi:
front 27 psi
rear 29 psi

Average speed over 11.5 mile course: aimed at 45 mph/achieved 42 mph.

Run 4 — Left hand side rear at half standard pressure. There was very noticeable static tire deflection, but the average motorist would have put the slightly ragged ride and handling down to indifferent road surfaces and would have ignored it. Average speed 42 mph.

Run 6 — Left hand side rear tire at 1/3 standard pressure. The average motorist would have noticed that there was something "odd" about the car's behavior in bends. Handling was still very controllable and similar to driving on a broken surface, standing height from the road to the center line of the wheel was down to 247 mm compared with 258 mm as standard tire pressure. Average speed 42 mph.

Run 7 — Running at 5-7 psi. With tire pressure at 5 psi on the left hand side rear the ride quality had noticeably degenerated. At speeds below 50 mph a slight pull to the left could be detected and on long bends the car began to snake. A pull to the left on braking was felt. All these phenomena could have been attributed to a crosswind as readily as to a suspected soft tire. The attitude of the car had changed. Running on the deflated tire caused pressure to rise to 7 psi.

Signs of "woolliness" were noticed on left hand bends for the first time, but vehicle control continued to be unimpaired. A crash stop did not produce a pull to one side nor any noticeable extra braking distance. Handling on fast corners was very good but a controllable slide now took place on negotiating very tight bends. However, the highest mean speed of all runs was achieved on this run 48 mph.

Run 8 — Running at 3-5 psi. When tire pressure was further reduced to 3 psi, resulting in a standing height of only 228 mm, the car pulled slightly to the left; handling response was "woolly", and the car had an uncertain gait, started to snake, and had a noticeable lurch on right hand bends. Change in handling, ride, and comfort, while it might mistakenly have been put down to poor road surfaces, would normally have induced the driver to stop and investigate tire inflation at the first convenient spot. The effect of gusts of wind on a car with fully inflated tires would have been little different.

The car was remarkably controllable — tire deflation effect could hardly be felt on a straight road, and the average speed attained was 44 mph. Inspection of the tire showed a 5/8" wide scrub mark on the mid sidewall about 3" from the rim. At the end of the run the pressure has risen to 5 psi.

Run 9 — Running without air in the tire. After a distance of 3½ miles there was a rise in pressure to 2 psi. It produced a sway — first noticed by the front seat passenger — comparable to towing a caravan at too high a speed. Reducing speed to weekend motorist's level immediately cancelled out this snaking on bends. The tire was warm to touch. The average speed was 41 mph.

Run 10 — Reinflated. When the tire was reinflated and pressure set at 27 psi front and 28 psi rear the ride, handling, and maximum speed through bends was exactly as in Run 1.

Test to Destruction: Rough Moorland Roads

The car was driven across rough moorland roads with gradients of up to 1 in 4. The left hand side rear tire was beyond all hope of partial reinflation as it had been deliberately damaged by two slits, made with considerable effort with a sharp knife, in the sidewall (at about mid height). This was undertaken to test the tire to destruction point under extreme conditions of torque reversal on the damaged tire carcass. The car was perfectly controllable and responded to steering and throttle in much the same manner as on the subsequent run with all tires correctly inflated.

The tire ultimately failed by throwing bits of rubber off, which had parted from the radial plies. The failure did not bring about anything more dramatic than the onset of a vibration some two miles before the carcass began to disintegrate, and the shedding of the lubricant, which deposited itself on the road and also spattered the windscreen of a car following.

TEST RUN SERIES IV

Self-Inflation After Standing Fully Deflated Overnight

All the air was removed by taking out the valve core. This was then replaced, and the car was left standing out in the open below 0°C overnight.

In the morning the tire pressure reading was zero.

It took several minutes before the deflated tire regained a round shape. Depending on speed of traffic and road surface temperature, it could take up to 2½ miles before a tire, which was once deflated to the point of puncturing the special lubricant canisters, was in full balance again, although it was reinflated to the correct tire pressure.

An out-of-balance vibration was noticeable particularly at about 40 mph. We attribute this to the delay in the equalization of the fluid film deposited on the periphery. This may be the only sign which a driver has after a puncture repair to indicate whether any new canisters containing the special lubricant have indeed been fitted.

The right hand front wheel position gave the greatest degree of sensitivity to imbalance throughout these tests.

Effect of Rain

Repeat of the earlier Lake Vyrnwy tests under drizzly rain conditions reduced the speed through bends, but throughout the tests, tire squeal was never of an order which was uncomfortable or would draw attention. The greater rolling resistance of a tire being steered when deflated could be detected; the front tire acted as an "extra brake".

Handling Stability

In contrast with the behavior of the car on standard crossply tires, easing the throttle after entering a curve too fast produced no slide or the start of a spin. This safety feature was noticed on both partially and totally deflated Dunlop Denovo tires, regardless of whether the tire was at the front or the rear. Surface texture as well as dryness of road had relatively little effect on this. Of course, one can enter any bend at such a speed that it is impossible to avoid loss of control. In the tests this could be countered safely by braking or by steering out of danger. The nearer the tire pressure to that recommended, the quicker the response to steering correction, and the more precise the course taken through a series of bends. Only if tire pressures (cold) were at less than ¼ of that recommended was there a distinct change in vehicle response.

Effect of Dry and Wet Roads

The repositioning of the deflated tire to the left hand rear gave similar results to those obtained previously in the dry, except that the effect of moisture on the road was noticeable in tight bends taken at speed. This resulted in a reduction of up to 23% in the safe cornering "g" loads. This is equivalent to a maximum speed reduction of 15% due to wet surfaces alone, or 30% due to a wet road surface

Typical Results Obtained on the Mid Wales Tests

Centrifugal acceleration = V^2/r where V = forward speed in ft/sec; r = radius of bend in feet; "g" = 32.2 ft/sec/sec. Limit of controllability before breakaway or snaking occurred.

Inflation of Dunlop Denovo condition	Radius	Road condition	V	Limiting lateral "g" loading
All fully inflated	90	dry high surface	34	.815
Left hand rear 5 psi only	90	dry high surface	29	.625
Left hand rear O psi only	90	dry high surface	28	.585
Left hand rear O psi only	90	wet high surface	25	.47
All fully inflated	160	tarmac, dry, slightly reverse camber	45	.85
Left hand rear 5 psi only	160	tarmac, dry, slightly reverse camber	41	.705
Left hand rear O psi only	160	tarmac, dry, slightly reverse camber	40	.67
Left hand front O psi only	160	tarmac, wet, slightly reverse camber	37	.585
All fully inflated	260	loose surface on top of tarmac, dry	55	.785
Left hand rear 9 psi only	260	loose surface on top of tarmac, dry	50	.65
Left hand rear O psi only	260	loose surface on top of tarmac, wet	47	.57

combined with running on a flat tire. There was a running-wide phenomenon and much less pull to the left. In effect, it is comparable to running in the wet with one bald rear standard tire. On the other hand a front flat Dunlop Denovo tire did not produce the same "wiggle" in the wet.

A surprising feature was that in the wet, even with the left hand side totally deflated, there was hardly any pull to one side. At speeds of 55 mph and above the car followed a straight course with both hands off the steering wheel. The faster we went the better the handling, except for an occasional side to side shuffle. It was easier to corner with power on, but this could result in oversteering if a corner were taken much too fast and correction attempted by easing off the throttle quickly.

Under heavy braking, 0.5 g – 0.7 g, even in the wet pull-up was dead straight with hands off the wheel. This is a dramatic improvement over the standard tire behavior. Under all stages of deflation, crash stop performance was good with either a flat front or rear tire.

Speeds of 65 mph were obtained on the short straight section under both dry and wet conditions even with totally deflated tires. Even so, the car could be controlled with ease with one hand. The fast run in the wet resulted in tire self-inflation values of between 3.5 and 4 psi.

APPENDIX II
FLAT TO COMPLETE DESTRUCTION TESTS IN DUNLOP DENOVO ON A FORD ESCORT 1100

Test Route

The roads in north Wales were dry throughout the test and carried normal mixed traffic, including buses and even pony trekkers. Traffic was intermittent, totally random, and in both directions. The roads were narrow, mostly between high hedges or dry stone walls with sharp changes in direction and camber. This called for frequent steering and check braking. It was impracticable therefore to achieve speeds in excess of 60 mph, regardless of whether the tire being tested was correctly inflated, or at low inflation pressure, running without air (by taking the valve core out), or damaged beyond repair. Driving was brisk, but the width of the road virtually prevented any attempt to drift the car round corners by applying rally techniques.

Test Procedure

On one such test the rear right side tire was deliberately damaged beyond repair by putting a 1″ wide knife cut at mid height right through the sidewall and its various rubber/cording layers. This immediately reduced tire pressure to zero. The size and nature of the cut being far more severe than an ordinary puncture, the special lubricant/sealant could not seal the cut to the point where a measure of self inflation would be operating.

Test Results

Characteristic stages of destruction test: right rear tire – After 18 miles the first slight oozing out was noticed, and there was also a slight side to side rock and imprecise steering round sharp left hand – but not the right hand – bends. All were perfectly easy to correct by the most inexperienced driver.

At 41 miles a further six cuts of 1″ width each right through the side wall were carried out at the roadside.

At 51 miles the slits were noticed to have elongated, the carcass began to break up, and the tire was hot to touch. Emergency braking from 40 mph now produced a slight "tail slide out", but controllability was still equal to that of a standard D 75 tired car (crossply) of correct inflation pressure all round.

After 58 miles a slight steering pull was noticed. Bump steer and rolling gait round corners had increased.

At 66 miles inspection showed that the slits had greatly increased in circumferential direction with the wheel. The feeling of some instability on left hand bends could be mistaken for partially defective shock absorbers. Negotiating right hand bends was still unaffected. An unfamiliar noise had developed. At 70 miles the noise had changed from a "flip flop" to a slapping sound transmitted through the body structure.

At 72 miles the car was still drivable on the straight with both hands off the steering wheel. On rough broken repair sections of the road this resulted in a slight pull to the left. The flapping sound was now quite loud.

At 74 miles the vibration became more severe, but even at 45 mph car held a straight course with both hands off the steering wheel. At 78 miles side to side flopping became distinctly noticeable at low speeds, peaking at 30 mph.

At 80 miles inspection showed that two of the 1″ slits were each about 8″ long. At 81 miles bits of the tread were hitting the inside of the wing, and severe body vibration was experienced. Drag of the damaged tire resulted in noticeable action of the differential gear; most of the power was transmitted through one wheel, but the car remained perfectly drivable, and as speed was increased from 30 mph (the shudder peak)

to 45 mph and over, the car became more pleasant to drive. The three slits had elongated to about 12″ each. At 88 miles the slits had joined up.

At 91 miles the noise was much worse; there was the impression of a "floating rear end", but the car was still perfectly controllable under all conditions of braking and maneuvering.

At 95 miles a sensation of imprecise steering round right hand as well as left hand bends was noted for the first time.

At 96 miles the car began to "waltz" and became distinctly throttle sensitive, power now being transmitted through the sound tires only.

At 98 miles it became apparent that the car behaved as a three wheeler with a driving wheel offset to the near side, the damaged tire merely acting as a steady support.

Tests were terminated in order to save the differential gear from being overloaded. The tire inboard sidewall was still intact, but the slit outside wall had almost totally separated.

No damage to the wheel rim had occurred since it had run on a layer of rubber even while undergoing the most severe maneuvers.

After the fitting of the replacement tire and wheel assembly no ill effects on steering, braking, or general behavior of the car could be detected.

Leftside Front Tire

The behavior of the front tire when approaching total destruction due to excessive heat build-up, and that of the totally deflated tire were different. To accelerate tire failure a no. 9 screw was screwed into the center of the tread and removed with a pair of pliers. This left a fair-sized hole, too large for the lubricant/sealant to seal. After about 25 miles a vibration of the front suspension/steering was detected, which peaked at 38 to 42 mph. With hands off the steering wheel, a pull to the left was observed, particularly at low speed, but one finger control of the steering was still achievable. The feel was that associated with a stiff steering knuckle joint rather than a flat tire. Due to the higher rolling resistance of the flat tire, braking produced a slight pull to the left similar to the effect of incorrectly aligned front wheels. Level crossings and humps in the road did not affect the steering. Running the car with this bias to the left, which was resisted by a constant slight pull on the steering wheel would have tired the ordinary family motorist after about 20-30 minutes. He would have stopped to investigate and would not have failed to notice that the tire was flat.

At 43 miles there was a change of note of the tire on the road – it now sounded like snow chains on soft snow.

At 50 miles, an outside visual check did not indicate that the carcass was breaking up, yet the lowering of the peak vibration from 38/42 mph to 34/38 mph, as well as tire squeal on sharp 90° turns, indicated that a change of tire structure was taking place. Uneven road surface and change in camber did not affect steering. Emergency braking from 50 mph was still in a straight line. More steering effort was required when negotiating right hand bends; on left hand bends, the tire tended to "squelch" out.

At 70 miles the passengers first notices occasional "bumps" when the tire ran hard on its wheel rim.

At 71½ miles the first audible warning "flap-flap" was noted. However, a defective left side front shock absorber would have given the same symptoms.

At 78 miles steering wheel shudder accompanied by the "flap-flap" noise of the tire was noticeable at around 40 mph. Emergency stops from 50 mph were still in a dead straight line.

At 80 miles a sharp 360° turn around a bollard was accomplished without difficulty, at moderate speed.

At 81 miles the passenger first noticed jolting and unusual tire noise.

At 91 miles there was a violent shudder, starting at 20 mph and peaking at 30 mph, and distinctly audible at 40 mph. The estimated point of carcass break-up was some 20 miles prior to this. The car was still capable of very precise maneuvers. The tire reaction was comparable to that of a conventional tire running flat on a well insert type special wheel rim, some 2 miles after tire-burst or total deflation had occurred. Violent shudder of the steering wheel could no longer be ignored.

At 92 miles, speeds had to be reduced since sidewall failure in the upper tread region resulted in some tire to wing contact, which was very audible.

At 95 miles pieces of tires were being flung off, but braking was still in a straight line. Raising the speed reduced the "flailing" noise.

At 98 miles the car became distinctly unstable when negotiating a 150° bend of about 300 ft radius at 40 mph. This still represents a limiting lateral force of the order of 0.30 g to 0.33 g, i.e., about 40% of the maximum attainable with all 4 Dunlop Denovo tires at their correct pressure or about 48% if standard crossply tires had been fitted. In other words on Dunlop Denovo of the correct pressure setting, the limiting speed would have been about 60-63 mph, and 55-58 mph on the standard crossply D75 type tires. The car was then braked to a standstill, which produced nothing "dramatic" – just a normal controlled change of lane and a steady slowing up.

Dunlop Denovo Tires on Ford Escort 1100

The MIRA Ride and Handling Course and the Steering Pad are accepted standard reference road sections which most UK vehicle or major accessory manufacturers and the AA use for vehicle model evaluation tests, since they incorporate most of the extremes of road hazard conditions likely to be met in Europe.

Two basic test series were carried out to determine:

1. time taken on the dry surface ride and handling course to complete a lap at maximum safe speed
2. steady state maximum speeds achieved, without applying opposite lock, to correct a breakaway tendency on 108 ft radius circle of flat uncambered steering pad.

These were each subdivided into stage by stage deflation of, in turn; the left front tire only, to be followed by the deflation of the left rear tire only.

Some typical test results **Conditions** **Dunlop Denovo Tires:**	**Observations**	**Measurements**
All 4 Dunlop Denovo tires at 22 psi (cold) 23 psi (hot)	Standard reference	Lap time: 1 min 46 sec equivalent mean speed: 47.2 mph
Left hand front tire deflated to 10 psi (hot)	No noticeable difference in handling characteristics.	Lap time: 1 min 48 sec Equivalent mean speed: 46.4 mph
Left hand front tire at vapor self-inflation pressure 4.2 psi (hot)	Ride comfort impaired; a little more steering correction required particularly on right hand bends and rapid change of camber sections.	Lap time: 1 min 54 sec Equivalent mean speed: 44 mph
Left hand front tire dead flat – 0 psi	Indifferent ride comfort slight lack of steering precision, but just as fast on short straight sections. Lack of precise positioning most noticeable on cambered sections.	Lap time: 1 min 56 sec Equivalent mean speed: 43.1 mph
Left hand rear tire at vapor pressure 4.2 psi	Distinctly imprecise handling, substantial oversteer induced on sharp turns and cambered sections which can best be controlled by closing throttle rather than by braking hard.	Lap time: 1 min 59 sec Equivalent mean speed: 42.1 mph
Left hand rear tire dead flat – 0 psi	Can still be driven fast on straight section but with noticeable loss of ride comfort; safe speed through most bends 10 mph lower than when fully inflated, distinctly snaky on bends, more so where change of camber or ripples are present.	Lap time: 2 min 8 sec Equivalent mean speed: 39.1 mph
Standard crossply		
(Dunlop D75 tires) All tires D75 crossply at manufacturer's recommended pressure settings 26 psi (hot) all round.	Ride not as comfortable, nor steering as precise as on Denovo tires.	Lap time: 1 min 47 sec Equivalent mean speed: 46.8 mph
Left hand rear tire at 15 psi	Distinctly "hairy" handling calling for more steering correction and speed of applying same, more than the average driver is capable of.	Lap time in excess of 2 min 8 sec, i.e. less than 39 mph mean speed

Comments

With a Dunlop Denovo tire controllability was assured right down to zero pressure with a deflating rear tire, but with standard tires at about half pressure in the critical rear tire there was real danger of loss of control of the vehicle.

Ride and handling tests on the crossply tires were abandoned as unsafe at about 12 psi (hot) setting.

The effect on steering of a deflating front tire is less noticeable with both Dunlop Denovo and standard crossply; on the other hand the car would be virtually uncontrollable with a rear standard crossply tire at really low inflation pressure, and at about 8-10 psi the tire would be rolled off its rim.

TEST SERIES (2)

Steering Pad Tests:
Step By Step Deflation Of Left Hand Rear Tire

There was remarkably little difference in the maximum steady speed which could be obtained on the Dunlop Denovo tires on the steering pad in dry and thoroughly wet conditions. All speeds refer to 108 ft radius.

Condition of Weight Transfer Due to Fast Cornering

- Maximum weight on deflated rear tire
- Minimum weight on deflated rear tire

Denovo Test Series					
Tire Pressure Condition	Weight Transfer on Deflated tire Max (A) Min (B)	Characteristic Speed Dry	Characteristic Speed Wet	Limiting Lateral Acceleration Dry	Limiting Lateral Acceleration Wet
(a) all tires at 23 psi		37 mph	36 mph	.85g	.81g
(b) left hand rear 10 psi	min weight	36 mph	36 mph	.81g	.81g B
	max weight	33 mph	32½ mph	.68g	.65g A
(c) Vapor pressure		35 mph	35 mph	.76g	.76g B
4.2 psi		31 mph	29 mph	. 6g	.52g A
(d) Dead flat 0 psi		35 mph	35 mph	.76g	.76g B
		30 mph	26 mph	.56g	.42g A
Test Series					
Standard Dunlop D75 crossply					
All tires at 26 psi		34 mph		.72g	
		rim almost touching tarmac			

Condition of rear tire and road surface	Loss in maximum safe speed	Loss in lateral stabilizing force
All tires at 23 psi (hot)		
wet	2½%	4½%
Deflated to 10 psi (hot)		
dry	11%(A) 2½%(B)	20%(A) 4½%(B)
wet	12%(A) 2½%(B)	23½%(A) 4½%(B)
Deflated to vapor pressure (4.2 psi hot)		
dry	16%(A) 5½%(B)	29½%(A) 10½%(B)
wet	21½%(A) 5½%(B)	39%(A) 10½%(B)
Deflated to 0 psi		
dry	19%(A) 5½%(B)	34%(A) 10½%(B)
wet	30%(A) 5½%(B)	50½%(A) 10½%(B)
Substituting 4 standard D75 crossply tires at 26 psi (hot) all round		
	8%	15½%

A most important safety feature is that there should be little difference in response to steering input between negotiating a right hand or a left hand bend when, without prior warning, a tire is deflating rapidly.

Expressing it another way the driver would have to reduce speed from the maximum safe speed by X% to be able to negotiate a 108 ft roundabout without applying negative lock, or taking a different line from the constant 108 ft radius – which may be impossible due to other traffic – or being in dange of spinning off. We used the fully inflated Dunlop Denovo tires all round as a reference standard, since this is largely determined by the car design characteristics.

Taking the dry condition with all Dunlop Denovo tires as their correct pressure as 100%, then the lowering of the safe speed and lateral stabilizing force would be as shown on the table.

APPENDIX III
DUNLOP DENOVO TIRE TESTING ON ROVER 3500

It was important to ensure that the test conditions were as representative of real life operation as possible, and hence they were designed to incorporate many features which the standard basically flat proving grounds could not provide. Above all there was a wide range of ambient and road surface conditions and the presence of an element of unpredictability of the behavior of other road users. Having obtained special dispensation from the appropriate Government authorities, some tests were carried out on different types of public roads to the point of total failure of the Dunlop Denovo tires.

Hot Weather Testing Conditions

One such series of extended testing on the Denovo tire – fitted to a Rover 3500 with a top speed capability of about 115 mph on the flat – took place under conditions of extreme heat in Italy. A 30 mile circuit in the Abruzzi Mountains and on the Adriatic Coast embraced the following features.

1. A new stretch of high speed motorway without speed restrictions, with a good surface, long sweeping fast bends, viaducts and cuttings which produced side wind effects.
2. Asphalt-covered trunk roads which imposed a frequent need to pull out and overtake slower traffic. The camber, while not as pronounced as in France, was more noticeable than in the UK.
3. Road building and repair sections having loose gravel and ill-defined edges, generally bumpy uneven road surface of varying texture and unpredictable "grip".
4. Town and village streets.
5. Steeply graded roads out of and into deep valleys involving many left hand and right hand bends often in rapid succession, and hair pin bends, as well as blind 120° turns.
6. Narrow but fast short straight runs across a mountain plateau.
7. Level crossings.
8. U turns.

In order to simulate a Continental type holiday, three adults plus a boot load of 350 lb were carried throughout the test and the petrol tank was kept at half full or above.

Ambient air temperature ranged from 35°C to 45°C.

Road surface temperature from 38°C to 50°C.

SUMMARY OF TEST RESULTS

Front Tire

Self Inflation From Starting Flat – Starting with a flat tire, it took some 3½-5 miles before self-inflation to just below 5 psi had taken place. The distinctive run flat tire noise which was most noticeable when braking hard then disappeared and steering response was normal.

Front tire gradually deflating: first signs of unusual ride and handling characteristics.

1/2 Pressure – A degree of bumpiness could be felt as soon as tire pressure had dropped to half normal pressure but even then at speeds up to 60 mph steering was neutral and all handling responses normal.

1/3 Pressure – At 1/3 normal pressure a slight pull towards the side on which the flat tire was fitted was noted, as well as a certain amount of roughness of ride and a slight side to side movement at speeds up to 60 mph. This slight pull was, however, easily controllable.

Increasing the speed virtually eliminated the pull. At speeds from 70 to above 100 mph the car travelled in a straight line with the driver's hands off the steering wheel.

Approaching corners and bends or executing U turns at speeds close to the limiting grip of a fully inflated tire did produce some lurching and a distinct tire squeal, but a little more steering effort easily redressed the balance.

Self Inflation Pressure 5 psi – A certain amount of tire imbalance was detectable particularly at higher speeds – which would induce a cautious driver to get out and inspect wheels and tires. He could not fail to see that the tire was flat.

Deflation From 5 psi To Zero Pressure – A significant change of ride and handling response took place when the valve core was removed. This resulted in a loss of the self inflation pressure build-up and a gradual oozing out of the special fluid. The lubricating/self-inflating/puncture-sealing characteristics were thus progressively lost over a distance of several miles. The effect was similar either to carelessly omitting to insert the valve core properly, or to very large punctures. The net result was a distinct pull to one side, which was, however, still controllable by single-hand steering. There was also a distinct and penetrating tire noise. This hum was most noticeable at low speed and it was impossible to disguise or swamp it, even at normal speeds by music volume on the car radio. Also a slight shudder was felt right through the steering wheel. Normal and emergency braking were still in a straight line, but in emergency braking the driver was required to grip the wheel to prevent the car from pulling to one side just before coming to rest.

When a large proportion of the special liquid had oozed out, the tire deflation was sufficient for the tire wall to leave its imprint on the melting tar and also on concrete surfaces, particularly when cornering hard or braking, but at no time did the wheel rim come in contact with the road surface.

Car controllability was surprisingly good. Taking a series of slight left-hand – right hand – left hand bends in succession and at speed it was possible to put in too much steering effort, resulting in some oversteering and snaking. All that was required of the driver to bring the car back onto a pre-determined and predictable course was a reduction of road speed. Generally easing off the throttle achieved that; in extreme cases check braking was needed. In no instance did we experience a skid, let alone a spinning of the vehicle, even when the tire outer wall temperature had risen nearly 60°C above road and ambient air temperatures to just below the boiling point of water.

Rear Tire

Gradually Deflating – The effect of a deflating tire on the rear was distinctly more pronounced and drew attention to itself much earlier.

½ Pressure – At half standard tire pressure setting, the impaired ride comfort was more noticeable to the rear passenger than to the driver. On a straight road the car held a dead straight line even with the driver's hands off the steering wheel. However, the driver had to use more judgment than usual when negotiating bends at speed.

A transient steering or check braking input produced an amount of weaving and tire squeal.

1/3 Pressure – At one third standard tire pressure setting the rear end "woolliness" was slightly more pronounced and distinctly different from the Rover 3500 in good trim. But it was no worse than on some softly sprung cars with four or five bar live back axle suspension.

A poor road surface and an imperfect or worn suspension damper could have produced similar ride and handling effects, particularly at speeds above 80 mph. But at 100 mph the car began to snake from side to side, particularly under the influence of cross winds and lateral road unevenness. Even so, fast bends did not call for a slackening of speed, and emergency stops from 112 mph were in a straight line. The car retained basic predictability of response to sensible application of steering and brakes when ascending or descending hair pin bends.

Self Inflation Pressure 5 psi – The degree of restraint required of the driver became more pronounced as tire pressure dropped even further. At 5 psi wallowing on bends called for gentler and more progressive application of steering and brakes, since precision of handling had deteriorated markedly. It was now relatively easy for the driver to overreact if he ignored the effect of weight transfer on tight corners, particularly when associated with a reversal of camber. He could be called upon to respond very quickly to correct the onset of a slide. Applying more power worsened the situation whereas closing the throttle easily restored the situation to one of safe progress.

Ride comfort was significantly altered on uneven road surfaces, due to the reduced "springiness" of the tire. An occasional bottoming of the tire sidewall was not uncommon.

Distress Warnings

While the rear passengers were the first to complain of the slight oscillating yawing motion, the driver received three distinct tire or mechanical distress warnings:

1. a vibration would set in at about 50 mph and finally smooth out at 75 mph
2. a distinct fall off on maximum speed attainable on gradients
3. a rougher than usual ride.

The smaller than standard rolling radius of the severely deflated tire of course accounted for most of these phenomena. Depending on road unevenness, camber, and side to side weight transfer while cornering, the deflating tire's radius was fluctuating significantly. Power flow on the straight would then no longer be one half to the left side the other half to the right side rear wheel. The more the tire deflated the greater the action of the differential in the back

axle and hence the "wander" of the back end of the car.

5 psi To Zero Pressure – A flat tire produced "woolliness" on the back end of the car when negotiating bends, and a more pronounced tire noise. At 25 mph the "wander" or "wobble" of the car could no longer be ignored, particularly by a back seat passenger. Sudden changes of direction, which were a feature of the mountain route, resulted in the car weaving as it emerged from such bends. Even if the driver had ignored the unusually skittish behavior of his car, the lack of steering precision and the shudder through the steering wheel could not have gone unnoticed. Yet at no time was the car uncontrollable.

Zero Pressure – When the self-inflating pressure had disappeared and the tire was truly flat, a characteristic and distinct lateral movement, a kind of shuffle of the rear tire on the road surface, could not be ignored at speeds from 20 to 30 mph and gave unmistakable indications of something being amiss. Increasing speed reduced the roughness of the ride – though at 35 mph the tire noise could not be mistaken or ignored by the average motorist.

At the same we found it possible, and safe, to travel for some 20 miles on the motorway at above 95 mph after ignoring the distinct vibration period (50-75 mph). However, we doubt whether the average motorist would have been prepared to accept, without stopping to investigate, the level of ride discomfort and the "waltzing" progress on fast bends, which called for concentration and rapid correction of the oversteering characteristic of the car.

At no time did the deflating tires bring about a loss of control which could produce a skid, spin, or failure to come to rest in a straight line.

Other Tire Conditions Which Could Be Mistaken For A Deflating Tire

1. Of course slight vibration transmitted through the car's structure is not confined to deflating tires alone. A dynamic imbalance of the tire wheel combination could also excite this.
2. The first indication of a partial separation of the peripheral steel braced tread from the rest of the tire carcass, which is a very rare occurrence and usually due to poor interlayer bonding, can produce similar vibration and noise characteristics.

   But side to side wallowing is a feature of a deflating tire and cannot be induced by the other tire conditions.

   The more familiar a driver is with the ride and handling characteristics of his car, the sooner he would note the distinct change which occurred once tire pressure had dropped to or below the self inflation pressure of about 4-5 psi.

   But even those who had never previously driven this type of car or tended to be insensitive to a most unusual extent would not be subjected to an unacceptable risk – other than needless aggravation, irreparable damage to the tire, or the eventual total destruction due to heat generation causing the tire's internal fabric to fail.

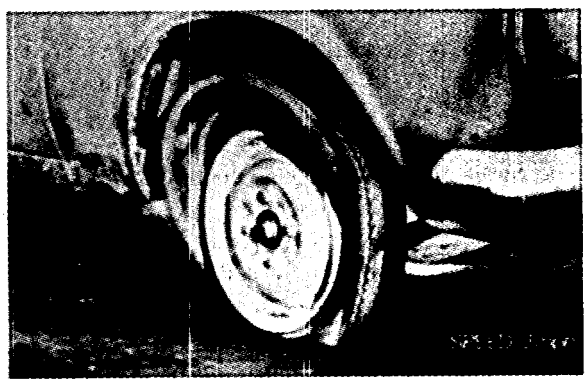


Figure 1. Conventional punctured tire badly "mangled" by conventional wheel rim. Note that outer bead of tire has sunk into wheel well.

Figure 2. Conventional tire and wheel assembly. Car has spun off in Slalom Test. Where has the tire gone?

Figure 3. Conventional tire and wheel combination: punctured wheel has lost all cornering force and is on point of being rolled off.

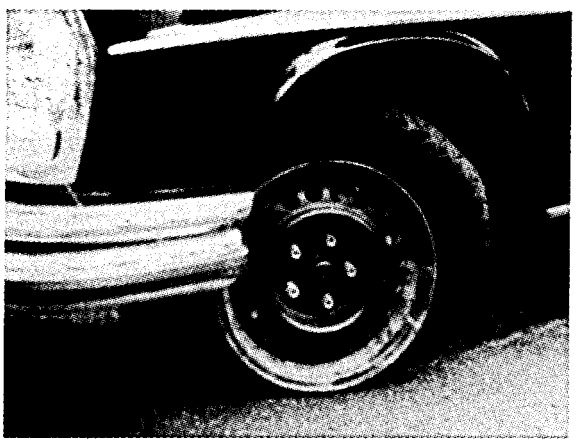


Figure 4. The tire is off! Conventional assembly — all steering and braking forces are resisted by rim in contact with the road surface.

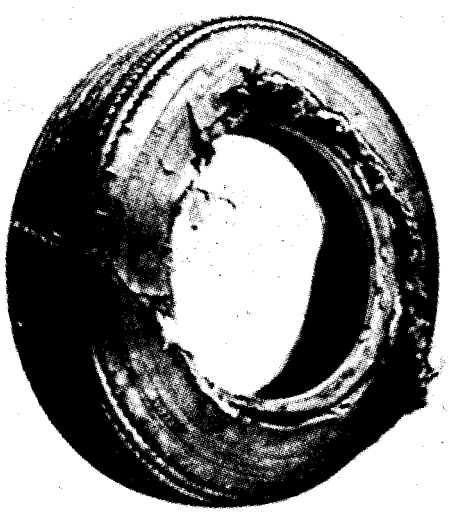


Figure 5. Curbing (concussion damage) leading to blow-out. Tire ran for 200 yards after blow-out.

Figure 6. Tire has been ripped and car has been driven a further 2 miles with wheel rim to road contact.

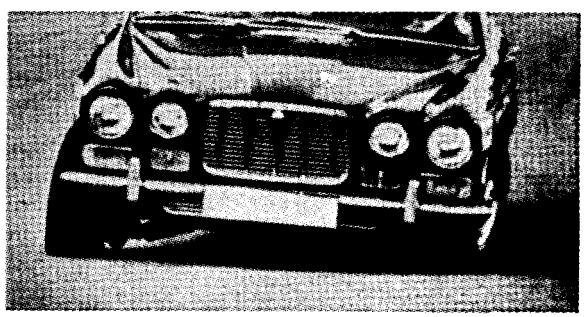


Figure 7. The Safety Wheel — Practically all cornering forces are provided by fully inflated tire. Punctured tire trails helplessly but does not roll off.

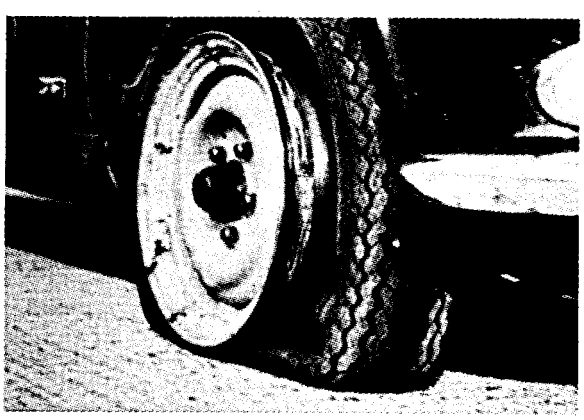


Figure 8. The Safety Wheel — The punctured tire is off! But beads cannot sink into wheel well. No rim to road contact.

Figure 9. Dunlop Denovo self inflation pressure 4.5 psi.

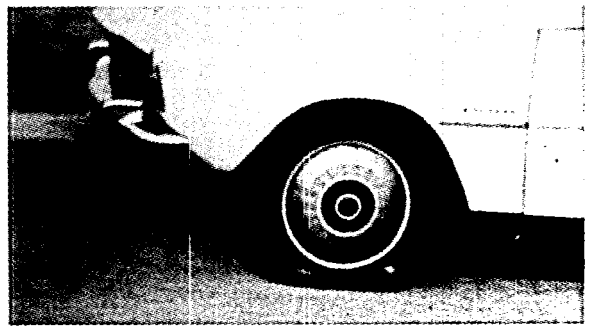


Figure 11. Dunlop Denovo running fully deflated (zero pressure) at 105 mph.

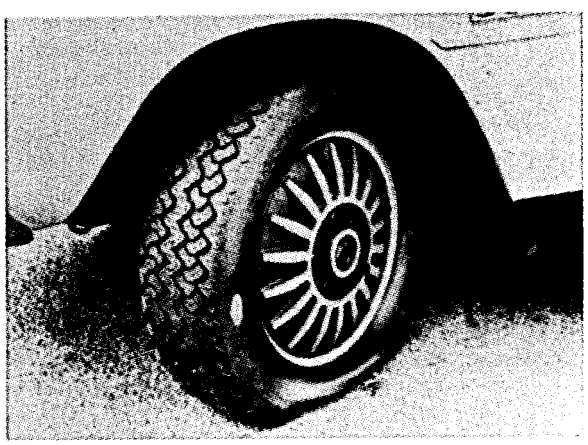


Figure 10. Dunlop Denovo at zero pressure.

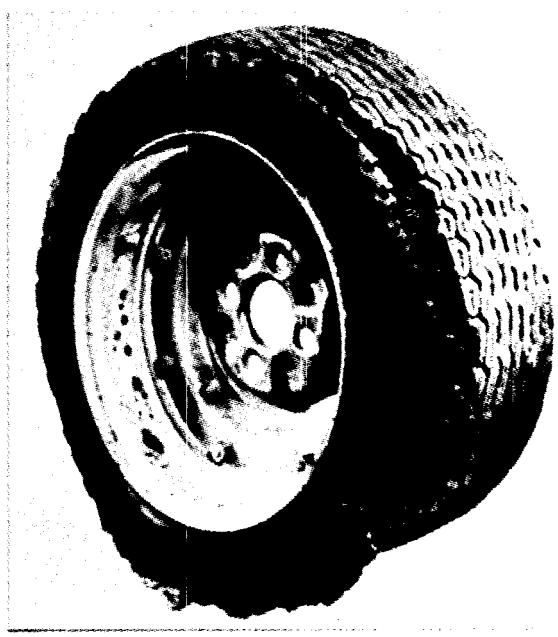


Figure 12. Dunlop Denovo After 100 miles running flat at zero tire pressure. Cause of failure: jagged hole in tread shoulder region.

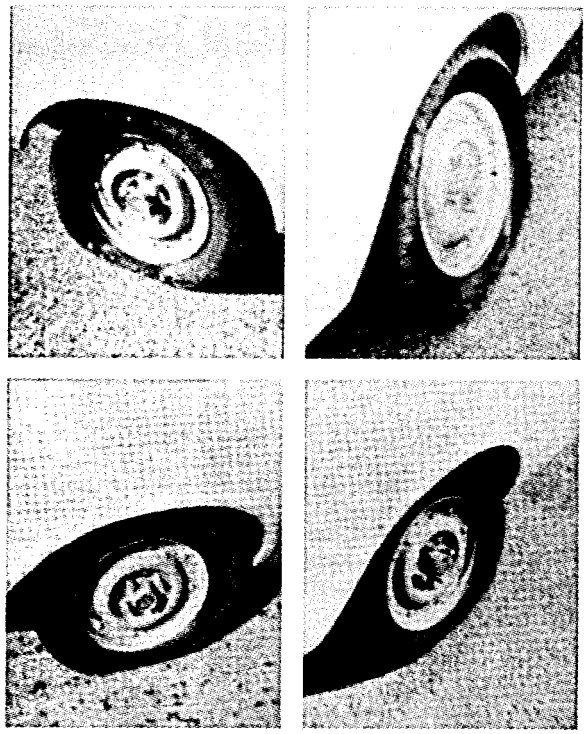


Figure 13. Dunlop Denovo with deliberately pierced side walls: run to total destruction at zero tire pressure. Stage 1: After 18 miles sealant/lubricant oozes out.

Stage 2: After 66 miles 1" cuts have elongated to 2".

Stage 3: After 81 miles 1" cuts have elongated to 12".

Stage 4: After 98 miles 1" cuts have almost joined up. Tread section is held to carcass by strands of Rayon cording only.

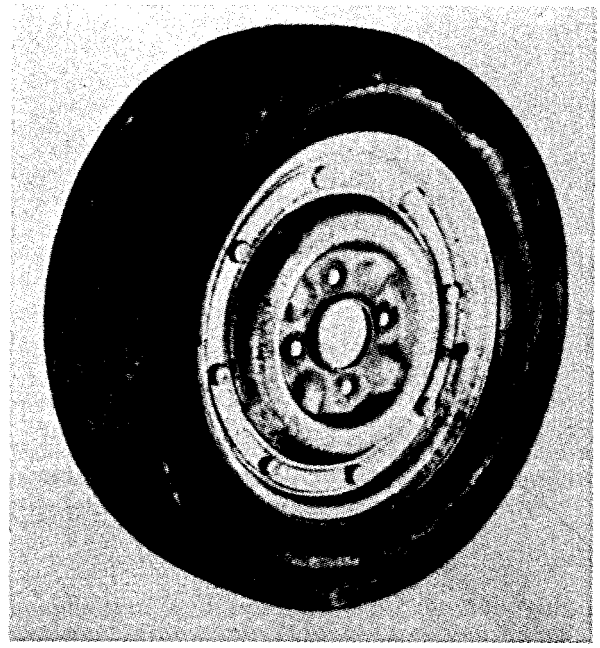


Figure 14. The End of the Story: Dunlop Denovo Tire. Severely damaged on side wall – 8 cuts of 1" each through side wall. After being run to total destruction in 98 miles it still provided some side force!

CHARACTERISTICS OF JAPANESE ESV TIRES

HIDEO SAKAI
Third Research Dept.
Japan Automobile Research Institute, Inc. (JARI)

INTRODUCTION

We have carried out experimental study on ESV tires in parallel with the accident avoidance tests of Japanese ESVs. We shall describe the outline of its results here.

Japanese ESVs set importance on steering and handling characteristics, and use large sized radial ply tires with 60 percent height-width ratio. To find out the characteristics of ESV tires, both indoor tests and on-the-pavement tests of four kinds of tires – two kinds of ESV tires, a conventional tire and a standard tire for pavement test under ASTM – have been conducted. Simultaneously, the brake system efficiency of ESVs has been studied.

Sizes and test loads of these tires are shown in Table 1, while their appearances are illustrated in Figure 1. I shall insist hereupon that the ESV tires are of high safety due to special structure.

TABLE 1
SIZES AND TEST LOAD OF TIRES

TIRE NAME	TIRE SIZE DESIGNATION	APPLIED RIM	CROSS-* SECTION WIDTH mm	OUTER* DIAMETER mm	TEST INFLATION PRESSURE AND LOAD	
					I.P. kg/cm^2	LOAD kg
ESV-1	DR60H14	6J X 14	621	621	1.7	300
ESV-2	BR60SR14	5½J X 14	193	610	2.0	350
CONVENTIONAL TIRE	6.45-14-4PR	5J X 14	172	634	1.7	350
ASTM STANDARD TIRE	7.50-14-4PR	5J X 14	183	690	1.7	500

*MEASURED AT TEST INFLATION PRESSURE

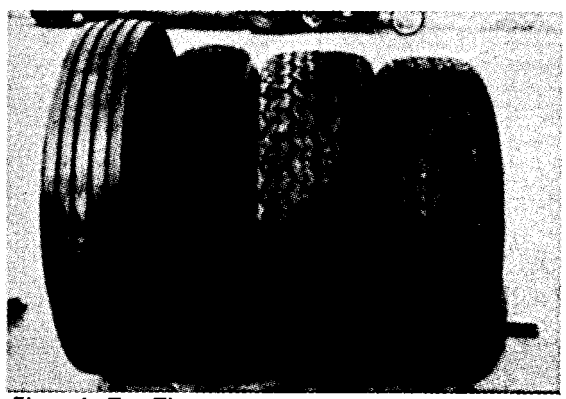


Figure 1. Test Tires

INDOOR TESTS

Tester

It seems that the use of an indoor tester is the most appropriate to conduct delicate comparative tests on tire cornering characteristics and braking characteristics. Figure 2 shows a dynamic tire testing machine of JARI for indoor test. Its general specifications are indicated in Table 2.

One of the most important points of an indoor tester is the properties of the drum surface. The drum surface of this tester can be easily replaced, but in our tests we used the most simple knurled and hard chromium plated surface which is shown in Figure 3.

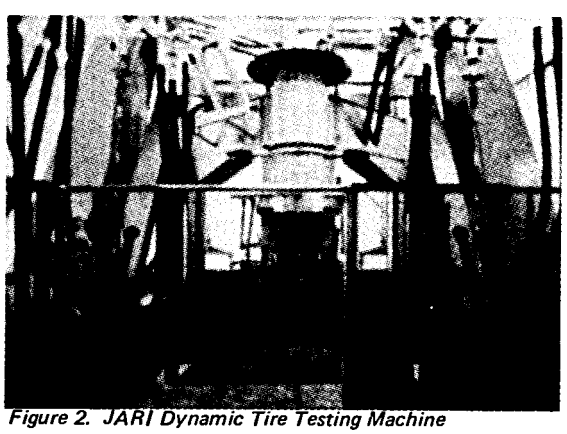


Figure 2. JARI Dynamic Tire Testing Machine

Test Results

Cornering Power. Cornering power is one of the most important features among cornering characteristics of tires. Figure 4 indicates the variation of cornering power coefficients with load. In the neighborhood of operative load, the ESV-1 tire has the largest coefficient, and next is the ESV-2 tire, a standard tire and a conventional tire in this order.

From the viewpoint of the changing rate of cornering power to load, the conventional tire is used in a load of a small changing rate, while ESV tires are

employed in a load of their considerable changing rate. This is because the latter are large in size.

Thus, Japanese ESV tires may be said to be capable tires and to bring satisfactory effect into the steering performance of the vehicles.

Cornering Characteristics at Large Slip Angles. To avoid confusion, I wish to explain what is meant by the following mentioned forces.

Cornering testers manufactured in Japan are mostly designed to measure rectangular force to the direction of wheel travel, therefore, cornering force, in many cases, is used as the term which means rectangular force to the direction of wheel travel.

SAE, however, defines cornering force as force in axial direction of tire wheels. If we define forces acting on the tire in horizontal plane as in Table 3, there are the following relationships among them (see Figure 5).

$$F = \sqrt{F_x^2 + F_y^2} \quad (1)$$

$$F = \sqrt{F_{x'}^2 + F_{y'}^2} \quad (2)$$

$$F = F_{y'}\cos\beta + F_{x'}\sin\beta \quad (3)$$

$$F = F_{x'}\cos\beta - F_{y'}\sin\beta \quad (4)$$

β = slip angle

TABLE 2
SPECIFICATIONS OF DYNAMIC TIRE TESTING MACHINE

TEST TIRES	CAR TIRE AND BUS & TRUCK TIRE
LOAD	100 kg – 3000 kg
DRUM DIAMETER	3500MM
DRUM AND TIRE DRIVE SYSTEM	OIL MOTOR 200PS
MAX. BRAKING AND DRIVING FORCE	1000 KG
VELOCITY	2 km/h – 350 km/h
SLIP ANGLE	-10°–0°–+90°
CAMBER ANGLE	0°– +20°
DRUM SURFACES	ALUMINUM HARD CHROMIUM PLATED KNURLED SURFACE SAFETY WALK ASPHALT ETC.
MEASUREMENT ITEMS	LOAD F_Z CORNERING FORCE $F_{Y'}$ ROLLING REGISTANCE $F_{X'}$ SELF ALIGNING TORQUE SLIP ANGLE SLIP RATIO CAMBER ANGLE
REMARKS	TIRES CAN BE ROLLED AT THE OUTER AND INNER SURFACES OF DRUM, AND HYDRO-PLANING CAN BE TESTED

Figure 6 shows the measurement results of cornering force $F_{y'}$ and rolling resistance $F_{x'}$. From this figure, it is clear that a tire with larger cornering power has their peak values at smaller slip angle.

