A MODEL OF VEHICLE-FIXED BARRIER FRONTAL CRASH AND ITS APPLICATION IN THE ESTIMATION OF CRASH KINEMATICS

Zuolong, Wei
Hamid Reza, Krimi
Kjell Gunnar, Robbersmyr
University of Agder
Norway

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ABSTRACT

The study of vehicle crash process is of great importance in transportation safety. The crash pulses of vehicles during the fixed barrier impacts can reflect the crashworthiness of the vehicle structure. In this paper, a mathematical model of vehicle kinematics during the frontal crash is investigated. This work is based on the analysis of crash response signals and vehicle structure. The proposed model uses piecewise linear functions to describe the trend of crash impulse and ignores the residual oscillations. To study the model variance, the crashes in various speeds and a full car crash in complex condition are compared. At the end of paper, the crash performance of a vehicle crash is predicted according to the proposed model and therefore demonstrates its effectiveness and usability.

INTRODUCTION

Crashworthiness is one of the core topics in the passive safety of vehicles and plays an important role in the condition that the impact cannot be avoided. Generally, the analysis of crashworthiness is based on the related crash responses, i.e. the displacement, velocity and acceleration, of critical parts of a vehicle in full car crash tests. However, these tests are required appropriate facilities, one or more cars with measuring devices, experienced staff and a long time to prepare. It means they are complicated, expensive, long-lasting and therefore not easy to realize [1]. This is especially true in the early stage of vehicle design. Therefore, vehicle designers and researchers made a lot of effort to build numerical models to describe the crash processes.
Up to now, various technologies are used to model the vehicle crash. Typical crash models may be classified into three broad categories [2]: 1) Detailed nonlinear finite element models. These models have excellent performance in the estimation of structural crashworthiness. However, before these crash models could be used, they usually require the details of the vehicle structure and materials. This limits the use of FE models in the design process. 2) Multibody models and multibody based lumped parameter models. As FE models, the multibody models also suffer the complexity. Consequently, the multibody based lumped parameter models make a compromise between the accuracy and complexity. Most of these models consist of energy absorbing (EA) elements with masses connected to both ends [3]. Reference [4 and 5] are typical studies on the lumped parameter models. 3) Functional approximation or response surface models. The functional approximation method is widely used in academia and industry. And reference [6] provides an overview of its use in the research of crashworthiness. To achieve better approximation, some advanced technologies are also introduced in this area, such as wavelet [7] and neural network [8]. Most of these models focus on the crash response signals themselves and can hardly be related to the vehicle structure. So the physical meaning of these modes is not clear.
In the proceeding of the study, a piecewise linear model is proposed to represent the vehicle-rigid wall frontal crash. Compared to existing models, this model is developed based on the analysis of crash responses and therefore can reflect the performance of vehicle structures in crashes.
The rest part of this paper is organized as follows: In the next section, the crashworthiness structure is introduced firstly. Afterwards, the proposed model is proposed. In this section, the modelling procedures will be presented in detail and the influences of crash condition are also discussed. After that, an estimation of vehicle kinematics is given as an application of the proposed model. The conclusion goes finally.

VEHICLE STRUCTURE

Most of modern commercial cars have unibody construction, i.e. a single entity forms a car's body and frame. The vehicle body is usually made of steel or aluminum that is stamped with the appropriate cross members and everything is mounted directly to it. In this paper, only the frontal crash of unibody construction is studied.

To improve the crashworthiness, the vehicles are meticulously designed. As an integration, the vehicle structure would transmit the impact load in a proper way, i.e. load-carrying path. In addition, there are some weak components arranged as crumple zone. During the crash, the crumple zone will deform and absorb energy and ensure the rest parts of the vehicle are safe. In the local design, the material and shape of beams, shells and connectors are also considered to achieve better energy-absorbing performance.

