

An Application of Occupant Deceleration Control by the Crash Load Control Structure to an Actual Vehicle

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ABSTRACT

In order to reduce the maximum value of occupant deceleration in a vehicle crash, a body structure was devised to approximate the optimal vehicle deceleration wave form by utilizing the difference of the deformation mode of the body. This concept of structure was successfully applied to an actual vehicle.

1. Introduction

When considering vehicle crash safety performance, optimizing the change in vehicle deceleration over time in a crash (hereafter referred to as the “vehicle deceleration wave form”) is an important issue for simultaneously securing survival space and reducing injury severity. Previous research mainly performed optimization using the sensitivity analysis method⁽¹⁾⁽²⁾, but the authors have derived and generalized mathematical solutions for the optimal vehicle deceleration wave forms for chest deceleration of the occupant through mathematical studies using a two-mass model⁽³⁾. These mathematical solutions have in common a three-part structure consisting of high initial deceleration, low or negative middle deceleration, and then constant deceleration later on.

However, vehicle deceleration is generally determined by the vehicle structure deformation load in a crash which is applied to the vehicle mass. This deformation load increases with the amount of vehicle deformation, which makes it difficult to reduce the deceleration during the middle period of a crash compared to that at the start of a crash. Therefore, the authors devised a fundamental model for approximating the optimal vehicle deceleration wave form by reducing the crash load during the middle period of a crash utilizing the difference in load characteristics of the deformation mode. In addition, application of this model to an actual vehicle structure was studied.

2. Mathematical Solutions for the Optimum Vehicle Deceleration Wave Form

Previously the authors had mathematically studied the vehicle deceleration wave form that minimizes the maximum occupant deceleration value using a two-mass model like that shown in Fig.1. Here, the authors introduced a transient function concept and obtained mathematical solutions for the optimal vehicle deceleration wave form⁽³⁾.

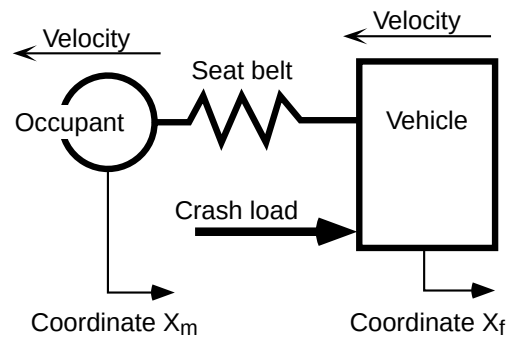


Fig.1 One-dimensional two-mass model

The transient function is the optimal occupant deceleration wave form that results in a complete ride-down condition during a certain short control time after the crash starts. Solutions of the function are not unique. A corresponding basic optimal vehicle deceleration wave form type (shown by equations (4), (5) and (6)) exists for each of the three basic types of transient functions shown by equations (1), (2) and (3). Here, X_m indicates the occupant coordinate, and X_f the vehicle coordinate. In these equations the time t_c when control ends is expressed using the transient velocity $\gamma = \pi/t_c$. Also, the deceleration wave form after $t = t_c$ is the constant value C for all cases.

$$\ddot{X}_m = -C \left[\frac{\gamma}{\pi} \right]^3 \left[2t - 3 \frac{\pi}{\gamma} \right] t^2$$

[Power polynomial type transient function] (1)

$$\ddot{X}_m = \frac{C}{2} (1 - \cos \gamma t)$$

[Cosine type transient function] (2)

$$\ddot{X}_m = C \frac{\gamma}{\pi} \left[t - \frac{1}{2\gamma} \sin 2\gamma t \right]$$

[Sine type transient function] (3)

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$$\ddot{x}_f = -C \left[\frac{\gamma}{\pi} \right]^3 \left\{ \left[2t - 3 \frac{\pi}{\gamma} \right] t^2 + \frac{6}{\omega^2} \left[2t - \frac{\pi}{\gamma} \right] \right\}$$