Figure 7 indicates sideway force F_y obtained from equation (3). From Figure 7 it is evident that sideway force is maximum at slip angle of 20-30 degrees and almost uniform at larger slip angles. We shall call the coefficient obtainable by dividing the maximum value of sideway force by load the maximum sideway force coefficient. This maximum sideway force coefficient differs according to the shape and conditions of road surfaces, and sideway force shall be, therefore, studied below.

TABLE 3
DEFINITION OF FORCES

SYMBOL	EXPLANATION	TERM
F	TOTAL FRICTIONAL FORCE IN HORIZONTAL PLANE	TOTAL FORCE
FX	FORCE IN THE LONGITUDINAL DIRECTION	DRAG
FY	FORCE IN THE AXIAL DIRECTION	SIDEWAY FORCE
FX'	FORCE IN THE OPPOSITE DIRECTION TO WHEEL TRAVEL	ROLLING RESISTANCE
FY'	FORCE IN THE PERPENDICULAR DIRECTION TO WHEEL TRAVEL	CORNERING FORCE
FZ	FORCE IN THE VERTICAL DIRECTION TO ROAD SURFACE	LOAD

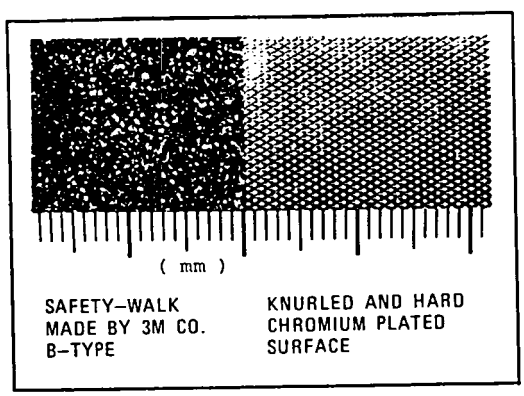


Figure 3. Surface of Test Drum

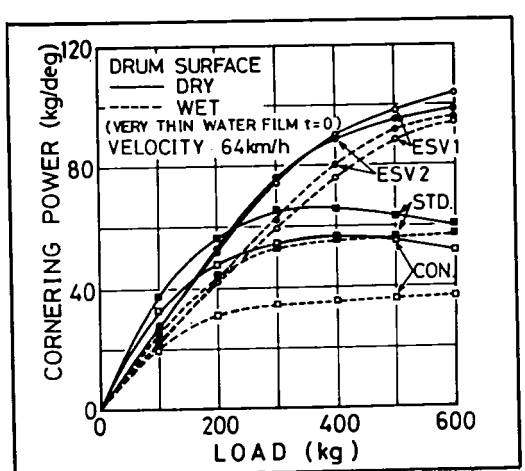


Figure 4. Variation of Cornering Power with Load

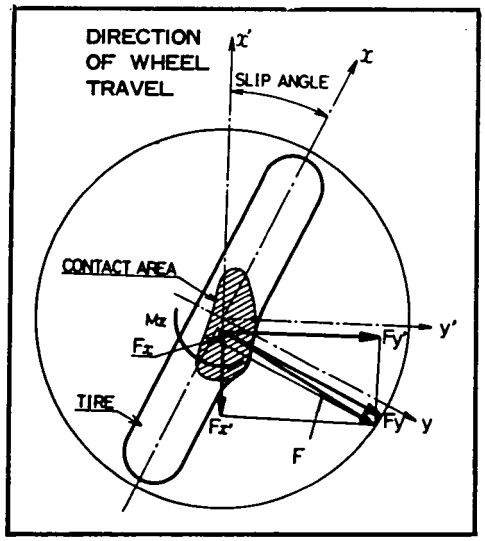


Figure 5. Relations Among Forces

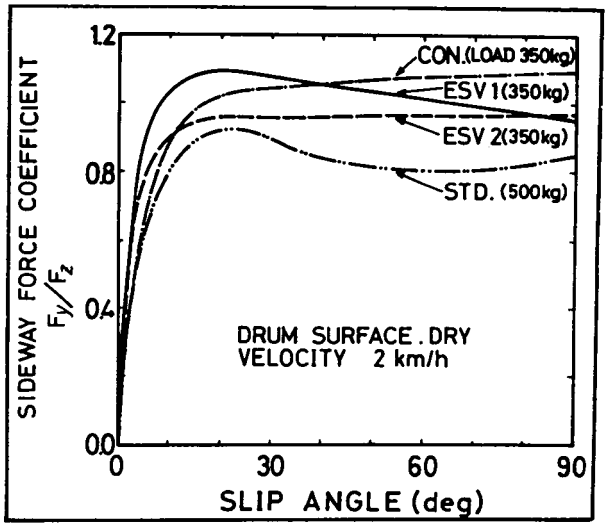


Figure 7. Sideway Force at Large Slip Angles

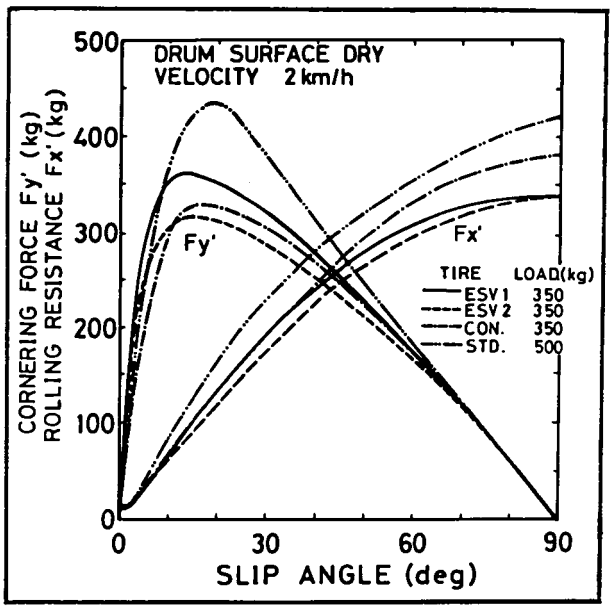


Figure 6. Cornering Force and Rolling Resistance at Large Slip Angles

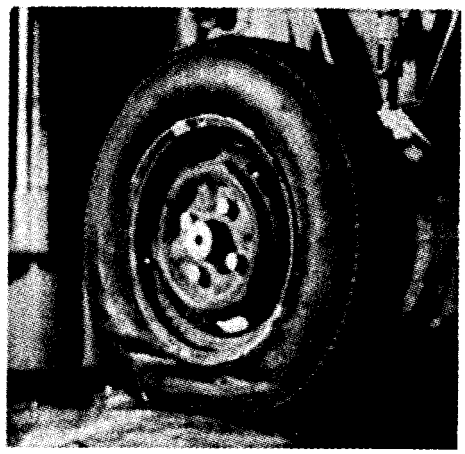


Figure 8. Buckling on the Sidewall at slip angle 30 deg. load 500 kg.

In the case of the conventional tire, buckling appeared on the sidewall, as shown in Figure 8, at large loads and large slip angles, and cornering force fluctuates. Buckling is feared to occur during quite sudden turning such as a J-turn test, and has a danger that it makes rim flanges contact the road surface. The large-sized ESV tires suffered no buckling even under load conditions of double operative load.

Cornering Characteristics at Large Camber Angles. A change of sideway force F_y due to a slip angle in the case when a camber angle is formed, is illustrated in Figure 9 [1]. Sideway force at zero slip angle is named camber thrust, and effects of camber thrust on sideway force are large at a small slip angle. However, a change of the maximum sideway force

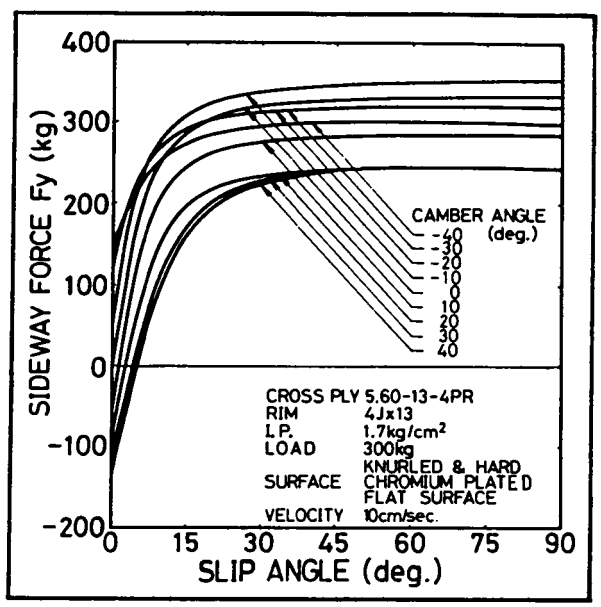


Figure 9. Sideway Force at Large Camber Angles

seems not to depend on the effects of camber thrust but the change of frictional coefficient itself due to camber angle. This change is shown in Figure 10.

The reason why the formation of a negative camber angle increases the sideway force coefficient is as follows; as shown in Figure 11, a tire is deformed by the sideway force and one side of tire tread apart from the road surface, such deformation is also caused by a camber angle, and the deformation due to the sideway force and deformation due to a negative camber eliminate each other to put the tire tread in the most ordinary contact state. Thus, the sideway force coefficient will become maximum at a small negative camber.

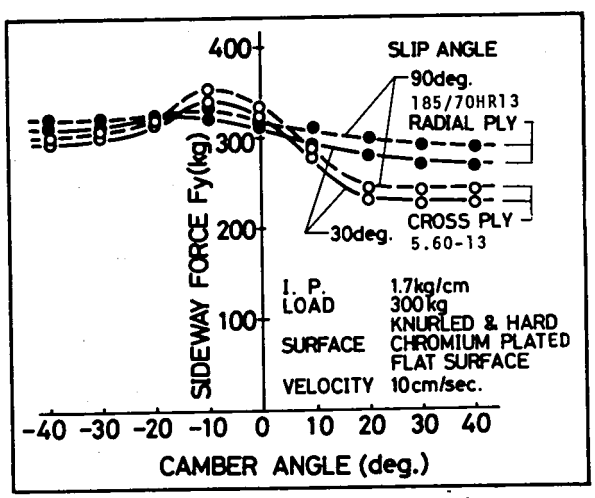


Figure 10. Sideway Force at Large Camber Angles

Figure 11. Deformation of Contact Shape by the Sideway Force

Figures 10 and 11 show a comparison between cross ply and radial ply tires. ESV tires of similar quality as this radial ply tire, but of different size, have a smaller change in the maximum sideway force coefficient due to a camber angle than a cross ply tire.

Characteristics When There Is Water Film. In Japan, there are comparatively many rainy days, and the rate of accidents on expressways on rainy days is larger than on fine days.

One of the main causes is sliding between tires and the road surface. Characteristics of tires, therefore, on the wet road surfaces must also be taken into serious consideration.

First of all, the effects of water film on braking and cornering characteristics, which were studied in the indoor test, and hydroplaning characteristics shall be examined. Tests were conducted by using dynamic tire testing machine, spreading water film over the inside of a drum and running tires on it, thereby measuring $F_{x'}$ and $F_{y'}$.

Figure 12 shows a change in cornering force $F_{y'}$ and rolling resistance $F_{x'}$ with velocity. As shown in

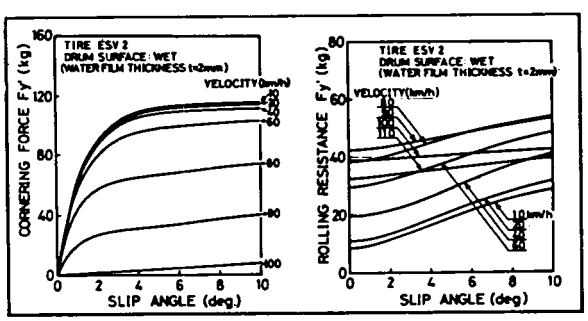


Figure 12. Variation of Cornering Force and Rolling Resistance with Speed on a Wet Drum Surface

Figure 12, the maximum value of cornering force decreases with velocity increasing, and rolling (running) resistance becomes considerably large, with velocity increasing, due to the addition of water resistance. The maximum cornering force coefficient μ_s ($F_{y'}/F_z$) and the cornering power coefficient (cornering power/load) change with speed as shown in Figure 13. In the case of very thin water film, they change slightly with speed. But, when water film becomes thick and speed approaches to NASA velocity $V = 63\sqrt{P}$ V; km/h, P; inflation pressure – kg/cm^2, where hydroplaning occurs, and the two coefficients suddenly decrease.

Figure 14 shows a change in the braking force coefficient (braking force/load) with speed. A change caused by velocity in maximum braking force and locked braking force is indicated in Figure 15. In this case, braking force includes such running resistance (water resistance) as Figure 12, it seems effective to study cornering characteristics in order to look into the effects of water film on the performance of automotive tires.

The drum surface, used in this test, is similar in properties to the polished concrete surface, which is

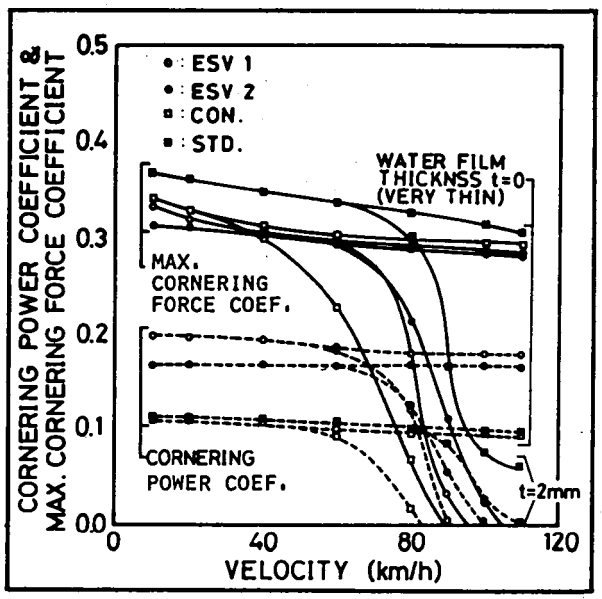


Figure 13. Influence of Water Film on Cornering Properties

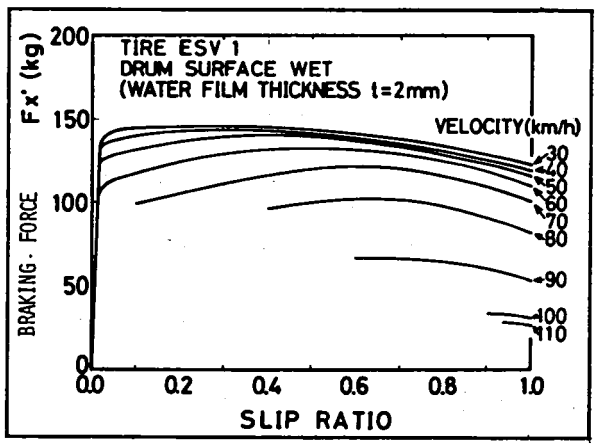


Figure 14. Variation of Braking Force with Speed on a Wet Drum Surface

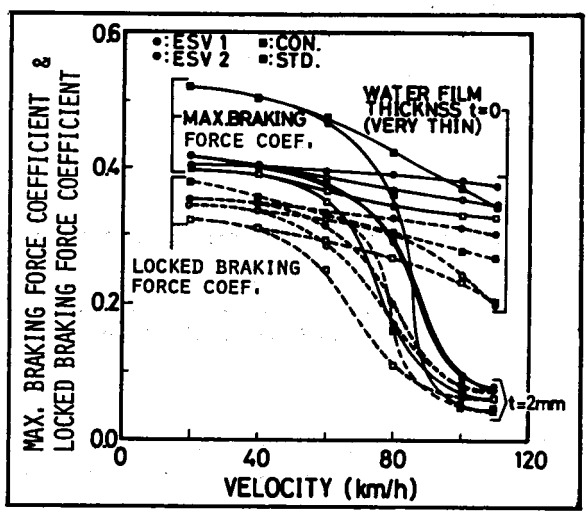


Figure 15. Influence of Water Film on Braking Properties

mentioned later. The roughness of drum surface is very important under wet condition. For your reference, test results which were measured previously by using the coarse drum surface are shown in Figure 16 [2]. This drum surface was sticked

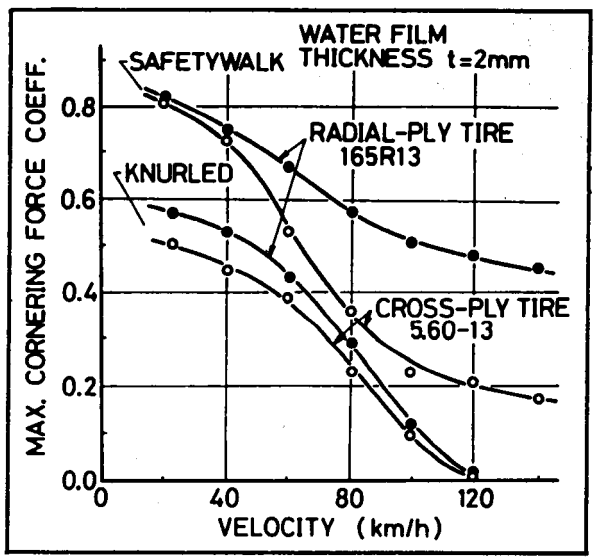


Figure 16. Influence of Water Film on Coarse Surface

with an emery paper tradenamed "Safety Walk" which is medium type and made by 3M Co. Tires used were those of the same tread pattern as ESV tires, but of different size, and a cross ply tire. It is clear from the figure that in the case of water film of 1 mm thickness, radial tires show better performance than cross ply tires.

A conclusion has been reached that although the drum surface is coarse, when water film becomes 5 mm thick, a difference in performance of tires is almost nil.

BRAKING CHARACTERISTICS ON ACTUAL PAVEMENTS AND BRAKE SYSTEM EFFICIENCY OF ESVs

An indoor tester is effective for delicate, cooperative tests, but an on the actual pavement test is more effective for the comparison of limit characteristics, namely the sideway force coefficient and the locked braking force coefficient.

Braking Force Tester

As shown in Figure 17, this tester is equipped with a test wheel as a fifth wheel under a truck, and tire rotation is conveyed, through gears and torque meter, to an oil motor. Tire rotation is controlled by adjusting oil flow with a throttle valve. Outlined specifications of the tester are listed in Table 4.

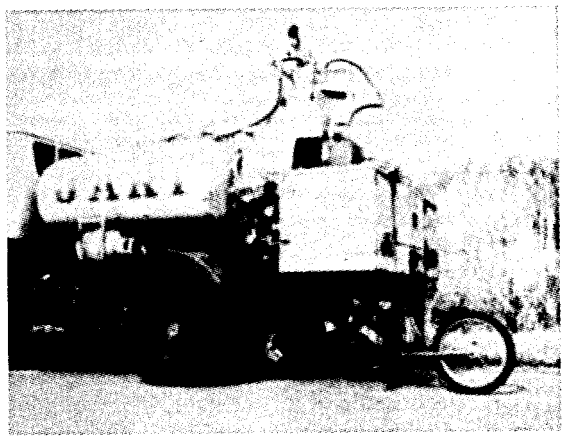


Figure 17. JARI Braking Force Tester

TABLE 4 SPECIFICATIONS OF THE BRAKING FORCE TESTER	
TEST TIRE	CAR TIRE
LOAD	300 kg – 500 kg
LOAD SYSTEM	AIR CYLINDER
VELOCITY	10 km/h – 90 km/h
SLIP RATIO	0 – +1.0
MEASUREMENT ITEMS	BRAKING TORQUE SLIP RATIO

Test Results of Braking Characteristics

Relations between the slip ratio and the braking force coefficient were measured regarding four kinds of tires and three types of road surface conditions by using the braking force tester. Three examples of the measurements are shown in Figures 18-20. Measurement was performed as follows: While the tester was running at constant velocity, tire rpm was lowered gradually to change the slip ratio from 0 to 1, thereby braking torque, and test tire rpm, were measured.

Figures 21-23 show comparisons among tires on a change of the maximum braking force and locked braking force coefficient due to speed.

Brake System Efficiency of Japanese ESVs

The brake system efficiency of ESV is defined as follows:

$$E = \frac{D_i}{D_a} \qquad (5)$$

where D_i = Ideal stopping distance
D_a = Actual stopping distance

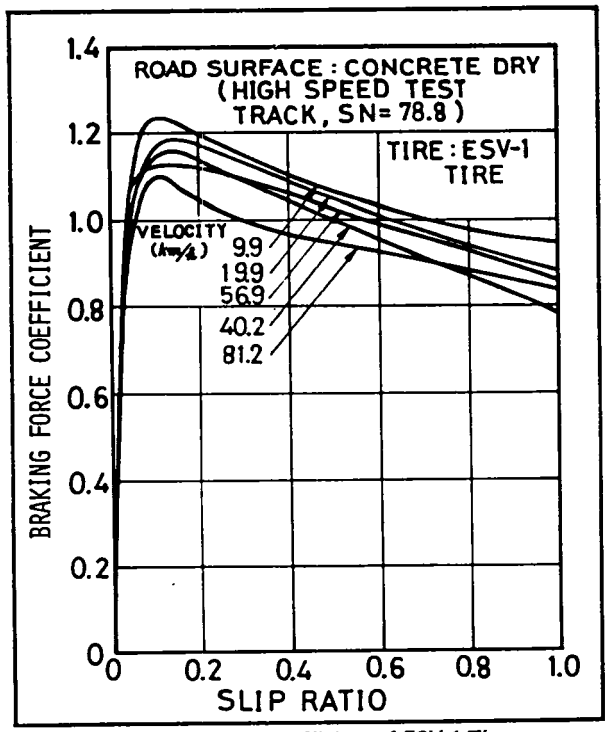


Figure 18. Braking Force Coefficient of ESV-1 Tire

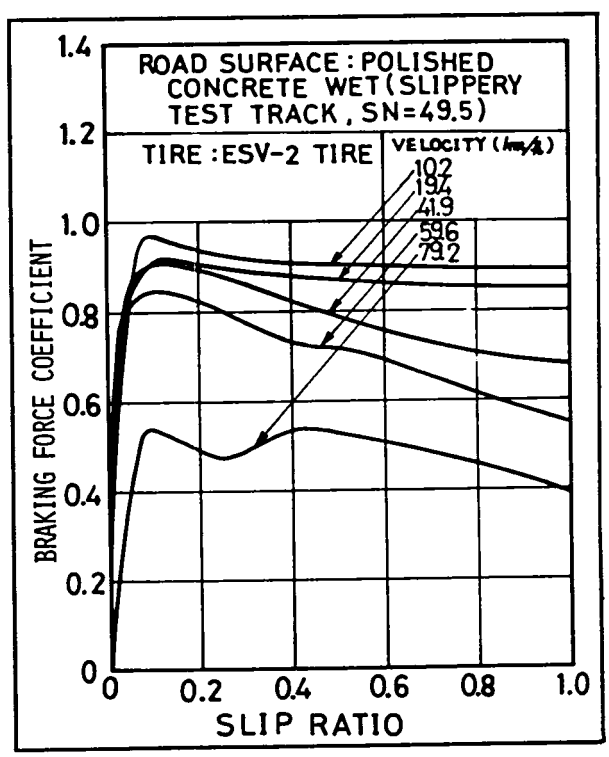


Figure 19. Braking Force Coefficient of ESV-2 Tire

When an empirical formula for the maximum braking force coefficient is represented by the following quadratic equation with regard to velocity, then the constants of each term are listed in Table 5.

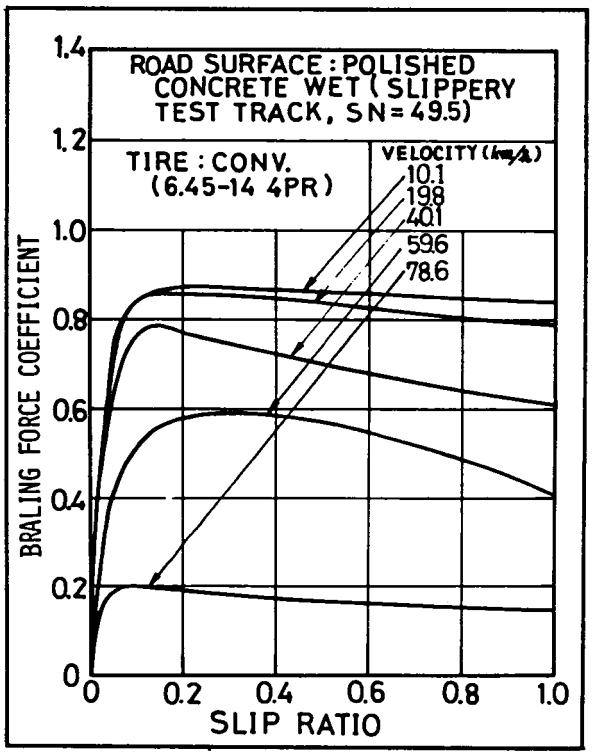


Figure 20. Braking Force Coefficient of Conventional Tire

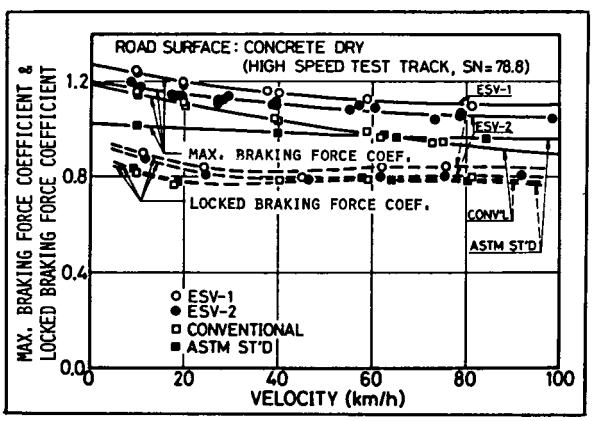


Figure 21. Maximum Braking Force Coefficient and Locked Braking Force Coefficient on Dry Concrete

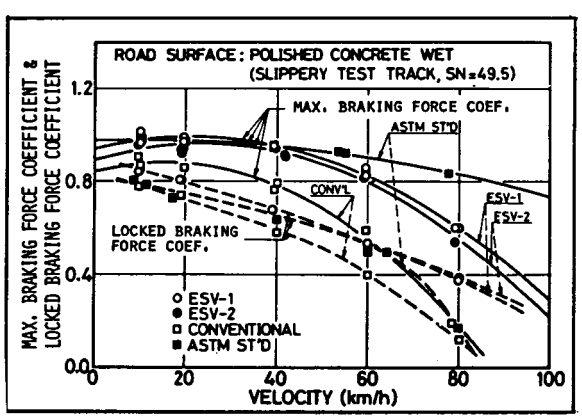


Figure 22. Maximum Braking Force Coefficient and Locked Braking Force Coefficient on Polished Wet Concrete

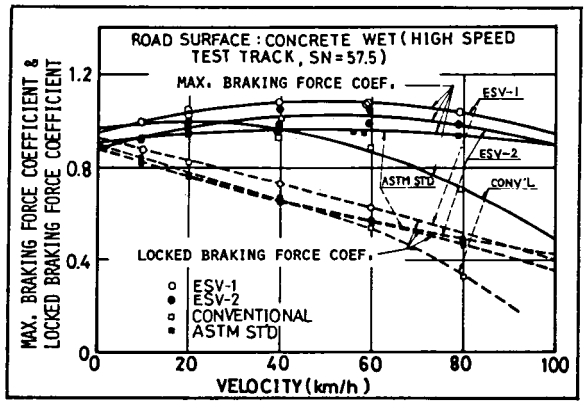


Figure 23. Maximum Braking Force Coefficient and Locked Braking Force Coefficient on Wet Concrete Surface

$$\mu_b(v) = aV^2 + bV + c \qquad (6)$$

where a, b, c, = constant; V = velocity.

The ideal stopping distance is a stopping distance in the case when a vehicle stops maintaining the maximum braking force by controlling the slippage of wheels. With a report of HSRI (Murphy) [3] as reference, we calculated as follows:

TABLE 5
CONSTANTS OF EQUATION (6)

TEST ROAD	CONCRETE DRY			CONCRETE WET			POLISHED CONCRETE WET		
CONSTANTS / TIRE	a ($\times 10^{-5}$)	b ($\times 10^{-3}$)	c ($\times 10^{0}$)	a ($\times 10^{-5}$)	b ($\times 10^{-3}$)	c ($\times 10^{-1}$)	a ($\times 10^{-4}$)	b ($\times 10^{-3}$)	c ($\times 10^{-1}$)
ESV-1	2.17	-3.81	1.27	-5.56	5.51	9.49	-0.899	3.74	9.51
ESV-2	1.35	-2.82	1.20	-5.54	5.69	8.81	-1.29	6.19	8.89
CONV'L. (6.45-14 4PR)	1.35	-4.29	1.19	-7.95	3.14	9.67	-1.68	5.24	8.40
ASTM ST'D (7.50-14 4PR)	0.515	-1.16	1.02	-2.54	2.44	9.07	-0.299	0.586	9.75

As shown in Figure 24, taking man's characteristics into consideration, the time needed from when a driver's foot touches a brake pedal to when maximum braking force is obtained, we adopted that this time is 0.15 sec. from a result of actual braking tests, and braking force during that time increases in proportion to the square of time, from a result of actual braking tests. Afterward, the vehicle stops keeping maximum braking force. Then we shall define the ideal stopping distance as the distance where a vehicle transfers from when the foot touches the brake pedal up to stopping.

Namely, the following equation is represented.

$$\frac{dV}{dt} = \begin{cases} -\mu_b \dfrac{(v') \cdot gt^2}{0.15^2} & (0 \leqslant t \leqslant 0.15) \\ -\mu_b (v) \cdot g & (0.15 \leqslant t \leqslant t_0) \end{cases} \qquad (7)$$

When $\mu_b(v')$ is the maximum braking force Coefficient at time t = 0.15 sec. and t_0 is the time

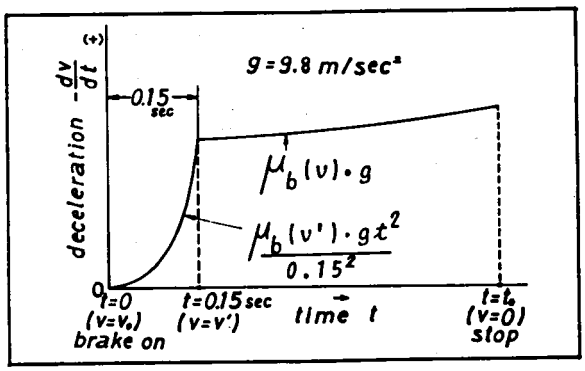


Figure 24. Explanation of Ideal Stopping Distance

when test vehicle stops. Thus, the ideal stopping distance is calculated by the following equation on a supposition that the coefficients are not changed with load.

$$D_i = \int_0^{0.15} \left(V_0 - \frac{\mu_b(v') \cdot gt^3}{3(0.15)^2}\right) dt$$

$$-\frac{1}{g}\int_{V'}^{0} \frac{V}{aV^2 + bV + c}\, dv \qquad (8)$$

where V' = positive root for the equation below
V_0 = initial velocity

$$\mu_b(V') = aV'^2 + bV' + c$$

$$V' = V_0 - 0.05\mu_b(V') \cdot g \qquad (9)$$

The brake system efficiency of ESV-1 (4-wheel antilock) and that of ESV-2 (rear-wheel antilock) is shown in Tables 6 and 7. For a bit of reference, brake efficiency at locked braking has been calculated, and an increase of efficiency when an antilock device is actuated, is also indicated.

Furthermore, the ideal stopping distance $D_{i.std.}$ when standard tires are used for ESVs and $D_{i.con.}$ when conventional tires are used for them, have also been calculated, and ratio of these to ideal stopping distance is expressed as tire factor.

$$\text{Tire Factor to ASTM Standard Tire} = \frac{D_{i.sts.}}{D_i}$$

$$\text{Tire Factor to Conventional Tire} = \frac{D_{i.\,con.}}{D_i}$$

To check the reliability of these tests results, locked wheel braking distances calculated on the basis of the locked braking force coefficient obtained by the tester and actually measured stopping distance were compared, and an error was found to be below one meter only.

RELATIONS BETWEEN SIDEWAY FORCE AND LONGITUDINAL FORCE COEFFICIENT

The maximum braking force and locked braking force regarding longitudinal slip have so far been described. What is most closely related when steering and stability of vehicles is sideway force coefficient. From this viewpoint, the sideway force coefficient was also measured by using a trailer type tire testing machine of JARI.

Trailer Type Tire Testing Machine

This machine is of the 2-wheel trailer type pulled by a large passenger car, and enables us to measure both braking torque and sideway force. An original model is a 2-wheel trailer for pavement tests under ASTM.

Figure 25 shows the tester equipped with a stabilization wheel at its rear end. This wheel is designed to increase stability when a slip angle due to toe-in to the test tires. On-the-pavement tests were

TABLE 6
BRAKE SYSTEM EFFICIENCY OF ESV-1

TEST ROAD SURFACE \ CONDITION & CHARACTERISTICS	CONDITION TEST LOAD (kg) & PERCENT	ACTUAL STOPPING DISTANCE (m)		IDEAL STOPPING DISTANCE (m)	CHARACTERISTICS				
		ANTILOCK ON	ANTILOCK OFF		BRAKE SYSTEM EFF. (%)	LOCKED-WHEEL EFF. (%)	INCREASE OF EFF. (%) (BY USE OF ANTILOCK DEV.)	TIRE FACTOR (TO ASTM STD TIRE)	TIRE FACTOR (TO 6.45-14 TIRE)
CONCRETE (HIGH SPEED TRACK) DRY (SN=78.8)	360/1 tire (100%)	40.09	45.25	34.26	85.46	75.71	9.75	1.14	1.14
CONCRETE (HIGH SPEED TRACK) WET (SN=57.5)	360/1 tire (100%)	43.92	62.14	36.59	83.31	58.88	24.43	1.10	1.30
POLISHED CONCRETE (SLIPPERY TEST TRACK) WET (SN=49.5)	360/1 tire (100%)	58.40	84.45	50.02	85.65	59.23	16.42	0.866	–

TABLE 7
BRAKE SYSTEM EFFICIENCY OF ESV-1

TEST ROAD SURFACE \ CONDITION & CHARACTERISTICS	CONDITION TEST LOAD (kg) & PERCENT	ACTUAL STOPPING DISTANCE (m)		IDEAL STOPPING DISTANCE (m)	CHARACTERISTICS				
		ANTILOCK ON	ANTILOCK OFF		BRAKE SYSTEM EFF. (%)	LOCKED-WHEEL EFF. (%)	INCREASE OF EFF. (%) (BY USE OF ANTILOCK DEV.)	TIRE FACTOR (TO ASTM STD TIRE)	TIRE FACTOR (TO 6.45-14 TIRE)
CONCRETE (HIGH SPEED TRACK) DRY (SN=78.8)	350/1 tire (50%)	44.04	45.48	35.62	80.88	78.32	2.56	1.10	1.10
CONCRETE (HIGH SPEED TRACK) WET (SN=57.5)	350/1 tire (100%)	50.79	65.65	38.51	75.82	58.66	17.16	1.04	1.23
POLISHED CONCRETE (SLIPPERY TEST TRACK) WET (SN=49.5)	350/1 tire (100%)	72.30	79.10	59.31	82.03	74.98	7.05	0.731	–

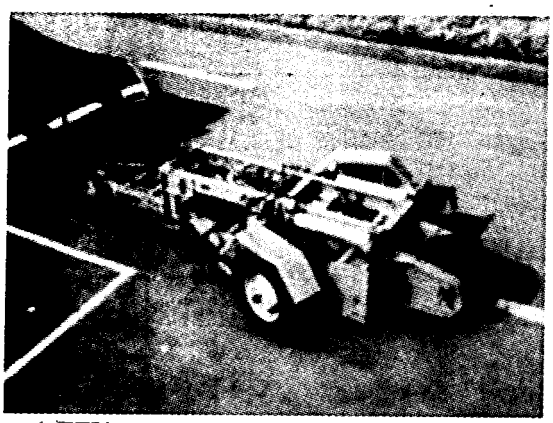


Figure 25. JARI Trailer Type Tire Testing Machine

conducted by removing the wheel. Summarized specifications on this tester are given in Table 8.

As stated already, it seems that running a tester on the actual pavement is the most effective method to measure and elucidate friction property between a tire and the road surface. Research by such methods will give us basic data for selection of the substitute road surface for an indoor tester.

TABLE 8
SPECIFICATIONS OF TRAILER TYPE TIRE TESTING MACHINE

TEST TIRE	MOTORCYCLE TIRE AND PASSENGER CAR TIRE
LOAD	250KG – 500 kg
APPLIED LOAD SYSTEM	AIR SUSPENSION
VELOCITY	2 km/h – 90 km/h
SLIP ANGLE	0° – 20°
MEASUREMENT ITEMS	BRAKING TORQUE SIDEWAY FORCE SELF ALIGNING TORQUE

Measurement Results of Sideway Force Coefficient

A change of sideway force on the asphalt pavement with a slip angle is shown in Figure 26. This shows that side force has the peak value at a certain slip angle. Maximum sideway force coefficient was dividing the peak value by load; a change in this coefficient with velocity on representative pavements is shown in Figures 27-28.

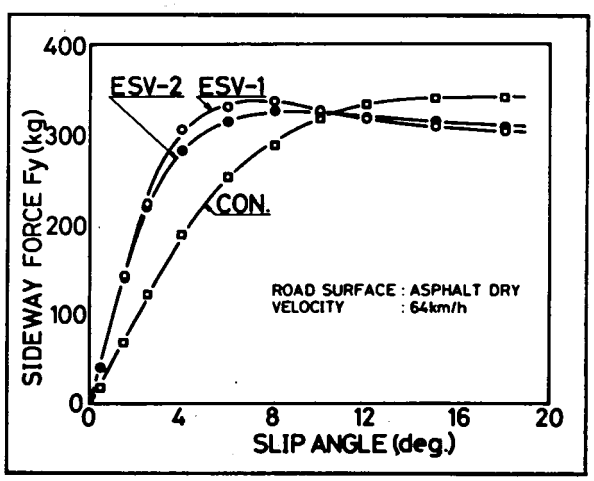


Figure 26. Sideway Force on a Real Road Surface

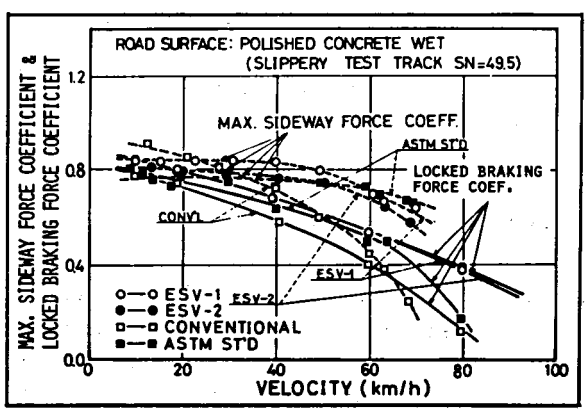


Figure 27. Maximum Sideway Force Coefficient and Locked Braking Force Coefficient

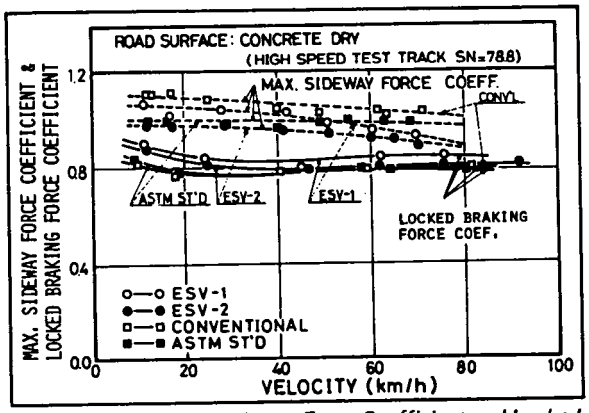


Figure 28. Maximum Sideway Force Coefficient and Locked Braking Force Coefficient

Relations Among Coefficients of Various Frictional Forces

It is clear from Figures 21-23 that linear relationships cannot be established between the locked braking force coefficient and the maximum braking force coefficient. This is also the same in the relationship between the locked braking force coefficient and the sideway force coefficient as shown in Figures 27-38. Comparatively similar tendency, however, can be seen between the maximum braking force and the maximum sideway force coefficient. The reason is as follows:

At the time of locked braking, the entire contact area of tread is slipping, and this slide velocity equals tire running velocity. However, same slip ratio where maximum braking force coefficient is obtainable, and at small slip angles (6-18 degrees) the maximum sideway force coefficient can be obtained, the adhesive region exists in contact area of tread and road surface, and moreover, tread slide velocity against the road surface is much less than tire velocity. Thus, from a reason, a similar tendency is observed in a change in the maximum braking force coefficient and the maximum sideway force coefficient with velocity, and both coefficients in the case of high velocity become larger than the locked braking force coefficient. Strictly considering, the above two maximum coefficients do not equal completely.

On the Change of Coefficient of Friction Due to Temperature

A change of skid resistance number and the sideway force coefficient with road temperature, which was measured on representative pavements, is shown in Figures 29 and 30 [4]. As a difference in road temperature between winter and summer exceeds 60° in Japan, correction of skid resistance number by temperature should be considered necessary.

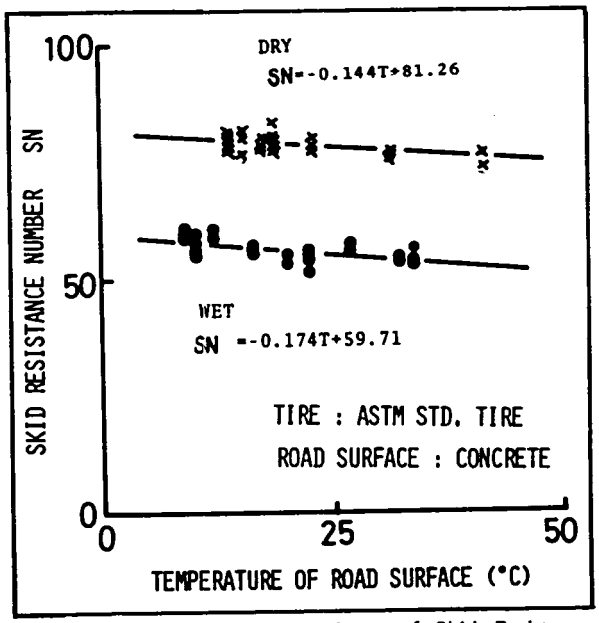


Figure 29. Temperature Dependence of Skid Resistance Number

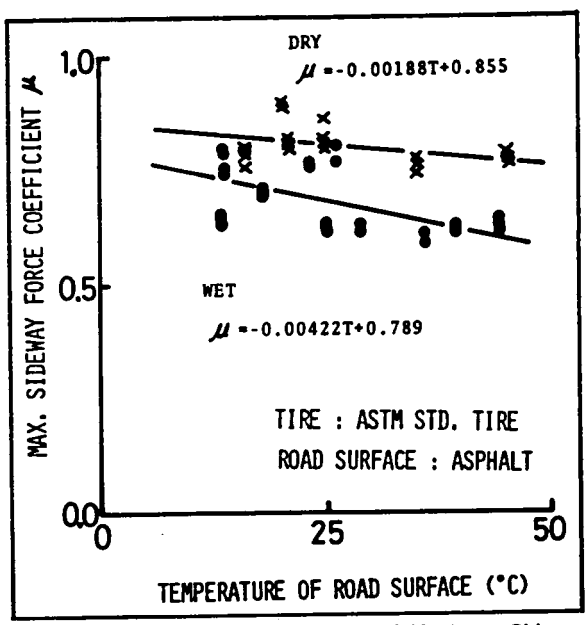


Figure 30. Temperature Dependence of Maximum Sideway Force Coefficient

CONCLUSION

I have described so far test results of braking and cornering performance of ESV tires. I am convinced that the outline of the characteristics of ESV tires is appreciated. In order to evaluate the whole performance of tires, other factors such as vibration, noise, driving comfortability, resistance to abrasion

and durability, endurance, safety when punctured, and cost, needless to say, must be considered.

Lastly, I wish to indicate the necessity of a standard tire to be used for measurement of the sideway force coefficient of pavement and for test of cornering characteristics. Such a tire must, first of all, be difficult to be abraded due to hard cornering. For this purpose, the value of friction is not important; tire tread rubber compound, which is not abraded in the case of hard cornering, must be selected; radial tire seems preferable as tire structure. As for the tread pattern, that, with square blocks arranged in a series, is considered.

Otherwise, such a type of tire would be used for comparison of test tires and for comparison of cornering test machines. Of course the tire will also be employed to conduct a braking characteristics test.

ACKNOWLEDGEMENT

The study reported in this paper was completed through the research on ESV tires and its related studies of the original researches on tires which have been conducted in JARI. This work was performed by the author, Mr. T. Furuichi and other members of 3rd Research Department of JARI. We had some comparable test data presented by both the Toyota and the Nissan Motor Companies. We should like to thank these members heartily.