Load-carrying Path and Crash Process

Generally, vehicles have a high relative velocity in the crashes. For this reason, the frontal structures always have enough space for the crashworthiness design. In many vehicles, there are three paths to transmit the impact load:

1) Path1: Accessories - Front bumper and Crash box - Front longitudinal beam-Engine - Firewall
2) Path2: Upper wing beam - A pillar - Guard beams of door;
3) Path3: Sub-frame - Sill beam

For most cases, the first path affords more than half of the total energy in crashes. Accordingly, the ideal crash process contains several stages:

1) Accessory crush. This is designed to protect the pedestrian. This stage is very short and nearly has no influence on the energy and velocity of vehicles.
2) Bumper crash and Crash box. The bumper is helpful in the low-speed crashes. Its ability for the energy absorbing is limited.
3) Longitudinal beam collapse. This part is responsible for crash compatibility, i.e. to protect the opposing vehicle in some extent and make the total lost lower in crashes.
4) Engine compression. Engine should be seen as a mass with limited deformation in crashes. In some high speed crashes, it will crush into the firewall and be compressed a little.
5) Firewall deformation. Firewall is much stronger than other components to protect the passenger cabin.

Crashworthy Structures

In most vehicles, longitudinal beams are designed as the crashworthy structures and employed for the energy absorbing. Although, the longitudinal beams can be various in terms of the material, section shape and processing technology, they follow the same requirement in design to optimise its performance:

1) No deformation in low speed crashes; Collapse progressively and absorbing the energy effectively in high speed crashes.
2) Having repeatable and reliable failure mode to ensure its performance in different crashes.

For this reason, the crashworthy structures of vehicles always experience a stables process of collapse during different crashes. In this process, the deformation follows a linear trend and have a series of oscillations.
To sum up, the crash processes of vehicles yield to an internal pattern, which is controlled by the load-carrying path and beam collapse mode. For this reason, a functional model is proposed to present the vehicle crash process.

CRASH MODEL

During a crash, the response signals contain several parts, which are corresponding to the crash stages. A piecewise model can be therefore identified by the accelerations and external force.

Piecewise Linear Model

There are some mathematical models of crash response in the literature, such as sine, triangular, and haversine [9]. However, none of them consider the vehicle structure and consequently are neither precise enough nor adjustable for different crash scenarios. To illustrate the proposed modelling process, NHTSA Test 5677, in which Yaris is crashing to a rigid wall in 56km/h, is employed for example.

Model structure The response of vehicle crashes can be recorded by the acceleration signals (crash impulses). Figure 1 shows the acceleration signal of the left rear seat during the crash and the proposed model structure.

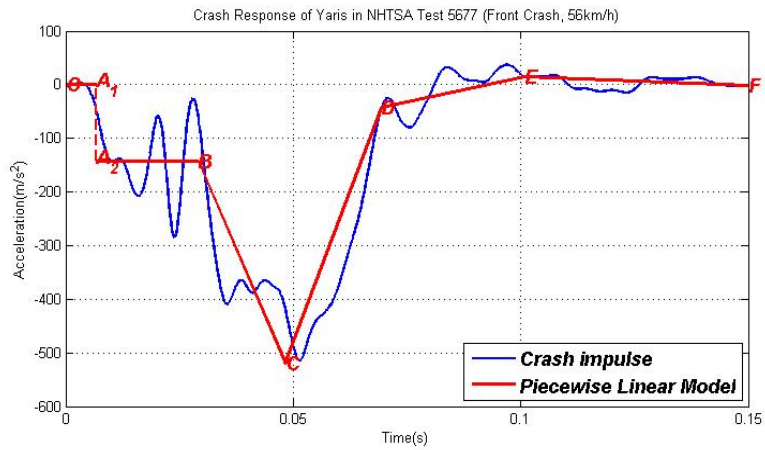


Figure1. Piecewise linear model structure for front crash of sedans.

Comparing with load-carrying path and crash stages presented above, the physical meaning of the model structure is as follows:

1) Original-A1 segment: The accessory are crushed and bumper deform. In this stage, there is no significant acceleration.
2) A2-B segment: The crash box and longitudinal beam are working. During this time, the acceleration is stable around a level.
3) B-C-D segment: The longitudinal beam keeps working and the engine is crashing to the firewall. The force worked on the firewall makes the sharp slowdown of the cabin.
4) D-E-End segment: Restitution process. The crumple area is fully compressed and most energy are absorbed. Some internal energy is released and leads the oscillation of velocity.