[Power polynomial type] (4)

$$\ddot{x}_f = \frac{C}{2} \left\{ 1 + \left[\frac{\gamma^2}{\omega^2} - 1 \right] \cos \gamma t \right\}$$

[Cosine type] (5)

$$\ddot{x}_f = C \frac{\gamma}{\pi} \left\{ t + \left[\frac{4\gamma^2}{\omega^2} - 1 \right] \frac{1}{2\gamma} \sin 2\gamma t \right\}$$

[Sine type] (6)

These wave forms are shown in Fig.2. The optimal vehicle deceleration wave form exhibits a curved form which differs for each type of function. However, what each wave form has in common is a three-part structure consisting of high initial deceleration, negative middle deceleration, and then constant deceleration later on.

3. Study of Crash Load Control Structures

3.1. Study of Target Vehicle Deceleration Wave Forms

When considering the vehicle deceleration wave forms shown in Fig.2 as actual target vehicle deceleration wave forms, creating a negative deceleration wave form for the middle period of a crash becomes an issue. Vehicle deceleration in a crash is generally produced by the vehicle deformation load (crash load) applied to the vehicle mass, so the crash load is limited to the direction opposite that of the velocity, and the deceleration tends to increase as deformation progresses. For this reason, it is difficult to produce stable low deceleration with conventional vehicle structures during the middle period of a crash. Furthermore, the crash load must be in the opposite direction of the vehicle mass to produce negative deceleration, making it even more difficult to achieve with this vehicle structure.

However, when for example the realistic values of $\omega \approx 701/s$ and $C \approx 400m/s^2$ are set in the mathematical solution for the cosine type vehicle deceleration wave form shown in equation (5), if the control time t_c is set to 35 msec or more, the negative region of the vehicle deceleration wave form disappears as shown in Fig.3.

When considering this type of vehicle deceleration wave form as the target wave form, it is possible for the actual vehicle deceleration wave form to approach the target by

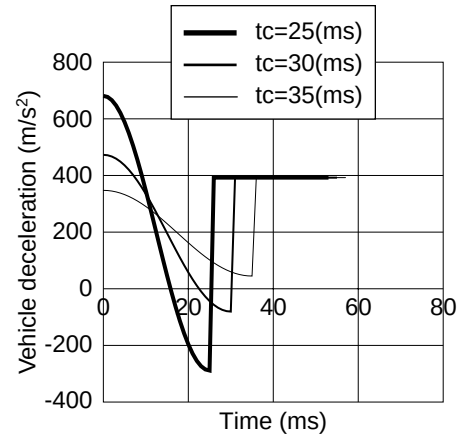


Fig.3 Deceleration curves for various t_c values

applying control so that the crash load is reduced during the middle period of a crash.

In addition, the linear combinations of the mathematical solutions for each different function type are also mathematical solutions, so various curve shapes can be considered as long as the optimal deceleration wave form has the common feature of the three-part structure. For this reason, even deceleration wave forms which change in a stepped manner can be approximate optimal deceleration wave forms. Therefore, the crash load does not necessarily have to be continuously controlled, and any means of reducing the crash load during the middle period of a crash can be considered effective for optimizing the vehicle deceleration wave form.

Thus, the target vehicle deceleration wave form was based on the cosine type and a realistic approximate optimal vehicle deceleration wave form was selected consisting of an initial high constant deceleration, a constant middle deceleration of approximately zero, and a constant later deceleration of value C . Next, methods for controlling the crash load with the vehicle structure were studied as a means of creating this wave form.