REFERENCES

[1] H. Sakai, K. Araki, S. Ogasawara; Cornering Characteristics of the Tire at Large Camber Angles and Large Slip Angles, Preprint of JSAE Lecture Meeting No. 731 May 1973

[2] H. Sakai, O. Kanaya, T. Okayama; Influence of Hydroplaning on Cornering and Braking Performance of the Tire, Preprint of JSAE Lecture Meeting No. 731 May 1973

[3] R.W. Murphy; A Procedure for Evaluating Vehicle Braking Performance, Second International Technical Conference on ESV Oct. 1971

[4] H. Sakai, H. Iijima; Trailer Type Test Vehicle for Measuring Tire Characteristics and Some Results measured by It about the Change of Skid Resistance Number of JARI Test Course Surface through the Year, JARI Technical Report No. 7 Jan. 1973

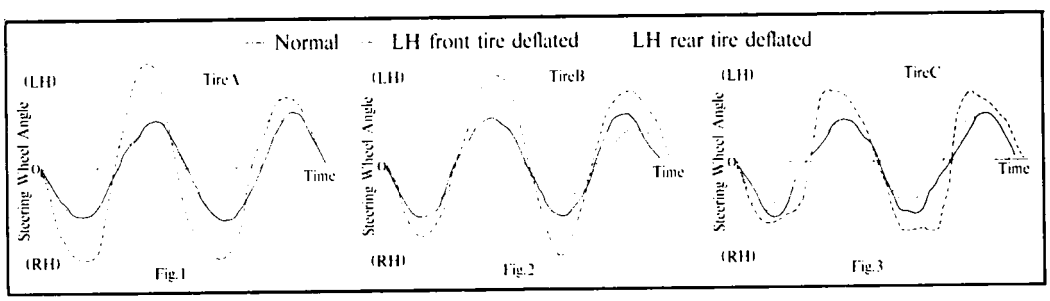
STEERING AND HANDLING CHARACTERISTICS OF A VEHICLE WHEN FAIL-SAFE TIRE IS DEFLATED

SHOICHI SANO
Honda Motor Co., Ltd.

1. INTRODUCTION

The development of fail-safe tires made it possible for a vehicle to travel even when the cornering capability of its front and rear wheels are greatly unbalanced, a state that had hitherto prevented it from running. In order to confirm safety of a vehicle in such a state, we first conducted slalom tests with actual vehicles. The results of these tests are shown in Figs. 1 through 3.

Steering Wheel Angles during Slalom



Tires used are three types of fail-safe tires that are entirely different in construction from one another, with two of them being 165/70 SR13 in size and the third one 185/60 R13; vehicles used consist of front drive test cars having the same specifications as Honda Experimental Safety Vehicle's. Tests were conducted under the severest conditions imposed on tires, by removing, for instance, the air valve from those of the kind which are intended to recover internal pressure created by vapour from the lubricant.

Figs. 1 through 3 show chronologically variation in the steering wheel angle of a test vehicle operated by a skilled driver when the vehicle slalomed at a velocity of 60 km/h between six pylons placed at intervals of 30 m, with irregular curves corresponding to the entry and exit omitted. From these figures you can see a substantial difference exists between tires of different types by noting that one indicates a large variation in the steering wheel angle during deflation of a tire as compared with that when the tires are normal while another shows a relatively small variation, that one indicates a large difference in the steering wheel angle between left and right turns while another shows a small difference, and that there are curves that already reveal saturation of the cornering force of the deflated tire.

Actual slaloming in a test course made us find that a large difference in the response and in the steering effort between left and right turns when a front tire is deflated constitutes a problem as well as a low lateral acceleration in the case of a rear tire deflation when spin-out occurs in a turn with the deflated tire as an outerside tire. However, these are problems that become evident only when lateral acceleration is relatively high; in running tests conducted later which are a simulation of public road driving, we acquired an anticipation that avoidance of drastic steering and high speed driving in recognition of a tire deflation would eliminate problems in actual driving.

In this case, however, it is naturally desired that the characteristics of a vehicle during a tire deflation should remain to be as close to those under normal tire conditions as

possible. With consideration given to the actual state of vehicle operations and items to be described below determined as evaluation items, we conducted the tests.

(1) Relation between steering effort and yaw velocity at high speed

This item has been selected as a result of consideration paid to maneuvers in the lane change and in the exit from an expressway when a tire is deflated. Since steering operations during high speed running are presumed to rely mainly on the feedback of steering efforts, it calls your attention to relation between the yaw velocity and steering effort within the range of small steering wheel angles. High speed particularly referred to here are those speeds which are in the neighborhood of the highest when a vehicle is running with its tire deflated.

(2) Relation between steering wheel angle and yaw velocity when steering wheel angles are large at low speed

This item is in contrast to item (1) and has been selected as a result of consideration paid to maneuvers at low speeds during turning a sharp corner or passing through a crank or S–shaped course. In this case, steering operations are presumed to rely mainly on the feedback of steering wheel angles.

(3) Maximum lateral acceleration

This means the cornering performance by fixed control and by manual control. A special emphasis is placed on the former which is not affected by the driver's skill.

(4) Stability at high speed

This is an item by which stability of a vehicle against disturbance, such as road surface irregularity and wind, while running at high speeds is evaluated; evaluation is made through acquirement of stability factor and considering critical speeds.

Evaluation criteria in these items are in any event "the smaller, the better" basis with respect to difference in the characteristics as compared with those under normal conditions. From a safety point of view, a special emphasis is placed on a characteristic difference between turns in the opposite directions.

In addition to these items, that for evaluation of the directional stability during braking, an important accident avoidance ability, is included. As for this item, only the test results are given at the end of this paper, without devoting discussion.

2. TEST RESULT

On the subject of evaluation items described in Section 1, the test results are given below:

2.1 Relation between steering wheel angle and yaw velocity at high speed

Shown in Figs. 4 through 6 are the measured results of relation between the tangential force on the circumference of a steering wheel and corresponding yaw velocity when the steering wheel is turned at a steering rate of 10°/sec, without changing the vehicle speed, from a straightforwarding at 60 km/h.

Each upper figure shows comparison between normal performance and performance with a front or rear tire deflated with respect to turns with the deflated tire as an outerside tire. Each middle figure shows comparison between normal performance and performance with a front tire deflated with respect to turns with the deflated tire as an innerside or outerside tire. Each lower figure shows the same comparison as each middle figure in the case of a rear tire deflation.

The presence of steering effort at position of zero yaw velocity indicates that steering correction is required while the vehicle is running straightforward; the effort is generally larger with a front tire deflation and smaller with a rear tire deflation. The quantity of steering correction differs greatly from one another among different types of tires. That which requires a larger steering correction has an effect of making the driver recognize a tire puncture through the steering wheel, but it is not desirable after all since spin-back

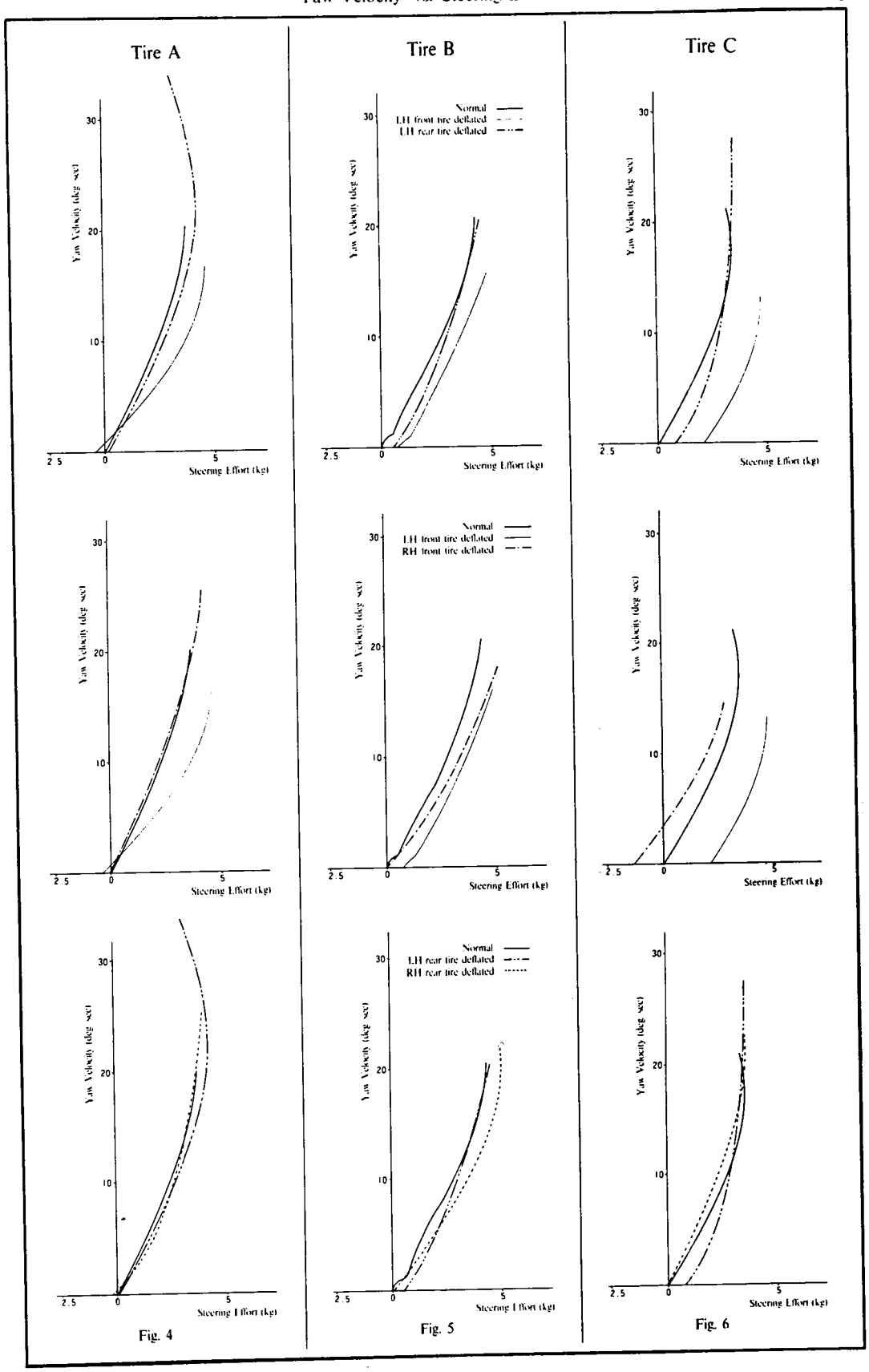


Fig. 4

Fig. 5

Fig. 6

of the steering wheel becomes abrupt when changing from a turn to a straightforwarding and tendency to overshoot in the opposite direction grows larger. The grade of a curve means an incremental yaw velocity per unit steering effort and this also differs greatly from one another among different types of tires as in the case of steering correction mentioned above.

2.2 Relation between steering wheel angle and yaw velocity when steering wheel angles are large at low speed

Shown in Figs. 7 through 9 are centripetal accelerations and corresponding steering wheel angles expressed in ratio to those at an extra-low velocity when a vehicle is subjected to cornering tests conducted at various speed along a circle of 15 m radius. The upper, middle and lower figures show the results obtained under the same conditions as given in paragraph 2.1 above respectively.

It can be seen that variation in the characteristics as a result of tire deflation is remarkable for one time and it is relatively small for another; however, those which differ greatly in the amount of variation in the characteristics between a front and a rear tire deflation do not necessarily differ remarkably in the same between turns in the opposite directions.

2.3 Maximum lateral acceleration

Shown in Figs. 10 through 15 are the results of the cornering tests conducted by means of fixed control and manual control along a circle of 20 m radius.

With fixed control, the limit is the lowest when a vehicle turns with a deflated rear tire as an outerside tire; and from a safety point of view, it is worthy of note that approximately 0.4 g is indicated in such a case with any of the types of test tires. Yet, with manual control, this situation may be improved substantially and the limit becomes the lowest when a vehicle turns with a deflated front tire as an outerside tire. Except for turnings with a deflated rear tire as an outerside tire, difference in the performance is small between fixed control and manual control. However, tires differ considerably from one another among their types in the performance of a vehicle with a tire deflation.

2.4 Stability at high speed

This item was intended to study relation between the centripetal acceleration and corresponding stability factor which was obtained from a yaw rate gain during each cornering when a vehicle was subjected to cornering tests conducted at a speed of 60 km/h with various turning radii. The stability factor K is a parameter that is defined by the following equation:

$$\left(\frac{r}{\theta}\right) = \frac{1}{1+KV^2} \cdot \frac{V}{l}$$

where $\left(\frac{r}{\theta}\right)$: yaw rate gain

V : vehicle speed

l : wheelbase

The results of the tests are shown in Figs. 16 through 18.

Cornering characteristics of each type of tires with respect to turnings performed under normal conditions and those with a deflated front or rear tire as an innerside or outerside tire are shown in the same figure.

These results also reveal that similarly to other items described before now, variation in the characteristics of tires due to deflation may be large for one and relatively small for another.

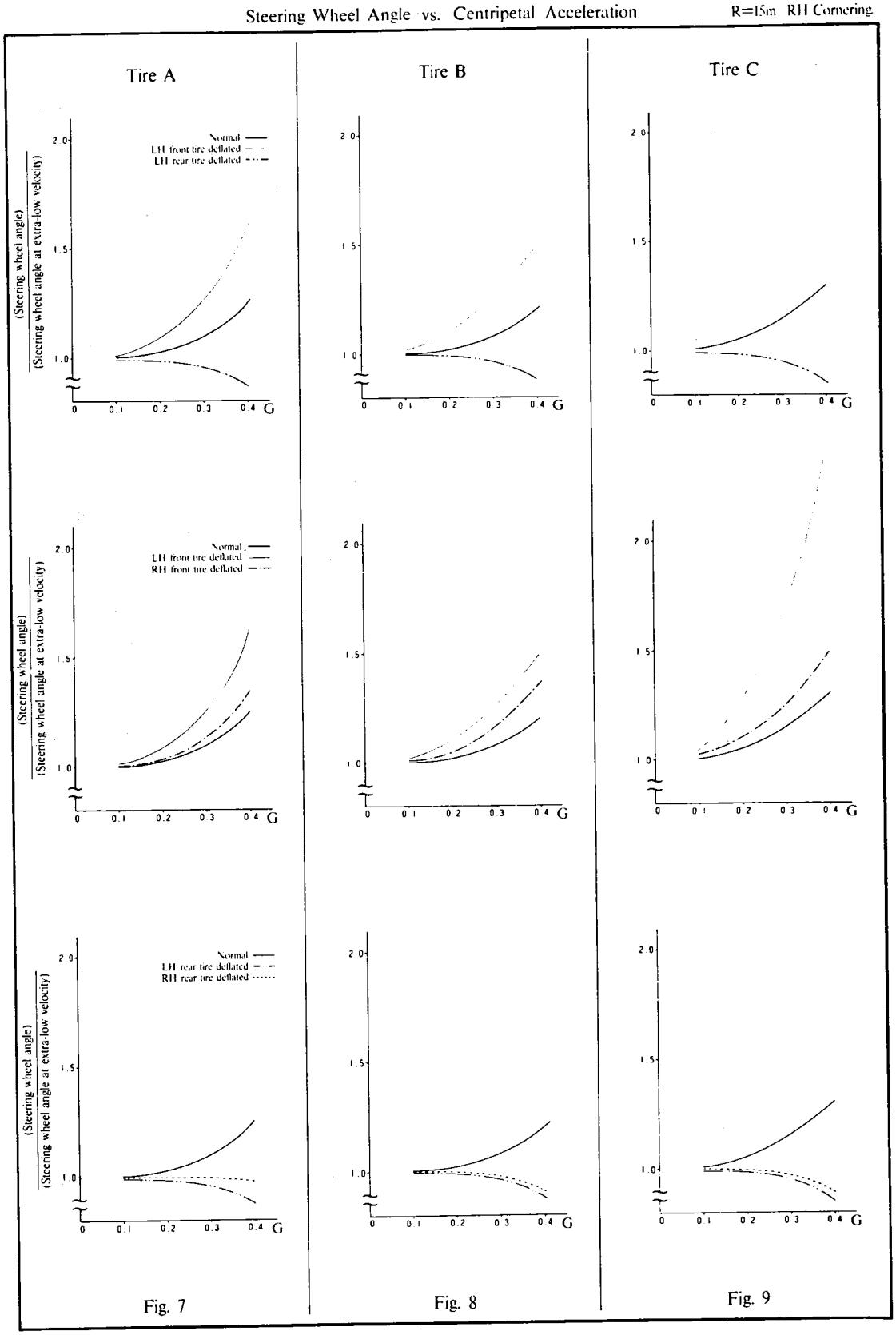


Fig. 7 Fig. 8 Fig. 9

Max. Centripetal Acceleration R=20m RH Cornering

Fixed control

Tire A

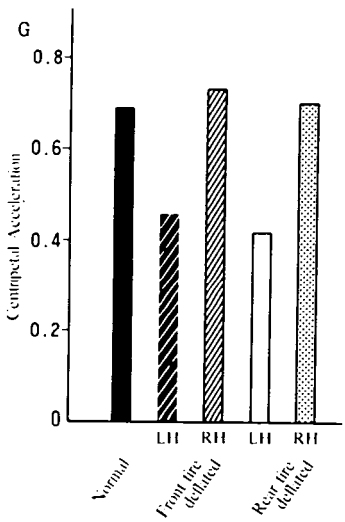


Fig. 10

Tire B

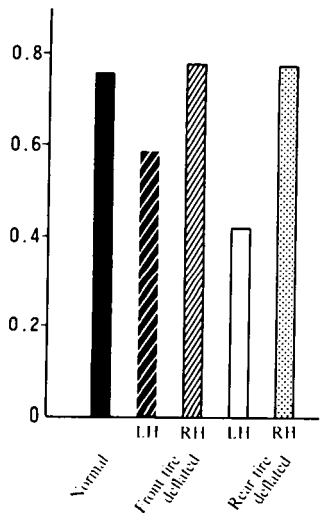


Fig. 11

Tire C

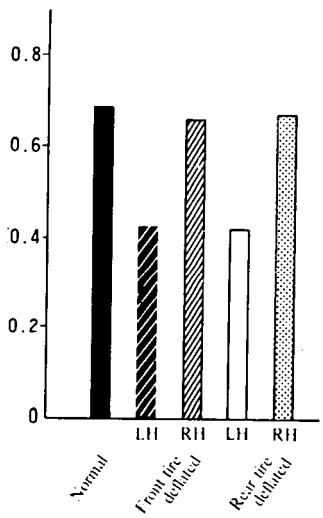


Fig. 12

Manual control

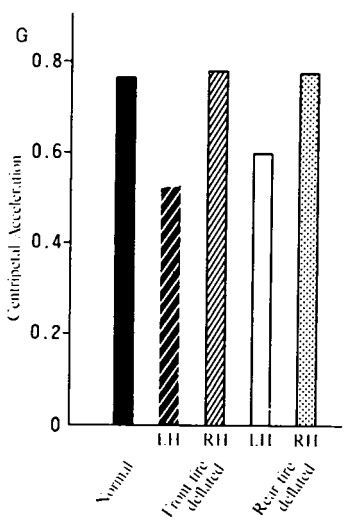


Fig. 13

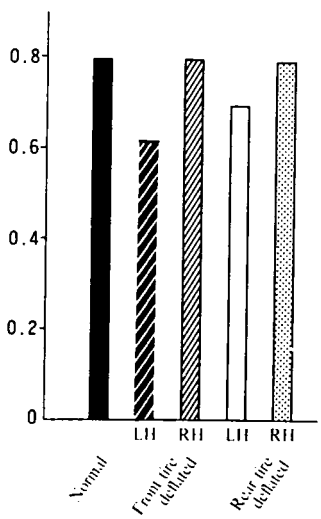


Fig. 14

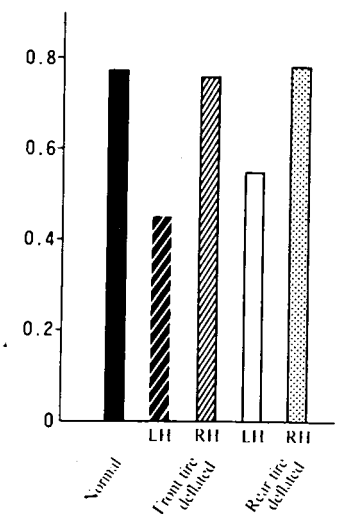


Fig. 15

Stability Factor vs. Centripetal Acceleration

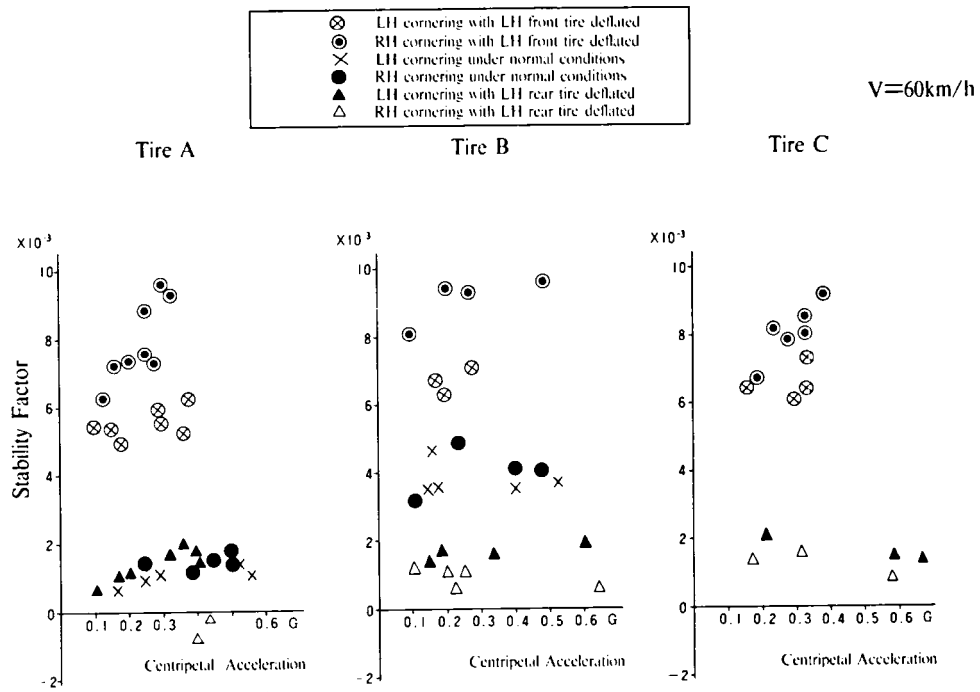


Fig. 16 Fig. 17 Fig. 18

3. ANALYSIS OF STEERING CHARACTERISTICS OF A VEHICLE WHEN FAIL-SAFE TIRE IS DEFLATED

According to the test results described in Section 2, these three types of tires differ from one another in their effect on vehicles when a tire deflation occurs. As for tire assemblies as a unit, they not only differ entirely from one another in construction, but also differ substantially in other characteristics. For example, with respect to deflection of a tire (change of static radius) when it is deflated, one deflates unnoticeably, another deflates to the point at which tire and rim come in close contact with each other, and the other lies at about the middle. Such other characteristics as cornering force, rolling resistance, self-aligning torque, camber force, etc. also differ from one another. Now, let us analyze by what mechanism these tire characteristics produce effect, in connection with the design parameters of a vehicle, on the vehicle's characteristics.

3.1 Symbols and axis system

Vehicle (constant)

a : distance from the center of gravity to the front axis (m)
b : distance from the center of gravity to the rear axis (m)
d : front and rear tread (m)
Ks : suspension rate of spring (kg m^{-1})
Ka : suspension rate of anti-roll spring (kg m^{-1})
l : wheelbase (m)
m : mass of the whole vehicle (kg m^{-1} sec^2)
k : $k = ks + ka$
suffix 1 : front wheel position at one side
2 : rear wheel position at one side

Vehicle (variable)

r : yaw relocity (sec^{-1})
R : turning radius (m)
V : velocity in the direction of X-axis at the center of gravity (m sec^{-1})
Z : vertical load on tire (kg)
β : side slip angle
δ : vertical displacement of wheel (m)
θ : front steer angle
ϕ : roll angle

Tire

AT: self-aligning torque (m kg)
C : cornering power (kg)
X : rolling resistance (kg)
Y : camber force (kg)
α : side slip angle of tire
γ : camber angle of tire
δ_0 : tire deflection when deflated (m)

suffix

None : sum of four wheels
1 : sum of front wheels
2 : sum of rear wheels
L : LH wheel
R : RH wheel

Axis system

The axis system employed is a right-hand system, with the center of gravity as the origin of coordinates, X-axis frontward, Y-axis leftward, and Z-axis upward the vehicle; the angle and moment are positive when it is in the direction of RH threads in reference to each axis.

Unless otherwise specified, a tire is to deflate that is in a positive direction of roll angle ϕ and the vehicle is to turn in a positive direction of yaw velocity r.

3.2 Change of attitude and vertical load on wheels of stationary vehicle when one of its tires is deflated

Disregarding the amount of displacement of the center of gravity due to change of the vehicle's attitude as a result of a tire deflation and considering deflection of the tire δ_0 due to deflation to be sufficiently small as compared with the wheelbase l and tread d, from the equilibrium of force in the direction of Z-axis, equilibrium of moment about X-axis and Y-axis, and from the requirement that the tire contact points should stay on the same plane, vertical displacement of each wheel δ and change of vertical load on each wheel ΔZ are obtained as follows:

For a RH front tire deflation:

$$\delta_{1R}=\frac{(2k_1+k_2+k_{a2})}{2(k_1+k_{a1}+k_2+k_{a2})}\delta_0 \qquad (1)$$

$$\delta_{1L}=\frac{(2k_{a1}+k_2+k_{a2})}{2(k_1+k_{a1}+k_2+k_{a2})}\delta_0 \qquad (2)$$

$$\delta_{2R}=\frac{k_{s1}}{2(k_1+k_{a1}+k_2+k_{a2})}\delta_0 \qquad (3)$$

$$\delta_{2L}=-\delta_{2R} \qquad (4)$$

$$\Delta Z_{1R}=-\frac{(k_2+k_{a2})k_{s1}}{2(k_1+k_{a1}+k_2+k_{a2})}\delta_0 \qquad (5)$$

$$\Delta Z_{1L}=-\Delta Z_{1R} \qquad (6)$$

$$\Delta Z_{2R}=-\Delta Z_{1R} \qquad (7)$$

$$\Delta Z_{2L}=\Delta Z_{1R} \qquad (8)$$

For a RH rear tire deflation:

$$\delta_{1R}=\frac{k_{s2}}{2(k_1+k_{a1}+k_2+k_{a2})}\delta_0 \quad (9)$$

$$\delta_{1L}=-\delta_{1R} \quad (10)$$

$$\delta_{2R}=\frac{(k_1+k_{a1}+2k_2)}{2(k_1+k_{a1}+k_2+k_{a2})}\delta_0 \quad (11)$$

$$\delta_{2L}=\frac{(k_1+k_{a1}+2k_{a2})}{2(k_1+k_{a1}+k_2+k_{a2})}\delta_0 \quad (12)$$

$$\Delta Z_{1R}=\frac{(k_1+k_{a1})k_{s2}}{2(k_1+k_{a1}+k_2+k_{a2})}\delta_0 \quad (13)$$

$$\Delta Z_{1L}=-\Delta Z_{1R} \quad (14)$$

$$\Delta Z_{2R}=-\Delta Z_{1R} \quad (15)$$

$$\Delta Z_{2L}=\Delta Z_{1R} \quad (16)$$

In the case of a LH tire deflation, it suffices to replace a suffix of R wich that of L. In any event, the vertical load on a tire located at a diagonal position from the deflated tire decreases and that on the other tires increases, with the amount of decrease proving to be equal to that of increase.

Roll angle ϕ_0 produced as a result of a tire deflation may be expressed as follows:

$$\phi_0=\frac{\delta_{1R}-\delta_{1L}}{d}=\frac{\delta_{2R}-\delta_{2L}}{d} \quad (17)$$

3.3 Equations of motion

Considering force Y in the direction of Y-axis that originates in the camber force due to the roll angle ϕ_0 produced as a result of a tire deflation and moment M about Z-axis that originates in the increase of the rolling resistance due to a tire deflation and camber force due to roll, from the equilibrium of force in the direction of Y-axis and equilibrium of moment about Z-axis during cornering, the following equations of motion are obtained:

$$C_1\left(\beta-\frac{ar}{V}+\theta\right)+C_2\left(\beta+\frac{br}{V}\right)-mVr+Y=0 \quad (18)$$

$$aC_1\left(\beta-\frac{ar}{V}+\theta\right)-bC_2\left(\beta+\frac{br}{V}\right)-\frac{\partial\Delta T_1}{\partial\alpha}\left(\beta-\frac{ar}{V}+\theta\right)-\frac{\partial\Delta T_2}{\partial\alpha}\left(\beta+\frac{br}{V}\right)+M=0 \quad (19)$$

$$Y=-\left(\frac{\partial Y_2}{\partial\gamma}\cdot\frac{\partial\gamma_2}{\partial\phi}+\frac{\partial Y_1}{\partial\gamma}\frac{\partial\gamma_1}{\partial\phi}\right)\phi \quad (20)$$

$$M=\left(b\frac{\partial Y_2}{\partial\gamma}\frac{\partial\gamma_2}{\partial\phi}-a\frac{\partial Y_1}{\partial\gamma}\frac{\partial\gamma_1}{\partial\phi}\right)\phi+\frac{d}{2}(X_{1L}-X_{1R}+X_{2L}-X_{2R}) \quad (21)$$

From these equations yaw velocity r is obtained as follows:

$$r=\frac{1}{1+\frac{m}{l}\cdot\frac{B-A}{E}V^2}\cdot\frac{V}{l}\left(\theta+\frac{C}{E}M+\frac{B-A}{E}Y\right) \quad (22)$$

where

$$A=aC_1-\frac{\partial\Delta T_1}{\partial\alpha} \quad (23) \qquad B=bC_2+\frac{\partial\Delta T_2}{\partial\alpha} \quad (24)$$

$$E=BC_1+AC_2 \quad (25)$$

Also, from $r=\frac{V}{R}$ (26) the steer angle θ is obtained as:

$$\theta=\frac{l}{R}\left(1+\frac{m}{l}\cdot\frac{B-A}{E}V^2\right)-\frac{C}{E}M-\frac{B-A}{E}Y \quad (27)$$

3.4 Steer angle and side slip angle during straightforwarding

Placing $r = o$ in the equations (18) and (19), the steer angle θ_0 and side slip angle β_0 during straightforwarding are obtained as follows:

$$\theta_0=-\frac{C}{E}M_0-\frac{B-A}{E}Y_0 \quad (28) \qquad \beta_0=\frac{C_1}{E}M_0-\frac{A}{E}Y_0 \quad (29)$$

M_0 and Y_0 are each the value when a tire is deflated and are obtained from the equations (17), (20) and (21).

3.5 Steer angle during cornering

Considering the state of cornering with a tire deflated to be the already acquired steer angle θ_0, side slip angle β_0, and roll angle ϕ_0 plus θ, β, and ϕ respectively and assuming that the amount of variation in the tire characteristics is so small when compared with that during straightforwarding with a tire deflated as to enable the second power thereof to be neglected, and that the variation is linear, the equations (27), (21) and (20) may be rewritten as follows:

$$\theta=\frac{l}{R}\left\{1+\frac{m}{l}\left[\frac{B-A}{E}+\Delta\left(\frac{B-A}{E}\right)\right]V^2\right\}-\frac{C}{E}\cdot\Delta M-\frac{B-A}{E}\cdot\Delta Y \qquad (30)$$

where

$$\Delta\left(\frac{B-A}{E}\right)=-\frac{\left(lC_2+\frac{\partial AT}{\partial\alpha}\right)B}{E}\Delta C_1+\frac{\left(lC_1-\frac{\partial AT}{\partial\alpha}\right)A}{E}\Delta C_2+\frac{BC}{E^2}\Delta\left(\frac{\partial AT_1}{\partial\alpha}\right)+\frac{AC}{E^2}\Delta\left(\frac{\partial AT_2}{\partial\alpha}\right) \qquad (31)$$

$$\Delta C_{\frac{1}{2}}=\left(\frac{\partial C_{\frac{1}{2}R}}{\partial Z}-\frac{\partial C_{\frac{1}{2}L}}{\partial Z}\right)\Delta Z_{\frac{1}{2}} \qquad (32)$$

$$\Delta\left(\frac{\partial AT_{\frac{1}{2}}}{\partial\alpha}\right)=\left(\frac{\partial^2 AT_{\frac{1}{2}R}}{\partial Z\partial\alpha}-\frac{\partial^2 AT_{\frac{1}{2}L}}{\partial Z\partial\alpha}\right)\Delta Z_{\frac{1}{2}} \qquad (33)$$

$$\Delta M=\left\{b\left[\frac{\partial Y_2}{\partial\gamma}+\Delta\left(\frac{\partial Y_2}{\partial\gamma}\right)\right]\frac{\partial\gamma_2}{\partial\phi}-a\left[\frac{\partial Y_1}{\partial\gamma}+\Delta\left(\frac{\partial Y_1}{\partial\gamma}\right)\right]\frac{\partial\gamma_1}{\partial\phi}\right\}\Delta\phi-\frac{d}{2}\Delta X \qquad (34)$$

where

$$\Delta\left(\frac{\partial Y_{\frac{1}{2}}}{\partial\gamma}\right)=\left(\frac{\partial Y_{\frac{1}{2}R}}{\partial\gamma}-\frac{\partial Y_{\frac{1}{2}L}}{\partial\gamma}\right)\Delta Z_{\frac{1}{2}} \qquad (35)$$

$$\Delta X=\left(\frac{\partial X_{1R}}{\partial Z}-\frac{\partial X_{1L}}{\partial Z}\right)\Delta Z_1+\left(\frac{\partial X_{2R}}{\partial Z}-\frac{\partial X_{2L}}{\partial Z}\right)\Delta Z_2 \qquad (36)$$

$$\Delta Y=-\left\{\left[\frac{\partial Y_2}{\partial\gamma}+\Delta\left(\frac{\partial Y_2}{\partial\gamma}\right)\right]\frac{\partial\gamma_2}{\partial\phi}+\left[\frac{\partial Y_1}{\partial\gamma}+\Delta\left(\frac{\partial Y_1}{\partial\gamma}\right)\right]\frac{\partial\gamma_1}{\partial\phi}\right\}\Delta\phi \qquad (37)$$

Now, defining the roll characteristics to be determined by the vehicle and transfer of vertical load on tire due to centripetal acceleration as:

$$\Delta\phi=K_\phi\frac{V^2}{R} \qquad (38) \qquad\qquad \Delta Z_{\frac{1}{2}}=K_{\frac{1}{2}}\frac{V^2}{R} \qquad (39)$$

respectively, and arranging the equations (31) through (39) and substituting them in the equation (30), and using $\Delta C_2=0$, $\Delta\left(\frac{\partial AT_2}{\partial Z}\right)=0$, $\Delta\left(\frac{\partial Y_2}{\partial\gamma}\right)=0$, and $\left(\frac{\partial X_{2R}}{\partial Z}-\frac{\partial X_{2L}}{\partial Z}\right)=0$ for a front tire deflation, the steer angle is obtained as follows:

$$\begin{aligned}\theta=\frac{l}{R}&+\left\{\frac{m(B-A)}{E}+\frac{K_\phi}{E}\left[\left(lC_2+\frac{\partial AT}{\partial\alpha}\right)\frac{\partial Y_1}{\partial\gamma}\frac{\partial\gamma_1}{\partial\phi}-\left(lC_1-\frac{\partial AT}{\partial\alpha}\right)\frac{\partial Y_2}{\partial\gamma}\frac{\partial\gamma_2}{\partial\phi}\right]\right.\\&\left.+\frac{CdK_1}{2E}\left(\frac{\partial X_{1R}}{\partial Z}-\frac{\partial X_{1L}}{\partial Z}\right)\right\}\frac{V^2}{R}\\&+\left\{\frac{mBCK_1}{E^2}\left(\frac{\partial^2 AT_{1R}}{\partial Z\partial\alpha}-\frac{\partial^2 AT_{1L}}{\partial Z\partial\alpha}\right)-\frac{mBK_1}{E^2}\left(lC_2+\frac{\partial AT}{\partial\alpha}\right)\left(\frac{\partial C_{1R}}{\partial Z}-\frac{\partial C_{1L}}{\partial Z}\right)\right.\\&\left.+\frac{K_1K_\phi}{E}\left(lC_2+\frac{\partial AT}{\partial\alpha}\right)\left(\frac{\partial^2 Y_{1R}}{\partial Z\partial\gamma}-\frac{\partial^2 Y_{1L}}{\partial Z\partial\gamma}\right)\frac{\partial\gamma_1}{\partial\phi}\right\}\left(\frac{V^2}{R}\right)^2\end{aligned} \qquad (40)$$

Similarly, for a rear tire deflation the following equation is obtained:

$$\theta=\frac{l}{R}+\left\{\frac{m(B-A)}{E}+\frac{K_\phi}{E}\left[\left(lC_2+\frac{\partial AT}{\partial\alpha}\right)\frac{\partial Y_1}{\partial\gamma}\frac{\partial\gamma_1}{\partial\phi}-\left(lC_1-\frac{\partial AT}{\partial\alpha}\right)\frac{\partial Y_2}{\partial\gamma}\frac{\partial\gamma_2}{\partial\phi}\right]\right.$$
$$\left.+\frac{CdK_2}{2E}\left(\frac{\partial X_{2R}}{\partial Z}-\frac{\partial X_{2L}}{\partial Z}\right)\right\}\frac{V^2}{R}$$
$$+\left\{\frac{mACK_2}{E^2}\left(\frac{\partial^2 AT_{2R}}{\partial Z\partial\alpha}-\frac{\partial^2 AT_{2L}}{\partial Z\partial\alpha}\right)+\frac{mAK_2}{E^2}\left(lC_1-\frac{\partial AT}{\partial\alpha}\right)\left(\frac{\partial C_{2R}}{\partial Z}-\frac{\partial C_{2L}}{\partial Z}\right)\right.$$
$$\left.-\frac{K_2K_\phi}{E}\left(lC_1-\frac{\partial AT}{\partial\alpha}\right)\left(\frac{\partial^2 Y_{2R}}{\partial Z\partial\gamma}-\frac{\partial^2 Y_{2L}}{\partial Z\partial\gamma}\right)\frac{\partial\gamma_2}{\partial\phi}\right\}\left(\frac{V^2}{R}\right)^2 \qquad (41)$$

4. STUDY

Indicating computation results on the basis of the results of these analyses, study effect of the tire characteristics and design parameters of the vehicle on the steering characteristics thereof when a tire is deflated.

Values for the test vehicle used for computation are as follows:

$a=0.989$	$k_{s1}=2740$
$b=1.211$	$k_{a1}=140$
$d=1.3$	$k_{s2}=1370$
$l=2.2$	$k_{a2}=0$
$m=111.2$	$K_1=18.58$
$\frac{\partial\gamma_1}{\partial\phi}=0.824$	$K_2=15.43$
$\frac{\partial\gamma_2}{\partial\phi}=0.805$	$K_\phi=0.00975$

4.1 Change of attitude and vertical load on tire of stationary vehicle when one of its tires is deflated

First, study change of the vertical load on tire which is considered to have effect on the tire characteristics. Compare the equation (5) with the equation (13), and you will find that if suspension rate of the anti-roll spring Ka is sufficiently smaller than suspension rate of the suspension spring Ks, the numerator of either equation becomes approximately $Ks_1 \times Ks_2$, so that the amount of change of the vertical load on tire due to a front tire deflation is nearly equal to that amount due to a rear tire deflation. Also, you will find that with respect to the amount of change of the vertical load on tire, the stronger the rear anti-roll spring is, the larger it will be when a front tire is deflated, and that in the case of a rear tire deflation, the stronger the front anti-roll spring is, the larger it will be.

As for displacement of the deflated wheel, you will find that by comparing the equation (1) with the equation (11), the stronger the suspension spring of the deflated tire is, the larger the amount of displacement or roll will be.

Shown in Table 1 is a computation example.

In this computation example, $Ks_1 \gg Ka_1$, so that the amount of change of the vertical load on tire when a front tire is deflated is approximately equal to that amount when a rear tire is deflated; and, since $Ks_1 > Ks_2$, the amount of displacement and roll is larger in the case of a front tire deflation. For a tire that deflects the most, the amount of change of the vertical load when a rear tire is deflated is 20.5kg; even in the case of a tire that is the largest in the second derivative of cornering power with respect to vartical load, $C_1 = C_{1R} + C_{1L} = 3187.6 + 3102.2 = 6289.8$, a negligibly small amount of change as compared with $C_1 = 6303$ under normal conditions. Therefore, it is our opinion that

with respect to running within the range of low lateral accelerations, effect of the tire deflection due to deflation could be disregarded.

Table 1

		δ (Wheel Displacement)		ΔZ (Change of Vertical Load on Tire)		ϕ_0 (Roll Angle)	
Tire	δ_0 (Tire Deflection)	Front Tire Deflated	Rear Tire Deflated	Front Tire Deflated	Rear Tire Deflated	Front Tire Deflated	Rear Tire Deflated
	(m)	(m)	(m)	(kg)	(kg)	(rad)	(rad)
A	0.0182	δ_{1R}^* = 0.0148 δ_{1L} = 0.0034 δ_{2R} = 0.0057 δ_{2L} = −0.0057	δ_{1R} = 0.0028 δ_{1L} = −0.0028 δ_{2R}^* = 0.0119 δ_{2L} = 0.0062	ΔZ_{1R}^* = −7.8 ΔZ_{1L} = 7.8 ΔZ_{2R} = 7.8 ΔZ_{2L} = −7.8	ΔZ_{1R} = 8.6 ΔZ_{1L} = −8.6 ΔZ_{2R}^* = −8.6 ΔZ_{2L} = 8.6	0.00874	0.00436
B	0.0333	δ_{1R}^* = 0.0270 δ_{1L} = 0.0063 δ_{2R} = 0.0104 δ_{2L} = −0.0104	δ_{1R} = 0.0052 δ_{1L} = −0.0052 δ_{2R}^* = 0.0218 δ_{2L} = 0.0115	ΔZ_{1R}^* = −14.2 ΔZ_{1L} = 14.2 ΔZ_{2R} = 14.2 ΔZ_{2L} = −14.2	ΔZ_{1R} = 15.7 ΔZ_{1L} = −15.7 ΔZ_{2R}^* = −15.7 ΔZ_{2L} = 15.7	0.01597	0.00799
C	0.0435	δ_{1R}^* = 0.0353 δ_{1L} = 0.0082 δ_{2R} = 0.0136 δ_{2L} = −0.0136	δ_{1R} = 0.0068 δ_{1L} = −0.0068 δ_{2R}^* = 0.0285 δ_{2L} = 0.0150	ΔZ_{1R}^* = −18.6 ΔZ_{1L} = 18.6 ΔZ_{2R} = 18.6 ΔZ_{2L} = −18.6	ΔZ_{1R} = 20.5 ΔZ_{1L} = −20.5 ΔZ_{2R}^* = −20.5 ΔZ_{2L} = 20.5	0.02089	0.01045

NOTE: *indicates a deflated wheel.

4.2 Steer angle and side slip angle during straightforwarding

As for the steer angle θ_0 during straightforwarding with a tire deflated, study of the equations (28) and (29) reveals that since Y_0 is always negative, if $M_0 < 0$, that is, moment that originates in the increase of rolling resistance as a result of a tire deflation is not canceled by moment that originates in the camber force due to roll as a result of a tire deflation, the first and second term in the equation (28) will the same in sign for a front tire deflation and opposite for a rear tire deflation, so that a significant difference is expected between a front and rear tire deflation.

As for the side slip angle during straightforwarding, according to the presumption described above, the first and second term in the equation (29) are always opposite in sign, and C_1 and A are approximately equal in the order and vary with the same tendency, so that the quantity of M_0 or Y_0 whichever is greater is considered to have a strong control over the result.

Shown in Table 2 is a computation example.

From the computation example, it can be seen that M_0 is occupied for the most part by a portion contributed by the rolling resistance as expected, and that M_0 and Y_0 are approximately the same in the order. Therefore, with respect to the steer angle, $\frac{C}{E}$ is larger in the comparison between $\frac{C}{E}$ and $\frac{B-A}{E}$, so that collectively speaking, M_0, in other words, rolling resistance is prevailing. In the case of a rear tire deflation, in addition to decrease in the rolling resistance as a result of decrease in the vertical load on the deflated

tire, because of a smaller roll angle than in the case of a front tire deflation, decrease in $|Y_0|$ and inverting of sign in $\frac{B-A}{E}$ make the steer angle reduce more substantially than when a front tire is deflated. All things considered, it can be said that those tires which are greater in the rolling resistance when deflated require a larger steering correction. As for the side slip angle, values we have obtained differ significantly from one another among the test tires. A noteworthy fact is that these values are not always negative, but may be positive in some case where the rolling resistance of the deflated tire is small and the camber force is sufficiently large.

Table 2

Tire	Location of Deflated Wheel	M_0		Y_0	$-\frac{C}{E}$	$-\frac{B-A}{E}$	$\frac{C_L}{E}$	$\frac{A}{E}$	θ_0	β_0
		$\left\{\frac{\partial Y_2}{\partial \gamma}\frac{\partial \gamma}{\partial \phi}+\frac{\partial Y_2}{\partial \gamma}\frac{\partial \gamma}{\partial \phi}\right\}_{\phi=0}$	$\frac{d}{2}(X_{1L}-X_{1R})$	$\left\{\frac{\partial Y_2}{\partial \gamma}\frac{\partial \gamma}{\partial \phi}+\frac{\partial Y_1}{\partial \gamma}\frac{\partial \gamma}{\partial \phi}\right\}_{\phi=0}$						
					x10^{-4}	x10^{-4}	x10^{-4}	x10^{-4}	(rad)	(rad)
A	Front	1.12	-13.68	6.52	-2.056	-0.678	0.857	-0.800	0.00348	0.00075
	Rear	0.58	-9.96	3.18	2.012	0.352	1.298	-1.251	0.00177	0.00082
B	Front	2.04	-6.79	13.90	2.019	0.843	0.783	-0.740	0.00207	0.00066
	Rear	0.09	-6.50	-6.76	2.020	0.274	1.299	1.225	0.00108	-0.00001
C	Front	0.71	-14.57*	-8.39	-2.505	-0.765	1.066	1.012	0.00411	-0.00063
	Rear	-0.45	11.49*	4.29	2.587	0.433	1.662	1.593	0.00290	0.00130

NOTE: *indicates an estimate by extrapolation.