Although some local oscillations (the high frequency component of acceleration signals) are lost in this model structure, it reflects the trend (base mode) of acceleration in a full crash. Because the integration of the oscillations approaches zero [10], this model can keep a good performance in the estimation of velocity and displacement.

<u>Time of model nodes</u> In the presented model structure, the time and value of each node (i.e. A1, A2, and B~F) are to be decided. For O-A_1 stage, the acceleration is small and the variation of velocity is tiny. The end of this stage can be set as:

$$T_A = t_0, \text{when } \Delta \mathrm{v}(t_0) = \int_0^{t_0} a(t)\,\mathrm{dt} = 1\% \times v_{init} \quad (1)$$

and

$$Acc_{A_1} = 0 \quad (2)$$

For node F, T_F is the end time of signals and the acceleration Acc_F is the value of end time. The times of other nodes are discussed in this part and the accelerations will be studied in the next part.

Comparing to A_2-B, the main feature of B-C-D process is that the engine is compressed by the firewall. The contact force slowdowns the cabin drastically. Correspondingly, the reacting force makes the deceleration of engine turn smaller (See Figure 2). So the time of B, T_B, is the time of minimum engine deceleration.

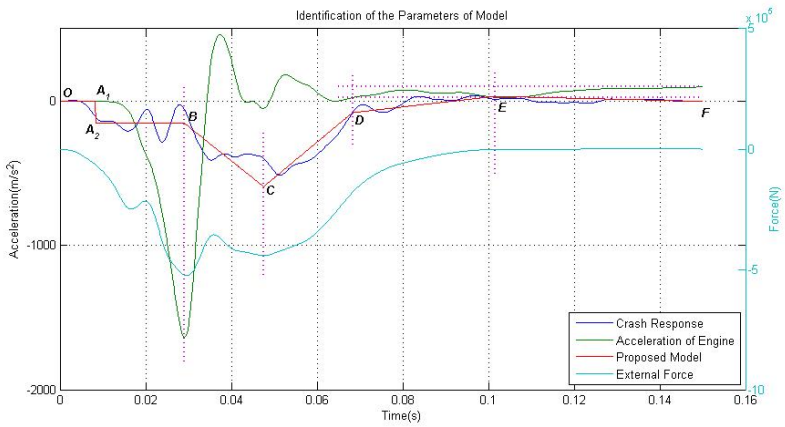


Figure2. Time Identification of Proposed Model Structure.

After T_B, the engine experiences a "step response" like process with the input of the reacting force from firewall and finally arrives the stable value when the time finished. So T_B can be set as the time when engine acceleration arrives the steady area. A recommended steady area can be seen from the maximum to the minimum value of engine acceleration after the T_E (will be given later).

Node C refers to the maximum deceleration in the crash process. Of course, it can be decided directly by the minimum point of crash response. However, in some cases, Acc_C is not significantly lower than the neighborhood and therefore T_C suffers a great uncertainty. This is because of the combined effect of the oscillations from engine and rest part of the vehicle. And neither of them plays a leading role. For these cases, an alternated method is given as below.

It should be noted that the crash response signal is measured from one point of the vehicle body. That means it cannot reflect the general response exactly perfect. To solve this problem, the external force, which works on the vehicle is to be studied (See Figure 2).

As shown, the first maximum force is corresponding to the node B and contributes to the maximum deceleration of the engine. The second maximum force is related to the Node C, as the engine acceleration is not significant at that period. So, T_C equals to the time of the second maximum force.

For the restitution stage, T_E is the time when external force falls to 1% of the maximum value.

<u>Accelerations of model nodes</u> The proposed model is hoped to make the error of acceleration and velocity small. For convenience in computation, the accelerations of B~E should ensure 1) Acc_B~Acc_E locate near the real crash response; 2) the interaction of the proposed model, i.e. the velocity of model, is same with the real crash response at times T_B, T_D and T_F. So we can set $Acc_D = a(T_D)$ arbitrarily and decide the rest accelerations as follows:

1) To ensure

$$\int_0^{T_B} Acc(t)\,dt = Acc_B * (T_B - T_A) = \Delta v(T_B) \quad (3)$$

Acc_B is set as

$$Acc_B = \frac{\Delta v(T_B)}{T_B - T_A} \quad (4)$$

2) To ensure

$$\int_{T_B}^{T_D} Acc(t)\,dt = \frac{1}{2}Acc_B * (T_C - T_B) + Acc_C * (T_D - T_B) + \frac{1}{2}Acc_D * (T_D - T_C) = \Delta v(T_{D-B}) \quad (5)$$

$$\int_{T_D}^{T_F} Acc(t)\,dt = \frac{1}{2}Acc_D * (T_E - T_D) + Acc_E * (T_F - T_D) + \frac{1}{2}Acc_F * (T_F - T_E) = \Delta v(T_{F-D}) \quad (6)$$

Acc_C and Acc_E are set as

$$Acc_C = \frac{1}{T_D - T_B}\left[\Delta v(T_{D-B}) - \frac{1}{2}Acc_B * (T_C - T_B) - \frac{1}{2}Acc_D * (T_D - T_C)\right] \quad (7)$$

$$Acc_E = \frac{1}{T_F - T_D}\left[\Delta v(T_{F-D}) - \frac{1}{2}Acc_D * (T_E - T_D) - \frac{1}{2}Acc_F * (T_F - T_E)\right] \quad (8)$$

Models for Various Speed

To show the variance of the proposed model, a series of crashes are simulated by the FE method. In these crashes, the 2010 Toyota Yaris is crashing to a rigid wall at the speed of 20km/h, 25km/h, 32km/h, 40km/h,

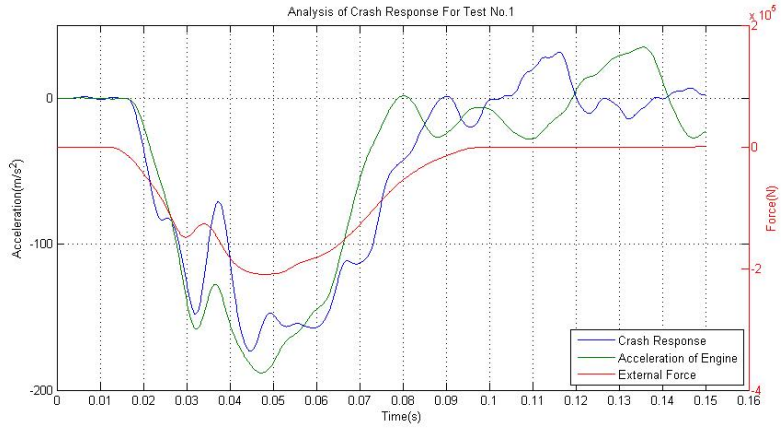
48km/h, 56km/h and 65km/h. The FE model of Yaris comes from the National Crash Analysis Center (NCAC) of George Washington University.
The parameters of the modelling results are shown in the table (See Table 1).

Table1.
Parameters of modelling results for crash simulations in various initial velocities.