3.2. Crash Load Control by Deformation Mode

A number of methods can be considered for controlling the crash load. Fig.4 shows the basic concepts of load control structures. In consideration of application to existing vehicle

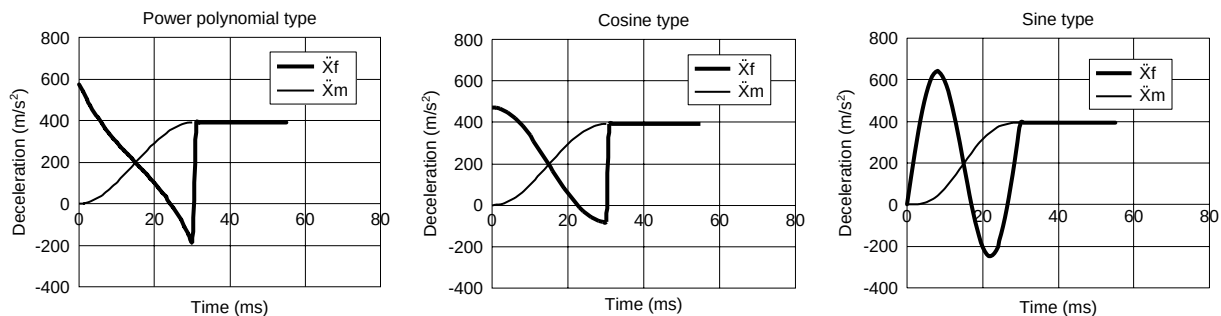


Fig.2 Wave forms of the mathematical solutions to the transient function forms

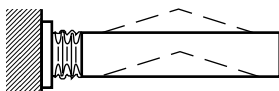
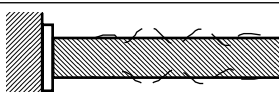
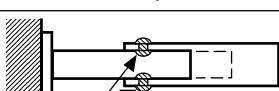
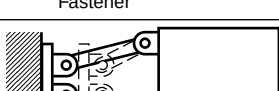
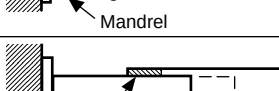
Control method	Basic construction
Combination of axial collapse and bending	
Controllable axial collapse	 Tailored composite material
Breakable fastener	 Fastener
Variation of the cosine component (geometry)	
Variation of the cross-sectional area	 Mandrel
Variation of the friction factor (from static to dynamic)	 Friction element

Fig.4 Methods for controlling the crash load

structures, a method which does not require mechanical elements is best. Therefore, the authors focused on methods utilizing the difference in load properties related to the deformation mode.

The vehicle structures which generate crash loads in general vehicles are hollow metal beam components such as front side members. When a load is generated on these structures compression collapse (hereafter “axial collapse”) or bending deformation is produced. Axial collapse is stable and generates a fairly constant load, whereas if bending deformation exceeds the maximum load the load decreases rapidly. Therefore, if the transition to bending deformation can be controlled, the load can be decreased at an optional time during the crash. This can be accomplished by a number

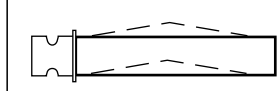
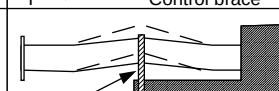
Control method	Basic construction
Difference in the maximum load between axial collapse and bending	
Control brace	 F Control brace
Breakable strap	 Breakable strap

Fig.5 Methods for controlling the transition to bending deformation

of methods as shown in Fig.5. Here the simplest structure is a combination of the axial collapse and bending zones in series. A structure was studied which maintains a large load at the start of the crash for a certain time and then rapidly decreases the load by providing a difference between the load generated in the axial collapse zone and the maximum bending zone load.

3.3. Fundamental Model of Crash Load Control Structure

Fig.6 shows the fundamental model of the devised crash load control structure. This fundamental model is a hollow beam component which hypothesizes a front side member, and is comprised of an axial collapse zone at the front and a bending zone in the middle. The axial collapse zone has a shape which concentrates stress to induce regular buckling deformation. The bending zone has a slightly angled bend shape in the center in order to direct the bending deformation in a stable manner. The cross section of each zone is set so that the axial collapse zone has a slightly smaller deformation load than the maximum load of the bending zone.