4.3 Steer angle during cornering

The steer angle during cornering with a tire deflated may be expressed by the equations (40) and (41). Observation of these equations as a whole shows that they consist of a constant term to be ruled by Ackermann geometry, a term proportional to $\frac{V^2}{R}$ or the centripetal acceleration, and a term proportional to $(\frac{V^2}{R})^2$ or the square of the centripetal acceleration.

As for details of each term, coefficient which is proportional to $\frac{V^2}{R}$ consists of a term to be controlled by the cornering force and self-aligning torque when a tire is deflated, a term to be controlled by the product of the camber force and roll rate, and a term to be controlled by the product of a difference in the rolling resistance coefficient between a deflated and a normal wheel multiplied by the roll stiffness distribution on the side of a deflated wheel.

Coefficient which is proportional to $(\frac{V^2}{R})^2$ consists of a term to be controlled by the product of a difference in the rate of change of the self-aligning torque coefficient with respect to the tire load between a deflated and a normal wheel multiplied by the roll stiffness distribution on the side of a diflated wheel, a term to be controlled by the product of a difference in the rate of change of the cornering power with respect to the tire load between a deflated and a normal wheel multiplied by the roll stiffness distribution on the side of a deflated wheel, and a term to be controlled by the product of a difference in the rate of change of the camber force coefficient to the tire load between a deflated and a normal wheel multiplied by the roll stiffness distribution on the side of a deflated wheel multiplied by the roll rate.

Under normal conditions, coefficients of $(\frac{V^2}{R})^2$ are all become o and so becomes the third term of coefficient of $\frac{V^2}{R}$, a fact that is in accord with a conventional linear theory. Also, the fact that these coefficients invert their signs according to change of locations of a deflated wheel shows that they are elements that control a characteristic difference between turnings in the different directions.

As for the level of contribution of each element, the equations are so complicated as they are now as to make estimation extremely difficult; therefore, let us proceed with discussion by referring to a computation example shown in Table 3 below.

In these tables, constant $\frac{l}{R}$ to be determined by Ackermann geometry is omitted since it is common to all equations. The computed results show that with respect to coefficient of $\frac{V^2}{R}$, the first term or a part to be controlled by the cornering power and self-aligning

Table 3 (1/2)

Tire	Location of Deflated Wheel	Coefficient of $\frac{V^2}{R}$: $\frac{m(B-A)}{E}$	$\frac{K_\phi}{E}$	$\left(\ell C_2 + \frac{\partial AT}{\partial \alpha}\right)\frac{\partial Y_1}{\partial \gamma}\frac{\partial \gamma_1}{\partial \phi} - \left(\ell C_1 - \frac{\partial AT}{\partial \alpha}\right)\frac{\partial Y_2}{\partial \gamma}\frac{\partial \gamma_2}{\partial \phi}$	$\frac{CdK_1^2}{2E}$	$\left(\frac{\partial X_2^1}{\partial Z}R - \frac{\partial X_2^1}{\partial Z}L\right)$
A	Normal	$\times 10^{-4}$ 8.45				0
	RH front LH front	75.36	$\times 10^{-8}$ 0.0217	2927	$\times 10^{-4}$ 24.8	−0.1165 0.1165
	RH rear LH rear	−39.17	0.0201	−2488	20.2	−0.0984 0.0984
B	Normal	18.42				0
	RH front LH front	89.16	0.0207	20230	24.4	−0.0055 0.0055
	RH rear LH rear	−35.13	0.0201	−12338	20.3	−0.0072 0.0072
C	Normal	15.10				
	RH front LH front	85.02	0.0327	———	32.4	———
	RH rear LH rear	−48.14	0.0332	———	25.9	———

Table 3 (2/2)

Coefficient of $\left(\frac{V^2}{R}\right)^2$: $\frac{m_A^B CK_2^1}{E^2}$	$\left(\frac{\partial^2 AT_2^1}{\partial Z \partial \alpha}R - \frac{\partial^2 AT_2^1}{\partial Z \partial \alpha}L\right)$	$\mp \frac{m_A^B K_2^1}{E^2}\left(\ell C_1^2 \pm \frac{\partial AT}{\partial \alpha}\right)$	$\left(\frac{\partial C_2^1}{\partial Z}R - \frac{\partial C_2^1}{\partial Z}L\right)$	$\pm \frac{K_2^1 K_\phi}{E}\left(\ell C_1^2 \pm \frac{\partial AT}{\partial \alpha}\right)\frac{\partial \gamma_2^1}{\partial \phi}$	$\left(\frac{\partial^2 Y_2^1}{\partial Z \partial \gamma}R - \frac{\partial^2 Y_2^1}{\partial Z \partial \gamma}L\right)$
	0		0		0
$\times 10^{-4}$ 0.628	0.000250 −0.000250	$\times 10^{-4}$ −0.828	−0.1550 0.1550	$\times 10^{-4}$ 0.405	0.0040 −0.0040
0.432	0.000108 −0.000108	0.599	−0.1715 0.1715	−0.338	0.0062 0.0062
	0		0		0
0.643	−0.01025 0.01025	−0.891	−0.0375 0.0375	0.418	0.0042 −0.0042
0.425	−0.01272 0.01272	0.581	−0.0953 0.0953	−0.334	0.0032 −0.0032
	0		0		0
0.919	−0.00120 0.00120	−1.190	0.0025 0.0025	0.484	——— ———
0.707	−0.00120 0.00120	0.975	−0.0443 0.0443	−0.432	——— ———

NOTE: 1. For those which have two symbols and/or two suffixes, the upper value is for front tire deflation and the lower value is for rear tire deflation.
2. ——— indicates those on which tire data and/or characteristic values are not available.

torque when a tire is deflated is by far the largest. The other two terms are smaller in the order by one digit, and it can be seen that in these terms, effect of the camber force is larger than that of a difference in the rolling resistance coefficient between a deflated and a normal wheel.

Although not shown in the tables, with respect to the first term of $\frac{V^2}{R}$ mentioned above, comparison between the cornering power and self-aligning torque for their effect shows that the self-aligning torque is smaller in the order by one digit than the cornering power and that the former can be disregarded without causing a significant effect in the consideration on the characteristics of a vehicle when a tire is deflated.

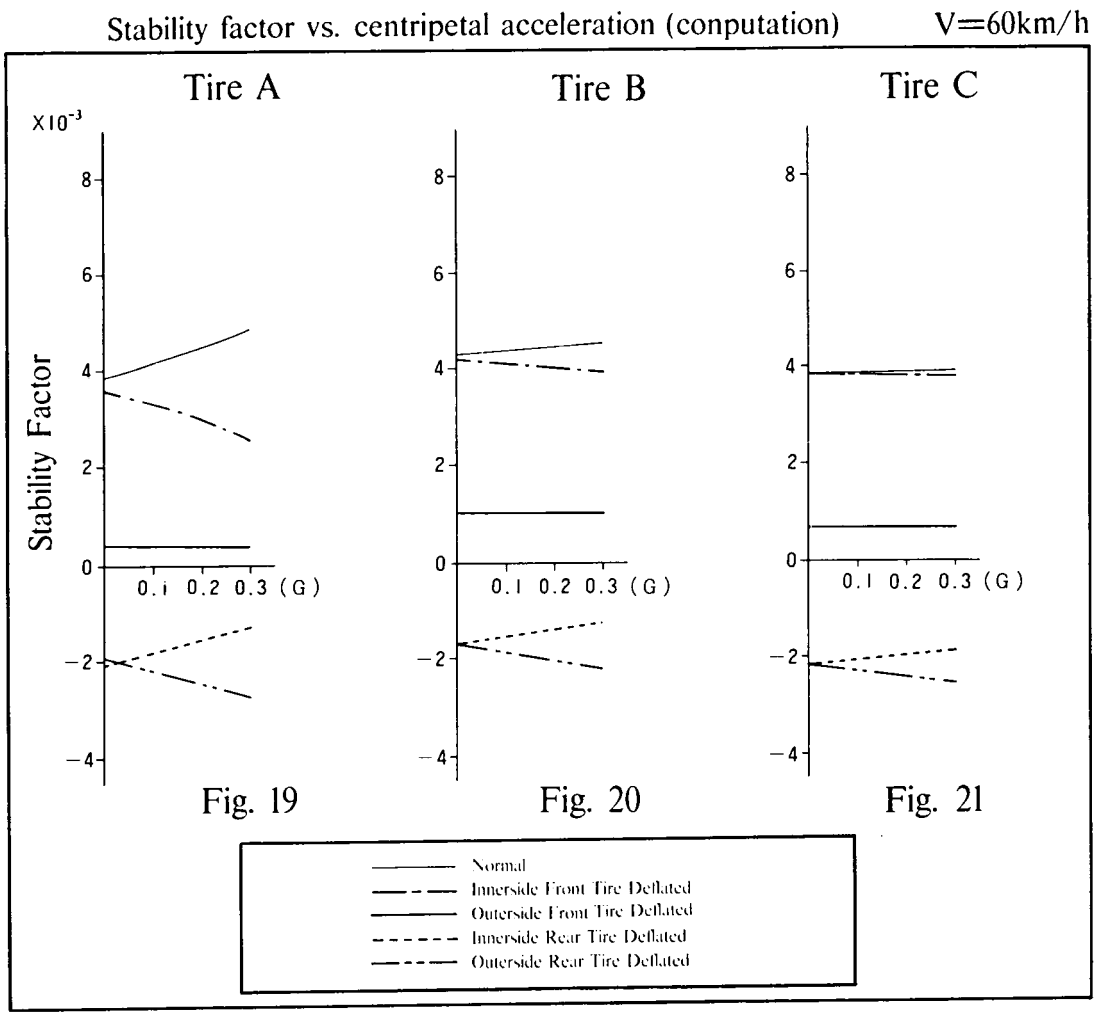
As for coefficient of $(\frac{V^2}{R})^2$, it can be seen that effect of the second term or a difference in the rate of change of the cornering power with respect to the tire load between a deflated and a normal wheel is remarkably larger and that the other two terms are by far smaller in the order.

Shown in Table 4 is what these results are summarized into a single equation. According to the findings of the study above, since those portions of tires on which individual data are not available are all found to have only a negligibly small effect, they are omitted in the process of development of the equation.

Table 4

Tire	Location of Deflated Wheel	θ
A	Normal	$\frac{2.2}{R} + 0.000892 \frac{V^2}{R}$
	RH front	$\frac{2.2}{R} + 0.008476 \frac{V^2}{R} + 0.000745 \left(\frac{V^2}{R}\right)^2$
	LH front	$\frac{2.2}{R} + 0.007897 \frac{V^2}{R} - 0.000745 \left(\frac{V^2}{R}\right)^2$
	RH rear	$\frac{2.2}{R} - 0.004231 \frac{V^2}{R} - 0.000598 \left(\frac{V^2}{R}\right)^2$
	LH rear	$\frac{2.2}{R} - 0.004628 \frac{V^2}{R} + 0.000598 \left(\frac{V^2}{R}\right)^2$
B	Normal	$\frac{2.2}{R} + 0.002286 \frac{V^2}{R}$
	RH front	$\frac{2.2}{R} + 0.009467 \frac{V^2}{R} + 0.000164 \left(\frac{V^2}{R}\right)^2$
	LH front	$\frac{2.2}{R} + 0.009224 \frac{V^2}{R} - 0.000164 \left(\frac{V^2}{R}\right)^2$
	RH rear	$\frac{2.2}{R} - 0.003753 \frac{V^2}{R} - 0.000354 \left(\frac{V^2}{R}\right)^2$
	LH rear	$\frac{2.2}{R} - 0.003782 \frac{V^2}{R} + 0.000354 \left(\frac{V^2}{R}\right)^2$
C	Normal	$\frac{2.2}{R} + 0.001510 \frac{V^2}{R}$
	RH front	$\frac{2.2}{R} + 0.008502 \frac{V^2}{R} + 0.000011 \left(\frac{V^2}{R}\right)^2$
	LH front	$\frac{2.2}{R} + 0.008502 \frac{V^2}{R} - 0.000011 \left(\frac{V^2}{R}\right)^2$
	RH rear	$\frac{2.2}{R} - 0.004814 \frac{V^2}{R} - 0.000248 \left(\frac{V^2}{R}\right)^2$
	LH rear	$\frac{2.2}{R} - 0.004814 \frac{V^2}{R} + 0.000248 \left(\frac{V^2}{R}\right)^2$

Shown in Figs. 19 through 21 are stability factors obtained from the result of Table 4 at an assumed velocity of 60 km/h, using the equation (25) and the equation shown in paragraph 3.2 respectively and indicated against corresponding centripetal accelerations.



Comparison of this result with the test result in Figs. 16 through 18 shows that a close coincidence exists between the two in the tendency of characteristic variation as a result of tire deflation. That difference exists with respect to the valves is due to the fact that the result with actual vehicles includes the roll steer characteristics, while that by computation leaves the roll steer out of count.

Now, let us touch again upon effect of the tire deflection described in paragraph 4.1. In that paragraph, effect of change of the vertical load as a result of tire deflation is described as negligible within the range of low lateral accelerations. Such a change of the vertical load as a result of tire deflation is, in the case of a tire which deflects the most, approximately equal to change of the vertical load during cornering with a centripetal acceleration of about 0.1 g; therefore, it is presumed that a wheel on the same side of the deflated wheel during cornering reaches as much earlier the saturation limit of characteristics, which, in cornering in the neighborhood of the maximum lateral acceleration, is to ease a difference in the characteristics between locations of a deflated wheel. Although it is not possible in this analysis to predict quantitatively such an effect, we would like to point out the existence of such a mechanism.

5. SUMMARY

The foregoing results may be summarized as follows:

5.1 Tire characteristics

(1) Increase in the rolling resistance as a result of tire deflation exerts a large effect on the steering correction during running with a tire deflated, but has only a small effect on the steering characteristics.

(2) The tire deflection as a result of deflation has only a small effect on the steering characteristics within the range of low centripetal accelerations. However, in cornering in the neighborhood of the limit, the deflection should have effect to ease a characteristic variation due to location of the deflated wheel.

(3) Difference in the cornering characteristics between normal performance and performance with a front or rear wheel deflated is determined mainly by the rate of decrease in the cornering power of a deflated wheel to a normal wheel.

(4) Difference in the characteristics between turnings in the different directions is determined mainly by the difference in the rate of change of the cornering power with respect to the load between a normal and a deflated tire.

(5) Other tire characteristics have only a small effect on the cornering characteristics of a vehicle when a tire is deflated.

5.2 Motor vehicle

(1) Variation of the characteristics due to the direction of turning with a tire deflated is controlled by the roll stiffness distribution on the part of a deflated tire, and the more the amount of roll stiffness contribution becomes, the more effect of a deflated tire will be.

(2) However, parameters of the vehicle have only a marginal freedom that may be altered for the sake of tires, so that it seems practically impossible to make up for a characteristic variation as a result of tire deflation by utilizing such a freedom on the part of the vehicle.

(3) However, for reason stated at the end of paragraph 4.3, there are possibilities to achieve an improvement in the characteristics of a vehicle when a tire is deflated by selecting tire capacity in such a manner as to utilize positively saturation portion of the tire characteristics or by doing something to make the roll understeer characteristics of a vehicle stronger. However, in this case, it is necessary to conduct a thorough study on how to balance the lowering of performance with such an improvement.

6. CONCLUSION

Here we described selection of items for evaluating the steering and handling characteristics of combinations of fail-safe tires and motor vehicle and the results of its tests. Also, based on the equations of motion of the vehicle's cornering, we made a quantitative prediction of effect of the tire characteristics and vehicle's design parameters on the cornering performance, and offered an image of desirable fail-safe tires.

However, as for elements to contribute the accident avoidance ability of a vehicle when a tire is deflated, there remain yet many other areas to be investigated, such as the critical characteristics, transient characteristics, effect on the steering system including driver, etc., requiring a further advanced study.

Shown at the end are the results of the tests of directional stability during braking when a tire is deflated, with a brief description thereof. (Figs. 22 through 24)

Max. Yaw Velocity during Deceleration vs. Deceleration during Braking (Fixed control) Vo=60km/h

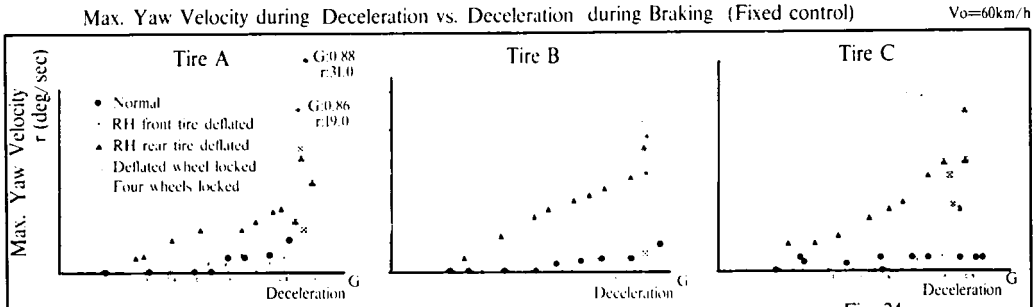


Fig. 22 Fig. 23 Fig. 24

Test Procedure

Using a test vehicle of the same specifications as given in the text, conduct the braking test without steering correction by a given deceleration at an initial velocity of 60 km/h under normal conditions and with a RH front and a RH rear wheel deflated, record the maximum yaw velocity developed and show it against corresponding deceleration.

Test Result

Little difference exists between normal performance and performance with a front wheel deflated. As for a rear tire deflation, the maximum value of yaw velocity larger than that under normal conditions was recorded, with difference between tires. However, in every test, the vehicle was observed to remain in a lane of 3.6 m.

DR. G. POCCI
Ministry of Transport

CLOSING REMARKS

We can perhaps draw a few conclusions from what we've heard this afternoon. I think I interpret all your views correctly, if I stress that safety, even if we limit ourselves to primary safety, is many sided in its aspects.

We've heard today things which seem to have nothing at all to do with the manufacture of the equipment of the vehicle. For example, the idea that the tire doesn't have to be changed might have safety aspects because we don't have to stop on the roads to carry out repairs; this doesn't involve manufacture, but it does have important aspects as far as safety is concerned. We've also seen that mechanical faults have to be avoided as much as possible; there's the question of up-keep, control, etc., which is necessary to insure that the vehicle really is a safe one.

Regulations and provisions have been mentioned on certain equipment, especially tires. I can tell you that now-a-days government authorities at the international level have already drawn up two regulations on tires; the first has been adopted in Geneva, and the second is to be adopted within the common market in Brussels.

Mr. Enke talked about a black box for cars as for airplanes. One of these days, that might make it possible to better judge the reasons for accidents. Something else that might be stressed in the following. Everything which, until yesterday, was judged subjectively, has now been presented objectively. For example, there was mention of the behavior of relationships between the vehicle and the driver. So far, road handling and vehicle behavior were judged individually. Now mathematical methods are being used and acceleration and so on is being measured, so the judgment achieved is much more objective.

As for lighting, we heard an Italian report, which stated that efforts are being made to find methods which make it possible to reproduce test and avoid individual judgments. All of this, of course, can be judged analytically in the laboratory, and it is possible to eliminate observers' mistakes.

In conclusion, safety must finally result in regulation. Tomorrow we're going to be talking about the application of safety ideas in regulations and standards. This will be the final results of all these ESV Programs. However, in Geneva, in the EEVC, or at European levels, work has been going on for years in vehicle safety. Most of the regulations deal with primary safety, that's why I said that primary safety was more advanced than protective or secondary safety. Now these problems are being dealt with as well, and I think that things will be improved as the ESV Program results become available, and this will be a major contribution to safety in vehicles.

SECTION 3

TECHNICAL SEMINARS

PART FIVE – OVERALL CAR REQUIREMENTS

Chairman: PROF. DR. ERNST FIALA
Volkswagenwerke AG
Federal Republic of Germany

Themes: Cost/Benefit for Safety Measures;
Interaction of Safety;
Environmental and Energy Factors

THE EVALUATION OF SAFETY, ENERGY, AND ENVIRONMENTAL FACTORS IN THE AUTOMOTIVE TRANSPORTATION SYSTEM

WILLIAM C. STEBER
Deputy Assistant Secretary
Systems Development and Technology
U.S. Department of Transportation

The United States Department of Transportation for several years has addressed the issues of automotive safety, environmental control, and more recently, energy conservation. The interplay of these three issues confronts us with one of the most complex systems analysis and cost/benefit trade-off problems in transportation history. The overall problem involves all transportation modes – land, sea and air. However, for the United States the heart of the matter lies in the automotive/highway transportation system.

- Automotive vehicles account for about 75% of the energy used directly by all U.S. transportation vehicles.
- In geographical areas that are subject to excessive incidences of photochemical smog formation, over half the tonnage of reactants can generally be attributed to motor-vehicle emissions.
- Finally, motor vehicles are involved in accidents which kill about 56,000 and injure about 4,000,000 people each year.

We are therefore confronted with the difficult question of how we can best reconcile the vehicle/highway systems engineering and integration requirements with the conflicting design goals of improved safety, reduced environmental impact, and optimized fuel consumption while maintaining at the same time reasonable life cycle cost to the consumer. I will direct my remarks today to the ongoing activities which we hope will lead us to rational decisions in the solutions to this automotive safety/energy/emissions/cost problem.

Automotive safety and the control of emissions have both received intensive study in the research and development activities of the Department of Transportation and the Environmental Protection Agency. More recently, emphasis has also been placed on automotive energy efficiency involving a comprehensive examination of all potential energy conservation measures throughout the manufacture and life cycle of the automobile. As a result of this examination, the conflict of safety and environmental control actions with energy conservation measures stands out clearly. For example, as many of you have already pointed out, occupant protection features generally increase weight with corresponding increases in energy needs for the manufacture and operation of the vehicle. Also, environmental protection devices tend to adversely affect fuel economy, creating higher operating energy demands.

The interactions and alternatives involved in these areas are currently under intense study by the Department of Transportation primarily through the Federal Highway Administration, the National Highway Traffic Safety Administration, the Office of the Secretary of Transportation and the Transportation Systems Center. In order to develop a total vehicle/highway system strategy and to establish the transportation safety and energy policies to support it, these organizations of the Department of Transportation, along with the Environmental Protection Agency and other agencies, have undertaken research and analysis programs to determine the technical and socio-economic trade-offs and options to guide the decision-making process.

One of the first steps in this direction was taken at the Fourth International ESV Conference in Japan over a year ago when we announced our intention to reorient the U.S. ESV program to address passenger vehicles weighing 3,000 lbs or less with safety performance predicated on requirements and conditions anticipated in the mid-1980s. As more recently defined, the Research Safety Vehicle (RSV) project objective is to provide data applicable to safety requirements for the 1980s, and to evaluate the compatibility of these requirements with

environmental policies, efficient energy utilization, and consumer economic factors. Consideration will be given to accident trends, automobile usage trends, anticipated roadway development and traffic regulations, and the possibility of traffic sub-environments which may render impractical the use of a single, all-purpose automobile concept. We fully expect this to be one of the most comprehensive examinations the federal government has ever made into the requirements of the personal automobile of the future in order to satisfy the safety as well as socio-economic needs of the American public.

Later today you will hear about the Environmental Protection Agency's program to evaluate selected alternatives to the current internal combustion engine for automobiles. This program started in 1970 to evaluate engines with the potential for significantly improved emission characteristics. The program was reoriented in 1972 to further emphasize those systems which offered the best possible fuel economy along with reduced emissions. In mid-1973 the program was expanded to study alternative fuels for both the internal combustion engine and its selected alternates. Plans are now underway to consider development of additional engine types. As you can see, the Environmental Protection Agency program and the Department of Transportation Research Safety Vehicle project are quite complementary, and resulting data will be useful to the RSV trade-off analyses as well as other DOT data base needs.

A major Department of Transportation management objective is to help achieve national energy goals by devoting a substantial effort to the conservation of energy consumed by the public and private transportation sectors. The Department has launched a Transportation Energy R&D program, a major part of which is the Automotive Energy Efficiency program. The primary objective of this program is to assess the technial capability of the automotive industry to increase the fuel economy of their production vehicles over the coming years. A technical data base and numerical simulation model computer programs which will analyze the technical and cost trade-offs to assist in identifying the best strategies to pursue in conserving energy in the vehicle/highway traffic system will be developed. This capability exists at our Transportation Systems Center at Cambridge, Massachusetts, and is being augmented for conduct of this program.

A number of studies have been conducted to quantify the relative importance of various techniques to increase vehicle fuel economy. One of these was a state-of-the-art survey of fuel economy technology for automobiles which indicated there are feasible gains in economy through improvements in fuel systems, ignition systems, transmissions, drive train components, and reduced vehicle weight and aerodynamic drag. We expect these surveys to be published shortly. The fuel economy of a small number of trucks and buses has been measured as a function of speed and operating conditions. Earlier studies of the overall automotive manufacturing process to assess lead time requirements have been expanded. Performance, manufacturing, and cost evaluations will be made of advanced engines and subsystems, alternate fuel and fuel mixtures.

A fleet mix model will estimate the fuel savings due to shifts to small cars, postulated improvements in engines, modifications to vehicles or introduction of new engines. It will be adapted to estimate the aggregate fuel consumption, emissions, and costs of future car fleets at several levels of emission control. A model of the overall automobile manufacturing process and data base on the manufacturing industry have been initiated to assess the skill mix, lead times, critical parts, etc., necessary for production of vehicles with improved fuel economy. Vehicle simulations are being used to estimate the effects on fuel economy of vehicle operations, substitute components, and driver performance. By building on the fleet mix model and developing a framework for an integrated Vehicles/Highway System approach, we are looking forward to the ability to make good judgments regarding the consequences of changes in vehicle technology on energy conservation, emissions, and safety.

The integrated Vehicles/Highway System approach to safety recognizes that contributions to the reduction of fatalities and injuries can be made through the driver, vehicles, highway, and traffic control elements of the overall system. It also recognizes the feasibility of incorporating safety, energy conservation, and pollution control into highway design and operational strategies for traffic movement. We know that highway design is becoming highly effective in the prevention of accidents, and the reduction of fatalities and injury severity. The design of our Interstate Highway System, which carries about 20% of all highway traffic has all but eliminated the frontal collision between vehicles by keeping the opposing traffic streams separated. The same result is now being accomplished on the primary roads in urban areas by installing a concrete median barrier on the roadway centerline. The fatal accident rate on Interstate highways is about half the rate on all other U.S. highways. In addition, about 13,000 vehicle hours are saved annually per mile because of its higher average travel speeds, and additional benefits accrue in reduced vehicle operating costs, in

tire wear, and fuel consumption because of fewer starts and stops.

During the past 6 years, U.S. State highway departments have participated in a safety improvement program based upon retrofitting system safety by use of improved traffic control and crash protection techniques and devices. As part of the program, rechannelization, pavement antiskid resurfacing, pavement marking and widening, installation of median barriers and other similar improvements have been implemented and evaluated. Careful records on both improvement costs and accidents before and after the improvements were obtained.

A commercial variation of the empty oil drum crash barrier installed in some 27 States was struck 92 times at speeds up to 70 miles per hour with only one broken bone injury resulting. The State of California finds it possible to save one life for each $35,000 spent on hazard elimination. The removal of roadside obstacles, use of pavement edge striping, selective use of guardrails and other protective barriers could average $5,000 per mile, or a total cost of $18.5 billion for the entire U.S. road system. Spread over 10 years of construction such improvements would require an annual investment of about $2 billion or about 4% of the societal cost of our highway accidents. Concentration on the high hazard locations can be even more cost effective. The Federal-Aid Highway Act of 1973 authorizes $550 million from the Highway Trust Fund over a three-year period for improvements at high hazard locations, elimination of roadside obstacles, and protection of railroad-highway grade crossings.

In this brief overview I have tried to describe some of the U.S. activities wherein we are attempting to analyze the costs, benefits and trade-offs involved in satisfying the conflicting requirements of safety, environmental control, and conservation of energy. As you can see it is a complex problem which we are attacking from a number of directions at this time. We are in the process of consolidating and integrating our analytical efforts in order to develop the capability to examine the alternatives, and make future decisions based upon an optimum balance between cost effectiveness and quality of life considerations.

I might add that I have been exceedingly impressed with the amount of information and the amount of data available for inputs to this study, as a result of the highway safety research activity that is taking place in all countries involved here. Presentations and displays at this conference have been very excellent and all of you are to be complimented.

AN OVERVIEW OF THE U.S. GOVERNMENT PROGRAM TO EVALUATE ALTERNATIVE POWERPLANTS TO THE CONVENTIONAL INTERNAL COMBUSTION ENGINES FOR AUTOMOBILES

JOHN J. BROGAN
Alternative Automotive Power Systems Division
Environmental Protection Agency

INTRODUCTION

There are three purposes for this paper. The first is to explain the basis for the U.S. government program to evaluate alternative powerplants in the automotive application. The second purpose is to describe the status of the program and the third is to show how this program relates to other U.S. government-sponsored research programs that involve safety, emissions control and conservation of energy for automobiles.

BASES FOR THE PROGRAM

In the decade of the sixties considerably increased concern was expressed by the U.S. public regarding the deteriorating air quality noticeable in major cities of our country. This was evidenced in many ways – as in the attention given to the problem in the popular press and other news media and in the increase in research by the federal government on causes of air pollution, its effects on health, and emissions control technology.

In 1965 Congress amended the Clean Air Act to authorize the Department of Health, Education, and Welfare to set exhaust emission standards for new light-duty motor vehicles to apply nationwide with the 1968 model year. California had started earlier in 1966. Although data on long-term health effects were not available on each significant pollutant from automobile exhausts (HC, CO and NO_x), projections made of the degree of emissions control required on this source indicated reductions on some pollutants by as much as 90% ultimately may be required.

There also was much concern within the scientific community that because of inherent deficiencies in the combustion process used in the conventional internal combustion engine, it may not be possible for this type engine to be adequately cleaned-up to meet these required future stringent emissions standards.* It was in late 1969 when members of the President's Office of Science and Technology, then headed by Dr. Lee DuBridge, expressed this concern to the President. It was felt then that it would be prudent for the nation to have available to it the technologies needed to produce alternative power systems for automobiles if they were needed and if they (the alternatives) were proven to be inherently clean and practical for this application.

A panel was formed to discuss these issues with the auto industry and to determine whether they, the U.S. auto industry, were developing or planned to develop any alternative engine system offering the potential of being virtually pollution-free. This panel concluded that serious development work was not underway – nor were there plans for such developments. As a result of these findings, a federal program was recommended, both to serve as a stimulus to industry and to provide the needed base of technology.

The President announced the program in his Message On the Environment early in 1970 and the program work formally began in July, 1970. This activity is called the Alternative Automotive Power Systems (AAPS) Program and is managed by the U.S. Environmental Protection Agency (EPA) from Ann Arbor, Michigan.

PROGRAM DIRECTIONS AND CHANGES

The purpose of the AAPS program is to evaluate alternative powerplants to the conventional internal combustion engine for automobiles. The program focused initially on investigating powerplants that offered high potential for being virtually pollution free. Five types of systems were initially under consideration: Hybrids such as heat engine/electrics and heat engine/flywheel types, battery powered electric systems, Rankine cycle and Brayton cycle systems. A year later, stratified charge versions of the internal combustion engine were brought into the program. Design studies were made for each system and in some cases component hardware was built and tested, all as part of the evaluation process. All of these systems were periodically ranked: based on updated knowledge of their potential for the automotive application, on the technical problems remaining to be solved and on their projected costs to develop. The ranking that resulted provided a basis for elimination of the less promising systems. At the same time this ranking provided a basis for selection

*These inherent deficiencies include use of an intermittent combustion process, with cyclic heating and cooling of chamber walls, a large number of variables influencing this process (such as cams, valves, springs, etc.) and expected difficulty in maintaining reasonable combustion efficiency during the life of the engine.

of the better systems which would receive development funding through system demonstration and test in automobiles, a final step in the evaluation process for these systems.

Hybrid systems were eliminated from the program because (generally) they did not provide the low emissions expected of them and they were found to be complex, very large in size and heavy compared with competitors. Further work on developing electric systems was deferred because the major weakness in this type system was found to be lack of a suitable battery. Development of this component appeared to be best achieved in other long-range programs being funded by other U.S. government agencies. In this instance, the National Science Foundation picked up the battery research on high temperature alkali-metal systems originally initiated in the AAPS program.

There were two versions of stratified charge engines that received development funding in the program – the PROCO by Ford Motor Company and the TCCP by Texaco. This development work was in support of the U.S. Army program which started in the mid-60s. Last year, the basic stratified charge engine development work of the Army entered a final development phase (soft tooling, fleet demonstration, etc.). At that point in its development, the stratified charge engine had progressed beyond the mission of the current AAPS program. The Army is continuing alone with the stratified charge development program.

For the reasons given above, the systematic elimination process left the Rankine cycle and Brayton cycle systems the two that would be developed through to systems demonstration and testing in automobiles by 1976. The evaluation report will follow those tests.

In 1972, the AAPS program was reoriented to emphasize specific work on improving the fuel economies of these two engines as well as to continue to develop their low pollution potentials. In mid-1973, the program was expanded further to study alternative fuels and to take a second look at the place of battery-powered electric vehicles in our nation's total transportation system.

The advent of the petroleum energy shortage last year highlighted the need for more intensive research and development on domestic energy sources, on improved energy conversion and on means to effect a reduction in energy consumption in our country. When it comes to the automobile and its role in our nation's energy consumption, it is a fact that the automobile consumes over 20% of all energy consumed in the U.S. and that the automobile alone consumes over 30% of all petroleum used in our country. At the request of the President, on December 1 last year, the Chairman of the Atomic Energy Commission (AEC) presented a 5-year, $10 billion energy R&D plan which included an expansion of the development activities embodied in the current AAPS program.

CURRENT STATUS

The Rankine cycle system development activities started in 1970 with design studies and technical problem identification. This was followed by technology development with application to solve the most elusive technical problems – designing a condenser to achieve a closed system and a lightweight and efficient boiler/burner. After gaining confidence that the technical problems associated with Rankine cycle components were solvable, detail system designs were initiated for four system versions of the Rankine cycle. These were followed by development of each system through complete system testing. The four competing versions were:

1. Steam with a reciprocating expander.
2. Steam with a turbine expander.
3. Organic fluid with a reciprocating expander.
4. Organic fluid with a turbine expander.

The technical team of contractors working on the Rankine program in various roles appears in Figure 1.

Based on the system test results (emissions, performance), on engine dynamometers and on the potential for further improvement, the four versions were narrowed down to a prime system: the steam version reciprocating expander (and a backup system, the organic version with a reciprocating expander). These two systems are currently continuing developmental testing. One version will be chosen this Fall to continue development for demonstration in automobiles during 1976.

The Brayton cycle system selected in the program was the open cycle, twin-spool regenerative gas turbine. The approach taken in the gas turbine development activity has been to attempt to solve the major problems facing introduction of this type engine as the prime mover in automobiles. Solutions would be demonstrated on available test bed (baseline) engines. The major problems are:

- High NO_x compared to the statutory standard.
- High cost of materials and manufacturing techniques.
- Poor fuel economy.

The overall approach being taken has been to bring to bear on these problems the knowledge of technical specialists from within the auto industry and from other industries and organizations skilled in various aspects of gas turbine technologies. Figure 2 shows

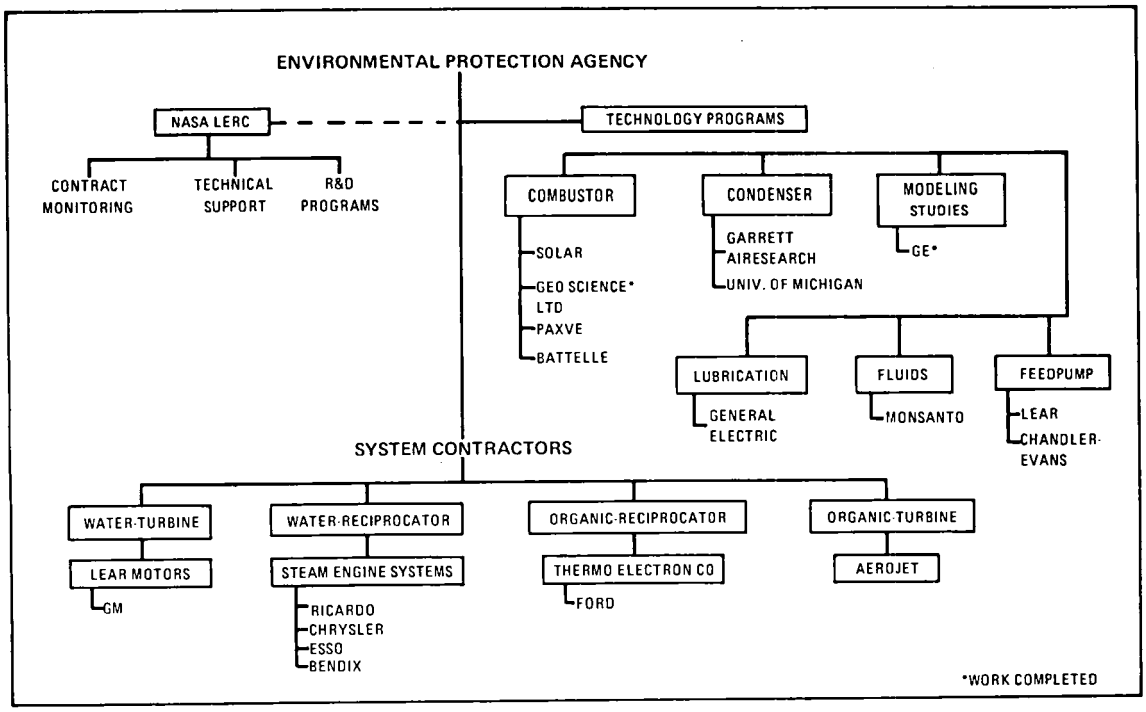


Figure 1. Rankine System Development Team

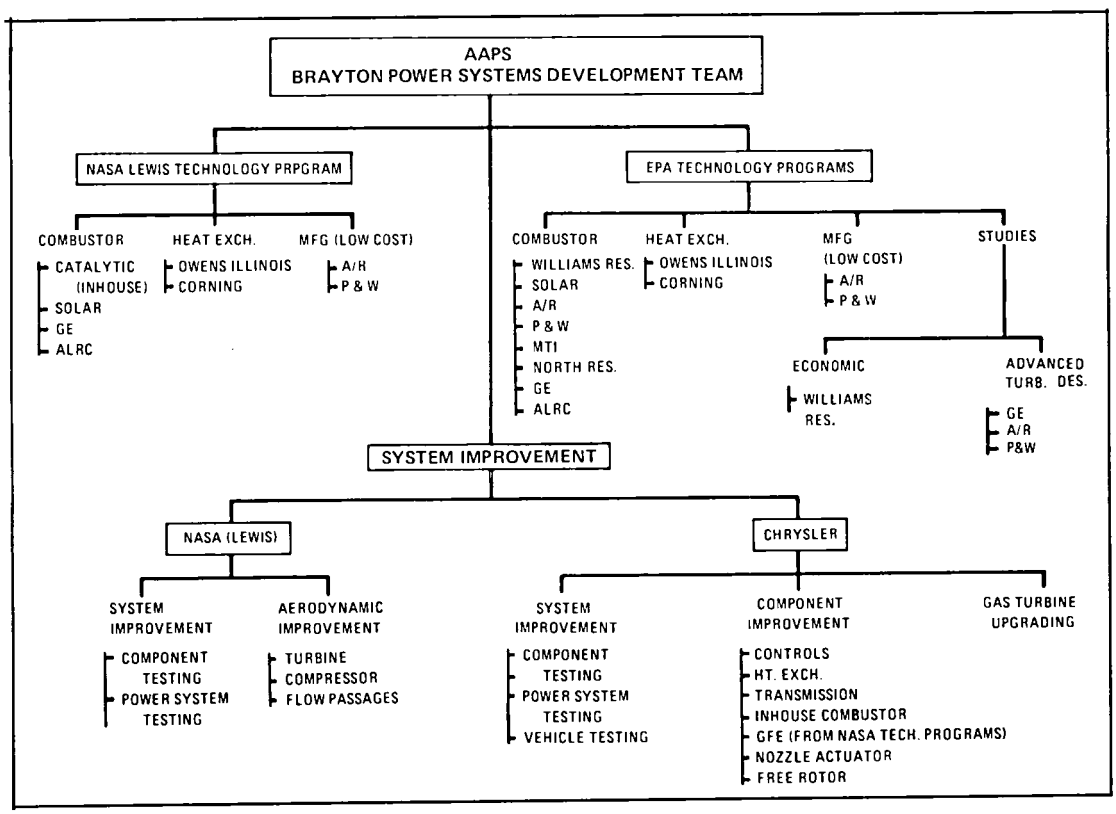


Figure 2

the different teams who have worked in or are working in the gas turbine program. The baseline engine, selected to demonstrate the research, is the Chrysler Corporation's sixth generation open-cycle gas turbine. This engine is an important element in the ongoing Brayton cycle program.

The NO_x emissions in gas turbine exhaust tends to be about half of the levels found in uncontrolled conventional internal combustion engines, while the HC and CO emissions from the turbine can meet the most stringent standard for these pollutants. Still, we are attempting to lower the NO_x emissions by employing combinations of approaches calling for shorter residence time, leaner mixtures and use of prevaporized and premixed air/fuel mixtures.

The manufacturing costs are influenced primarily by the use of critical metals in the rotating parts and by the lack of availability of the means to produce close tolerance rotating machinery with conventional casting techniques. The program work in these areas emphasizes use of both metallic and ceramic turbine wheels and regenerators, and development of new and low cost manufacturing processes for both the metallic and ceramic materials.

The fuel economy of available gas turbines is about 75% of the values of the conventional internal combustion engine of 1973 vintage for similar vehicle weights and power/weight, etc. The multifaceted approaches being taken to make the necessary improvement in turbine fuel economy performance appear in Table 1. Achievement of the 50% improvement noted would make the turbine match the conventional engine on fuel economy while not requiring any exhaust aftertreatment. The latter is vitally necessary for the conventional engine.

The Rankine cycle system and the gas turbine will be tested in automobiles in 1976. We expect both systems to meet the most stringent federal emission standards and to do so without exhaust aftertreatment and while matching the fuel economy of the conventional engine with its treated exhaust emissions.

As stated earlier, in mid-1973, the program expanded into investigations of alternative fuels for automotive applications. This work is showing that the most cost effective fuels available from abundant domestic resources other than those derived from petroleum include gasoline-like fuel from shale and from coal and methanol derived from coal. Further work will start soon to: better understand the economic, environmental and social impacts on our country of use of these fuels and to determine their compatibility with different engines; determine the products of their combustion, determine effects of these products on photochemical smog; and investigate means to control any hazardous pollutants appearing from the combustion process.

We expect that the new program emanating from the AEC energy R&D plan will expand the scope of the current AAPS program to permit serious consideration to developing such potentially efficient powerplants as those employing the Stirling and Diesel cycles. In addition, work will be directed toward improving energy consumption in automobiles regardless of the type of powerplant; for example, in reducing losses involved in accessory

TABLE 1
FUEL ECONOMY IMPROVEMENT ITEMS FOR BASELINE GAS TURBINE DEVELOPMENT PROGRAM

IMPROVEMENT ITEM	IMPROVEMENT DESCRIPTION	IMPROVEMENT GOAL	IMPROVEMENT IN TANK MILEAGE
1. Water Injection (Upstream of Compressor)	Reduced Engine Size for Same HP	10% HP	5%
2. Variable Inlet Guide Vanes	Reduced Engine Size for Same HP	12% HP	6 to 8%
3. Ceramic Regenerator	Better Effectiveness	+4 pts	6%
4. Higher Cycle Temperature	Reduced Engine Size for Same HP	7% HP	5%
5. Recover and Reduce Heat Loss	Reduced Parasitic Losses	50%	6 to 10%
6. Power Turbine	Higher Efficiency	+2-4 pts	7%
7. Compressor Turbine	Higher Efficiency	+4 pts	8%
8. Compressor	Higher Efficiency	+1 pt	2%
		TOTAL	45 to 51%

drives and transmissions. Other new projects are to be defined but will include investigations of turbocompounding and the use of bottoming cycles to improve fuel economies for long haul Diesel trucks.

This overview of ongoing and new work was intended to provide some insight as to the direction this work is going. A complete program will be defined by the Fall of this year.

RELATIONSHIP TO OTHER U.S. GOVERNMENT PROGRAMS

There have been numerous U.S. government-sponsored programs relating to the automobile other than the AAPS Program. Three of these programs sponsored by the U.S. Department of Transportation are:

1. Experimental Safety Vehicle Program – to design, fabricate and demonstrate vehicular systems which meet stringent safety requirements.
2. Research Safety Vehicle Program – to design, fabricate and demonstrate vehicular systems of 3,000 lb maximum weight and maximum possible safety features built-in.
3. Automotive Energy Efficiency Program – to develop a data base for use in determining the best strategies to pursue in conserving energy in the highway traffic system.

The basic purpose of these programs is to provide a reservoir of factual knowledge for federal policymakers, regulating agencies, Congress and the American public on those characteristics of the automobile which can affect safety, environment and energy consumption. The information generated from these programs generally has been obtained by working with the affected industry and this knowledge often "feeds back" to the individual members of industry and shows up in improved products without resorting to regulation.

Availability of this knowledge also may serve an additional purpose, including promulgation of sensible regulation in areas of safety, exhaust emissions, and/or fuel economy. It is the lack of knowledge in these areas and not its abundance that leads to faulty and noncost effective regulation.

RESEARCH SAFETY VEHICLE CRASH EFFECTIVENESS METHODOLOGY

JOHN J. EDWARDS
Ford Motor Company

This morning I will summarize a paper written by Dr. John Versace and Mr. Sherman Henson of our RSV Program Office. Time does not permit a presentation of the complete paper, but copies are being made available to all conference participants.