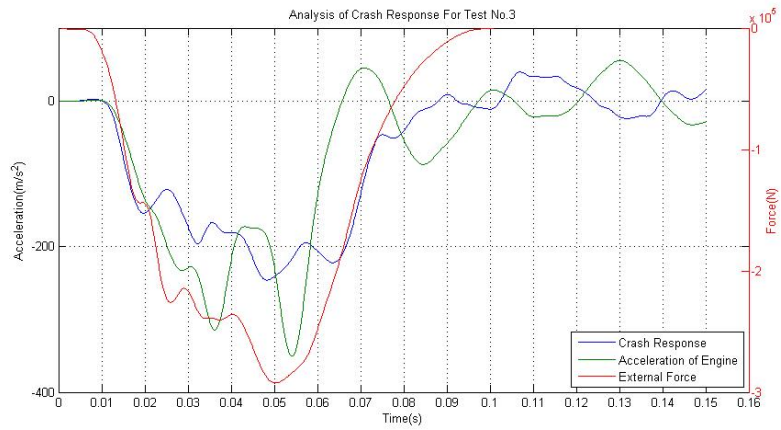
No.	Initial Velocity	At(s)/ A1(m/s^2)	Bt(s)/ B(m/s^2)	Ct(s)/ C(m/s^2)	Dt(s)/ D(m/s^2)	Et(s)/ E(m/s^2)	Ft(s)/ F(m/s^2)	Note
1	20km/h	0.0203 0	*0.0473* *-114.4992*	*0.1159* *-127.7229*	0.0736 -93.6547	0.1026 22.6533	0.1499 2.0482	Bad
2	25km/h	0.0172 0	0.0368 -125.1086	0.0569 -199.2543	0.0749 -87.6170	0.0985 20.3931	0. 1499 -0.5908	Good
3	32km/h	0.0145 0	*0.0541* *-178.0036*	*0.1058* *35.2196*	0.0933 -3.2184	0.0989 -1.1863	0. 1499 14.3759	Bad
4	40km/h	0.0125 0	0.0403 -187.4030	0.0516 -285.4251	0.0774 -84.9810	0.1030 37.3893	0.1499 -17.9486	Good
5	48km/h	0.0111 0	0.0337 -195.5635	0.0578 -379.2445	0.0745 -110.9681	0.0927 25.4419	0.1499 17.3587	Good
6	56km/h	0.0102 0	0.0330 -195.0890	0.0474 -484.4808	0.0873 40.1901	0.0963 40.0699	0.1499 -12.1832	Good
7	65km/h	0.0096 0	0.0278 -202.8010	0.0456 *-399.5320*	*0.1042* *44.3387*	0.1049 *21.1624*	0.1499 -22.8631	Bad
8	20km/h	0.0203 0	0.0323 -97.2563	0.0481 -172.514	0.0736 -93.6547	0.1026 22.6533	0.1499 2.0482	Good after adjustment
9	32km/h	0.0145 0	0.0362 -150.5192	0.0503 -229.4992	0.0933 -3.2184	0.0989 -1.1863	0.1499 14.3759	Good after adjustment
10	65km/h	0.0096 0	0.0278 -202.8010	0.0456 -500.0374	0.0905 38.564	0.1049 35.1197	0.1499 -22.8631	Good after adjustment

This table shows that:

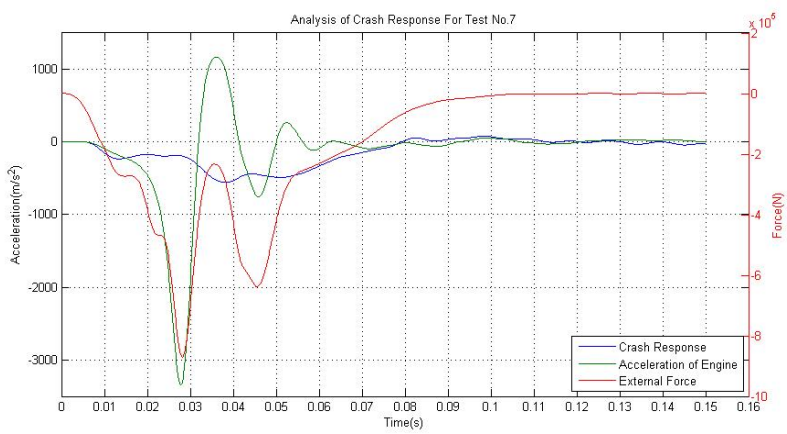
1) Generally speaking, the model has good applicability for the crashes with the initial velocity from 25~56km/h.
2) For the No. 3, the model has a wrong identification of T_B and T_C and therefore fails to fit the crash response. By studying the crash responses of 32km/h crash (See Figure 3b), it can be found that the engine acceleration at 0.0362s is the first local minimum value with the abrupt turn of trend. According to the physical meaning, this abrupt turn refers to the contact between the engine and firewall and T_B should be 0.0362s. Consequently, T_C is 0.0503s. After this adjustment (as No. 9), the model fits the crash response well. This is because of the less compression of the engine in the lower speed crashes. So for the crashes with initial speed lower than 32km/h, it is highly recommended to check the model again according to the physical meaning.
3) No. 1 suffers similar problems with Test 3 and can be adjusted (as No. 8) to achieve good performance. However, the minimum value is not much lower than other values in this crash response (See Figure 3a). Specifically, the process B~D (i.e. the engine influence) is not as significant as other crashes. This means the crash process, as well as model structure, is different from high speed crashes. For this reason, the proposed model may cannot represent the responses well for the crashes with initial velocity lower than 20km/h.
4) No. 7 is for the 65km/h crash and have a problem in the identification of T_D. In the crash responses (See Figure 3c), there are oscillations with big amplitude in the process D~E and lead the wrong identification. Obviously, in this crash, the vibration of the engine is much stronger than 56km/h crash. An adjustment can be made manually to improve the model (as No. 10). But it should be mentioned that for a higher speed crash, the model can hardly present the crash process very well.