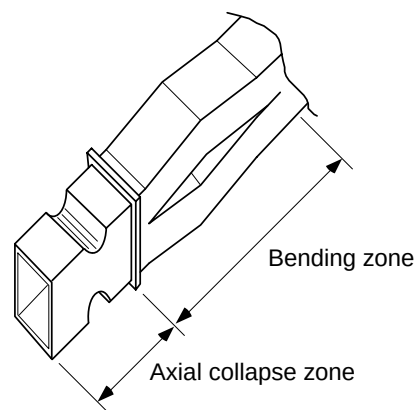


Fig.6 Fundamental model of a crash load control structure

Fig.7 shows the deformation process in the fundamental model. Fig.7-A shows the 1st stage. First, the stress centralizing shape of the axial collapse zone causes it to start deforming immediately after the start of the crash. A fairly constant load is produced after the axial collapse zone starts to deform, but the load increases as the regular buckling deformation advances over the entire axial collapse zone, and eventually reaches the maximum load of the bending zone. Fig.7-B shows the 2nd stage. When the bending zone exceeds the maximum load, it immediately bends significantly and the load drops to a fraction of that in the 1st stage. Fig.7-C shows the 3rd stage. As the bending deformation progresses, the beam bottoms out and the load increases again at the end of the crash.

Fig.8 shows the results of obtaining the change in crash load over time through a simulation using a finite element model created based on the fundamental model. This simulation attached a mass corresponding to a vehicle to the

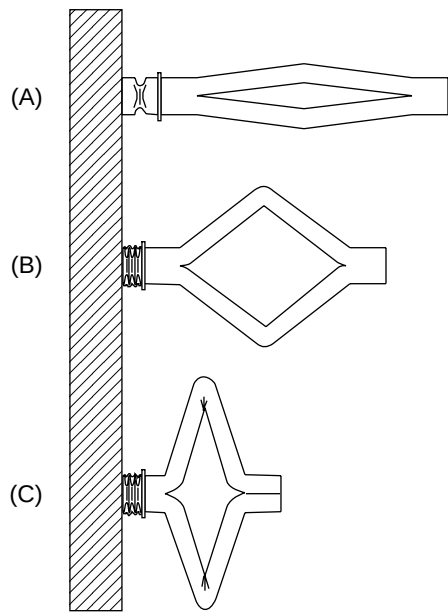


Fig.7 Deformation process in the fundamental model

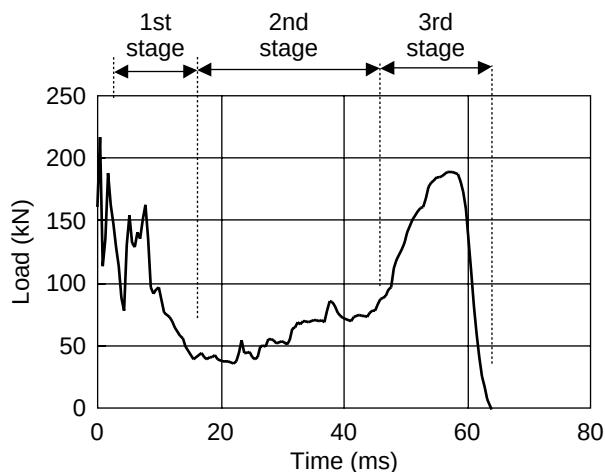


Fig.8 Change in crash load over time for the fundamental model

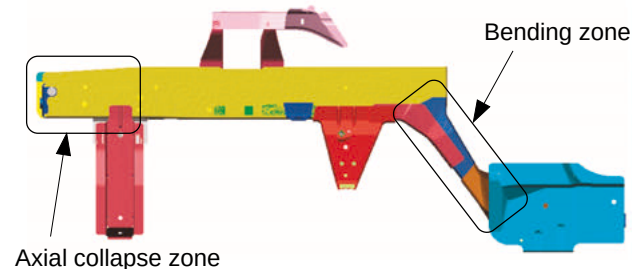
rear end of a beam, crashed the beam into a rigid wall with an initial velocity of 15.56 m/sec, and obtained the load generated on the rear end of the beam. The graph shows that a large load is maintained for a certain time at the start of the crash, the load decreases during the middle period, and then rises again toward the end of the crash, thus confirming crash load control based on the fundamental model. These results indicated that it was possible to achieve deceleration close to the target vehicle deceleration curve without significantly changing the conventional vehicle structure of actual vehicles.