Ford Motor Company is one of five contractors awarded a first phase contract by the United States Department of Transportation for the development of a research safety vehicle. The first phase, which is 15 months long, will develop performance specifications and a preliminary design for a 3,000-pound RSV. Our goal is to develop a concept for a 3,000-pound car which will optimize safety benefits within the mixed-vehicle population of the mid-1980s.

The overall approach will:

1. Predict the traffic environment in the mid-1980s,
2. Integrate the predicted accident data from this environment into a system model,
3. Use this model to identify the design concept required for an optimized vehicle in this traffic environment,
4. Develop a preliminary vehicle design which reflects the model output,
5. Develop performance specifications based on the optimized design concept.

Establishing performance specifications for a research safety vehicle will result from a thorough system analysis to determine the trade-offs in system features, looking toward the greatest safety payoff and given practical constraints on size, weight, carrying capacity, and probable cost. Allowance must also be made for those additional product features which the consumers will demand in a practical, marketable car. The vehicle is to be viewed as a system, but one which is embedded within a larger system – that of the driving environment of the mid- and late 1980s.

There are two approaches which may be taken to increase safety to a maximum level. The first approach would specify the requirements, leaving open the method for their achievement. This approach is open-ended, without a ceiling. It does not prevent the escalation of requirements to an impractical level of costs and other penalties. The emphasis is on increasing safety and letting costs and complexity fall where they may. There is no formal way in this approach of setting a ceiling where diminishing returns begin to set in.

In the second approach, the hardware, rather than the performance specifications, is the initial focus of interest. The characteristics of various hardware arrangements are established, or provisionally assumed, and a determination is then made of the system's probable effectiveness by calculating occupant dynamic and biomechanical response to crash inputs. The emphasis in this case is different from the first: costs and complexity will not be allowed to fall where they may, but limits will be established in advance and safety maximized within those limits.

The Ford RSV analysis will be based on the second of the above two approaches. Because there are several ways of producing traffic casualties, there are also several ways of reducing them. The system optimization task then becomes one of determining how to allocate a limited resource of countermeasures among the various elements of the system in such a way that the final system response – the casualty measure – is minimized. The countermeasures are limited because the total weight cannot exceed 3,000 pounds and exotic, impractical solutions are ruled out by considerations for producibility, marketability, and cost.

The system analysis starts by considering the probability distribution of exposure to accident events, as projected to the 1985 time frame. The output of the analysis is an overall measure of casualties. The method will involve the repeated calculation of system output as product performance assumptions are incrementally varied. That list of variations resulting in the minimum casualty level is then searched for.

Figure 1 shows the general method of analysis in our design-oriented approach. First, a large number of accidents selected to be representative of the population are each used in turn as inputs to the analytical model. The model is made up of individual simulations for frontal, side, etc., types of accidents. Parameters of the vehicle model, (e.g., masses, stiffness, dimensions, deflections, pressures, moduli, force-deflection curves, etc.) are varied in successive trials. Analysis will then proceed as these parameters are systematically varied within the limits of allowable combinations and defined constraints.

Major elements of the system model include a trade-off analysis procedure, a vehicle model, an occupant model, a crash avoidance model, and

probable accident profiles which the RSV will encounter in the mid-1980s. It is the tool for the overall optimization process and can be regarded as perhaps the major analytic task in the entire program. Optimization – the trade-off between what we think the traffic ambience will require in terms of safety levels; what our analysis indicates is the most efficient use of the vehicle's weight; and what can be practically developed – will continue through most of the project and its output will be reflected in the preliminary vehicle design.

The approach to injury scaling used in the model will focus on the conversion of dynamic response into a probability of survival – the chance the occupant would have of surviving a particular level of deceleration, rather than some graded magnitude of injury. The "lethal dosage" form of severity scale that biologists customarily use in describing tolerance to stressing agents is a probabilistic type of scale. Scales that related the probability of survival to the amount of head deceleration were proposed by biomechanics researchers at the University of Michigan Highway Safety Research Institute (1971).

Figure 2 shows such a fatality threshold curve for chest deceleration. It is important to note in this tolerance curve that while survival probability may rapidly decline as the g-level rises, there is no abrupt cutoff to zero survivability at some particular g-level. It is not true that 60 g is survivable while 61 g is not. People and circumstances are individually different in an infinite number of ways, leading to the kind of probabilistic threshold curve seen here rather than an abrupt live-or-die cutoff at some g-level.

Probability of survival can also be related to barrier-equivalent speed for a particular system. By combining this with the curve in the previous figure, we can show the joint relationships between crash input level, occupant dynamic response, and probability of survival. This is discussed in the paper.

Effectiveness of a frontal crashworthiness system, for example, must take into account the probability of being exposed to a crash at any given speed. Relatively few cases of 50-mph barrier-equivalent collisions occur, and relatively few persons are fatally injured in 15-mph crashes, despite their far greater frequency. Accident data are relied upon to determine the statistical distribution of barrier-equivalent speeds at which fatalities occurred. As shown in the lower part of Figure 3, these fatally injured persons then constitute the candidates available for saving.

System effectiveness is determined by multiplying the relative effectiveness, or probability of survival given that the crash takes place, *the upper curve*, with the probability of being exposed (in this case, the probability of being a fatality in the absence of the system, *the lower curve*). The product of these two functions is integrated across the whole speed spectrum, yielding the relative number of lives that would be saved.

It is very important to note a significant point here: it is the hardware which actually saves the lives, not the standard. Therefore, the likely characteristics of the hardware must be taken into account when evaluating a test requirement. It was for this reason that a simulation of actual system dynamics is being used as the basis for calculating effectiveness. The likely characteristic of any real-world hardware will always involve a "roll-off" in effectiveness, not an abrupt cutoff at some designated barrier speed. There will always be some effectiveness above the required test speed. Because of the "roll-off", the idea of partitioning the exposure distribution at the mandated test speed and assuming that only those persons below that speed are available for saving must be rejected as incorrect.

System effectiveness must be evaluated for each of the crash modes every time any change in vehicle parameters are considered. Thus, attempting a trade-off in front structure for increased side strengthening would equate to less forward protection and more side protection – given that total vehicle mass must remain constant. Net savings in lives are obtained by summing the component (front, side, rollover, etc.) savings. This sum is the overall benefit.

Ford Motor Company is very pleased to be a part of the RSV Program. This paper indicates our approach to a determination of overall effectiveness which we are striving to maximize within the established limits of weight and with appropriate consideration for practicability and producibility.

INTRODUCTION

This paper describes several aspects of safety performance evaluation. The specification of performance requirements for a research safety vehicle will be developed from a thorough systems analysis that determines the tradeoffs in system features, looking toward the greatest safety payoff, given practical constraints on size, weight, carrying capacity, and probable cost. Allowance must also be made for those additional product features

which consumers will demand in a practical, marketable car. The research safety vehicle is viewed as a system, but one which is embedded within a larger system, that of the driving environment of the mid and later 1980's.

There are two approaches which may be taken to increase safety to its maximum level: (A) The first of these would specify the requirements, leaving open the method for their achievement. This seems like the most logical way to do it, but it is open-ended, without a ceiling; it doesn't prevent the escalation of requirements to an impractical extension of costs and other penalitites. The emphasis is on increasing safety and letting costs and complexity fall where they may. There is no formal way, in this approach, of setting a ceiling where diminishing returns begin to set it. (B) On the other hand, an alternative approach is to circumscribe the allowable bounds of design, production difficulty, and cost and then search for the configurations within those bounds which result in the smallest number of fatalities and injuries. The emphasis in this case is different from the former: Costs and complexity will not be allowed to fall where they may, but limits on them are established in advance and safety is maximized within those limits. However, in that formulation, the smallest number of resulting fatalities and injuries may still be greater than we might wish for.

Taking the systems approach focuses attention on the final goal, minimization of traffic casualities, rather than on components or intermediate measures. This has been referred to as looking toward the "bottom line." Because there are several ways of producing those traffic casualities, there are also several ways of reducing them. The system optimization task then becomes that of determining how to allocate a limited resource of countermeasures among the various elements of the system in such a way that the final system response, the casuality measure — the "bottom line" — is minimized. The countermeasures resources are limited because the total weight cannot exceed 3000 lbs, and exotic, impractical solutions are ruled out by considerations for producibility, marketability, and cost.

The systems analysis would start with a consideration for the probability distribution of exposure to accident events, as projected to the 1985 time frame. The output of the analysis should be some overall measure of casualties. The method will involve the repeated calculation of system output as product performance assumptions are incrementally varied; that list of product variations resulting in the minimum casuality level is then sought. The product variations selected for each simulation trial would be specified by reference to our considerable design and crash test experience and adjusted to meet preset weight, size, cost, durability and reliability limitations.

Accident avoidance aspects of vehicle design may be traded off with occupant protection features. However, neither that nor the effect of changes in highway designs and changes in the rules of the road will be considered in this paper. It should be noted, however, that most highway and traffic matters are not within the purview of the vehicle designer or producer, and thus it is appropriate to view them as constants, as determiners of the environment in which the vehicle system will operate, but not subject to optimization trade offs.

OVERALL ANALYSIS CONCEPT

System analyses can proceed in at least two different ways. First, a panoply of possible performance requirements, without reference to the kind of hardware for meeting them could be set out, as is generally done in the NHTSA standards and as was done in the previous ESV projects. Judgments as to the probability of being effective in saving lives might then be made for each of the requirements, and for each element in the matrix of input (accident) conditions. While this kind of evaluation was not made of the ESV specifications, it is the kind of procedure used by NHTSA (Cooke, 1971) in an effectiveness evaluation of the FMVSS 208 "air bag" standard. In this method, emphasis is on such things as judgmentally established effectiveness factors — factors that range between 0 and 1.0 which are used as multipliers on accident frequencies — rather than on dynamical or biomechanical considerations.

In the second approach, the hardware rather than the performance specifications is the starting point. The characteristics of various hardware arrangements would be established or provisionally assumed and a determination then made of the system's probable effectiveness by calculating occupant dynamical and biomechanical response to crash inputs. (Grush et al, 1971).

The Ford RSV analysis will be based on the second of the above two approaches. See Figure 1. There are two principal, broad concepts.

The first broad concept, the exposure distribution, is illustrated in the lower part of the figure. People in accidents are more often exposed to collisions at low severity levels than at high severity levels. The distribution of barrier-equivalent fatality speed is shown here as a measure of exposure. (It is used here for illustrative purposes; it is not necessarily the most appropriate measure of crash severity.) A protective system should serve to save as many of these as possible; a perfect system would have 100 per cent effectiveness, it would save all.

The second broad concept, the vehicle's relative safety effectiveness, is illustrated in the upper part of the figure. A given vehicle will generally have a declining capability for protecting the occupants at the higher levels of crash severity than at the lower. Vehicles of different design presumably will differ in the shape of this curve, in the relative protection they offer at low crash severity as compared to higher crash severity.

The concept of "relative performance effectiveness" — the probability of survival given the crash of that severity — could be referred to as the "severity response" characteristic of the system, being somewhat analogous to the "frequency response" characteristic of dynamical systems. It can be characterized by similar parameters, such as that severity level on the roll-off portion where some designated survival probability is crossed, e.g., say at the 75 per cent point.

Another parameter might be the roll-off rate. The concept of relative performance effectiveness is further described in Figure 2.

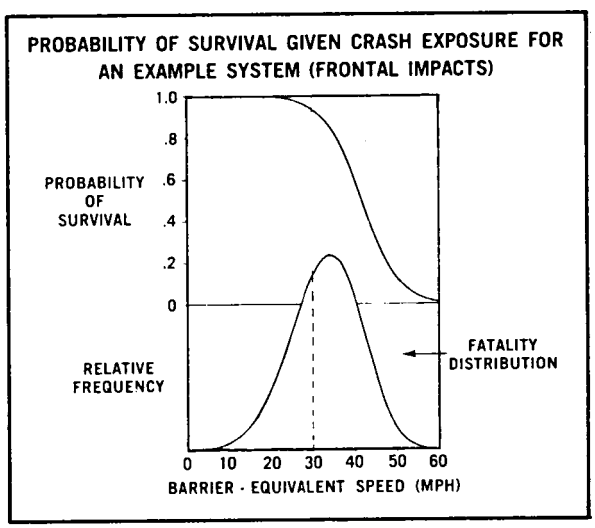


Figure 1

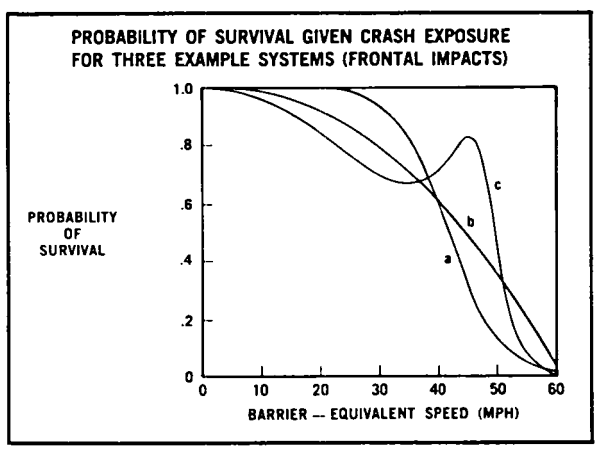


Figure 2

The three curves in Figure 2 represent different ways the vehicle system might offer protection, as a function of the collision severity. The hypothetical curves shown might represent systems with these properties:

a. A large measure of protection over a limited severity range, with rapid roll-off above some collision severity level.

b. A continuously declining performance as severity increases, but no sharp drop in performance, so that it outperforms "a" by a wide margin at the higher input severities.

c. A performance which is "tuned" to be particularly effective at some high severity value, making it superior to the others at that level, but at the expense of some effectiveness loss at lower levels.

The dimensions and units of measure for both coordinates — "collision severity" and "relative performance effectiveness" — are not firmly established at this point and continue to be an object of study. The barrier-equivalent speed is perhaps the most likely type of variable for the measure of collision severity — especially if it is adjusted to reflect some of the dynamic aspects of the struck object. Relative performance effectivesness is not readily definable in terms of dynamic performance, so it will have to be based on a biomechanical calibration of dynamic occupant model output into survival probability.

VEHICAL PERFORMANCE CHARACTERISTICS

System optimization studies are carried out by varying the characteristics of the system until the maximal effect occurs. The system characteristics that are varied can be any defining or descriptive features, usually design features: e.g., masses, stiffnesses, dimensions, deflections, pressures, moduli, force-deflection curves, etc. Analysis could proceed as these are systematically varied, within the limits of allowable combinations and defined constraints.

Figure 3 shows the general method of anaylsis in a design-oriented approach. First, a large number of accidents selected to be representative of the population are each used in turn as inputs to the analytical model. The model is made up of individual simulations for frontal, side, etc. types of accidents. Parameters of the vehicle model are varied in successive trials, either by some predetermined schedule of trial values, or by an appropriately derived scheme of improvement chasing. There is also an occupant simulation, whose make-up depends on the type of input under consideration (i.e., crash mode, size percentile, seated position, etc.) Dynamic response of the occupant is the dependent variable in the simulation runs.

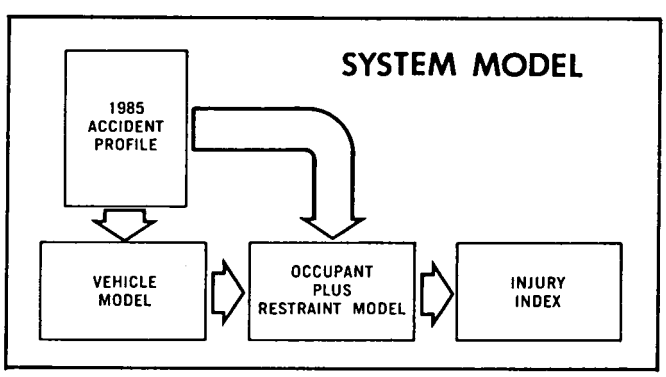


Figure 3

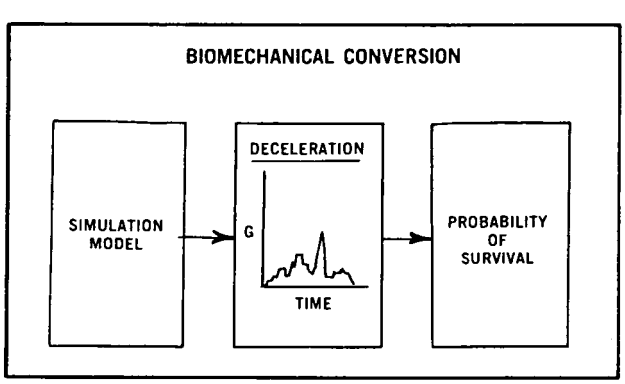


Figure 4

The area of simulation best developed at this time is for frontal crashes using, for example, the Cornell 11-degree-of-freedom, 2-dimensional occupant model (Segal and McHenry, 1968.) This model is adaptable to belt configurations of various types, and its

adaptation to air bag types of restraints is also becoming available. Such simulation yields results in dynamic terms: head and chest deceleration. But these must be converted to human tolerance terms by some kind of biomechanical conversion. See Figure 4.

The deceleration output of the simulation model, shown in the center box of Figure 4, will of course depend upon the crash input level to the simulation, upon the nature of the vehicle and the restraint, upon the size of occupant, etc. For any type of occupant and type of vehicle/restraint, the deceleration level will depend on the crash input severity.

A calibration is needed which, for a given vehicle structure and a given restraint system, describes how the estimated deceleration changes with input severity level. Different structures and different restraint systems would require different calibration curves.

BIOMECHANICAL SCALING OF INJURY

One line of approach to injury scaling has been the quasi-quantitative scaling found in accident reports, for example the 10-point Abbreviated Injury Scale (AIS) developed by the American Medical Association. Similarly, there is the 5-point KABCO scale generally used by police reporting agencies, in which scale values roughly correspond to fatal, serious, moderate, minor and none. It is rather difficult to see at first how to go from the kind of response obtained from crash testing or from dynamic simulation — chest deceleration, for example, in Figure 4 — to a corresponding value on one of these injury scales.

A second line of approach to injury scaling would focus on the conversion of dynamic response into a probability — the chance the occupant would have of surviving that level of deceleration. The measurement scale, then, would be probability of survival rather than some graded magnitude of injury. The "lethal dosage" form of severity scale that biologists customarily use in describing tolerance to stressing agents is a probabilistic type of scale; Versace (1971) suggested application to trauma. Scales that related the probability of survival to the amount of head deceleration were proposed by biomechanics researchers at the University of Michigan Highway Safety Research Institute (1971). Grush, et al (1971), used fatality threshold curves. It is important to note that in tolerance curves survival probability may rapidly decline as the g-level rises, but there is no abrupt cut-off to zero survivability at some particular g-level. It is not true that 60 g is survivable while 61 g is not. People and circumstances are individually different in an infinite number of ways, leading to some kind of probabilistic threshold curve rather than an abrupt live-or-die cut-off at some g-level.

Use of such threshold curves should not obscure the fact there is quite imperfect quantification of the biomechanical basis of injury in real accidents. The type of injury criteria applied to crash test dummies in the tests required to meet NHTSA standards have not been demonstrated to have one-to-one relation with actual field experience. This is particularly true of belt restraints, which have shown excellent field effectiveness while often yielding large crash test readings. Among other things, measurements of this sort do not take into account any load distribution effect on occupants (Warner, 1972) — still an essentially unquantified effect at this time — nor do they account for type of injury. Accurate and pertinent human tolerance data are scarce.

RELATIVE EFFECTIVENESS

Each vehicle/restraint system can be characterized by the relation between the crash input level and the resulting dynamical response — the occupant deceleration. But given the occupant deceleration, the probability of survival still needs to be estimated. What would be needed is some calibration of the joint relationships among crash input level, occupant dynamic response, and probability of survival. Different protective systems would have different calibrations among these variables.

The relative effectiveness curve in Figure 5 assumes an occupant is exposed to the crash. It gives the probability of survival, given a crash of that severity. Overall effectiveness in a societal sense, however, must also take into account the probability of

being exposed to a crash at that speed. Relatively few cases of 50-mph barrier-equivalent collisions occur, and relatively few persons are fatally injured in 15-mph crashes despite their far greater frequency. Accident data are relied upon to determine the statistical distribution of barrier-equivalent speeds at which fatalities occurred, as shown in the lower part of Figure 6. These fatally injured persons then constitute the candidates available for saving. (Strictly speaking, it is the distribution of exposure to collision which is appropriate — regardless of whether the injury was fatal or even if there was no injury. The fatality distribution is really the outcome, not the input, and its shape will change as systems are improved. However, the fatality distribution determined from cars predominantly of the mid-1960's, as is the case shown here, can be viewed as an exposure distribution — for fatalities — relatively invariant under system changes. The seriousness of the discrepancy would depend on the shape of the relative effectiveness curve — for example, if it were like curve "C" in Figure 2. The more correct approach is to take the exposure to the collision rather than just the instances of fatality, but establishing such a distribution will be much more difficult because of data lacks. However, it is being pursued in Ford's Research Safety Vehicle project.)

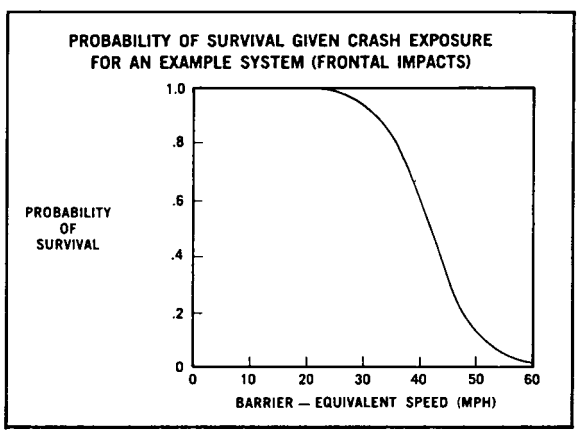


Figure 5

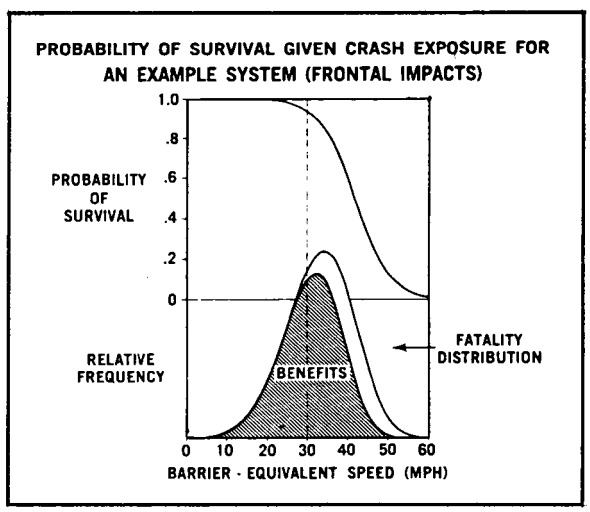


Figure 6

DETERMINING EFFECTIVENESS

The fatality distribution is shown in the lower part of Figure 6 together with the portion of it — shown shaded — for which the occupant protection system is effective. The effectiveness is arrived at by the procedure outlined in Figure 1, by multiplying (a) the relative effectiveness, or probability of survival given the crash, and (b) the probability of being exposed (in this case, the probability of being a fatality in the absence of the system). The product of these two functions is integrated across the whole speed spectrum, yielding the percent effectiveness (or the number of lives that would be saved by using the actual frequency distribution of exposure instead of the relative distribution.)

> It is very important to note a significant point here: It is the hardware which actually saves the lives, not the standard. Therefore, the likely characteristics of the hardware must be taken into account when evaluating a test requirement. It was for this reason that a simulation of actual system dynamics is being used as the basis for calculating effectiveness. The likely characteristic of any real-world hardware will always involve a "roll-off" in effectiveness, not an abrupt cut-off at some designated barrier speed. There will always be some effectiveness above the required test speed. Because of the roll-off, the idea of partitioning the exposure distribution at the mandated test speed and assuming that only those persons below that speed are available for saving must be rejected as incorrect.

There will be a different effectiveness in each of the various crash modes, and the overall effectiveness is the sum of them. Unfortunately, simulations for non-frontal crashes

are not well-developed and reliable, so effectiveness determinations for non-frontal crashes may have to be made largely from reported accident data. Inasmuch as restraints will have relatively less effect on side impact or rollover protection as such, most of their effectiveness in such accidents relates to the degree they can control ejections.

The overall safety effectiveness of the whole vehicle must be determined every time any change in vehicle parameters is tried. Thus, trying a trade-off in front structure for side strengthening means that there will be less forward protection and more to the side, given that total vehicle mass remains constant. Overall benefit, or net savings in lives, will be determined by summing the savings and losses resulting from each component change.

RSV IMAGE

The procedure described in this paper for maximizing safety is largely design-oriented: that is, variations in the basic design elements of the vehicle are tried by simulation, until that combination of variations is found which predicts the maximum safety performance achievable. However, the design variations to be tried are limited by a constraining envelope of vehicle characteristics defined by the vehicle image.

The vehicle image is a necessary starting point for the creation of a practical car. There is an infinite number of possible configurations of vehicle components and design elements. To consider exhaustively all possibilities would be a virtually impossible and wasteful procedure. Thus, one must start with known combinations of components and designs in existing vehicles and work by iteration toward a realistic convergence of safety and practicability.

To meet the 3000 lb. weight limitation within the bounds of proven automotive technology dictates a Pinto to Maverick size. Once overall size is determined, cost and other marketing considerations establish the limits of operational characteristics, noise, vibration and harshness, durability and reliability, and customer option availability. Any realistic limitation on vehicle weight must include provision for the optional equipment that customers buy. If the objective is to have all production cars of a given size class grouped around a target weight of 3000 lbs., one must include the weight of the optional equipment that about 50 per cent of all customers order. In this way, some cars will weight less and some more but most will be about 3000 lbs.

While the procedure that has been described in this paper is largely design-oriented, it is not the objective of this project to arrive at an RSV design. Rather, the objective is to arrive at a set of performance specifications that would increase safety to the maximum level. Obviously, performance specifications could be established merely on the basis of wants, with no reference to the means for achieving them, as described under (A) in the Introduction to this paper. However, to establish realistic safety performance specifications, i.e., requirements that are achievable in a producible and marketable 3000 lb. car, requires that the overall limits of the car be determined first. Then, the maximum safety performance of which that car would be capable becomes its specification.

REFERENCES

1. Cooke, C. H., Safety Benefits of the Occupant Crash Protection Standard, NHTSA, January 1971, General Reference File, Item 43, NHTSA Docket 69-7.

2. Grush, E. S., Henson, S. E. and Ritterling, O. R. Restraint System Effectiveness, September 1971, General Reference File Item 119, NHTSA Docket 69-7. (Later published in Highway Research Record No. 420, Highway Research Board, January 1972.) 16-24.

3. McElhaney, J. H. Et al. Door Crashworthiness Criteria, Proceedings of 15th Stapp Car Crash Conference, 1971, 489-517. Also see: Biomechanical Aspects of Head Injury, Human Impact Response, Plenum (N.Y., 1973) pp. 85-112.

4. Segal, D. J. and McHenry, R.R. Computer Simulation of the Automobile Crash Victim, Revision 1. Cornell Aeronautical Laboratory, Inc., CAL Report VJ-2492-VI, March 1968.

5. Versace, J. A Review of the Severity Index, Proceedings of 15th Stapp Car Crash Conference, 1971, 771-796; see Appendix.

6. Warner, C. Y. Discussion of Grush, et al, Restraint Effectiveness, Highway Research Record No. 420, Highway Research Board, January 1972, 24-26.

BENEFIT/COST ANALYSIS AS A BASIS FOR DECISIONS IN THE AUTOMOTIVE INDUSTRY

DR. ERNST FIALA
Volkswagenwerke AG

A SURVEY

In recent years benefit/cost analysis has proved itself to be a suitable method for the assessment of competing safety measures. Various governments have already decided, or at least announced, their intention to use it as a deciding factor for the basis of safety regulations. The time has come, therefore, to report in detail on the work done to date and on the problems connected with the benefit/cost analysis as applied to safety measures in road traffic.

With respect to traffic safety, the benefit/cost analysis commences with the extent of damage which experience has shown would occur in the period under observation if no countermeasures were used. These countermeasures call for a certain outlay which is known as the "cost". The "benefit" is represented by the reduction in damage which is achieved. This definition is very important because there has occasionally been misunderstanding about it. "Cost" was thought to be overall costs and "benefit" the total benefit to road traffic. To repeat: the cost is the additional outlay for safety measures, and the benefit is the reduction of the damage otherwise incurred. At this stage it is immaterial whether the costs are private or public.

In general, the monetary assessment of the costs causes no special difficulty. It is important to include all expenses incurred (e.g., depreciation of investments). For example, in the U.S. Exterior Protection Standard MVSS 215, which stipulates that no damage may result to a vehicle colliding with a fixed obstacle at 5 mph, it was forgotten that, because of increased weight and spoiling of front end streamlining, the fuel consumption is increased and this can more than double the cost of the measure.

Recent investigations comparing average repair costs on the 1972 and 1973 models of otherwise identical vehicles in the U.S. with and without the improved bumpers have shown that, despite the bumper standards, the average repair costs have decreased only $1. The purchaser, however, has to pay $65 to $100 more for the new bumper systems.

The assessment of nonmaterial values is also difficult. It is, for example, hard to find a monetary value for the inconvenience of putting on seat belts or for the loss of liberty, namely the freedom to wear the belt or not, if wearing of seat belts becomes mandatory. Another difficulty is the assessment of the proportion of life time which, for example, vehicle occupants spend on uninteresting roads as a result of speed restrictions. Factors which should introduce an objective assessment in this problem are known from Peschel, Funck, Juergensen, and Altrup. In such papers an optimal driving speed is given which adjusts itself in trouble-free traffic conditions so that the sum of the time value and the operating costs is a minimum. Numerical connections between time values and this optimal speed have been put together by Niklas (Figure 1).

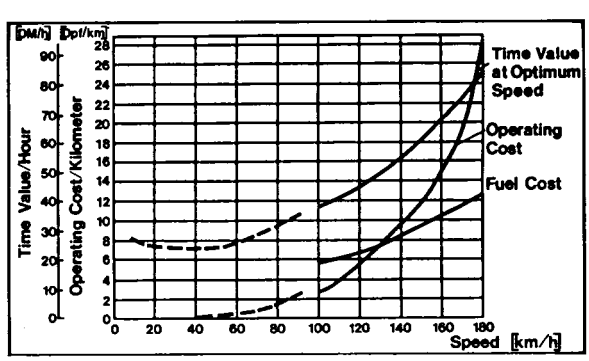


Figure 1

The money assessment of the benefit is always difficult because the major benefit, namely the saving of human life and the reduction of suffering and pain, is not accessible for a direct monetary assessment. In general, only the material public loss as the result of injury or death is quoted, for example, by Niklas (Figure 2).

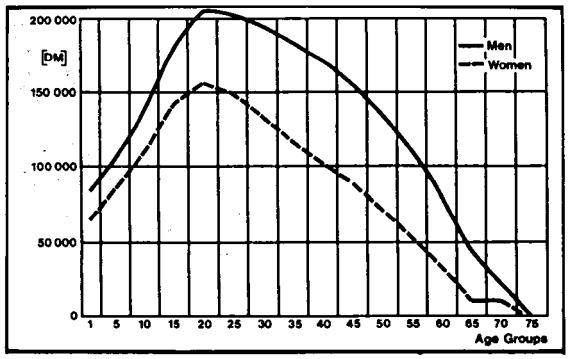


Figure 2

In a report of societal costs of motor vehicle accidents issued by the U.S. authorities in April, 1972, the figure given for the public loss incurred by a traffic death was $200,700 and $7,300 for an injured person.*

Niklas describes the monetary evaluated benefit in the form of a "general benefit/cost factor." In order to make possible a differentiated observation parallel to the pure monetary assessment, he introduces an additional "special benefit/cost factor." This encompasses all costs necessary to protect a person from death or from serious or minor injuries.

Another way of getting over the problem of giving the benefit itself a monetary assessment is the closely related "cost-effectiveness" method. It is, for example, difficult to evaluate the damage caused by oxides of nitrogen in the air. The cost of the measures on cars, power stations, and household fires required to reduce the oxides of nitrogen concentration can, however, be compared. I should just like to mention this "cost-effectiveness method" here as an example for a possible expansion of the cost/benefit method.

The benefit/cost function normally is a degressive curve (Figure 3). This is because at first small expenditures bring a noticeable benefit whereas,

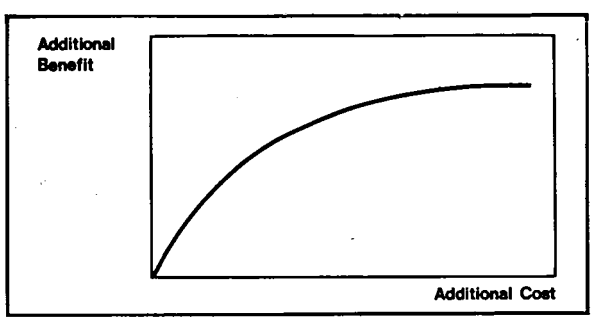


Figure 3

*Source: Societal Costs of Motor Vehicle Accidents, Preliminary Report, April, 1972, U.S. Department of Transportation, NHTSA.

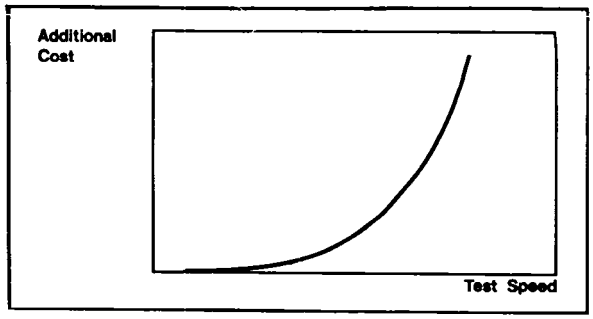


Figure 4

finally, even a very high expenditure does not help (Figure 4). This problem is explained in more detail with the example of the measures described as the "survivable test speed." The injury criteria used are accelerations, or values extracted from them, measured on dummies in the test vehicles during the crash tests. If one increases the test speed continuously with fixed injury criteria, the costs, due to the more elaborate construction and the higher fuel consumption caused by the greater weight, increase more than in linear proportion to the test speed (the energy to be absorbed by the structural deformation determines the constructional outlay and is proportional to the square of the speed).

In contrast to this, the benefit increase is less than proportional because, as experience has shown, accidents corresponding to higher test speeds are becoming rarer. Consequently, the ratio of benefit to cost is a degressive curve (Figure 5).

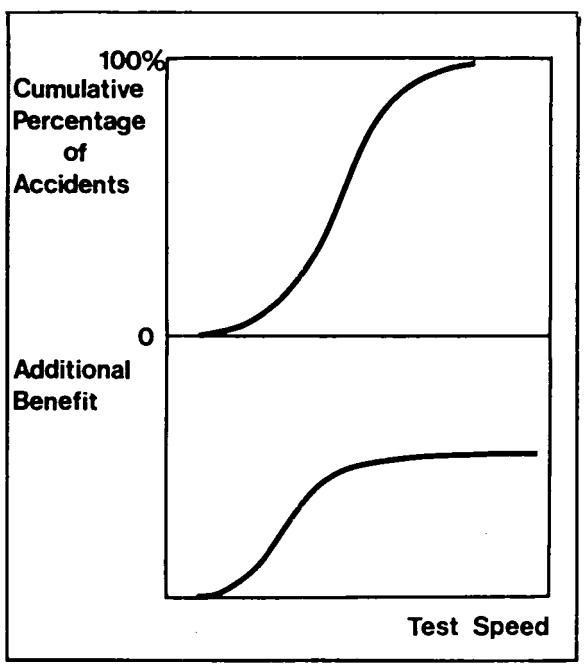


Figure 5

As an example, the benefit/cost curves for three different measures are now examined, viz., the occupant protection in frontal vehicle impact, in rear end collisions, and in side impacts (Figure 6). The

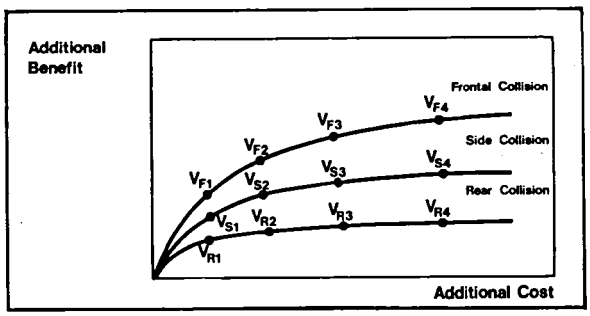


Figure 6

benefit is relatively high in the case of the frontal impact since this type of accident has a high frequency. The curve is lower for the side-on collision because the expenditure for strengthening the vehicle sides is relatively great and because these collisions are less frequent. If the maximum possible benefit is to be achieved for a certain outlay, this will obviously only be feasible provided the cost for the individual measures are so related to each other that the benefit/cost function for all three measures has the same gradient (Figure 7). If, therefore, an exception

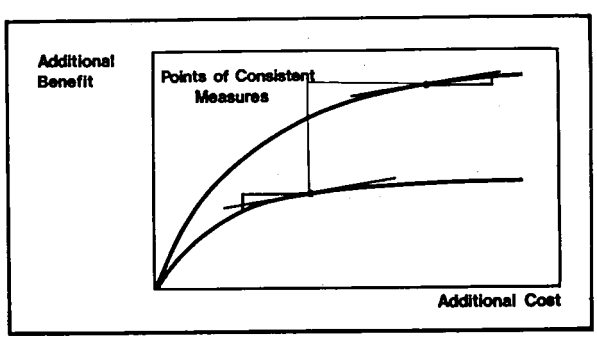


Figure 7

were to be made for one of the measures and the cost of this increased by a certain amount, then the increase in benefit would be less than the loss on another measure on which the same cost had been cut. The rule which leads to "consistent measures" may thus be quoted: "The demands on the various measures must be so selected that the optimum benefit is obtained when the gradient of the benefit/cost function has the same value for all measures" (Figure 8.)

The consistent measures rule is generally applicable and is not limited to the speeds of frontal, lateral, and rear impacts but also applies to such

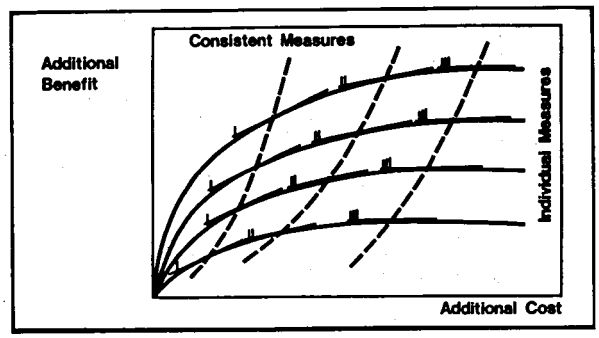


Figure 8

diverse measures as speed limits, installation of a fire extinquisher, elimination of dangerous intersections, or rescue of traffic casualties by helicopter.

An additional difficulty in the optimization process lies in the interdependence of the various measures. Thus, for example, if one measure lowers the number of accidents, the benefit of another measure which lessens the consequences of the accident is thereby reduced.

The harmonizing of many possible measures in the various fields of activity to increase traffic safety is the essential product of the benefit/cost analysis. In the calculation of the benefit and cost of the range of measures, it is seen that the benefit/cost ratio differs widely (Figure 9). Therefore, the absolute precision

Safety Measure	Benefit/ Cost Ratio	Reduction in Number of Fatalities p.a	Serious Injuries p.a	Minor Injuries p.a	Decrease in Property Damage in Million DM p.a
Helicopter Rescue	3.4	400	0	0	0
4 Safety Belts/Vehicle Belt Wearing Quota 100%	1.5	3 007	34 133		0
4 Safety Belts/Vehicle Belt Wearing Quota 70%	1.1	2 735	23 893	66 422	0
2 Safety Belts/Vehicle Belt Wearing Quota 30%	0.7	879	7 679		0
Driving School	0.7	3 151	26 545	60 145	329.7
4 Headrests/Vehicle	0.04	50	600	2 400	0

Figure 9

of estimating benefit and cost is not so important as simultaneous consideration of all possible measures for car, road, and driver as consistent ones. Only in this way can it be ensured that the greatest possible benefit is derived from the cost incurred and that the maximum number of persons is protected from injury or death (Figure 10).

On the other hand, the benefit/cost analysis cannot produce any objective statement about the "right" amount of cost to be expended. The decision as to the extent of the outlay for increases in traffic safety remains a political one. It is, however, possible to utilize benefit/cost analysis in other spheres for

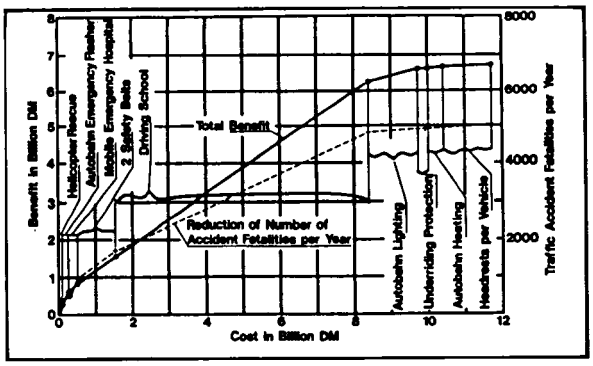


Figure 10

objective descriptions such as the reduction of accidents in the home or in public places, the erection of hospitals, or measures against environmental pollution and to arrange these in an order which can be compared with the measures for increasing traffic safety. The consistent measures rate is, after all, applicable to all sectors.

The following consideration can provide an aid in decisions on the extent of cost expenditure (Figure 11). The benefit/cost curve may be regarded as reflecting the sum total of all consistent measures. It will also be degressive. The straight line AC of cost equal benefit has been drawn in. It is now possible to discuss the points A, B, and C. A means there is nothing to do; there is no cost, but no benefit. As the cost rises, the benefit increases even more rapidly up to point B. After this point the benefit increases more slowly than the cost, even after point C, where the benefit and cost are equal.

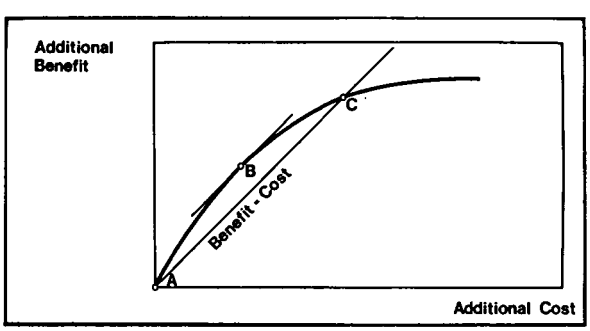


Figure 11

It must be pointed out, however, that this is a consideration of political economy. On the other hand, with regard to many measures, cost is more likely to occur in the politically relevant economic concerns of individuals. It must again be remembered that in the benefit only the monetarily assessable portion of the damage reduction is considered. The truly humanitarian part of the curve is that which starts to the right of C. It is only there that the material expenditure to reduce deaths, pain, and suffering becomes greater than the material benefit.

SUMMARY

Based on the expectations of damage estimated on experience, the benefit/cost analysis offers the possibility of finding the maximum benefit for the planned outlay of cost, that is to say, the maximum possible reduction of damage for a given cost. Variations in the anticipated damage (i.e., through the energy crisis) influence the benefit/cost analysis, just as material shortages resulting in price increases influence the cost aspect.

Even so, in the sphere of changing peripheral conditions, the benefit/cost analysis is also the best aid presently known for decisions on road safety measures. By employing this analysis systematically, it is possible in the long term to achieve continuous progressive improvement on a rational basis where today decisions are often emotionally colored.

COST-EFFECTIVENESS STUDY

N. MUHLRAD
Organisme National de la Securité Routiere
ONSER

INTRODUCTION

Cost-effectiveness studies are designed to facilitate the choice of the most effective measures within the scope of limited budgets. As concerns road safety, the measures are not all financially interchangeable inside a given package, as the costs thereof may, in fact, be borne by different economic entities (Government, local communities, users, automobile manufacturers, etc.), even though, in the final analysis, such costs are always passed on to the citizen-consumer. Despite this incomplete interchangeability, the compatibility of the effectiveness of measures financed by distinct sources should be ensured. It is in fact difficult for the Government to impose on other entities, in the name of road safety, charges which it is itself unwilling to assume.

The first road-safety cost-effectiveness studies were carried out in France and were based on road safety installations. Accordingly, road-safety cost-effectiveness studies are customarily carried out according to the calculation methods developed in respect to such installations. The safety benefits are calculated in 1974 on the basis of monetary equivalents fixing a dead person's earnings at 320,000 francs.

320,000 francs is thus the amount which the Government is in theory ready to spend to avoid the future road accident death of a statistically unknown person.

This value was arrived at on the basis of detailed socio-economic assumptions and is therefore relative to a certain state of the economy and to a certain philosophical concept. It is, however, noted that the values adopted in the U.K. and certain American states are very close to the French value.

In relation to the theoretical bases of the cost-effectiveness theory, we discuss the pertinence of the use thereof according to whether the intrinsic suitability of a measure is assessed or whether a choice is made among several measures.

COST-EFFECTIVENESS STUDIES

Everyone seems to agree on the need for a cost-effectiveness study of the various safety devices under consideration, in order to grade them and thus select the most expedient and, also, the most suitable time to place them in use.

Thus formulated, the problem seems simple, but in reality it raises numerous difficulties, which may be practical in nature (gathering of data, etc.) as well as theoretical and even doctrinal (the value of life, economic assumptions, etc.). The solutions proposed are all imperfect and all call for additional research. It can, nevertheless, be considered that cost-effectiveness studies based on actual results are useful decisional aids in providing rational criteria for a principled grading of safety devices.

In point of fact, it is noted in practice that most of the cost-effectiveness studies now being made in the vehicle-safety field serve, not for the establishment of an order of priority among the measures under consideration, but to bolster a decision to be made, or to assess (before the fact) the effectiveness of a safety device without getting into alternative choices.

This *modus operandi* is unsatisfactory insofar as it is based on a definition of benefits (expressed in monetary units) established on the basis of a purely comparative use and the compatibility of which with the costs – properly so-called – of the measures has not been verified.