a) Crash Responses and References of No.1 Simulation



b) Crash Responses and References of No.3 Simulation

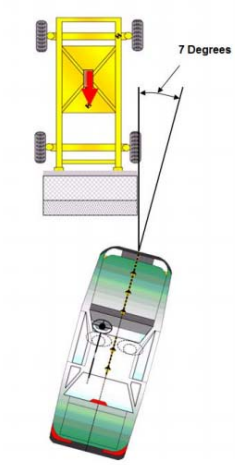


c) Crash Responses and References of No.7 Simulation

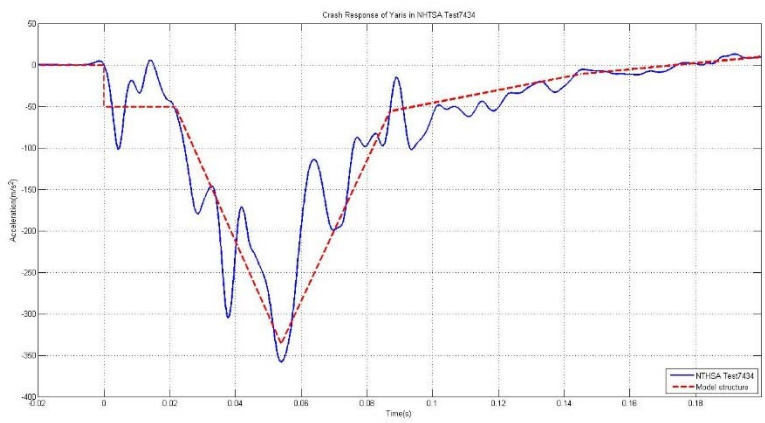
Figure3. Crash Responses and References (a. No.1; b. No.3; c. No.7).

Complex Condition

In this subsection, a full car crash test will be studied. In this crash, the Yaris is crashed by a moving deformable barrier (RMDB) with a target speed of 86.7 km/h. The crash mode is 7° angle and 20.6% offset (See Figure 4a). The test data come from NHTSA Test 7434.



a) Configuration of NHTSA Test 7434



b) Crash Responses and Modelling of NHTSA Test 7434

Figure4. NHTSA Crash Test 7434 (a. Crash Configuration; b. Crash Response)

From this response, we can find that:

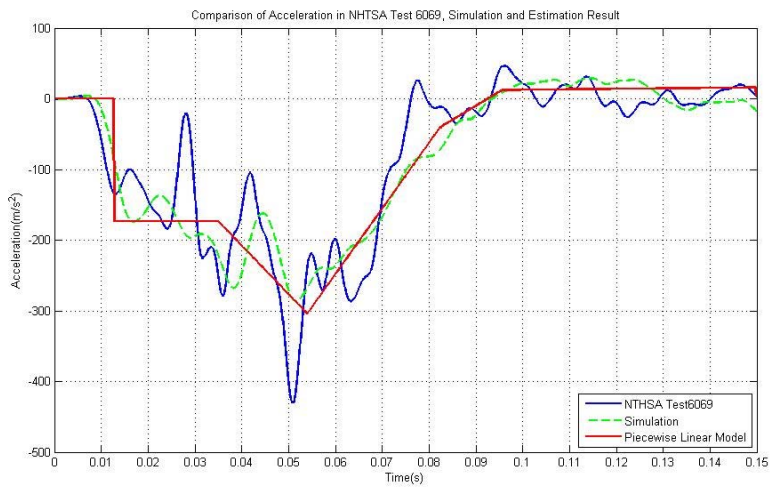
1) Because of the deformable barrier, the maximum acceleration in this test is 358.3208m/s^2 in negative direction, which is near the value in 40km/h rigid wall crash. So it is possible to have an experience that the maximum acceleration of a deformable barrier crash can be estimated by the rigid wall crash with half speed. Of course, this is a rough estimation and the barrier should have similar mechanical characteristics as the vehicle body.
2) In 0~0.02s, the bumper, crash box and longitudinal beam are deforming. But due to the offset, only left half part of the crush zone is crushed and therefor this crash response during this time is different from 100% overlap crash. The most important difference is the average value in this period is about 50m/s^2, which is about only 1/ the Acc_B in No. 4 of Table 1. In other words, the absorbed energy is much lower than the full overlap crash
3) There is also a process like B-D in the model, which indicates that the engine is also crushed into the firewall and compressed. This shows all the crashworthiness structures will work in the crashes with an offset.