4. Study of Crash Load Control in an Actual Vehicle Structure

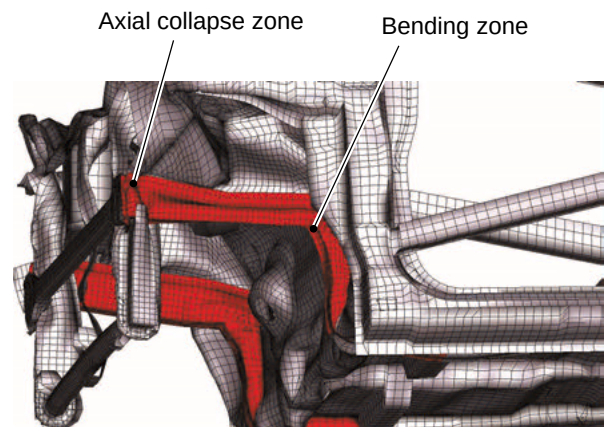
The fundamental model shown in Fig.6 in Section 3 has an extremely simple structure. However, in an actual vehicle

the engine mount is normally established near the midpoint of the horizontal part of the front side member, so the bending deformation of this part is controlled by the mass of the engine and the transmission. For this reason, it is difficult particularly in compact vehicles to adopt a design which provides a bending zone shaped like the fundamental model in the horizontal part of the front side member. Therefore, the existing bent shape in the connection between the cabin and the rear end of the front side member in the conventional vehicle structure was used as the bending zone, and an attempt was made to achieve the same functions as the fundamental model by making the maximum deformation load of this zone slightly larger than the axial collapse load of the front side member.

Fig.9 shows the structure of a production compact vehicle which performs vehicle deceleration control by introducing this approach. In this example there is almost no change to the vehicle structure, and reduction of the dummy chest deceleration was verified through barrier crash tests (Fig.10).

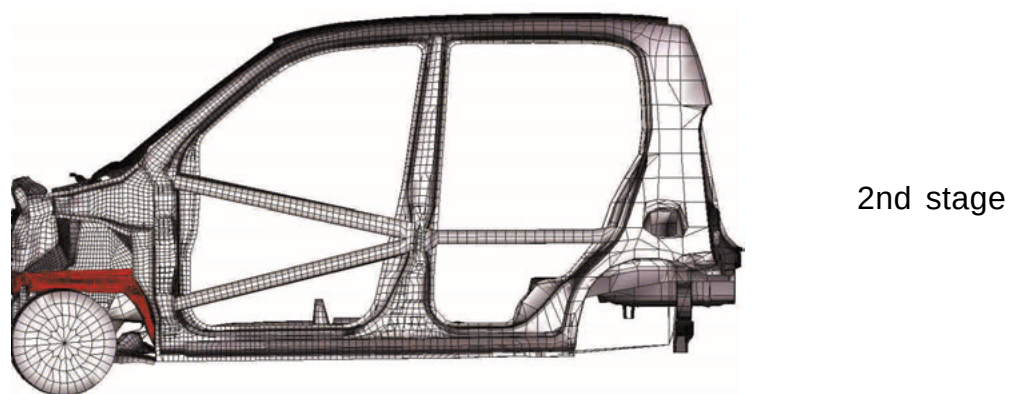
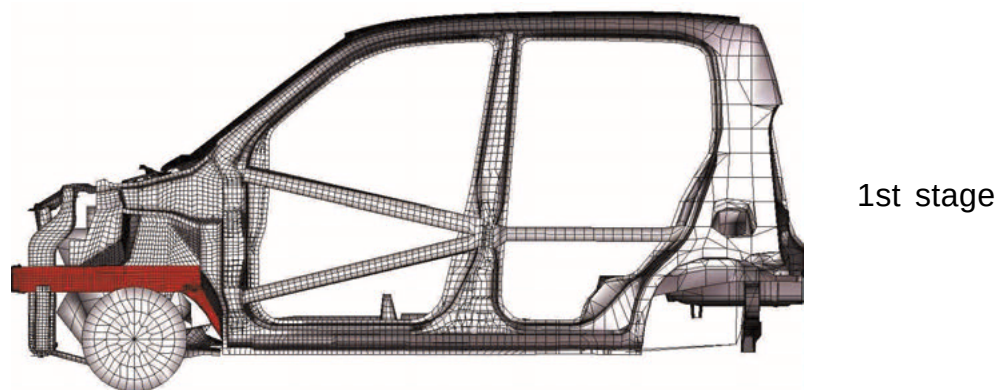