We will here seek to demonstrate more precisely the various aspects of the question.

PRACTICAL PROBLEMS RAISED BY COST-EFFECTIVENESS STUDIES

We refer here to studies which are actually to be used for principled grading of safety measures. The procedure followed in such a study can be broken down into three phases: (1) establishment of the costs (properly so-called); (2) assessment of the effectiveness; and (3) appraisal of the total effect in order to obtain the benefit.

Costs

The basic problem at this level is to demonstrate the portion of the investment which must be assigned to safety. The solution may be relatively simple; i.e., in the case of a safety device to be attached to a standard model vehicle, the cost of the device per vehicle is marginal. But the problem may be more difficult; i.e., when it involves a safety device which cannot be incorporated in an existing model but requires a new vehicle design (or for example, in another sphere, when it is a question of building a new road meeting certain safety criteria). It is then impossible to assign the entirety of the investment made to safety; and the proportion thereof can be estimated only with a certain degree of arbitrariness.

It also becomes awkward to estimate such proportion before the fact and hence to quantify in advance the cost of the safety measure in question.

Other problems may arise in connection with the appraisal, necessary for the calculation of an overall cost of the sphere of application of the measure. This can, for example, be expressed (if safety devices to be incorporated in vehicles are involved) as the total number of cars to be equipped or as the number of cars to be equipped per year, etc. (or in the case of road equipment, as the number of miles to be processed in a year or overall, etc.). In principle, this sphere should be simple to delimit, depending on the way in which the safety measure is formulated; but such formulation may not be precise, or definitive (and may even be a factor to be tested by the cost-effectiveness studies) when before-the-fact work is involved. There, accordingly, remains a choice to be made at this point; it is noted that it thus entails the way in which the total effectiveness of the safety measure will be calculated and that the simplest choice from the cost standpoint may not be the best.

Effectiveness

The procedure usually followed to assess the effectiveness of a safety device or measure consists of:

- The selection of a period. This is either simply determined by the formulation of the measure itself, or established consistently with the sphere of application chosen for the calculation of costs.
- A survey of the accident situation to which the measure relates and an appraisal of its scope, in number of deaths and injuries, in the geographical area in which the measure will be applied.
- An estimate of the "percentage of success" of the measure, i.e., the reduction in deaths and injuries which application of the measure is predicated as achieving in the corresponding accident situation and in a given period of time.
- An assessment of the total effect in relation to the reference period selected, the "percentage of success" previously established, and the scope of the relevant accident situation.

The problems which arise at this level are manifold and are basically data-gathering problems. The official accident statistics may suffice for appraisal of the accident situation to which a measure relates, but in many cases these may prove to be too general, necessitating the results of more detailed investigations made on the basis of samplings assumed to be representative (or adequately adjusted).

Similarly, in the estimate of a measure's percentage of success, it is necessary to limit the degree of arbitrariness so far as possible, and to do this, adequate experimental data or accurate accident investigation reports are required. Such data or reports are obviously obtainable only if a device – the use of which would be required by the measure under consideration – already exists in suitable number in the field before the measure is enforced. Otherwise, the formulation of the experiments to be used and the interpretation of the results may prove to be extremely complicated.

Finally, there must be taken into account in such percentage of success, any prejudicial effects of the measure to be evaluated. It is usually quite difficult to identify these and even more difficult to quantify them: such effects being, for the most part, slight in scope, it is difficult to grasp them statistically. Nevertheless, they cannot always be disregarded.

We have considered here only the principal problems involved in before-the-fact evaluation of safety measures. The question can obviously be simplified when, in particular, one of the measures to be considered has already been placed in effect elsewhere and when its effectiveness can be appraised after, rather than before, the fact, on the basis, for example, of reductions in the number of victims as may appear from the accident statistics. It is thus seen that the advantage of examining a given measure after the fact is far from being inconsiderable, but it is then necessary to make sure that the improvements observed indeed result from such a measure and not from other, concurrent factors.

Benefits

The total effectiveness is quantified in the number of deaths and injuries avoided, due to the measure in question. The benefit is calculated in such terms as to be comparable with the costs already established. It is accordingly necessary to assign a price to life by giving a certain value to persons killed and injured.

Such value has been established on the basis of assumptions which vary from one country to the next. But in all cases the value is an average one – which is warranted in the case of a death, since the victim is a statistical person, but can raise problems in the case of an injury. While certain measures (incorporation in cars of an improved brake system) permit avoidance of injuries sufficiently varied for the "statistical injured person" to be representative thereof, others (restraining devices for example) may contrarily have an impact on a certain type of injury, requiring the calculation of a specific injured-person cost, probably quite different from that of the average injured person. In this connection a very clear inadequacy of all of the cost-effectiveness studies now being undertaken is noted.

The other problems connected with calculation of benefits are much more theoretical in nature and basically relate to fixing a value to life.

THEORETICAL PROBLEMS RAISED BY COST-EFFECTIVENESS STUDIES

We continue to refer to the case in which the cost-effectiveness studies are designed to be used in connection with establishment of an order of priority of the measures under consideration.

The basic problem here is valuation of life and it arises at several levels: the point of view according to which such value is fixed, the various factors to be taken into account in the calculation, and the a priori socio-economic assumptions made.

Overall Point of View

As concerns road accidents in France, the value of a dead person has been considered to be "the amount which the community is agreeable to spend to save one of its members." The value which a human being may put on his own life has thus been disregarded. Estimates of the value of life have been made in spheres other than that of road safety (air transport, medical field, industrial safety, etc.) and have frequently given very different results. The theoretical grounds for these variations cannot always be demonstrated. It does, however, appear that the value of a life lost in a road accident is one of the lowest which have been calculated, because it appears as a "probable statistical death" (as to which, assignment of liabilities is quite diffuse), whereas in other cases, the death can be "certain" (case of a life saved at sea for example), as to which the responsibility is more directly incumbent on the community (air transport).

It is not clear that these differences are justified and this point must be clarified. It could, for example, be advantageous to study the factor of "risk assumed by the individual," to determine its influence on the figures now used, and to establish the degree to which it should actually be taken into account in placing a value on life.

Factors in Estimating the Cost of Road Death

The factors which have been generally considered in fixing the value assigned by society to a person killed in a road accident are: loss of the victim's future gross production, loss of his future consumption, the actual (medical or administrative) costs, and the so-called "affective" costs (pain and suffering, disfigurement in the case of a serious injury, emotional costs sustained by the family in the case of a death).

In France, out of a concern for objectivity and for simplification, the factor adopted is that of loss of the victim's future production. This selection calls for certain comments:

- Medical costs can be disregarded because they can be considered negligible in relation to the loss of future production. The same thing is not true in the case of an injury in which the medical expenses may be substantial. If injuries are classified as "serious" or "slight," a slight injury (one without permanent disability) costs the community nothing.
- No account is taken of "effective" costs because of the arbitrary way in which damages for pain and suffering and for injury sustained by the family are assessed in France. For practical considerations of estimation of actual costs, account is taken only of an individual's yield factors and none of a life-value factor which may be all the more substantial when the economic situation is more stable and the "well-being" of individuals is therefore more broadly taken into account by the community.
- Finally, the victim's future consumption is not subtracted from the loss of gross production. This is justified on the ground that production can currently be considered as the most important index of community life and that it is difficult to isolate a marginal loss of consumption having practically no impact on a corresponding loss of production. Moreover, taking into account the victim's future consumption would result in negative costs in respect to old or retired persons, which would depreciate accordingly the average value of the dead or injured person, and which well demonstrates that the amount which the community is agreeable to paying to save a victim cannot be expressed in the victim's net loss of production.

It seems, finally, that there has been a tendency, because of the factors of calculation selected in France, to slightly underestimate the value of road victims.

Socio-Economic Assumptions

Calculation of the loss of gross future production of an average road victim in France involves the following steps:

1. For this purpose account is taken of the individual's age, his sex, his socio-occupational category and the average income of such category in relation to the individual's share of the net national product.
2. Calculation of the loss of an individual's cumulative future production, taking into account

his age, sex, and socio-occupational category. For this purpose it is necessary to make assumptions as to the discount rate to be used, the growth of the net national product, the mortality tables, the rates of activity according to the socio-occupational class, and the probabilities of actual employment. It is also necessary to assign to unemployed young persons a future socio-economic category, while retaining the same overall distribution with reference to the population of road victims.

3. Calculation of the average cost of death on the basis of the structure by age, sex, and socio-occupational category of road victims.
4. Calculation of the average cost of injury. For this purpose the cost of the average serious injury is calculated on the basis of the cost of death previously obtained and of an average percentage of partial permanent disability, with slight injuries involving no loss of production (or rather a loss of production deemed to be negligible), there is finally obtained the cost of the average injury on the basis of division of road injuries between these two classes.

The figures obtained by this method which are currently used are 320,000 francs for a death and 14,000 francs for an injury. They represent a purely monetary revaluation, as of 1973, of values established some fifteen years ago.*

The assumptions here call for a few comments:

- In considering that an individual's production is proportional to the average wage of his socio-occupational category, it is tacitly assumed that the amount of money that the commumity is disposed to spend to save an individual's life depends on the amount of his wages. It is thus considered that wage and socio-occupational class are valid indices of an individual's "usefulness" to the community. This point seems debatable. This assumption also raises a problem concerning certain categories of unemployed persons such as children (retirees, etc.). Is the community ready to spend nothing to save them? The difficulty was finally gotten round by taking into account only the average cost of the person killed, whatever his socio-economic characteristics.
- The assumptions as to rates of activity and probabilities of employment can only be highly arbitrary insofar as gross production is cumulated over some sixty years. It thus seems unnecessary to refine the distributions of socio-occupational categories and of wages. It is in fact found that *the values of the cost of death calculated by this method are very sensitive to variations in the selected rates of growth and discount.*** This is not serious insofar as these results are used only to establish cost-effectiveness ratios of various measures for comparison *inter se*. The same thing does not follow if the intrinsic worth of a measure is under consideration.

In view of the questions raised by the values thus obtained, a more recent study has been undertaken at ONSER*** with a view to establishing new costs of death and injury. The same procedure has been followed, but a certain loss of production has been assigned to all unemployed persons (estimating the loss of future production as being equal to that of low-income persons in the same category). Medical and administrative expenses and more recent distributions of socio-occupational categories have been used, and, finally, a component representing the property damage caused by the accident has been incorporated in the cost of death and injury. The resulting figures are higher than those officially used: 390,000 francs for a death, 63,000 francs for a serious injury and 2,000 francs for a slight injury. The official figures again seem slightly underestimated. In view of the use made of these appraisals, the ratio of the cost of death to the cost of injury is basically used, not the absolute values of these two "costs to the community."

COST-EFFECTIVENESS STUDIES IN REALITY

Despite all the failings of the values used and all the practical problems encountered in their use, it is helpful to classify the various safety measures under consideration in order of priority and thus to underpin the decisions to resort to the comparative results of cost-effectiveness studies. This permits a measure of circumvention of arbitrariness. But here one is either in a situation in which an aggregate financial package is posited which cannot be exceeded and which must be disbursed as judiciously as possible, or else in a situation in which all of the measures proposed will be applied and it is only a matter of establishing which are the most urgent.

If one is in neither of those situations, the grading of the measures implies the elimination of some which are "too unprofitable," in absolute terms or relative to the others. Adoption or rejection of a measure may be proposed in such case by comparing the cost-benefit ratio of such measure with the ratio of the less "profitable" measures already adopted. Consistency in decisions is thus ensured.

*See appended Bibliography.

**The sensitivity to variations in these rates is given in Appendix 1 per varying values of the rates of growth in national production and of discount.

***See Bibliography in Appendix 2.

These cases are the most frequent in actual studies; the reasons are various. First, consideration is generally given (at the level of formulation of national policies) to a well-defined safety measure and that a choice among several different measures is rarely presented a consequence, in particular, of the lead time required for the preparatory studies adequate to justify them. Second, it is difficult, in actuality, to refer to a specific financial package – such may tacitly exist but the corresponding costs are in fact incumbent on various entities (the Government, the automobile manufacturers, ordinary persons, etc.).

The conclusion is that, in the present state of affairs, the currently used cost-benefit concept is partly unsuited to the needs and that it is necessary, in the vehicle-safety sphere, to seek other appraisal procedures (for example, application of the multi-criteria analysis method.).

Future Research

In future, it seems advisable:

1. to develop, on the basis of the values of the present cost of human life, comparative studies of the various possible safety measures;
2. to go more thoroughly into the definitions given and the assumptions made and to revise them on the basis of the use to be made thereof, in keeping with the work in fields other than that of road safety;
3. insofar as it seems advisable to justify a particular measure on the basis of its cost-effectiveness ratio, to check the consistency of the concepts used and the validity of the results.

APPENDIX 1

SOCIAL COSTS OF DEATHS AND BODILY INJURIES

The results obtained (in 1968 francs and with reference to 1968 data) depend very appreciably on the discount rate (DR) and on the rate of growth in national production (RGNP) selected for purposes of making the estimates.*

RGNP %	DR %	Death	Serious Injury	Slight Injury	"Average" Injury	Property Damage Per Accident
2	6	286,300	46,000	1,660	16,400	5,400
2	8	236,400	38,400	1,660	13,900	4,500
4	6	375,000	59,400	1,660	20,900	7,000
4	8	288,800	46,400	1,660	16,600	5,400
4	10	238,500	38,700	1,660	14,000	4,600
6	8	377,100	59,800	1,660	21,000	7,000
6	10	291,300	46,800	1,660	16,700	5,500

The listed values are average values for the victims in relation to age, sex and socio-occupational categories.

APPENDIX 2

BIBLIOGRAPHY

1. A. Levy and S. Verdier, "Analysis of the Social Costs of Accidents Involving Deaths and Bodily Injuries with Reference to Road Safety." ONSER, 1973.
2. C. Abraham, and J. Thedie, "The Value of a Human Life in Economic Decisions." *Revue Française de Recherche Opérationnelle*, 1960.
3. S. Ch. Kolm, "On Public Safety." 1969.
4. G. Morlat, "A Model for Certain Medical Decisions." Research Conference on the Economics of Uncertainity.

*The values currently used for purposes of the VIth Plan are DR = 10% and RGNP = 5.8%. All of the figures in the foregoing table are expressed in francs, except as otherwise noted.

STUDY ON COST-EFFICIENCY RATIO BASED ON "BASIC RESEARCH VEHICLE"

P. VENTRE
Head of Crashworthiness Department
The Renault State-Owned Works

To make a choice between the different technical solutions that are possible, the cost-efficiency ratio must be calculated.

Evaluating the price of various solutions poses no serious problem; but it can be difficult to evaluate efficiency, especially if one wants to avoid rough and subjective estimates.

The method adopted to evaluate efficiency is as follows:

1. Study of French accident survey as regards the parameters that are decisive for occupant protection: type and mass of vehicle; type of obstacle; impact localization of vehicle. This work has been carried out using statistics compiled by the French Office National de la Securite Routiere (ONSER). Selected were eight vehicle types classified by mass and engine position; four types of obstacles, passenger cars, vans and trucks, fixed and rigid obstacles, and other obstacles; four collision areas, front, side, rear, and rollover.
2. Starting from those data, samples were gathered of accidents taken from the Peugeot-Renault Accident Survey. These samples accurately represent the situation described by the statistics; all parameters of the accidents in the survey are precisely known. The first list of samples comprises 500 vehicles.
3. Each sample accident was studied on a case-by-case basis to evaluate the potential result of each of the possible safety devices. This analysis was carried out by a multidisciplinary staff, including specialists on crashworthiness, restraint devices, accident surveys, and biomechanics.

Each individual accident being described in full detail (thanks to numerous documents), the staff is able to analyze the influence that would have been exerted by various envisaged protection devices. Those devices are: (1) For frontal impact: safety belts either conventional or more or less sophisticated, structure reinforcements, various interior fittings, steering columns, passive restraints (such as air bags); (2) For side impact: door reinforcement, structure reinforcement, interior panel fittings; (3) Rear collision: head restraints and structure reinforcement; (4) Rollover: all the above items, plus all devices meant to prevent doors opening.

SYNTHESIS OF RESULTS

Starting from these evaluations, one can figure out the effects that would be obtained by each device, either alone or combined with others. When analyzing efficiency, account must be taken of a coefficient representing the probability of the device being operational (e.g., rate of correctly worn safety belts). Given the fact that belt wearing is compulsory in France, a coefficient was taken of 0.5 for plain 3-point belts and 0.75 for automatic inertia reel belts.

Cost is expressed as retail price – including taxes as paid by the customer – and it was assumed that there is a car population replacement at an annual rate of 10%. Efficiency is expressed by the total number of deaths avoided and of severe and light injuries avoided.

CONCLUSION

In our opinion there is no overall protection device having the same efficiency as those incorporated in the BRV with a cost-efficiency ratio as low as the 3-point belt. When one compares vehicles fitted with inertia reel belts to those fitted with conventional 3-point belts, one finds an acceptable cost-efficiency ratio for the former, since cost increase is not prohibitive and usage coefficient is higher while proper belt adjustment is more frequent.

Solutions retained for the Renault BRV give a maximum saving of casualties, but they already reach the level where efficiency tends to taper off sharply; i.e., cost tends to increase much faster than efficiency (Figure 1). Air bag efficiency was not listed since it falls outside the scope of this investigation.

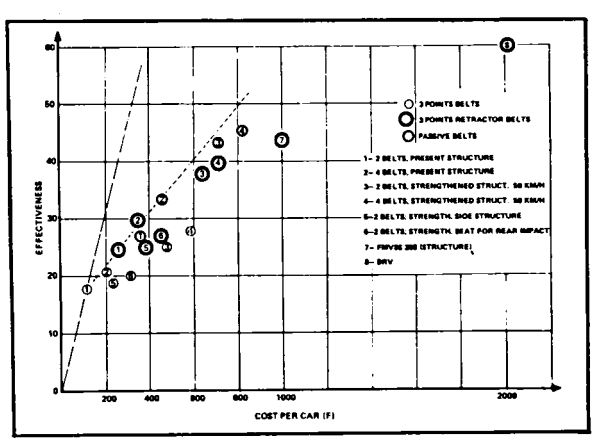


Figure 1

AN OVERVIEW OF TRRL PROGRAMS

DR. JOHN FITCHIE
Head of Department of Transport Systems
TRRL

I have been asked to mention, briefly, some of the programs we are carrying out at TRRL outside the safety field, but which may be of some interest to you in shaping your plans for the future.

The topics I have selected are mainly in the environmental field and the energy resources area. The requirements in these two areas will often be in conflict and are also likely to conflict with safety considerations and the overall wish to minimize the use of our resources.

I do not have the time this morning to go into detail on any of our projects, but if any delegate would like more information, will they please write to us at Crowthorne.

In deciding the balance of our program we have to take into account the conflict that arises in the centers of our most congested cities between transport requirements and environmental considerations – and within the transport requirements – the competition between private cars and other road users. We have also to take into account the likelihood that even by the turn of this century the majority of the population will not have access to a private car.

You will not be surprised, therefore, when I tell you that programs aimed at improving public transport are high on our list of priorities. We are involved in all forms of such transport and not just those which are road based.

We continue to devote an appreciable effort at TRRL to collecting data, much of it by taking measurements in the field and by setting up experiments – for example, to monitor changes arising from changes in transport operations and land use.

We have recently started to measure energy consumption patterns for certain road vehicles, but we already have a sizeable bank of data on environmental effects.

A program has also been running for some time whose aim is to place economic values on social and environmental factors. One of the aims of the work I have just mentioned is to replace subjective and often emotional views by hard factors. A second aim is to use the data to support the mathematical models we build to predict future situations. One model, for example, will predict changes in environmental factors, while another can be used to examine the differing effects from a range of investment policies.

Another aim of our predictive modelling work is to show whether improvements in transport can be brought about best by operational changes, by technological developments, or some combination of both. For example, on the basis of optimizing consumer surplus, one can choose between various management schemes for reducing traffic congestion. Such schemes and local improvements to traffic flow at road junctions would produce reductions in air pollutants but no noticeable reductions in noise levels.

This leads me on to our program on quiet vehicles. Our first project is aimed at producing two demonstration vehicles – these are both heavy articulated trucks. This is a collaborative program involving the Southampton University Institute of Sound and Vibration, the Motor Industry Research Association, British Leyland and Fodens – the latter are employing Rolls-Royce as their engine subcontractor. Virtually every aspect of the truck design is involved in the program, since the basic aim is to determine to what degree a vehicle can be made quieter – while at the same time remaining commercially viable to the manufacturer and operator and meeting other requirements on the basis that these results could also be used to influence decisions on future noise level regulations.

A difficult area is truck tire noise above, say, 40 mph, particularly in the wet where there is a real conflict between safety and noise level. We are grateful for the advice we are getting from the tire manufacturers, but we are also considering the effect of the road surface. Some years ago, TRRL developed a porous road surface, which not only considerably reduces splash, but leads to a marked reduction in wet road tire noise. A field trial on a road near Birmingham is progressing well and this may be a good cost-effective solution, where relatively high speed traffic passes close to residential areas.

As for energy considerations, we completed the first phase of our study of the energy problem just over two years ago; this was concerned with estimating fossil fuel reserves, predicting future energy requirements (which of course are closely linked with predictions for Gross Domestic Product), and the economics of synthesizing petroleum type fuels. Other programs have looked at energy storage systems, including various forms of electrical batteries, and the total system economics (in resource terms) of novel systems and possible developments in more conventional transport systems. We have also done a little work on alternative energy sources.

Currently, we are examining investments that would be required for some of these proposals.

A FEASIBILITY STUDY ON SMALL SAFETY VEHICLES

YOSHIO SERIZAWA
Nissan Motor Co. Ltd.

1. A REFLECTION OF ESV DEVELOPMENT

As brought forth at the Fourth International ESV Conference held in Kyoto in March of last year, Nissan Motor Company, at an expense of nearly three years and approximately two billion yen, succeeded in developing a small, lightweight ESV for four passengers.

The specifications of the Nissan ESV were essentially the same as those of the U.S. ESV regarding fulfillment of the requirements in 80 km/h frontal barrier collisions, etc., except that some minor changes were made; specification requirements for 80 km/h rear-end collisions, for example, were modified to 65 km/h as a result of a close review of accident analysis and other related data.

Experimentally, our ESV has satisfied all ESV specifications and requirements; when examined as a practical vehicle, however, it leaves many questions unanswered and many problems unsolved, as was also brought forth at the International Conference in Kyoto.

1-1. Cost Evaluation of Occupant Crash Protection

In order to make an extensive study of occupant crash protection in high-speed collisions, we tried different combinations of vehicle construction characteristics and occupant protection systems, and, as a result, developed the following two ESV prototypes:

E_1 Equipped with a monocoque body plus an air bag system

E_2 Equipped with a broach-construction body plus a seat belt system

The E_1 had a combination of the current conceptual body construction and a rapid-deployable type air bag; its target was occupant protection in 80 km/h collisions. The E_2 was targeted to deliver the same safety performance as the E_1 by means of a seat belt system and by its capacity to achieve a flatter vehicle deceleration curve.

Of course, to attain 80 km/h collision safety, we had to take into consideration many interwoven factors as shown in Fig. 1.

To improve overall reliability of all safety factors in 80 km/h collisions, the reliability of each individual factor must be improved. To our regret, however, we found that the statistical probability of all ESV functions operating as specified was not as great as expected. The results of many tests carried out indicated that the probability of the injury levels of the head, chest and femur simultaneously meeting the ESV specification requirements was about 25% for the E_1 and about 15% for the E_2.

However, the ESV injury criteria are considered to be too severe when compared with the levels of injuries sustained in actual collisions. Moreover, due to malfunctions and poor repeatability of dummies used in the tests, the foregoing probability rates turned out, it seems, to be unreasonably low. Particularly, in the E_2 test the chin-chest contact of an Alderson dummy caused a high g value.

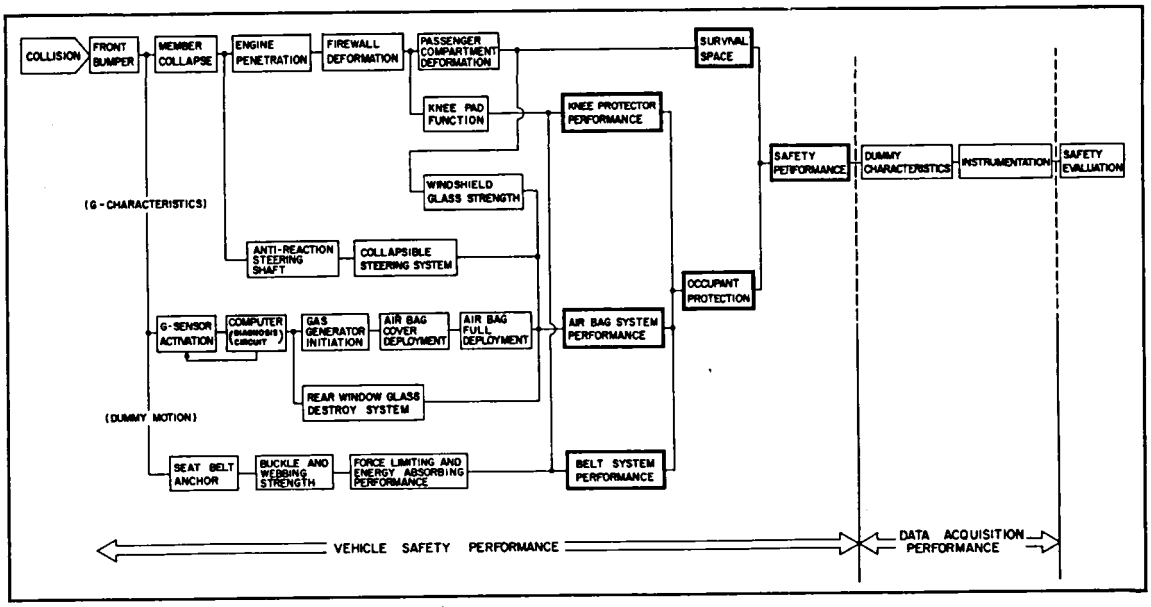


Figure 1. Occupant Crash Protection System — Driver at 80 km/h Frontal Collision

The cost of the E_1 was approximately 80% greater than that of the Datsun 610. This cost increase analyzed by performance requirements is as indicated in Fig. 2. Of the 80% cost increase, 43% was attributed to improvement in occupant protection and vehicle crashworthiness, and the remaining 37% was for accident avoidance.

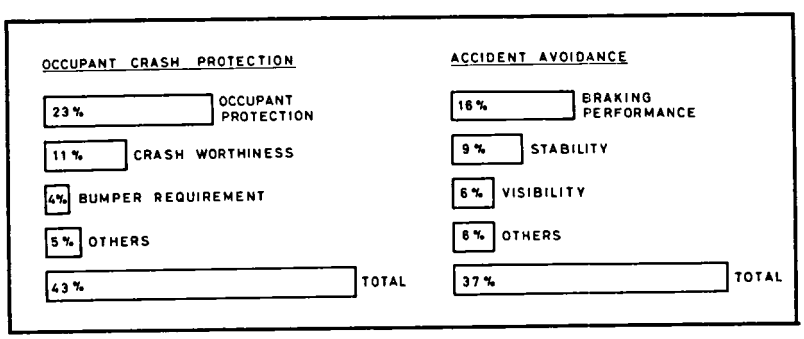


Figure 2. Cost Increments of the Nissan ESV

A detailed breakdown of the 43% occupant protection and vehicle crashworthiness cost increase is as follows: 23% to occupant protection, and 11% to vehicle crashworthiness. This indicates that to secure occupant survival space alone, an 11% cost increase will be sufficient, but that to ensure occupant safety, a cost increase of 23% plus 11%, or a total of 34%, will be necessary. As stated above, however, the degree to which the safety ensured for dummies coincides with protection for human beings is uncertain in many respects. It also remains unknown whether this 34% cost increase will contribute to safety in actual collisions.

1-2. Cost Evaluation of Accident Avoidance

Of the 37% cost increase for accident avoidance shown in Fig. 2, a considerable portion (16%) is allocated to braking performance. This is mainly for installing an anti-lock device on the rear wheels. The evaluation of the anti-lock's effect on safety is of great significance.

For improvement of steering and handling performance, a 9% cost increase was required. This is chiefly for improvement in tires. The cost increase in this respect will be reduced if the run flat performance can be lowered.

On the other hand, difficulty in meeting the ESV specification requirements will most likely vary according to vehicle size, and the relationship between ESV requirements and accident avoidance is itself questionable in some respects. Further, the ESV specification requirements for the most part regulate the characteristics of vehicles alone, and it remains doubtful whether this is effective at all when the driver's feedback is added. This should be clarified in the future as a question of a man-machine system.

1-3. Vehicle Crashworthiness in Collisions at Lower Speeds and Aggressiveness

In order to increase energy absorbability in high-speed impacts, the front end of our ESV has been considerably strengthened in comparison with that of small cars in current use. This means that in barrier collisions at low or medium speeds, the ESV receives a greater impact than the other vehicle and that in car-to-car collisions, the other vehicle suffers greater damage.

Collisions between vehicles of different sizes will be discussed in detail later.

1-4. Remaining Unsolved Problems

The weight increments of the ESVs developed until this time in various countries are indicated in Fig. 3. The increments shown are by weight difference between the ESV and the basic vehicle, or a car which has the same-sized passenger compartment and the same engine displacement.

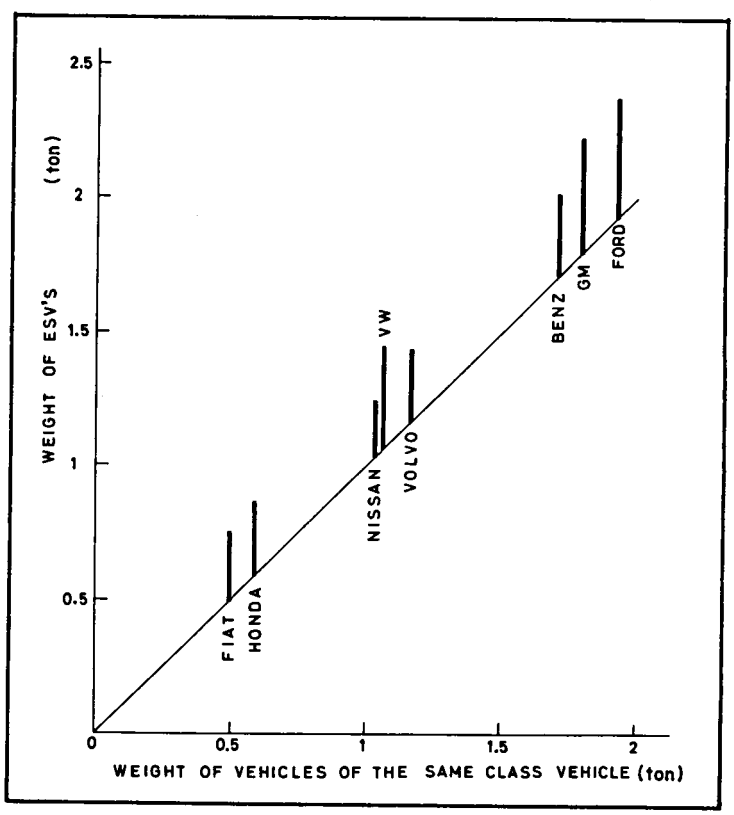


Figure 3. Weight Increment of ESVs

In comparison with the weight increments of the ESVs of other countries (Fig. 3), the Nissan ESV, only 200 kg heavier, may be said to be extremely light in weight. We are proud of our efforts which

have resulted in this achievement, and we would like to give a brief summary of the consideration we have given to safety in developing our ESV, and of the problems remaining to be solved.

1) In the Japanese ESV specifications, it is specified that for securing occupant survival space, the passenger compartment intrusion should be less than 125 mm in the portions related to safety. In this regard, we thought, for example, that the compartment intrusion at the lower-limb level in a lateral collision would not affect safety, even if that intrusion were substantial.
2) Reviewing actual collision cases, we considered that, as in the case of 65 km/h rear-end collisions, occupant protection in frontal pole crashes would be sufficiently attainable at 65 km/h. Accident analysis in greater detail will be required in the future.
3) Concerning rear-end collisions, we designed the rear end, so that the moving barrier would come to a stop just short of the fuel tank, and thus we succeeded in shortening the total length of our ESV. In actual rear-end collisions, however, a sharp-edged section or luggage in the trunk compartment often causes damage to the fuel system, and in collisions between different-sized vehicles, it sometimes happens that the car colliding from behind overrides the other car. Further study in this respect should be undertaken also.
4) With regard to roll-over accidents, we considered that deformation around the A-pillar would not affect safety if space around the B-pillar were secured (as stated in the Japanese ESV specifications). This concept is regarded as valid especially in the case of an occupant restrained by an air bag or a 3-point seat belt.
5) For the determination of injury levels, we utilized 50 percentile dummies and the injury criteria of the ESV specifications. The injury criteria of the ESV specifications are different from those of MVSS, and there are new comments and proposals regarding injury level determination. In actual 80 km/h frontal barrier collisions, the Nissan ESV would have obtained somewhat greater probability of occupant survival than in tests with dummies. The definition of the human tolerance limit should be internationally standardized, and along with this, the actual rate of collision survivors should be definitely established. Otherwise, the design of vehicles will be distorted.

2. CONSIDERATIONS OF FUTURE SMALL CAR SAFETY

In considering vehicle safety for the near future, we must carefully review the achievement of our ESV development, analyze the patterns of actual automotive accidents, and take into account the shortage of natural resources, the economy of energy, and other universal problems.

Needless to say, vehicles can exist only when accepted by the majority of people. As the history of automobiles shows, anything that forces inconvenience upon us, even though able to force its way for some time, will disappear in the long run.

In this light, the U.S. DOT setting up the RSV plan and starting the study of Phase 1 at the beginning of this year is a very welcome development.

Last spring, when we completed the final design of our ESV, we started activities for the investigation of a small safety vehicle as our next subject; and ever since, we have been proceeding with related study, development, and trial manufacture. We wish to give a partial outline of our activities herein, with a view to receiving comments from those concerned.

2-1. Lessons from Accident Analysis

The patterns of vehicle collisions in Japan are as shown in Fig. 4. As can be seen from this figure, frontal collisions represent about 39% of the automotive accidents involving death and injury. This indicates that occupant protection in frontal collisions is as important as ever.

Each collision pattern, converted into the equivalent barrier test speed, is shown in Figs. 5 through 7. The converted velocity has been derived by the equalization of changes in the velocity of colliding vehicles (i.e., changes in momentum), and this is considered to be essentially reasonable in the evaluation of occupant safety. Approximately 95% of the frontal collisions will be covered by the equivalent barrier test speed of 65 km/h.

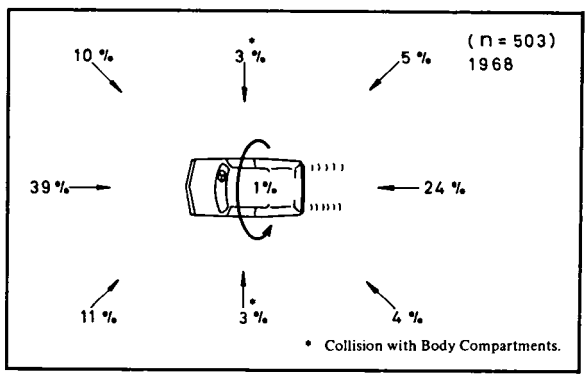


Figure 4. Direction of Collision

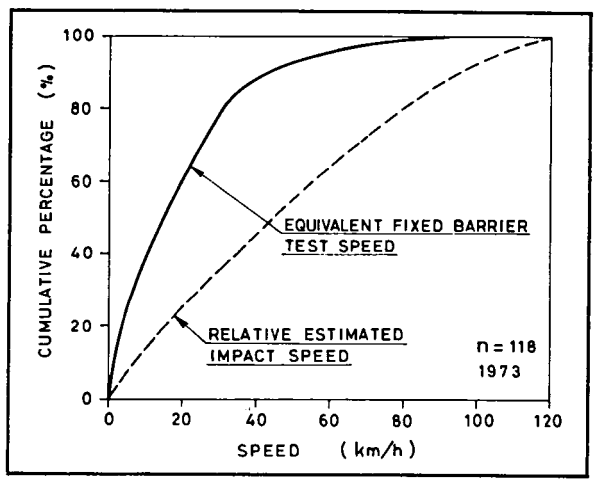


Figure 5. Comparison of Speed (Frontal Collision)

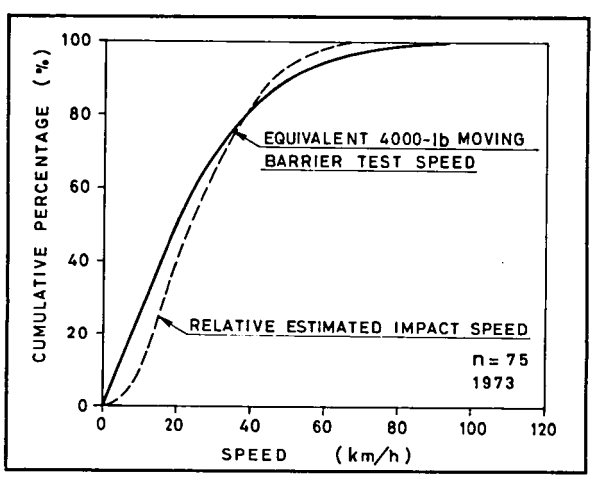


Figure 6. Comparison of Speed (Rear Collision)

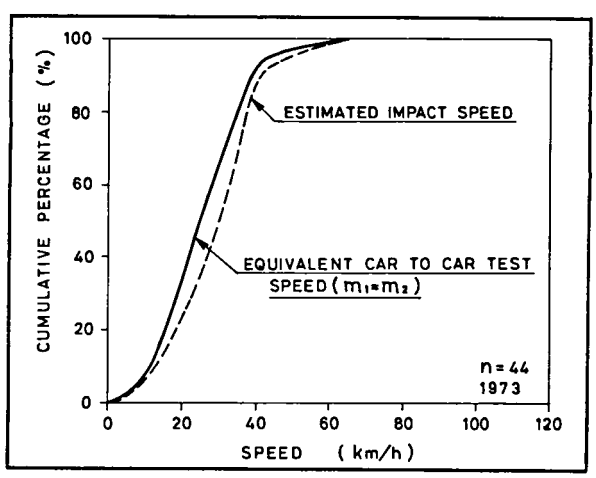


Figure 7. Comparison of Speed (Lateral Collision)

An equivalent 4,000-lb moving barrier test at 50 km/h will cover approximately 90% of actual rear-end collisions and the equivalent equal-weight vehicle test at 50 km/h will cover approximately 95% of actual side collisions.

Examination results in the U.S.A. indicate that the equivalent barrier test speed of 65 km/h covers about 75% of the frontal collisions involving death, and approximately 90% of those involving injury (see Fig. 8).

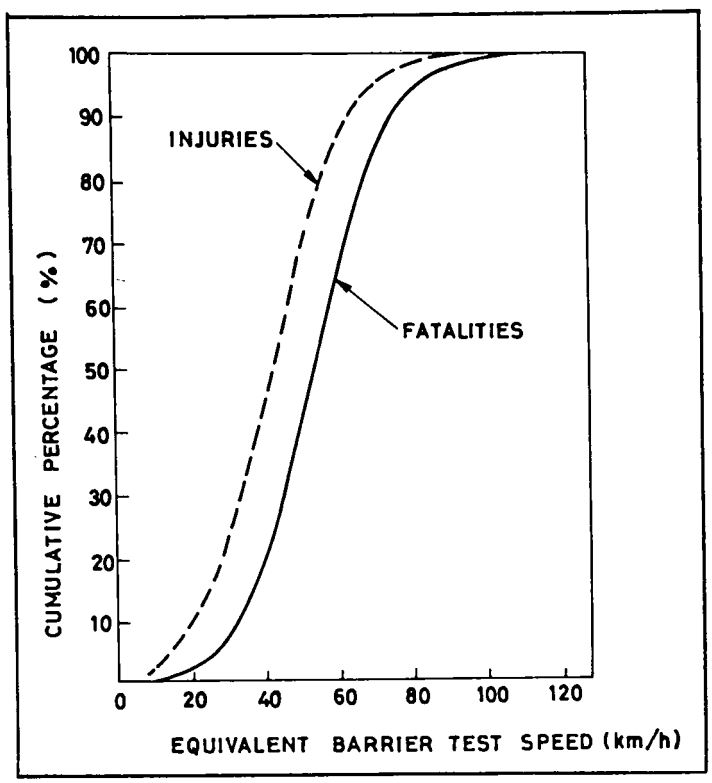


Figure 8. Cumulative Percent Fatalities Injuries Within Equivalent Test Speed Range (Source: SAE Paper No. 720445)

2-2. Car-to-Car Collisions

In car-to-car collisions, the weight of the other car, and the energy absorbing ratio between both cars come into question.

Your Car	m_1
The Other Car	m_2
Closure Speed	V_C

Taking the above, if the restitution coefficient is 0, from the momentum theory, the equivalent barrier test speed, V_E, of your vehicle is expressed as follows:

$$V_E = \frac{m_2}{m_1 + m_2} V_C .$$

How V_E/V_C changes by mass ratio m_2/m_1 is indicated in Fig. 9, and basically, this is used in evaluating occupant restraint safety.

However, this is based on the assumption that occupant survival space is sufficiently secured in the passenger compartment. The equivalent barrier test speed, in which deformation of the vehicle is noted, should be expressed in the following equation:

$$V_E = \sqrt{K_1 \frac{m_2}{m_1 + m_2}}\, V_C ,$$

where K_1 denotes the ratio of the amount of energy absorbed by your vehicle to the entire energy. K_1 expressed as a parameter is shown in Fig. 10.

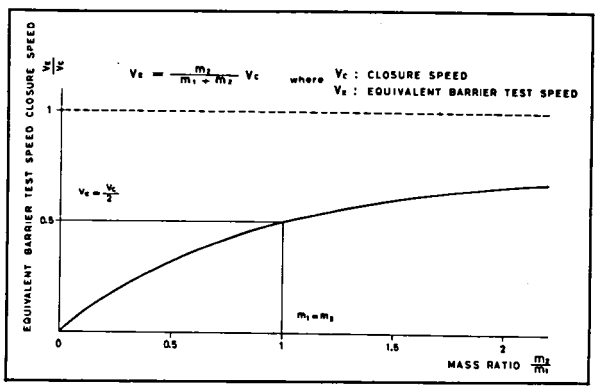


Figure 9. Equivalent Barrier Test Speed (Momentum) to Vehicle Mass Ratio

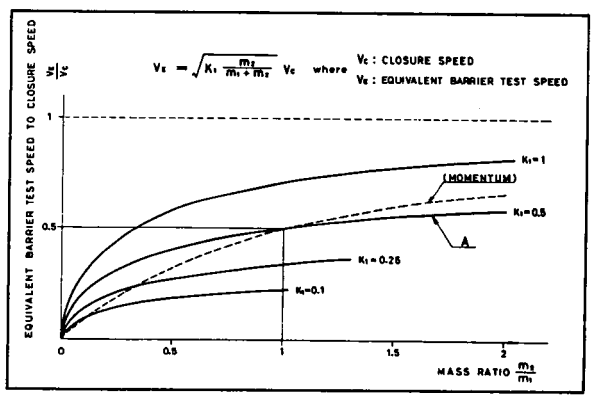


Figure 10. Equivalent Barrier Test Speed (Energy) to Vehicle Mass Ratio

When the reaction-force characteristic of the vehicle's front end is supposed to take the shape of a triangular wave, and the crush stroke in a barrier collision is supposed to be extended in proportion to the vehicle weight raised to 1/2 power as indicated in Fig. 11, it can be said that the barrier collision reaction force to be obtained by dividing kinetic energy by the crush stroke is also in proportion to the vehicle weight raised to 1/2 power. Then, it may follow that the vehicle's reaction-force characteristic, i.e., the inclination of a triangular wave, seems to have nothing to do with the vehicle weight, and that the amount of energy absorbed by each car in a car-to-car collision is supposed to be 1/2, i.e., K_1 is 0.5.

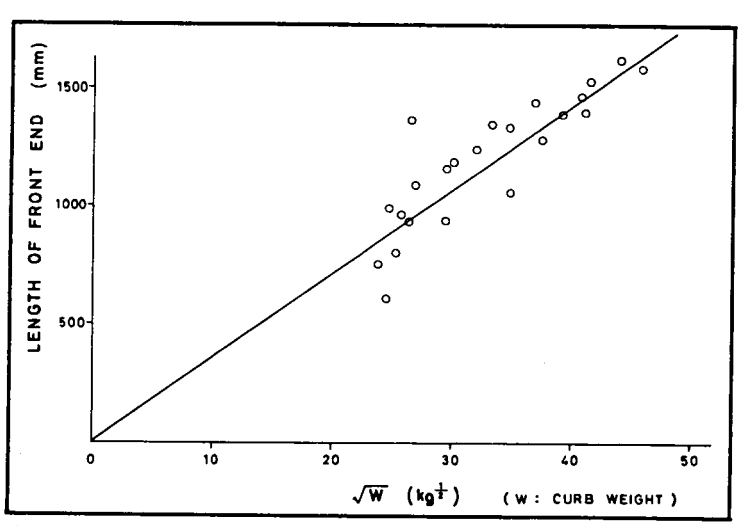


Figure 11. Curb Weight and Front End Length

In this computation, the equivalent barrier test speed of actual car-to-car collisions is assumedly indicated by curve A in Fig. 10.

What we can see from this figure is that, if provided with a safety device effective in a collision at the equivalent barrier test speed of 65 km/h, your vehicle will be safe in a collision with a vehicle twice as heavy at a closure speed of approximately 110 km/h.

As the characteristics of the actual vehicle body are different from the above assumption, in consideration of mixed traffic, vehicles heavier than the average should be of a soft construction, and conversely, those vehicles lighter than the average should be hardened, in order to balance cars of different weight. With the small car, the extent to which the front end can be hardened is naturally limited.