In conclusion, this crash test can also be described roughly by the proposed model structure, shown as the red dash line in Figure 6b. However, some local performances are different, such as the mode of B-C stage. And consequently the related parameters should also be revalued.

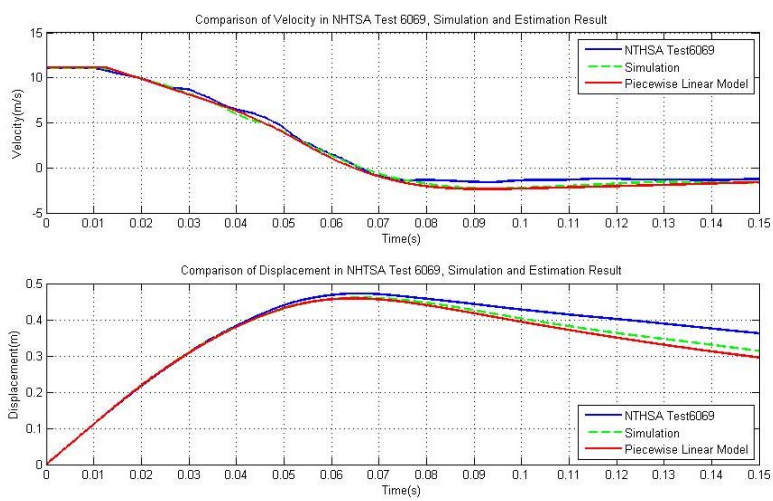
ESTIMATION OF VEHICLE KINEMATICS

The proposed model can be used for the estimation of vehicle kinematics. The NTHSA Test 6069 can be used as an example. The crash condition is: 39.6km/h full overlap crash to a rigid wall barrier. To make an

estimation, the parameters of the proposed model can be set as the average value of No. 5 and No. 9 in Table 1. That is $A_t = 0.0128$, $Acc_A = 0$, $B_t = 0.0350$, $Acc_B = -173.0414$, $C_t = 0.0541$, $Acc_C = -304.3719$, $D_t = 0.0825$, $Acc_D = -40.4034$, $E_t = 0.958$, $Acc_E = 12.3273$, $F_t = 0.1499$ and $Acc_F = 15.8673$.
Note: the T_D of No. 9 is obviously higher than other T_D. This may be because of the uncertainty of simulations. To get better estimation, the estimation of T_D and Acc_D should be adjusted as the average of No. 2~6, as the 39.6km/h is the average of the corresponding velocities.
The estimated model will be compared with the Test 6069 and simulation result (See Figure 5).



a) Acceleration Signals



b) Velocity and Displacement Signals

Figure5. Validation of the estimation (a. Acceleration; b. Velocity & Displacement)

As shown in Figure 5, the estimated results fit the test and simulation very well. The displacement error (about 0.0668m) at the end of time, which is similar with the simulation (about 0.0460m).

CONCLUSIONS

Comparing to finite element models and multibody modes, the mathematical models have advantages on conciseness and usability. For this reason, the mathematical models can be used in the early design of vehicles, as well as accident reconstruction. This paper presents a novel modelling scheme of crashes, which is based on the acceleration signals and vehicle structure. The proposed model can reflect the crash process clearly and therefore describe the crash response exactly. In addition, this model suits for the crash in various conditions

by adjusting the parameters. At the end of paper, an estimation of vehicle kinematics shows the good performance of the proposed model for a frontal crash at 40km/h.

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