On the other hand, a greater reaction force in a lower speed collision is not desirable, since it will reduce occupant safety and increase aggressiveness. Therefore, it is necessary that reaction force be in proportion to the crush stroke, and that a strong passenger compartment be made to serve as the ultimate inhibiting factor of the stroke in order to secure occupant survival space.

In actual collisions, it often happens that a larger car overrides a smaller one. A similar case frequently occurs in frontal collisions between a truck and a small car. A small car the nose of which is forced down in braking is liable to be overridden in a collision. Further, in a partial overlapping or oblique collision, damage tends to be greater in extent. However, it is advisable that the same crashworthiness be obtained wherever force is given.

2-3. Pedestrian Safety and Low-Speed Collisions

Needless to say, pedestrian safety is an important subject. The results of an analysis of traffic accidents involving pedestrians along roads and streets in Japan are charted in Fig. 12. The actual speeds in collisions between pedestrians and vehicles are approximated as indicated by the heavy lines in the figure. If pedestrian protection in collisions up to 40 km/h became a reality, approximately 80% of the pedestrians involved would be saved.

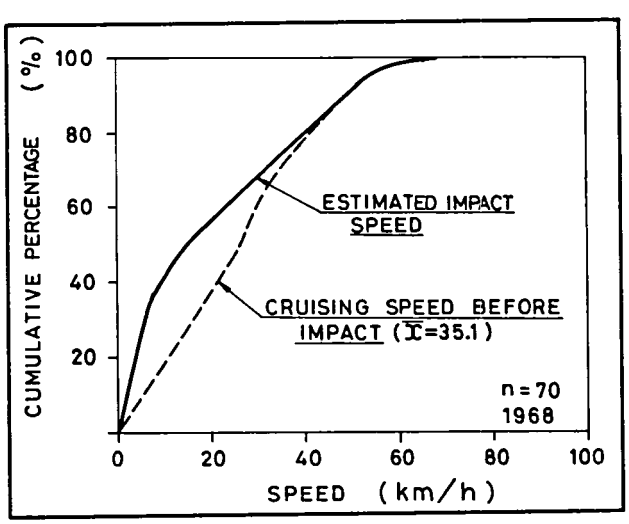


Figure 12. Comparison of Speed (Pedestrian Accidents)

It is desirable that the front-end of a vehicle have a damping force greater than the critical damping rate, in order to receive the pedestrian softly in a collision, without repelling him the next moment.

In a collision, the pedestrian should be received by the front end of the vehicle as widely as possible, in order to minimize injury due to concentrated reaction force. Moreover, it is preferable that the resultant reaction force be applied near the center of gravity of the pedestrian in order to reduce the

rotational impact on the pedestrian's head. Even if the head is stricken upon rotation, the impact should be reduced to the minimum possible extent.

From the above, it is considered desirable that the vehicle's front end be made of plastic or a similar material which has an appropriate spring constant and a suitable damping force. Such a characteristic, when properly utilized, can also be effective in a low-speed barrier collision or car-to-car collision. In this case, by selecting an appropriate geometry of contact with the other car, one can obtain a non-linearity that is initially soft and gradually hardened.

This idea should be applied not only to the front end but also more or less to all sides of the vehicle. This would serve to reduce the level of injury sustained by anyone involved in a low-speed rear-end collision as well as reduce property damages resulting from contacts with a bicycle or a car.

When the above concept is brought together with the previously mentioned notion of safety in medium or high speed collisions, one example of the "stroke to reaction-force" characteristics desirable in the small car is as indicated in Fig. 13.

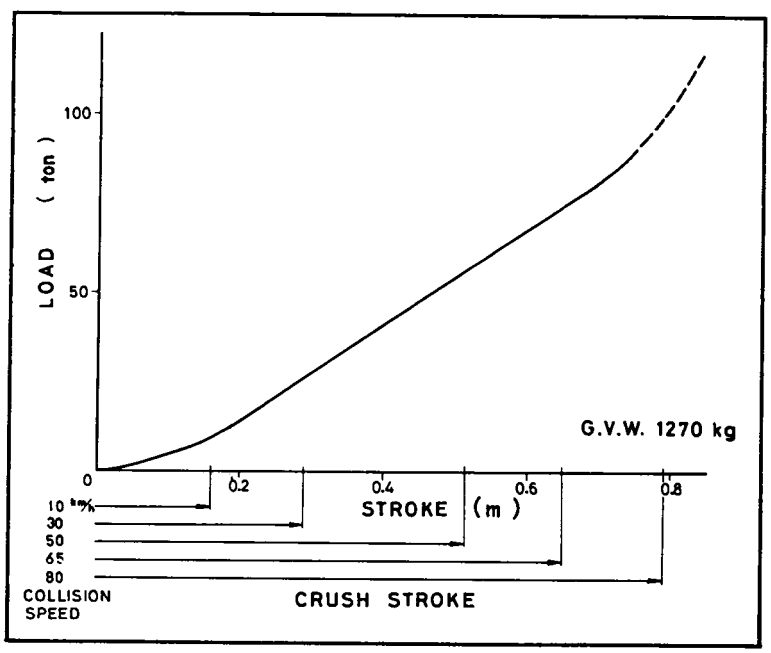


Figure 13. An Example of Desirable Load-Stroke Curve

2-4. Requirements for Accident Avoidance

Generally it is said that drivers are responsible for 99% of the automotive accidents occurring in Japan. According to American data, 73% of the drivers are found to have been directly responsible for, and 96% to have had some connection with, the causes of all automotive accidents taking place in the U.S.A.

Regarding the causes of accidents in Japan, shown in Fig. 14, those related to driver behavior account for 106%; those related to driver inattention 93%; those related to driver physical or mental condition 14%; and those related to other factors 13% (the total percentage exceeds 100%, since two or more causes of a single accident are counted).

Representing a large percentage of the accidents resulting from driver behavior are speeding and reckless driving; ranking first and second in the accidents resulting from improper physical or mental condition are impatience and intoxication. Further, when fatal accidents are arranged by cause, we find that inattention, intoxication and dozing are again ranked at the top.

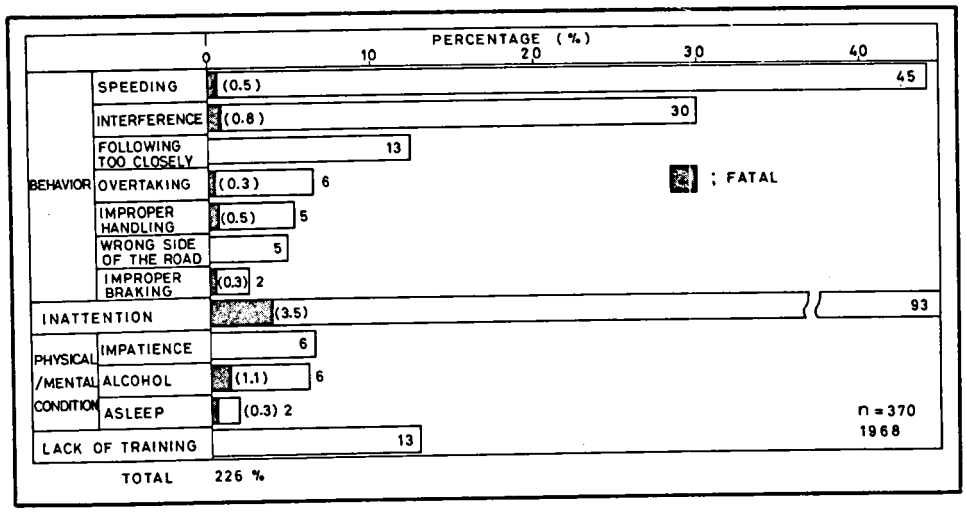


Figure 14. Human Causal Factor Rates at Accidents

Most of these should be understood as a question of man-machine relations. Accordingly, automotive dynamic systems should be developed so that the vehicle may best respond to the driver's natural operation at a critical moment.

On the other hand, it is necessary that the vehicle be equipped with something to help the driver make proper judgment. An alcohol interlock device and an anti-doze device are examples of instruments to meet such a purpose. We might find that a low consciousness situation while driving has some points in common with the pattern of dozing.

Further, to increase the driver's feeling of unity with the surrounding environment, it would be effective to widen the vehicle's forward visibility extensively, for example, by eliminating the A-pillars.

3. THE VEHICLE ENVIRONMENT

Vehicles of the future should be based on safety harmonized with the social environment. In designing a safety vehicle, we must take into consideration some points which will be discussed in the following subsection:

3-1. Problems of Energy and Natural Resources

The economy of natural resources is a world-wide problem at present. Needless to say, the smaller the vehicle, the fewer the resources consumed in its manufacture will be. Let us consider fuel consumption by vehicles. In the urban areas, vehicles consume fuel in proportion to their weight, because inertia resistance, rolling resistance and grade resistance are all a function of weight (see Fig. 15).

The oil consumed in the manufacture of one small car can be estimated at 1,400 liters (according to a survey conducted within Nissan Motor Company). Driving this small car for 100,000 km results in the consumption of an additional 10,000 liters of petroleum.

Now, suppose that the car weight is reduced by 50%; total fuel consumption will be reduced by about 35%. Therefore, the logical goal is a light safety vehicle of the smallest practical size.

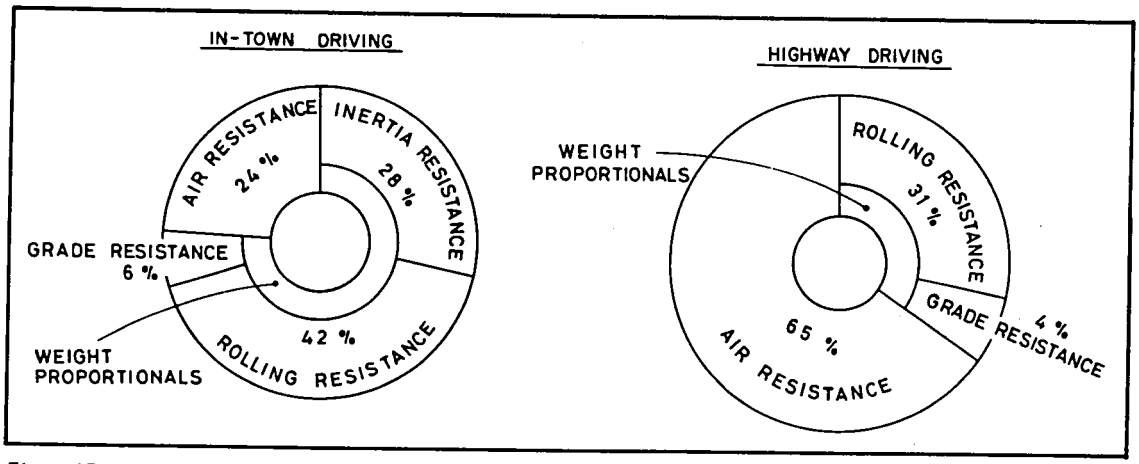


Figure 15. Running Resistance

3-2. Traffic Environment

The traffic congestion and parking problems presently being experienced in the cities of the world not only cause loss of time, but also have adverse affects on fuel consumption, emission control and noise control. For the alleviation of these undesirable conditions, we foresee the development of public transportation facilities, a reduction in mixed traffic, and an increase in the average number of passengers per vehicle; the smaller vehicles will also play an important role.

The speed limit control enforced in various countries since last winter's energy crisis proved to be a great contribution toward a reduction in the frequency rate of traffic accidents.

Further controls of the maximum allowable speed, and compulsory installation of speed warning devices should be continued in the future.

3-3. Other Problems Concerning Vehicles

In some countries, there are some severe problems other than those mentioned above; the emission control in Japan, for example, and noise control in some European countries.

The emission control law in Japan is an especially serious regulation. It entails a great increase in car production cost, a considerable increase in car weight, strict temperature control and the increase of fuel consumption.

A 10 dB noise reduction would also be quite serious. We are afraid that this might cause a deterioration of vehicle performance like severe emission control.

We hope that in rulemaking, a sufficient condition will be given in cost benefits and the difference between the experimental possibilities and their productive or social possibilities of vehicles. We will endeavor to this direction.

A futuristic objective in a certain country might become a stern reality in other countries, and perhaps some day spread from there to the rest of the world. Regarding safety at least, we do not wish to see such a thing happen.

The vehicle should be designed as a total, well balanced system, and of course we are going to continue our efforts toward manufacturing such a vehicle with the international cooperation.

4. A FEASIBILITY STUDY ON SMALL SAFETY VEHICLES

Based on the ideas mentioned above, we are pursuing a definite study for the development of a small safety vehicle. We wish to outline hereunder our current plans and progress.

4-1. Target Performance of the Nissan Safety Vehicle

To be sure, the higher the target performance is set, the better it may be, but our experience with the development of the Nissan ESV showed us that because the aim was too high the ESV lost practicability as a motor vehicle. As already mentioned, a small car should be selected as the safety vehicle of the future, and we have established the target performance of the Nissan safety vehicle that is realistically attainable. This is performance sufficiently effective from the accident analysis results we have referred to above.

The target performance of the Nissan safety vehicle is indicated in Table 1, in comparison with the Nissan ESV specifications, and the 3,000-lb ESV specifications that provide reference data for the RSV Plan. The typical features of the Nissan safety vehicle are as follows:

1) Occupant Protection in 65 km/h Frontal Barrier Collisions
2) Fire Prevention in Car-to-Car Collisions in Addition to Safety in 50 km/h Moving Barrier Rear-End Collisions
3) Reduction in Vehicle Aggressiveness in All Speed Ranges
4) Improved Pedestrian Safety
5) Better Visibility Including Frontal Vision without A-Pillars
6) Steering Stability in Compliance with Driving Behavior
7) Devices for Preventing Drivers from Intoxication and Dozing
8) Emission Control and Noise Control in Harmony with Safety

Besides the above, we are studying an automatic braking system controlled by radar, though this may not become a reality until the distant future.

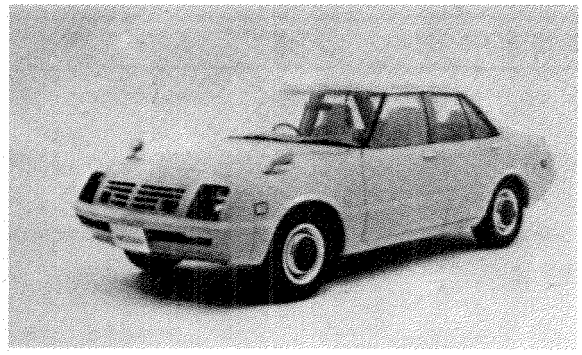
The main specifications of the trial model of the Nissan safety vehicle are indicated in Table 2, and the pictures in Fig. 16.

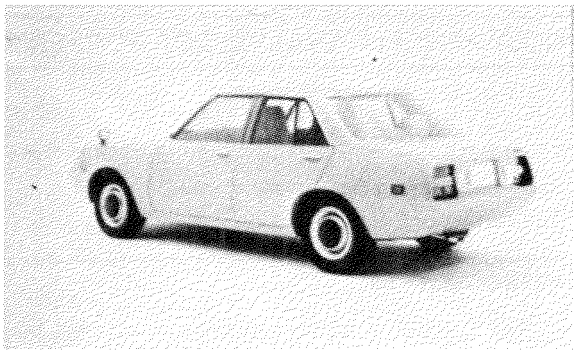
TABLE 1 - TYPICAL TARGET PERFORMANCE OF THE NISSAN SAFETY VEHICLE (NSV) AND OTHERS

		NISSAN NSV	3,000-lb ESV	NISSAN ESV
Body Style Curb Weight Size		4-Door Sedan 1,000 kg 4,150 x 1,595 x 1,345 (Trial Model)	4 or 2-Door Sedan Max. 1,360 kg (3,000 lbs) Max. 4,700 x 1,780 x 1,420	4-Door Sedan 1,250 kg 4,415 x 1,640 x 1,385
Accident Avoidance	Braking Performance	o Satisfying the Requirements for the 3,000 Pound ESV o Improvement in Accident Avoidance Performance Through Consideration of Human Characteristics	Stopping Distance Max. 53 m Test Speed 95 km/h 100% and Light Load	Max. 50 m 95 km/h 50% Load
			Stopping Distance Max. 26 m in a 109 m Radius Lane Test Speed 65 km/h 100% and Light Load	No Requirement
	Overturning Immunity		[Drastic Steer and Brake] No Overturning at a Specified Combination of Rapid Steering and a Panic Locked Wheel Braking	[J-turn] No Overturning, Test Speed 110 km/h, Steering Rate Min. 500°/sec Steering Angle Min. 180°
	Others		Requirements for Steering and Handling	←
Visibility	Direct Visibility	Max. 4° (Horizontal)	Obstructions' Angular Width (in Zones I and II) Max. 6° (Horizontal)	No Requirement
		No C-Pillar in a 270° Arc (Zones I through V)	Max. 4 Pillars in a 270° Arc (Zones I through V)	←
		4 Corners	Requirements for Corner Reference Points (2 Corners)	No Requirement
	Others	→	Lighting etc.	←
Occupant Crash Protection	Frontal Collision	65 km/h	Fixed Barrier 80 km/h	←
	Rear Collision	50 km/h	4,000 lbs Moving Barrier 45 km/h	65 km/h
	Car to Car (Frontal)	Compatibility Requirement	Requirements for Energy Absorbing Characteristics in Collisions with Various Weight Vehicles	Equal Mass Vehicles 120 km/h
	Injury Criteria	→	HIC Max. 1,000 Chest Max. 60 g-3 m·sec Femur Max. 770 kg	Head Max. 80 g-3 m·sec ← Max. 640 kg
Pedestrian/Cyclist Protection	Energy Absorption	Control of Energy Absorbing Characteristics of Front Nose and Side Body	Body Block Impact Test Speed 40 km/h Acceleration Max. 60 g-3 m·sec	Capability for Dispersing Impact Force
	Exterior Protrusions	→	Elimination of Protrusions or Crushable Protrusions	Elimination of Sharp Protrusions

TABLE 2 - PRINCIPAL SPECIFICATION OF THE NISSAN NSV TRIAL MODEL AND THE NISSAN ESV

	NISSAN NSV (TRIAL MODEL)	NISSAN ESV
Type	→	4-Door Sedan
Body Structure	→	Monocoque
	No A-Pillar	
Capacity	→	4 Persons
Overall Length	4150 mm	4415 mm
Overall Width	1595 mm	1640 mm
Overall Height	1345 mm	1385 mm (Without Periscope)
Wheelbase	2420 mm	2550 mm
Tread (Front)	1275 mm	1365 mm
(Rear)	1265 mm	1375 mm
Curb Weight	1000 kg	1250 kg
Engine	1400 cc	1800 cc, 105 ps
Energy Management	Monocoque Body	Monocoque Body (E_1) Broach Unit (E_2)
Occupant Protection	o Air Bag and Seat Belt or Seat Belt Only (Front) o Seat Belt (Rear) o High Back Seat (Front) o Pop Up Head Restraint (Rear)	o Air Bag and Seat Belt (E_1) o Seat Belt With Preloader } (E_2) o Automatically Retractable Steering Column } (E_2) o High Back Seat (Front, Rear)
Pedestrian Protection	o Soft Safety Nose o Soft Side Protector	o Soft Padded Bumper o Energy Absorbing Hood
Other Safety Device	o Safety Monitoring Device o Head Lamp Wiper (Retractable) o Automatic Brake o Alcohol Interlock Device o Safety Tire (Single Chamber Type) o Human Characteristic Suspension	o Periscope o Central Warning Device o 3-Beam Pattern Headlamp o 2-Level Tail Lamp and Trail Lamp o The Nissan Safety Tire (Double Chamber Type)





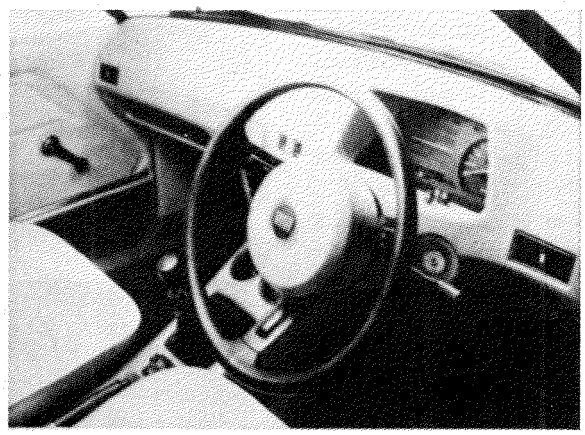


Figure 16

4-2. International Cooperation in Development of Small Safety Vehicles

The development of a small safety vehicle is no easy matter. The attainment of all target performance criteria in the Nissan safety vehicle will in some respects involve more difficulties than were experienced in the development of the Nissan ESV.

However, efforts to overcome such difficulties are indispensable for the future of the automobile. We expect that such work will also be started in other countries and that we will be able to secure mutual technical cooperation. Without international cooperation, it would be impossible for us to develop the following important points.

1) Investigations and analyses of automotive accidents in various countries, to obtain data which can be fed back to vehicle design.
2) Study of the driver's steering behavior and response to stimuli from outside the vehicle.
3) Establishment of more realistic criteria for injuries sustained by human beings. Besides the fatality limit, criteria for determining the severity of injury should be set up from a social viewpoint. (It should be mentioned that cadavers cannot be used for tests in Japan.)
4) Dummies with which we can determine the above criteria and level of injury; and the repeatability of dummies as measuring instruments.
5) Unification of requirements for vehicles which come from the social conditions of various countries. In the future, safety control in the U. S. A., noise control in Europe and emission control in Japan should be internationally well balanced.

With regard to all the above, we hope to secure the cooperation of those concerned in each country. Together we can go forward, intensifying our efforts toward the improvement of vehicle safety throughout the world.

COST-BENEFIT ANALYSIS OF BUMPER SYSTEMS FOR SMALL CARS

TEIJI IIDA
Body Engineering Design Department
Toyota Motor Co., Ltd.

INTRODUCTION

The bumper – which was once an external body component mainly used to finish off the appearance of a vehicle – has been undergoing extensive changes as a result of public demand, insurance company involvement, and legislative action. In particular the National Highway Traffic Safety Administration, U.S. Department of Transportation, enforced the Motor Vehicle Safety Standard (MVSS) No. 215 in 1972 and its requirements have been getting stricter. Now they are proposing a no-damage bumper standard for 5 mph impacts under Title I of the 1972 Motor Vehicle Information and Cost Savings Act.

Insurers seem to have welcomed this proposal and urged strengthening the standard in the future to cover 10 mph impacts, corner impacts, and so forth. On the other hand, automakers have objected to the proposal and some asked for more time to study the effect of the current bumper system in the real world.

So, we studied various bumper systems, especially those used for small cars, from a cost-benefit standpoint.

A COST-BENEFIT ANALYSIS OF THE BUMPER SYSTEMS FOR TOYOTA COROLLA

We compared the bumper system of the 1974 model Toyota Corolla with that of the 1972 model from a cost-benefit standpoint.

1972 and 1974 Models: Structural Differences Between the Bumper Systems

Figures 1 and 2 show the bumper system of the 1972 model Toyota Corolla and Figures 3 and 4 show that of the 1974 model Toyota Corolla.

The structure of the 1972 model Toyota Corolla bumper system was simple. Figure 5 shows a cross section through the bumper system. The U-shaped bumper bars were attached to the body structure by four rigid stays.

We changed the simple 1972 model bumper system into a rather complex one for the 1974 model. Figure 6 shows the structure of the bumper system for the 1974 model Toyota Corolla. We reinforced the bumper bars using high-strength back bars and we installed energy-absorbing units instead of rigid stays.

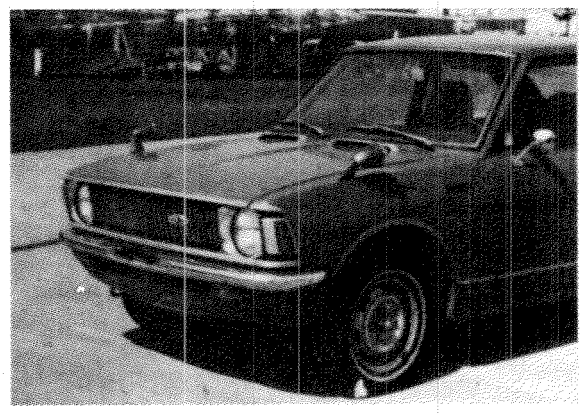


Figure 1. The bumper system for the 1972 model Toyota Corolla (front).

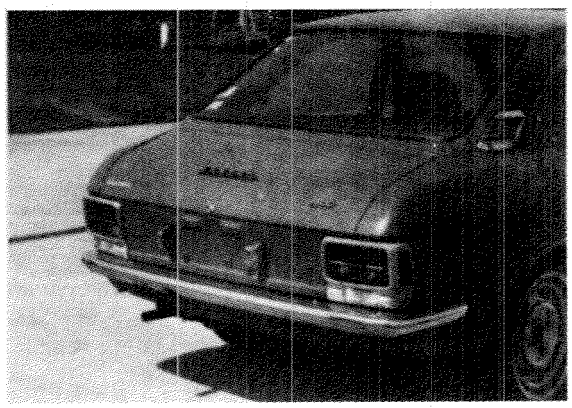


Figure 2. The bumper system for the 1972 model Toyota Corolla (rear).

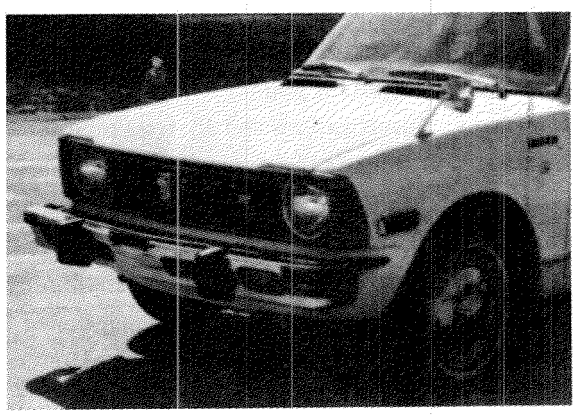


Figure 3. The bumper system for the 1974 model Toyota Corolla (front).

The 1974 model bumper system meets the requirements of the MVSS No. 215, and its optional bumper system with rubber guards on the face bars meets the no-damage requirements for 5 mph barrier impacts.

Figure 4. The bumper system for the 1974 model Toyota Corolla (rear).

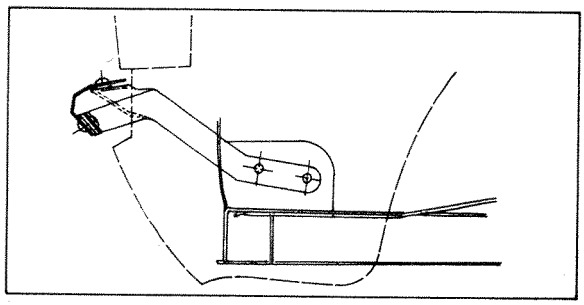


Figure 5. The structure of the bumper system for the 1972 model Toyota Corolla.

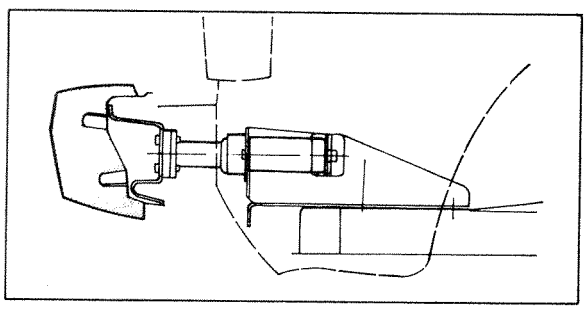


Figure 6. The structure of the bumper system for the 1974 model Toyota Corolla.

Weight and Cost

The weight of the 1974 model Toyota Corolla bumper system was increased by 24 kg over that of the 1972 model bumper system; the cost increase was 60 dollars.

A Cost-Benefit Analysis of the 1974 Model Toyota Corolla Bumper System

Table 1 shows how the benefit/cost ratio of the 1974 model Toyota Corolla bumper system was estimated, taking that of the 1972 model to be the standard.

To estimate the benefit/cost ratio, we used our own cost data and we also accepted, without modification, the data from the report on bumper systems which was published by the Insurance Institute for Highway Safety (IIHS) in the U.S.

Table 1
BENEFIT/COST RATIO COMPARISON

Model	Weight Increase (kg)	5 mph Barrier Crash Repair Cost (U.S. dollars)	Cost Increase (U.S. dollars)	Benefit/Cost Ratio
1972	–	466.24	–	1.00
1974	24	181.39	60	4.51

Claim frequency per 100 vehicle years = 10%.
Average Life span of passenger vehicles in U.S. = 9.5 years.
Benefit/cost ratio of the bumper system for the 1974 model = (466.24 - 181.39)/60 × 9.5 × 10/100 = 4.51.

For the repair costs, we adopted the IIHS report data of the 5 mph barrier crash test results. In the real world, the repair cost due to the low speed vehicle accidents seems to be higher than this data, considering the data of another IIHS report which shows the average loss payment per claim is higher than the 5 mph barrier crash repair cost. Nevertheless, it was reasonable for us to use this data to estimate the ratio of benefit/cost.

Concerning the claim frequency, we presumed it was 10% per year, since the data of the IIHS report showed it was around 10% in 1972 and 1973.

According to the report, "Cumulative Regulatory Effects on the Cost of Automotive Transportation (RECAT)," prepared for the Office of Science and Technology, the average life span of passenger cars used in the U.S. was 9.42 years in 1970 – we rounded it off to 9.5 years.

The benefit/cost ratio of the 1974 model Toyota Corolla bumper system was 4.51, taking the ratio of the 1972 model system to be the standard. In other words, we could say our bumper system for the 1974 model Toyota Corolla is more economically beneficial to the consumer than that of the 1972 model.

The ratio of our optional bumper system with rubber guards will be higher, considering our 5 mph barrier impact test results show no damage.

NO-DAMAGE BUMPER SYSTEM AT 10 mph IMPACTS

At the previous ESV International Conference in Kyoto, we presented test results which showed that we had developed a no-damage bumper system at 10

mph barrier impacts. Figure 7 shows the Toyota ESV and Figure 8 shows the structure of its bumper system. The bumper bars are made of thick stainless steel and the energy-absorbing unit's maximum stroke is 200 mm.

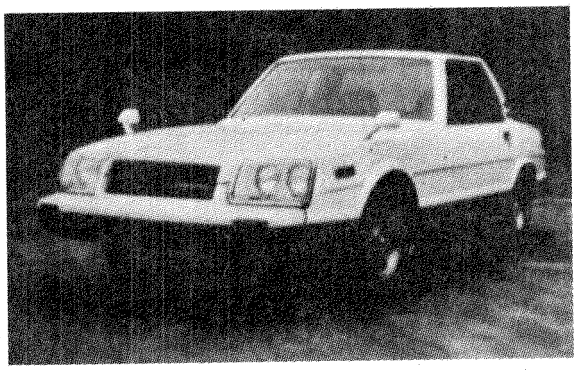


Figure 7. The bumper system for the Toyota ESV.

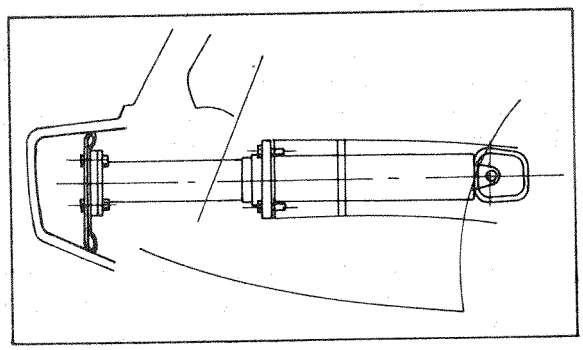


Figure 8. The structure of the bumper system for the Toyota ESV.

We considered how the 10 mph impact no-damage bumper system would work on our small car, Toyota Corolla.

No-Damage Bumper System Considerations

When we considered the bumper system restricting the force generated during 10 mph impacts to almost the same level as that of the 1974 model bumper system, we discovered that the weight of the 10 mph impact no-damage bumper system had to be increased by 60 kg over that of the 1972 model system and the cost increase would be 150 dollars.

But the benefit/cost ratio of the 10 mph impact no-damage bumper system would be higher than that of the 5 mph impact no-damage bumper system for the 1974 model because the repair cost of the 1972 model system for the 10 mph barrier impact would be extremely high.

The stroke of the shock absorbers, however, would be 230 mm to absorb the impact energy which is 4 times as great as that of the 5 mph impact. The stroke of the 1974 model Toyota Corolla is 50 ~ 55 mm. A long stroke of 230 mm, as Figure 9 shows, would cause an extremely protruding bumper system and it would give the small car an unacceptable appearance.

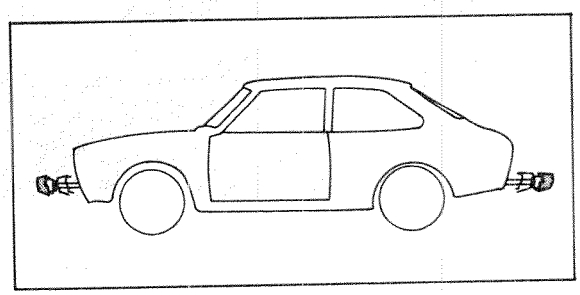


Figure 9. A 10 mph impact no-damage bumper system.

So, if we restrict ourselves to a bumper system with shock absorbers, we must take steps to reduce the length of the stroke to the level of the 1974 model. We have to reinforce the body structure on the same scale as we did on our ESV and, at the same time, we have to develop methods to protect safety related components. As a result, the deceleration on the floor of the car during 10 mph barrier impacts would be 23 g, which is almost the same deceleration as the 1974 model Toyota Corolla sustains during its 25 mph barrier impacts. The weight and cost increase would be higher than those previously mentioned and the benefit/cost ratio would be decreased. It is apparent that such a large weight increase would reduce performance of the small car, resulting in poor handling and poor fuel economy.

NO-DAMAGE POLYURETHANE BUMPER SYSTEM AT 5 mph IMPACTS

We have been developing and evaluating no-damage polyurethane bumper systems at 5 mph barrier and pendulum impacts. We have been evaluating two types of polyurethane bumper systems. In one, the impact energy is absorbed by compression of polyurethane foam; and in the other, the energy is absorbed by buckling of the walls of polyurethane. Figure 10 shows one of our experimental polyurethane bumper systems. Figure 11 shows the structure of the compression type (*uniflex*) bumper system and Figure 12 shows the buckling type (*hiflex bumper*).

Polyurethane Bumper Considerations

Our recent test results of polyurethane bumper systems show the bumper system meets the 5 mph no-damage requirements of proposed MVSS No. 215 for the 1976 model.

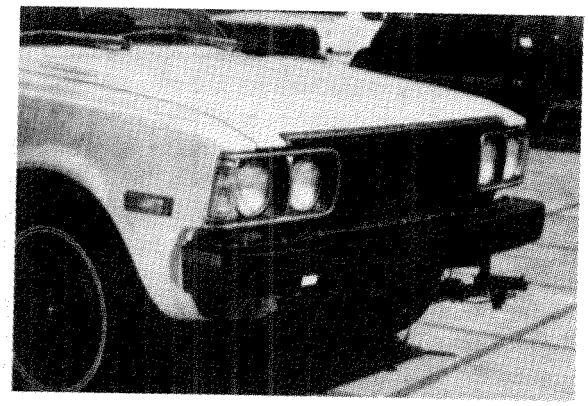


Figure 10. An experimental polyurethane bumper system.

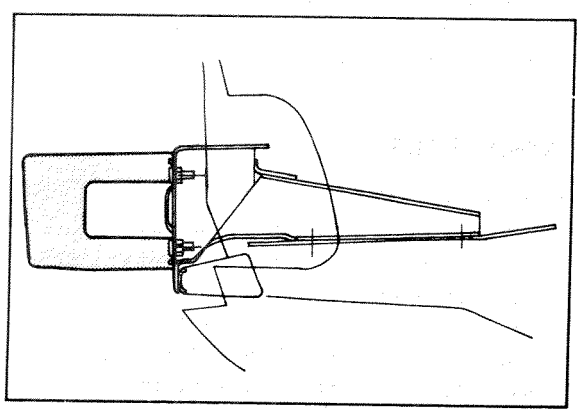


Figure 11. A structure of the compression type polyurethane foam bumper system.

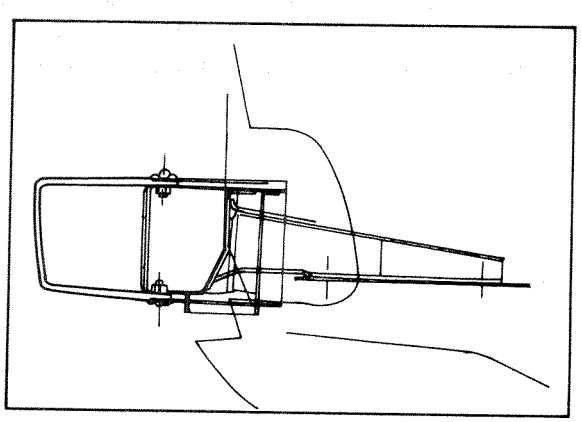


Figure 12. A structure of the buckling type polyurethane bumper system.

It has been generally said that this kind of bumper system is liable to generate higher force during the barrier impacts, resulting in damage of safety related components; and on the other hand, the system is liable to fail to meet the requirements of the present standard because excessive stroke of polyurethane during the pendulum impacts may cause the bumper bars to contact the surface of Plane A of the pendulum specified by MVSS No. 215.

However, we discovered – through our series of tests and evaluations – that we could control the deflection of polyurethane bars during the barrier or pendulum impacts at almost the same level and we could keep the force of barrier impacts almost two times as great as that of pendulum impacts. Then, we could make the bumper system meet the conflicting requirements of either the barrier or pendulum impacts, if we took certain technical steps.

We used the plan view shape of bumper bars, as shown in Figure 13, to lessen the force during the barrier impacts.

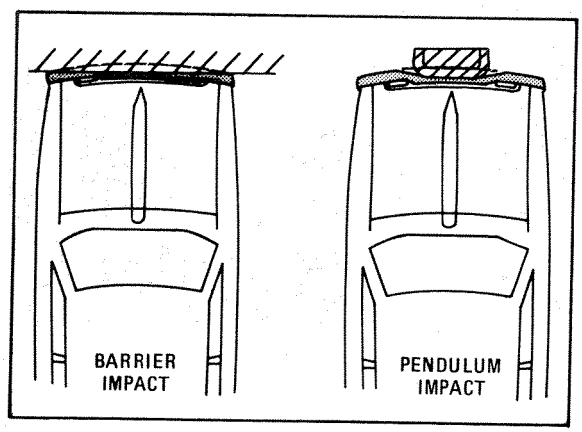


Figure 13. Deformation of a polyurethane bumper bar.

In the case of the compression type bumper system, to keep the force generated during any different height pendulum impact at almost the same level, we chose the proper balance of thickness between the pendulum contact areas and other areas of polyurethane foam; and we kept at least 30 mm vertical contact between a bumper bar and a pendulum, as shown in Figure 14.

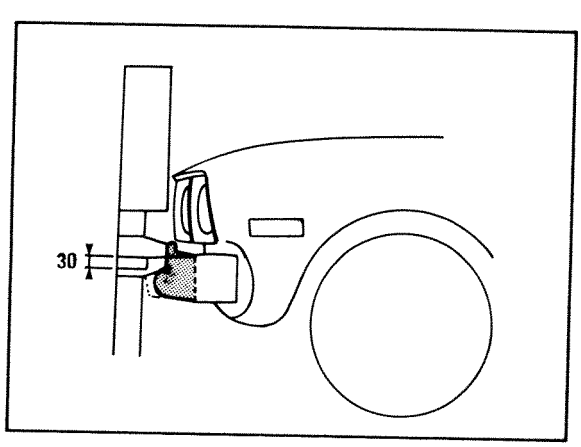


Figure 14. Vertical contact between a bumper bar and a pendulum.

As for the buckling type, we have not yet arrived at a conclusion concerning the above subject when we consider the polyurethane bumper bar which has enough deflection capability during impacts not to permit any damage to the vehicle. We still have to find a proper balance of the energy-absorbing capability between the shells (upper and lower areas of the section) and ribs of the bumper bar.

Figure 15 shows a view of pendulum tests.

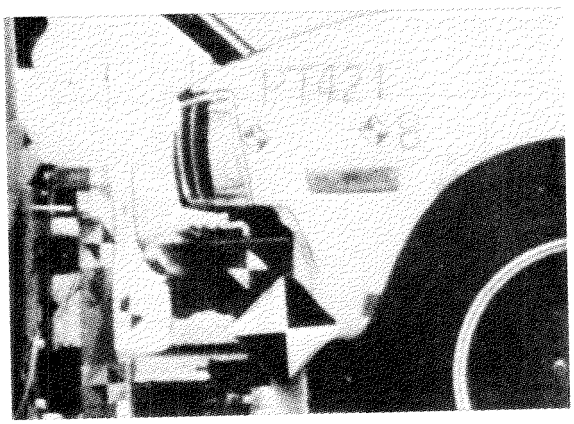


Figure 15. A pendulum test of a polyurethane bumper system.

We still have to confirm whether we can properly control the energy-absorbing characteristics of polyurethane during mass production of this bumper system. We have to consider the variance of the energy-absorbing characteristics in mass production and the effect of climatic conditions, especially low or high temperature. In the case of the compression type bumper system, we have to study how water or moisture absorbed in polyurethane foam affects its energy-absorbing characteristics and also establish its economic and manufacturing feasibility.

It seems, however, we can develop the 5 mph impact no-damage bumper system by adopting polyurethane – with more advantages to the consumer – than we can by adopting the system using shock absorbers.

We can make a light-weight bumper system using polyurethane. From a vehicle performance standpoint this factor is essential for the small car with a smaller engine.

Using this material, we expect to establish a bumper system more economically beneficial to the consumer than that of the 1974 model because of its lower damageability and lower cost.

CONCLUSIONS

1. From a cost-benefit standpoint the bumper system for the 1974 model Toyota Corolla seems to be economically more beneficial to the consumer than that of the 1972 model.
2. The bumper system which sustains no damage during 10 mph impacts would not work on the small car from a vehicle performance standpoint and a cost-benefit standpoint.
3. In the near future it seems likely that, with new technology, it will be possible to develop an economically beneficial no-damage bumper system, effective in 5 mph impacts, for the small car.

MR. BUREAU (France):

My question is directed to the Japanese speaker who gave us a calculation of cost effectiveness for bumpers which enable one to reduce the damage when you have impacts at 5 mph. One of the elements in the calculation was obviously the likelihood of accidents of this type. Now this probability, the speaker said, is at present ten percent a year; but obviously, if the material cost of an accident at 5 mph diminishes to a great extent, the drivers will pay less attention and the likelihood of accidents at low speeds will increase very considerably. The ratio of cost/benefit will, therefore, be much lower, it would appear to me, than the figure of 4.5 put forward by the speaker at the end of his calculations.

MR. IIDA (Japan):

May I say that the ten percent is an estimate. It may be five percent or so, as appropriate. I found this ten percent figure in the report of IIHS and thought it was a little high. But to compare the ratio, I thought it was reasonable to take it as it is.

MR. BUREAU (France):

If I have understood the reply, it would appear that my question was not understood. I did not wish to discuss the exactness of the figure of ten percent or to learn the source. I think it is quite accurate. All I wanted to say was this: the likelihood of ten percent is the probability today, but if the material cost of accidents at low speeds diminishes, the probability of their occurring will increase. As this likelihood is one of the elements of the calculation, the cost/effectiveness ratio will diminish to the same extent. Therefore, the figure of 4.5 for the cost/effectiveness ratio, a figure which was the result of the calculation, would probably in reality be much lower.

DR. FIALA (Chairman):

If my information is correct, there is an investigation in the United States concerning the practical value of this bumper system. Perhaps I may add something from our experience. We found in one of our models that that the air resistance increases about three percent through the bumper system because the air resistance is higher there. It also will increase the weight of the car by about three percent or so. Of course, the fuel consumption goes up by about three percent and you have to add in your calculation the three percent fuel consumption increase. Nobody, I'm afraid, thinks about this influence when we start such a project.

DR. LINCKE (Federal Republic of Germany):

My question is to Mr. John Edwards from Ford, and I'm referring to your presentation of survival probability versus human body loads. I totally agree that it isn't a step function, but where did you derive the precise shape of that function from?

MR. EDWARDS (United States):

You may recall that we referenced the Highway Safety Research Institute's work at the University of Michigan, and based on that information, plus the application of a normal distribution to the probability of survival versus barrier equivalent speed, we arrived at this conclusion. Now, I would hasten to add that the particular curve and the particular ordinate and absissa of values are illustrative. They're not intended to be precise. So, I would enjoin all delegates when they read our paper not to start plotting specific numbers from that particular curve. That particular curve is intended only to show the concept.

DR. FIALA (Chairman):

Thank you, Mr. Edwards. I think this is a very important remark, because anyone who attempts to make a cost-benefit calculation will find that it is very difficult to have precise numbers. On the other hand, the difference in cost/benefit ratios of different measures is so large that it isn't necessary to have the last two or five percent of accuracy for the assumptions.

MR. MELA (United States):

One of the questions concerned the possibility of changing the behavior of drivers through the addition of an improved bumper. Many of us have speculated on the possibility that improvements in handling, stability, and braking will, in fact, cause drivers to drive differently and possibly negate some of the benefits. My question is whether any of the speakers or anyone in the audience has had any experience or knows of investigations that have attempted to measure or estimate the change in driver behavior that could occur as a result of the improvements in accident avoidance capability, such as improvements in brakes, steering, handling or even the bumper.

MR. JACOBSEN (United Kingdom):

We have recently carried out some tests, in conjunction with a number of interested parties, to find out to what extent drivers, under normal operating conditions on a route to and from work which they're familiar with, adjust themselves to improved handling capabilities of a vehicle. We found that, on the average, it's taken between three or four days, and six days in some instances, before they use the advantage offered by higher performance limits. These limits in terms of better road holding and predictable course maneuvers. They then become progressively more aggressive in the driver behavior so the benefits are nullified in about seventy percent of instances. This sample represents about thirty people in a matter of a week or so.

MR. HALPERN-HERLA (France):

We had an opportunity to undertake a number of studies which have shown that the behavior of the driver got worse in certain cases when one improved the objective safety conditions. I can give you two examples. The first is a study of the effects of horizontal white signs painted on the roads. We ran an experimental test. We placed these horizontal signs on a stretch of road previously unmarked and measured the effect as far as the behavior of the driver was concerned. These studies showed that, without having more police surveillance, these signs led to an increase in speed on the part of drivers with the fastest cars.

The second example is seat belts. This was referred to by Mr. Bluet. In 1972, the wearing of a safety belt was not compulsory in France. The largest proportion of those wearing belts were young people who drove more modern cars – cars which were above average in performance. So, if one were to estimate the number of people killed in relation to the miles driven in 1972 among belt wearers and those who were not wearing them, one found that in 1972 the risk of being killed had not been reduced in the case of seat belt wearers. The seriousness of injuries in accidents was reduced by a factor of two, but the involvement in accidents had doubled. On the other hand, when the measure became compulsory in 1973, it affected a very large number of people and was effective.

MR. BUREAU (France):

My question relates to your own presentation, Mr. Chairman. You stressed the importance of considering, in the choice of measures, those which correspond to the cost-effectiveness curve; cost is equal to effectiveness. In other words, the ABC area in your graph. Now, in the numerical examples which you gave us in another graph, you are still below the right cost equals effectiveness. In other words, none of these measures would have an advantage which is greater than the cost. What conclusion is one to come to? Is it simply because one hasn't taken into account sufficiently the human factors which you also refer to, or is the situation this: that the estimate of human lives or injuries are being underestimated? I'd like to have your views on this.

DR. FIALA (Chairman):

On the graph that I showed you, this is true. It gives only a cost/benefit ratio larger than one for four measures. These would be the helicopter rescue, the Autobahn emergency flasher, the mobile emergency hospital and two safety belts per car. I also said that this graph was derived from the assumption of only the future income of the killed and injured car occupants. So, if you take into account the moral value of life or the pain or suffering of all sorts, it is clear that we can, and we should, introduce measures whose cost-benefit value is well below one. But again, if you compare the graph that I presented with some others where the cost for passengers killed is up to more than 200,000 dollars, instead of about 200,000 German marks, in my graph, you will see that this gives a wide variation. We shouldn't discuss monetary values of a human life; what is important is the relative values. For only on the basis of relativity can you compare different cost-benefit investigations.

DR. ENKE (Federal Republic of Germany):

In the graph that you just mentioned, there was one figure which horrified me, namely, the effectiveness of driver education.

The fact is that most accidents are caused by mistakes made by the driver. Even a very small

percentage improvement would have the biggest effect in absolute figures. Are we tackling these problems with a great degree of intensity? We are engineers. Perhaps it would be a little different if we were psychologists or doctors or educationists or some other type of scientist of this kind. I think we are biasing ourselves on driver training as it has evolved traditionally, where drivers are taught to attend to certain road signs and all that sort of thing. I would really like to suggest that research into training should be increased and that a hundred thousand pounds, or some such sum, should be devoted to a study of how such a driving school should be organized to reach the ordinary person and to stimulate common sense.

There's one more point I'd like to raise. If you train a driver to overcome a dangerous situation, then he remembers it for a time but, thereafter, I'm afraid he forgets the technique. We should teach drivers to recognize dangerous situations. I am convinced that many drivers are totally unaware of danger and they rush headlong into a trap which they could avoid if they realized that this was what they were doing. There are a lot of them who are simply in a hurry to get home after work and it's not their intention to drive dangerously.

There might, perhaps, be methods which could be accepted, and I would suggest that we could have very good results by doing research along these lines.

DR. FIALA (Chairman):

Thank you very much, Dr. Enke. It is true, the cost-effectiveness figure of zero point seven for this measure and the Niklas compilation for West Germany is very high. Zero point seven is about the same as equipping all vehicles with two safety belts with use quota of thirty percent, or the cost effectiveness is zero point zero four. Somewhere near about one-twentieth of the measure of driver education.

Perhaps this should be made more explicit. I should explain precisely what we mean. This is not a better driving school or stricter driving test. It is continuous training for all who hold a driving license or all who make use of a driving license. This figure, zero point seven, I must add, is one that I cannot judge myself. It's taken from U.S. investigations and there must be psychologists at the back of this figure. I, myself, am not in a position to judge. Let me just say that I consider it as an uncertain figure – without wishing to insult the psychologist. On the other hand, the usefulness of this measure is very large, much larger than that of any other individual measure. However, costs are enormously high, nearly seven billion German marks a year, that is about five percent of the national budget. Do you think our government would be willing to spend this amount of money? Where would the buildings and teachers be found in the near future? These are questions which, of course, immediately come to mind in connection with this measure. You are aware, no doubt, that first one should take a very careful look at the basis of all this, but this is not the responsibility of the automobile industry.

MR. CHILLON (France):

I'd like to give some sort of reply to the question that was asked a moment ago on the effectiveness to be expected from an objective improvement in safety conditions. This is often questioned because of the behavior of the user. There is no a priori reason why an improvement in qualities of the car, or of the road itself, should not lead to a systematic improvement in safety, since drivers can use the improvement either to improve security or to improve the acutal performance. It has been shown that in certain cases, for instance, if you get rid of dark areas, the safety increase has been considerable. Again, by improving road-holding qualities, you have drivers who go faster. Then, if they do have an accident, and there may not be more accidents, they are more serious. In these conditions I would merely say that one has to take into account the fact that a vehicle exists to serve and to serve well. Therefore, in the balance sheet you have to introduce the quality of the service and the speed comes into this. If one does not take this into account, you will have only a partial view of the problem, and it's also how satisfied the driver is to increase safety. I think one has to introduce other parameters apart from the road and the vehicle which relate to the driver – which might be as Dr. Enke was saying, parameters of instruction, and there I think there's a great deal to be done. What might help could be speed restriction or improving signals on the road, but this is not all. There's also the question of the driver.

DR. FIALA (Chairman):

I think this is a very important remark. If nighttime driving speed increases as a result of having better information for the driver, that's positive. It's not negative. The other question is whether the accident rate goes up. We want to arrive at the point where we have the benefit of higher transportation speed without the losses of more accidents. That's a very crucial thing.

SECTION 3

TECHNICAL SEMINARS

PART SIX – APPLICATION OF RESEARCH RESULTS TO PRODUCTION CARS AND FUTURE ESV ACTIVITIES

Chairman: LAWRENCE R. SCHNEIDER, Chief Council
National Highway Traffic Safety Administration
United States Department of Transportation

Themes: Application of Research Results to Standards; Future Activities

THE RESEARCH SAFETY VEHICLE PROGRAM AND U.S. MOTOR VEHICLE SAFETY STANDARDS

DR. JAMES B. GREGORY
Administrator
National Highway Traffic Safety Adminstration

It is a pleasure for me to present the United States position on the application of automotive safety research results to the development of motor vehicle safety standards.

As this group knows, the National Highway Traffic Safety Administration began in 1966, when Congress enacted the National Traffic and Motor Vehicle Safety Act and the Highway Safety Act. This legislation charged the Department of Transportation with the responsibility of reducing the number of traffic fatalities in the United States and authorized it to set standards to accomplish this end. Recognizing that the automobile-driver-environment system is a complex one that can easily defy ready analysis, these Acts also authorized the National Highway Traffic Safety Administration to perform such research as necessary to support the rulemaking activity. Since our safety standards apply to all cars sold in the United States, I would like to review our problem with you and indicate some of the progress we are making.

In 1966, some 53,000 people were killed in traffic accidents in the United States. The fatality rate that year was over 5.7 deaths per hundred million miles and rising. We calculate that if that surging rise of the early 1960s had continued, close to 78,000 persons would have died in traffic accidents during 1972, or about 36 percent more than the nearly 57,000 deaths that did occur that year. In addition to the fatalities, approximately four million people were injured. More than two million of those were disabling injuries.

Our figures for 1973 indicate total fatalities were 56,000, down over 1,000 from 1972, and down almost 2,000 from our projected figure for the year. This level of death on our streets and highways is still appalling, but it is noteworthy that the fatality rate in 1973 was on the order of 4.3, the lowest in our history and about 22 percent lower than the fatality rate in 1966, seven years ago, when our programs were started. This was accomplished in the face of a significant increase in the number of vehicles, licensed drivers, and miles driven on our nation's highways. Comparing 1973 with 1972, alone, the number of vehicles increased by about 5 percent, licensed drivers increased by 3.5 percent and miles traveled increased 3 percent. The annual increase in miles traveled, we have typically seen, was almost certainly reduced to some extent in 1973 by the energy situation which existed during the last two months of the year.

We experienced a sudden reduction in fatalities in November of 1973. This reduction can be attributed almost entirely in our judgment to lower speed limits and reduced highway travel resulting from the energy shortage. Figures for January and February, 1974, showed more than a 20 percent reduction in the numbers of fatalities compared with January and February a year ago. We are also finding that the fatality rate is continuing to go down. Now, I am the last one to say that, as a result of the energy problems, we are enjoying all of today's constraints on driving. I hasten to add, however, that we are continuing to be very much aware of the need to conserve our energy resources, and I assure you that this problem, as well as others I will discuss later, are being considered in our rulemaking activities.

The energy shortage can also have a negative effect on our safety problem. The shortage and the increase in the price of motor vehicle fuel have resulted in an increasing demand for smaller, more efficient passenger vehicles. Based on recent estimates, the number of subcompact cars registered annually is expected to increase on the order of 50 percent by 1976, whereas, the heaviest class of passenger cars will undoubtedly decrease by a percent or two.

All passenger cars are expected to increase by perhaps 10 percent. Our projections say that a potential 2,000 additional two-car collision fatalities could occur nationwide in 1976, if the vehicle mix

changes as expected. This figure may, of course, be reduced with continued lower speeds and reduced driving.

The indicated increase in fatalities is primarily due to the weight disparity of large and small cars involved in multivehicle collisions. That is, a small vehicle experiences a more severe crash than a large vehicle when the two collide. The small vehicle must also be looked on as having an inherent disadvantage because of the shorter distances available to manage the crash energy. We must not, however, lose sight of the fact that for the foreseeable future, we will have vehicles with many different sizes and weights on our highways, and that these vehicles must be compatible from a safety standpoint.

We began to study problems associated with the vehicle mix in 1968. Many of you may have seen data from our large car/small car baseline tests and from our unmodified large car/modified small car tests, conducted in 1969 and 1970. We have continued to study the problems associated with the vehicle mix, addressing vehicles in the 2,000 to 6,000 pound weight range. Our studies have been principally analytical in nature.

We believe we have gone about as far as it is reasonable to go using analytical procedures and techniques. Some of you have followed our progress in this area. Information on our studies will be presented at the Third International Congress on Automotive Safety in San Francisco next month, and reports will be available at that time. These analyses generally define the structural collapse characteristics of the different weight vehicles found in our vehicle mix. Vehicle aggressiveness and restraint system compatibility were the prime considerations. The studies are obviously quite complex, and many assumptions had to be made along their course.

Phase I of our RSV program will aid us greatly in further addressing the problem of vehicle compatibility and will bring together a variety of expert independent viewpoints on the level of safety that, based on current and anticipated technology, can be built into the passenger cars of the 1980s as we visualize them today.

I want to make it clear that we intend the relationship between the RSV program and rulemaking to be a direct one. Indeed, it is intended in large part to generate data relating to the recently issued Advance Notice of Proposed Rulemaking on Federal Motor Vehicle Safety Standard No. 208 requiring 45 or 50 mph crash protection.

We have many parallel programs, also, in direct support of Standard No. 208, such as Restraint Systems Development, Barrier Development for Compliance Testing, Biomechanics, or Human Tolerance Data, etc. In a very real sense these NHTSA programs can be considered an integral part of the RSV program as well.

We are very interested to see what ideas will be generated in RSV -- Phase I. Some of our contractors may suggest that we proceed by modifying existing systems or vehicles rather than designing and fabricating all new hardware; some may suggest we have one or more parallel programs relating to vehicle design, including addressing the pedestrian problem. We want to keep you informed, and we want to receive your comments as we proceed.

At a later point, either this fall or early in 1975, NHTSA will hold a public meeting to outline our future vehicle rulemaking plans from a multiyear standpoint. All interested parties will be invited to attend and participate in this meeting. As we approach difficult decisions, we feel that effective communications with other governments and industries are more important than ever. We recognize that, in order for us to achieve our goals, there must be proper planning by industry.

We discussed the energy shortage earlier. We must also consider the environment and the economy. All three areas are factors in the future design of vehicles, and we want to keep them in mind as RSV proceeds. In discussing safety, energy, environment and economy, I have called this concept the "S3E," and I would be very interested to receive your thoughts about such an approach.

I believe we would all agree that there are many problems in many areas of vehicle safety that we need to address. Some will require large expenditures of resources. I urge everyone to participate to the extent their resources permit. I assure you that Secretary Brinegar is vitally interested in our International Vehicle Safety Program and that he looks forward to your participation.

APPLICATION TO REGULATIONS OF STUDIES MADE PURSUANT TO THE ESV PROGRAM

A. OSSELET
Chief Engineer
Mines Department

INTRODUCTION

Since 1970, the French Government has been engaged, within the purview of the ESV program, in a programmed, thematic vehicle safety campaign (ATP), intended to stimulate and sustain the efforts of automobile and equipment manufacturers to produce vehicle subassemblies in order to determine the industry capabilities to improve vehicles (especially secondary safety).

The French experience, as well as what we have learned in regard to the accomplishments of the various countries taking part in the ESV program, has shown that very appreciable improvements can be made. Nevertheless, the ESV program could not be self-contained and it has been necessary, concurrently, to undertake thorough accident studies and to increase our knowledge of biomechanics. It is necessary to make sure that the new requirements which the vehicles must meet will entail a reduction in the number of deaths and injuries such that they will exhibit the best cost/benefit ratio and that such a ratio will attain a level acceptable to the community. It is for this reason that, to date, France has not deemed it realistic to advance concrete proposals as to future regulations of passenger cars. It considers that it is now possible to lay down the principal rules applicable in 1980, which are to have a substantial impact on automobile manufacture.

Important work for unification of the regulations of the various European countries, centered on seeking out of solutions leading to greater safety, has been done within the WP 29, which is under the authority of the Economic Commission for Europe (Geneva) and within the European Economic Community (Brussels). This standardization work has not yet been completed, but it can be stated that the major part of it has already been done – at least as concerns the countries belonging to the European Economic Community.

More recently, the EEVC has facilitated identification of the common positions of six European countries in pursuance of studies carried out in those countries, either under the ESV program or in connection with accident or biomechanical surveys.

France, for its part, fully accepts the recommendations made by Working Group No. 2 of the EEVC, especially as concerns impact tests and restraint devices. It feels that those recommendations must be implemented relatively quickly, so that, taking account of the efforts which will have to be made by the automobile industry, as large part of the European production will, in 1980, be in compliance with the contemplated new regulations. We propose to discuss here the principal problems presented and the solutions which could be found.

CONTENT OF THE NEW REGULATIONS

As stated in the report of the EEVC, we do not have sufficient information now to formulate the complete content of the new regulations. Nevertheless, a complete list of the measures which can be given consideration at this time for application in 1980 must be drawn up very promptly. Among such measures, at least the following must be selected:

- Passenger cars must pass the front end impact, lateral impact, rear end impact and rollover tests described in broad outline in the EEVC report.
- Vehicles will have to be equipped with seat belts for all of the passengers (or with some other restraint device of equivalent effectiveness), with the front-seat belts being automatically adjustable.

PREPARATION OF THE NEW REGULATIONS

In view of the commitments undertaken by France, the new regulations will have to have been

discussed with her fellow members of the European Economic Community and form the subject of Community directives. It is, accordingly, eminently desirable for this question to be placed as promptly as possible on the EEC agenda and that it be discussed within the Economic Commission for Europe (WP 29) so as to form, if possible, the broadest consensus at the European and world levels, with a view to eventual adoption of the same regulations in the countries in which European-size cars are in use.

A number of questions remain in abeyance in the formulation of the tests as given by the CEVE. These questions are of various types:

1. First of all, there are questions of principle relative to the tests; for example, is the rollover test to be static or dynamic?
2. Then, as always, the *modus operandi* of the tests should be laid down with precision.
3. Finally, it is provided that observance of the limits of biomechanical tolerance is one of the performance standards to be satisfied.

The problems in categories 1 and 2 are of the same nature as those which have already been discussed and resolved within the framework of the WP 29 and the European Economic Community. The finding of a precise European formulation as to these two points should present no particular difficulty.

The matter is more complex concerning the limits of biomechanical tolerance and the injury criteria, as we have become accustomed to call them, and warrant special examination.

INJURY CRITERIA

As concerns the injury criteria, we are faced with a substantial range of proposals relative to the various parts of the human body. First, these proposals are not of equal importance; and second, some of them which concern important points are subject to debate or involve tests in which value and repeatability are arguable in view of the available test equipment or test equipment which is scheduled to be available in the near future.

It does not seem possible to adopt injury criteria for all of the parts of the body, because the vehicle approval tests would involve material difficulties out of proportion to the safety improvements which would result from adoption. This is especially true if the results obtained are valid under the conventional test conditions which are selected but are not necessarily so in the infinity of situations which are encountered in actual accidents. It follows that, in the final analysis, we are obliged to adopt only certain criteria which permit the forming of an opinion as to the quality of the structure and interior outfitting of the vehicle and of the restraint devices used.

In these circumstances, there should be laid down three, or at the outside, four injury criteria for the front end and lateral impacts, such criteria not necessarily being the same in both cases.

It, accordingly, seems absolutely necessary for meetings of European biomechanics experts to be held to establish these criteria promptly and for those experts to be asked to prepare the proposals which can serve as a basis for the future European regulations. The question of determining the framework in which such meetings of experts are to be held remains to be answered. It would be highly desirable for such preparatory work to be done within the framework of the Working Group No. 3 of the EEVC. The group would be reactivated and the representatives of the automobile industry who have worked on these questions would be invited to take part in an advisory capacity. In view of the urgency of this problem, this group would be asked to make proposals by July 1, 1975.

PROCEDURES FOR APPLICATION OF THE NEW REGULATIONS

The scheduled date for effectiveness of the new regulations could be October 1, 1980, and it would be desirable for all passenger cars placed on the market from that date to be in compliance with the new requirements. Most of them could be applied without great difficulty to the vehicles manufactured before that date. On the other hand, it is highly probable that the vehicle models now in production and which will continue to be manufactured at October 1, 1980, will not be able to comply with certain new rules. Among such new rules which it will be difficult, if not impossible, for existing vehicles to comply with, are at least the rules applicable to the behavior of the vehicle in front end and lateral collisions. It is accordingly proposed that such new rules be applicable: to the new passenger car models placed on the market from January 1, 1979; and to all passenger car models marketed from October 1, 1983.

APPENDIX
IMPACT TESTS DESIGNED BY THE EEVC

PERFORMANCES TO BE REQUIRED FOR THE TESTS

1. Compliance with biomechanical tolerance limits.
2. No bursting open of doors during impact.

3. Possibility, after collision, of opening at least one door without using tools.
4. Possibility, after collision, of removing the complete dummies.
5. No fuel spillage and no fire.

FRONTAL IMPACT TEST

Test Procedure

An asymmetrical impact test has been considered more representative of real accident conditions than the usual head-on test. A choice will have to be made between two possibilities:

- Impact against barrier angled at 60° to the vehicle main axis.
- Offset impact against barrier with radiused edge; the impact will involve half of the vehicle front, on the steering wheel side.

Test Velocity

50 km/h.

Test Conditions

Vehicle in running order; two dummies (50th percentile, male) in the front outboard seating positions; restraint systems in the normal position and conditions of use, ready to react on the dummies.

Requirements to Be Met

As specified in items 1 to 5 above.

SIDE IMPACT TEST

Test Procedure

The vehicle to be controlled will be struck on its side by the front end of an identical vehicle; the relative velocity vector of the striking vehicle will be angled at 75° to the main axis of the struck one; when the collision occurs, the H point of the driver's position in the struck vehicle will be in the main vertical plane of the striking one.

Test Velocity

The relative velocity of the striking vehicle to the struck one will be 40 km/h.

Test Conditions

As specified for the frontal impact but for the seating positions of the two dummies, which are here adjacent to the struck side.

Requirements to Be Met

As specified in items 1, 2, 3 and 4 above.

ROLLOVER TEST

A choice will have to be made between a rollover test with two full rotations at an initial speed of 50 km/h, and a dynamic impact test.

REAR IMPACT TEST

The stationary test vehicle will be struck from the rear by a 1,100 kg barrier, moving at a speed of 35 km/h.

The requirements to be met are specified in items 2, 3, 4 and 5 above.

CONTRIBUTION OF THE GERMAN MOTOR INDUSTRY

DR. G. BRENKEN
Principal Manager
Verband der Automobilindustrie e.V.

After the publication of the first specification for the design of Experimental Safety Vehicles (ESV) and after the First ESV Conference in Paris in January, 1971, the worldwide exchange of knowledge on ESV development organized by NHTSA permitted the acquisition of valuable knowledge in a relatively short time. This knowledge in the field of vehicle safety can be considered common knowledge today.

At the Fourth ESV Conference in Kyoto – and especially in this Fifth ESV Conference – many contributions have clearly shown that a certain completion of the design of complete ESV has been attained.

The main target of future tasks should be concentrated on partial research in the following fields:

- Intensification of biomechanical research.
- Improvement of cost-benefit analyses for traffic safety measures.
- More detailed accident research and systematic interpretation of accident reports using internationally unified criteria which have to be developed.
- Improvement of pedestrians' protection.

This list does not claim to be complete.

Furthermore, the harmonization of the international legislation in the field of vehicle safety should be accelerated to serve as a basis for the specification which will be prepared for Phase I of the RSV project.

As long as results of partial research tasks are not available, no data concerning impact speeds should be publicly given or fixed by law. They would generate confusion, and at the present time they are not yet based on technically and scientifically founded results. Values of impact speeds should only be given again when all research work is interpreted and when differentiated specifications have been prepared for different vehicle categories.

According to our experience, these research tasks will take several years. The accident history of all roads in our world does not give us any more time. Immediate measures must be taken worldwide.

From this understanding the German Motor Industry proposes that – from this conference – all governments are urged to give orders by law to use the safety belts in this present year. Conventional barriers should be dropped in doing this.

At the present time, the safety belt is the publicly best known and technically best matured restraint system for the protection of vehicle occupants. It has the further advantage of lowest cost with highest efficiency.

The safety belt reduces accident consequences considerably in 90% of all cases, and drastically lowers the number of fatal accidents, as well as the number of injuries. This has been clearly demonstrated by the Australian experience and by the present status of accident report interpretation.

A further advantage of this proposal would be the gain of time for research in the partial research fields.

All politicians of all countries are summoned for the realization of this proposal. The common result of this conference of government representatives, researchers and experts from the industry, in cooperation with the NHTSA, should be the important advice given with this proposal. The proposal should be transmitted to press, radio and television.

TRANSLATING THE RESULTS OF THE EXPERIMENTAL SAFETY VEHICLE PROGRAM INTO PRODUCTION CARS

R. J. A. SHARP
Department of the Environment

As Mr. Mulley, the Minister for Transport, said when he opened this conference on Tuesday, the United Kingdom has been anxious from the start of the ESV program to obtain practical results and to obtain them reasonably quickly. This meant spending some time initially in defining the target areas for research as carefully as possible. For without defining with some precision the problem or problems to be solved, a great deal of wasted effort and expense could have resulted. The problems were identified by the, now, very familiar method of analysis of accident data. This enabled the Transport and Road Research Laboratory to place research contracts with industry on a collaborative and cost sharing basis in such a way that effort was concentrated on improving features of existing vehicles without adding greatly to cost, weight or fuel consumption and without making the interior unacceptably claustrophobic.

This approach did not lack critics, especially within the U.K. Nevertheless, we were considerably encouraged when, following the Fourth ESV Conference in Kyoto in 1973, some of our partners in the European Experimental Vehicles Committee suggested that the time was ripe to take stock of the mass of data accumulated as a result of ESV work and to give it as practical an orientation as possible. It seemed appropriate to begin this task at a European level because of the similiarity of the problems faced in the various European countries and also because of the long established collaboration in harmonizing vehicle regulations in the Economic Commission for Europe and the European Economic Community. A vary valuable feature of the working groups set up by EEVC to carry out this stocktaking was the positive contribution made by industry in supplying new data, both about accidents and about test procedures.

The outcome is available in the EEVC Report already presented to the conference. As the report says, the conclusions do not represent the exact points of view of all participants. It may therefore be helpful to indicate the U.K. position more precisely and to offer a few general comments. Without necessarily being committed to all the details, the U.K. government generally endorses the EEVC Report as a fair summary of the state of progress on vehicle safety research, particularly as regards occupant protection, and as indicating the possibilities for applying a new generation of regulations for impact tests for passenger cars produced in the early 1980s. As far as one can make a preliminary judgment, the papers presented to the conference, and the European and Japanese safety vehicles displayed at Crowthorne, tend very much to confirm the EEVC conclusions. In a word, we can make significant improvements in the safety of typical European production cars within the next few years. This is certainly our objective.

A few more specific comments may be in order. Accident investigation and analysis is vital and must be a continuing process. Nevertheless, we need to concentrate in the short-term on pulling together of the mass of data already at hand – and to discover, if there are differences, whether these represent different real conditions or different methodologies.

There has been much mention of cost-benefit evaluation. This is right. Cost-benefit is, however, a good servant but a bad master. Its value is in helping us to rank available measures in order of priority – not in calculating to the last penny whether so many lives are or are not worth saving. And it is, in my view, a blind alley to try to use the techniques in deciding how much should be spent on road safety as compared with hospitals or old peoples' homes or art galleries. This is because, the wider you spread the field of comparison, the more unquantifiable factors enter in and answers become elusive. When we approach vehicle regulations, none of us – in government or industry – should be under any illusion that when all the analysis and calculation has been done – and it should be done – the element of judgment and intuition has to come in and decisions made accordingly. We cannot hand over rule making to computers.

Turning to the impact tests themselves, our assessment is that the four listed in the EEVC Report, none of which is precisely defined, represent the likely shape of the package. While it might be tempting to require all the possible alternatives within each main type of test, it seems highly doubtful whether the results would yield much extra benefit or that the additional trouble and expense of multifarious tests would be justified. The need, therefore, is to find quickly which are the most representative and repeatable tests – especially for front and side impact – and to agree on the speeds to be used. In this connection I had better say that we in the U.K. are not sure whether 60 or 55 kph rather than 50 kph might be appropriate for frontal impact. But we can say with a fair degree of certainty that we

see little benefit in considering any speed higher than 60 kph for this test. Similarly it would be worth examining the pros and cons of slight adjustments in the speeds suggested for the other tests. As far as side impact is concerned, we should like to see immediate international consideration of the possibility of a standardizing bumper and sill heights at a level which will minimize injury to predestrians. The BLMC/TRRL contract has demonstrated the substantial safety benefits this will achieve for virtually no weight penalty.

We must also emphasize that this fresh generation of regulations, using anthropomorphic dummies in crash tests, rests on a clear assumption that seat belts will be worn. So far as unbelted car occupants are concerned, it is simply not possible to envisage the safety improvements proposed without paying enormous and unacceptable penalties. This link with seat belt wearing may not be generally understood and it is perhaps a factor to be emphasized in future discussions about the compulsory wearing of seat belts. In any case it is vital that work on improving the effectiveness of belts in reducing injuries and making them acceptable to wearers of all shapes and sizes continues apace.

We entirely endorse the emphasis the EEVC Report places on the need for urgent agreement on human tolerance levels in road accidents. Clearly this means concentrating on establishing broadly agreed criteria for those areas of the body most at risk in accidents producing severe injury, for example the head, the chest, the adbomen and the hips.

Finally, a word about procedure. The member states of the EEC are committed to agreeing to common vehicle standards in Brussels but they normally lay the basis for such standards in the wider European forum of the ECE in Geneva. It would be desirable to settle a timetable for the implementation of the EEVC proposals in Brussels and Geneva at an early date. Nevertheless, we believe it would also be advantageous, both in the light of the growing similarity of the problems we face and the worldwide nature of motor vehicle exchanges, if the United States and Japan felt able to join in some way the dialogue on these matters. This might be done in the sort of working groups set up by EEVC to study different aspects of the problems – the groups include government, industry and academic experts. A further possibility would be plenary meetings of government representatives of all the countries concerned, where common ground on the application of results to standards might be established. I think we have to say, frankly, that one cannot do this sort of work in big ESV Conferences of 400-500 people, however valuable they may be for other reasons. Since our expert manpower resources are scarce and the need to reduce the toll of accidents is urgent, our strong preference is to give priority to the more restricted type of gathering, which has proved highly effective in the European context.

APPLICATION OF RESULTS TO STANDARDS – FUTURE ACTIVITIES

DR. POCCI
Ministry of Transport

The position of the Italian Government has already been clearly expressed in the presentation of a few days ago. We would like, however, to thank the organizers of this conference for this opportunity to be more specific on certain points. Particularly, we would like to illustrate our technical standards and what our plans are for the future.

The Italian technical standards for vehicles are fairly complete. Since 1959, our traffic code, together with other technical regulations, have required vehicles to be considerably safer than pre-1959 cars. Nevertheless, Italy recognized that as regulations became more and more severe, they would be put at an international level. Therefore, we agree with the presentation of our French colleagues and in part with what our U.K. colleagues have said. We accept the importance of what is being done within the framework of this ESV program. We are also very interested in what is being done within the EEVC.

We believe this work can give us adequate guidelines for the development of technical regulations along international lines and, therefore, help those who are working toward this end. We're thinking of the work that is being done in the ECE U.N. organization and what is going on in the CEE, first in Geneva, and second, of course, in Brussels.

As for WP-29, they have already drawn up 40 technical regulations. The oldest ones deal with active safety, followed by the legitimate requirements for passive safety. Therefore, certain regulations which are concerned with structures have been applied. Now these regulations, when we know the general orientation that EEVC will give, can be improved and integrated. We may even have new regulations to take into account the new requirements. Of course, this will call for a generalization, because what you can expect from an experimental vehicle you cannot expect from production vehicles. Therefore, our administration, although it is aware of the need to create safer vehicles, cannot ignore the requirement that these vehicles must be made available at prices the general public can afford.

We also think that these measures should be extended internationally to all countries, even to those which, for economic reasons or because they are very young countries, have been unable to share in the work. The problem on the international level was taken up by the ECE in November 1970 within its Transport Subcommittee. This subcommittee stated that it was prepared to accept research results to improve the regulations which already existed. These regulations affect the common standards in the CEE which embraces nine countries, most of which are represented here. The regulations of the ECE are considered within the framework of some 28 countries.

There already exist basic documents, such as the 1968 Vienna Convention, which provide that in the light of certain requirements vehicles must be subjected to tests and control.

I repeat, one has to use these studies in order to improve the general production. Nevertheless – this has already been said but I would like to go into a bit more detail – the task has been made harder because you only have to think about a vehicle having either active or passive safety consideration apart from the compromise of the contrast between passive requirements and active requirements, when here a compromise is necessary. You add to this new requirements which the vehicles must have such as: production and use costs neither excessively expensive nor expensive to operate and this, of course, as a result of the recent energy crisis. Then the problem that the vehicle should not pollute the environment either in terms of exhaust or noise. All this is quite hard because the trade offs are very difficult ones.

We are now in England. I would recall Aldous Huxley who said it's easier to be a good citizen than a saint or a hero. Therefore, the experimental safety vehicle was a hero, saint. The safety vehicle as we see it today is a good citizen, a good father, that is more difficult. The view of the Italian Ministry of Transport which we have said is to continue on the studies within the EEVC for the safer vehicles.

Unlike what we heard from one or two speakers, there are lots of results which could already be exploited and which will enable us to move towards certain solutions and certain standards without, of course, hurrying, so as to enable the manufacturers to fall into line. The second thing is that once these facts are available, we must utilize them within the international bodies which already exist, and avoid setting up others. As far as the national field, it would seem that we preach to the world but behave badly ourselves. There is a recent law in Italy, however, that applies all the directives of the CEE to national law and imposes certain obligations which previously did not exist. Moreover, there is a draft law which gives the framework about the requirements of passive

safety; as well as tests of lateral or front impacts, head supports, seat belts, etc. Of course, there's the active part, but here – luckily – we're doing quite well because this field is well controlled in Italy. I think this can be quite frankly said. This law will also see that these requirements are translated into technical regulations, which will be absorbed into the Italian regulation of international standards.

NORIYOSHI UNO
Director of Engineering Division
Motor Vehicles Department
Road Transport Bureau

INTRODUCTION

It gives me a great pleasure to present a report at this conference on the application of results from the ESV program to safety standards and the future activities of Japan.

THE PRESENT ESV PROGRAM OF JAPAN

The ESV program of Japan has been proceeding, based on the memorandum concerning cooperation in the development of ESV between Japan and the United States, which was formulated in 1970 in response to the request of the United States. While the specification for the 4,000 pound class ESV was set for the United States, the one for the 2,000 pound class was set for Japan, and after establishing the specification, a notice was sent out immediately to all automobile manufacturers in Japan to participate in this program. Three manufacturers: Toyota, Nissan and Honda, agreed to participate in this program in May, 1971. For the ESV development, Toyota built 100 units of trial vehicles with an outlay of 2,000 million yen; Nissan built 250 units of trial and remodeled vehicles with an outlay of 2,000 million yen; and Honda has built 40 odd units of trial and remodeled vehicles with an outlay of 1,000 million yen. After going through the researches and developments based on the Japanese ESV specifications, Toyota and Nissan completed ten final trial vehicles each of which were delivered to the Japanese Government in September, 1973.

At the request of the Government, the Japan Automobile Research Institute, Inc. (JARI) undertook evaluation tests of these vehicles on their visibility, steering, handling, braking, crushworthiness, and rollover performance since the fall of last year, and the results were reported by the JARI earlier in this conference.

PRESENT MOTOR VEHICLE SAFETY STANDARDS OF JAPAN

Before touching on the future motor vehicle safety standards, I would like to speak about the present motor vehicle safety standards of Japan.

The safety standards for the structure and equipment of vehicles are provided in the Road Vehicles Act and the Safety Regulations for Road Vehicles enacted in 1951, which also includes the standards for exhaust gas and noise.

However, in order to meet the changes in traffic environment and the actual usage of vehicles, the safety regulations have been amended thirty times since 1951. In the last several years these regulations have been amended frequently to meet the requirements of the new ages that are caused by the rapid increase of the vehicle population, the aggravated traffic congestions, the environmental pollution and the speedier traffic with the extension of expressways and highways.

The latest amendment was made according to the first target set by the "Program Plan for Reinforcing Motor Vehicle Safety Standards." I will speak in detail on this amendment, later. In this amendment twenty items are included, for example, conspicuity improvement of the clearance lamps, tail lamps, reflex reflectors, stop lamps and direction indicators; installation of a dual brake system, more seat belts, more head restraints, and collapsible outside mirrors for passenger cars.

These regulations are the standards applicable to:

1. Design of automobile manufacturers;
2. Maintenance at garages;
3. Periodical inspection compulsory to every motor vehicle;
4. Inspection of motor vehicle users.

RESULTS FROM THE ESV PROGRAM AND THE FUTURE SAFETY STANDARDS

To formulate ideal and effective standards for motor vehicles, there are three important things – one of them is to establish plans on a long-range basis

in an integrated form. To implement this conception, technical measures for safety of motor vehicles were referred to the Council for Transport Techniques, whose membership is composed of men of experience and learning, specialists in the fields of medicine and engineering, and representatives from automobile manufacturers and allied organizations. The long-range program was recommended to the Minister for Transport by this Council. The works on the future safety standards in Japan are to be conducted based on this long-range program (which was presented by the Japanese Government authorities at the Fourth International Technical Conference on ESV held last year in Kyoto, Japan). In this program sixty-three items have been submitted to be achieved from 1972 to 1977.

This program is broadly divided into three parts:

1. Measures to avoid accidents caused by the vehicle structure and function by reducing factors connected with accidents;
2. Measures to reduce casualties of occupants and pedestrians as far as possible;
3. Measures to minimize fire hazards.

The second important thing in formulating the safety standards is to clarify the correlation between the traffic accident and the vehicle structure. As there were not sufficient data to clarify this relation in the past, a committee was organized in the Ministry of Transport in 1973 to collect data and analyze the correlation between traffic accidents and the vehicle structure.

Another important thing in further strengthening the safety standards is that they should not be limited to what can be achieved under the existing industrial technology, and that they should be established with a goal representing social needs and in the direction of promoting necessary, technical development. In this respect, the development of the ESVs has a significant meaning, and I am happy to note that the developed ESVs met the early target. I also would like to commend those who developed them and to recognize that some of the developed techniques from the ESV program, for example: the air pressure warning device for the tire and the concentrated warning device have been applied to the market vehicles in Japan.

FUTURE ACTIVITIES OF JAPAN

I would like to speak about some problems of the ESV program and future activities of Japan.

The first problem is the aggressiveness of the ESVs. It is necessary to study their compatibility in the traffic with other types of vehicles, motorcycles and pedestrians. The second problem is the cost-benefit analysis of the ESVs. The assessment of value is important, but very difficult. The third problem is the accident analysis. As to this, it is necessary to reflect different conditions existing in all countries.

The fourth problem is the environmental protection. In Japan, recently motor vehicles are considered a main factor causing air pollution. Public criticism is rising against the exhaust gas emission and enforcement of a rigid emission control is scheduled for 1975.

The fifth problem is the energy resources crisis which cropped up last year.

In addition to these, there are a number of problems to be resolved, such as the improvement of the environment for driving from the standpoint of human engineering, and the pursuit of the optimum size of vehicles related to the traffic jams and parking issues in the cities. Because of these problems, it is unreasonable to connect the ESVs directly with the actual safety standards and, due to the present conference, I have a deep interest in the future of the RSV program.

DR. ULRICH W. SEIFFERT (Federal Republic of Germany):

Isn't there a conflict in your RSV work and to future rulemaking which is just now going on? For example, your request in Docket 74-15 for impact protection at 45 and 50 mph.

MR. SCHNEIDER (United States):

No, I see no conflict whatsoever in the rulemaking proposal. It is an advanced notice of proposed rulemaking, which for those of you that have been involved in our rulemaking over the years know, is quite different from a notice of proposed rulemaking which is a further step along the line to the creation of a safety standard. This procedure attempts to elicit the views, attitudes, data, knowledge and criticisms, if you will, of affected and interested parties on the wisdom of having 45 or 50 mph frontal crash protection in a timeframe quite distant from now insofar as the rulemakers are concerned. As I recall, the proposal talks in terms of September 1, 1980.

DR. SEIFFERT (Federal Republic of Germany):

Here in Europe, specifically during this Fifth International ESV Conference, it was shown by the European manufacturers that they prefer safety belts instead of air bags. Several manufacturers have also used active belt systems. The Federal Republic of Germany has decided that there should be a mandatory seat belt law. It is my understanding that this would be possible in the U.S. as well. Is it the intention of the NHTSA to go further in this direction or do they intend to stay with the request for passive restraints?

MR. SCHNEIDER (United States):

Our policy is to seek an enactment by the States in the U.S. for mandatory seat belt use laws. You may recall that about a year or so ago, the Secretary of Transportation asked the Congress to pass a resolution urging the States to enact mandatory seat belt use laws. That request resulted in a law which set up a program whereby the States were given an incentive. They would get a sum of money up to 25 percent of the money that the Federal Government gives them for their regular highway safety programs if they enacted seat belt use laws patterned after the model law which we in NHTSA developed. It is unfortunate, in my judgment and the judgment of the Department of Transportation, that other than the Commonwealth of Puerto Rico, no State has as yet enacted a mandatory use law. Several States came very close during their last legislative sessions. There have been a total of 22 or more States that have introduced in their respective legislatures bills which would call for the mandatory use of seat belts. We fully recognize that this is, as some speakers have said, the most cost-effective way of dealing with the problem. We fully recognize the Australian experience and it has taught us a great deal and has led in part to our seeking mandatory seat belt use laws. But we believe that the two dovetail together and that as of today, our policy is the rulemaking most recently enacted on 208. Not the advanced notice – but the notice – to still require passive protection in frontal, angular, lateral, and if unable to meet rollover requirements passively, a manufacturer could choose to use a sequential warning lap belt. We are firmly committed to proceeding in both directions, Dr. Seiffert, and believe that they are compatible.

SECTION 4

CONCLUSION

CLOSING REMARKS

ALEXANDER SILVERLEAF
Director
Transport and Road Research Laboratory
United Kingdom Department of the Environment

DR. GENE G. MANNELLA
Associate Administrator for Research and Development
National Highway Traffic Safety Administration
United States Department of Transportation

SECTION 4

CLOSING REMARKS

ALEXANDER SILVERLEAF
Director
Transport and Road Research Laboratory
United Kingdom Department of the Environment

I would like to make some concluding remarks on behalf of the host government. It has been a privilege and an honor for the United Kingdom's government to act as host for this important international event. We hope that the representatives of the United States government, the sponsors of this conference, and those of all other participating governments, as well as all those others who have attended, have been satisfied with the arrangements we have made. As you know, these arrangements here in London for the conference and for associated technical and social events, and at Crowthorne for the exhibition, have been organized jointly by the Transport and Road Research Laboratory of the United Kingdom, Department of the Environment, and by the Society of Motor Manufacturers and Traders of Great Britain. This is typical of the close, friendly and fruitful collaboration between government and industry which now characterizes the ESV program worldwide.

It is natural to expect the Fifth ESV Converence to mark a significant milestone in the progress of this international project which, whether or not it achieves its original aim, will have a major effect on the lives of many millions of road users throughout the world. We hope you share our feeling that this conference and the safety vehicle exhibition do indeed show that important developments, indeed encouraging developments, have occurred since the first conference little more than three years ago. Perhaps the most important aspect is that passenger cars which meet realistic safety criteria, particularly for occupant protection, can be produced so that they are generally acceptable to road users, to industry and to government. Further, it is clear that there is growing international agreement on the major problems to be solved and recognition that there are several different but acceptable ways of solving them. What is perhaps most encouraging is that there seems to be a steady move towards broadly similar solutions. All this indicates to us, and I hope to you all, a strong move toward realism. Perhaps I may now mention briefly some particular points.

First, the arrangements for the conference and the exhibition. Clearly, success depends very largely on the collaboration of all participants. And we, in the host country, would like to pay tribute here to the way in which everyone has collaborated. We are most grateful for the extremely helpful way in which everyone has worked together to overcome many awkward problems, both relating to the conference here and to the exhibition at Crowthorne. Technically, the conference has, we believe, shown a trend towards greater technical discussion – more technical discussion; more depth to the discussions exemplified by the seminars and technical sessions which have dominated these past two days. And in those technical sessions, particularly, and in general discussion, there has been, we feel, a much closer and more precisely defined identification of important technical issues, both those which have been solved to some extent and those which still remain to be solved. And, of course, on the technical front, as I have already said, the important emergence of development cars – not just research vehicles – which seem clearly suitable for production within the next few years.

If I may just say a word on procedure, without I hope, saying anything that cuts across the arrangements for the conference itself, which are determined, of course, by the sponsor country, it is clear from your discussions that the EEVC has emerged as a contributor to the conference by pooling European knowledge and experience. This, of course has been pointed out several times and is particularly relevant because of the growing trend, worldwide, toward smaller vehicles.

Secondly, a point has been made by several contributors this afternoon in particular, the scope

for meetings and, indeed, the need for meetings of relatively small groups of specialists similar to the EEVC, but on a world regional basis, to include the United States and Japan. If I may be permitted to say so, perhaps the United States government may consider this a useful basis on which to plan perhaps the next or following conferences in this series. Finally, the outputs of the meetings in particular are not only the discussions that you have held here, in this hall within the last few days but, of course, the personal contacts which you make, renew and strengthen. Then later the proceedings, which are published by the United States government. In the past they have been published with amazing speed following each conference and we very much hope and, if I may dare say so, expect that our American colleagues will maintain this remarkable record and produce the proceedings of this conference as quickly as they have in the past.

DR. GENE G. MANNELLA
Associate Administrator for Research and Development
National Highway Traffic Safety Administration
United States Department of Transportation

We are now concluding the Fifth International Technical Conference on Experimental Safety Vehicles. During the course of this meeting, much valuable technical information has been presented, several interesting proposals have been put forth and the cause of improved road and traffic safety has been well served. My hopes at the outset for a frank and open dialogue were certainly well founded.

On behalf of the American delegation, I would like to express our thanks to the host Government, Minister Mulley, the Society of Motor Manufacturers and Traders, the GKN Group, the Transport and Road Research Laboratory, and all others who contributed to such an impressive and memorable conference. I would also like to congratulate the Conference Services Limited, the organizers of the meeting, the lovely ladies who ran the Conference, and last but not least, our breathless translators.

In closing, I would like to make the following short statement. The challenge to all of us is clear, to find ways to build small, safe vehicles. Whether one approaches the problem from the S3E point of view of the United States, or perhaps the social ESV concept of the Japanese, the point we are making is a common one, we must provide the technology to produce a socially acceptable vehicle in a time of extreme concern about energy, inflation and ecology. The United States is committed to the concept that by creative application of technology we can indeed develop ways to solve the sometimes contradictory requirements of personal mobility, safety, energy conservation and ecology. We invite and urge you to continue to work with us and to share in the solutions of these problems. As has been stated earlier the U.S. RSV phase of the ESV Program is a clear reflection of our intensions.

I perhaps share the view of many of you when I note that the status reports are now largely technical in nature, since the time and location of the next conference are at this time still undecided, we will bear this in mind in communicating with you in the future.

In any event I feel strongly that this conference will continue in an evolutionary manner to move towards a systems oriented technical conference, with a somewhat reduced amount of time devoted to Government policy statements.

Ladies and Gentlemen, the Fifth International Technical Conference on Experimental Safety Vehicles is now adjourned.

Thank you for your attendance and participation.

☆ U.S. GOVERNMENT PRINTING OFFICE : 1975 O—567-692