Front side member (before crash)



Front structure (after crash)

Fig.9 Application of the crash load control method to a production vehicle



Rear end of the front side member where bottoming occurred in the final stage

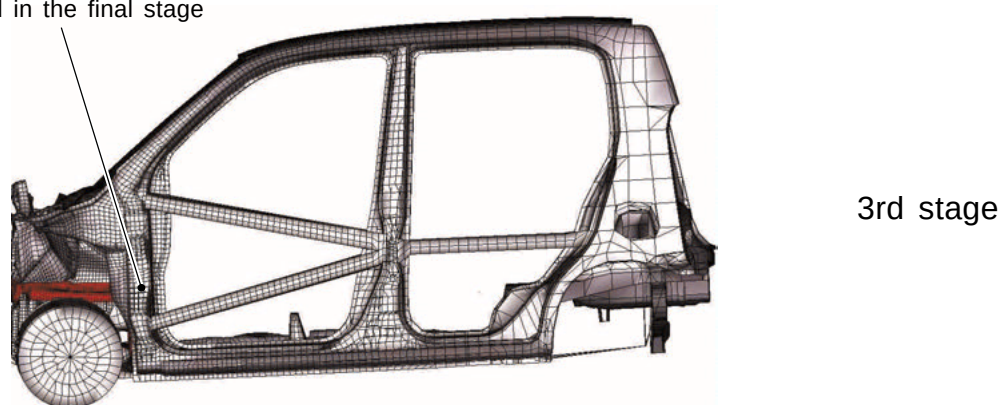


Fig.10 FEM simulation of barrier crash test

5. Conclusions

- (1) Focusing on the difference in load properties due to the hollow beam deformation mode, a fundamental model was devised for creating low deceleration during the middle period of a crash, which is a feature of the optimal vehicle deceleration wave form. This was achieved by controlling the change in crash load due to the transition from axial collapse to bending deformation by making the maximum deformation load of the axial collapse zone slightly smaller than that for the bending zone.
- (2) A finite element simulation model was created based on the fundamental model, and the change in crash load over time was obtained. The crash load was confirmed to decrease during the middle period and then rise again toward the end of the crash, thus verifying the functionality of the fundamental model.
- (3) Studies were carried out to apply the concept of the fundamental model to the bending deformation of the front side member of an actual vehicle. As a result, the dummy chest deceleration in barrier tests could be reduced without significantly changing the conventional vehicle structure.

Further studies will be carried out in the future on structures which optimize vehicle deceleration for various crash modes.

References

- (1) Takahashi, K., et al.: Optimization of Vehicle Deceleration Curves for Occupant Injury - The 1st Report, Basic Discussions with Sensitivity Analysis, Transactions of JSAE Vol24, No.2, p.98-p.102(1993)9303753 (Written in Japanese)
- (2) Suzuki, N., et al.: Optimization of Vehicle Deceleration Curves for Occupant Injury - 2nd Report, Application of Sensitivity Analysis to Occupant Simulation, Transactions of JSAE Vol24, No.4, p.109-p.112(1993)9307443 (Written in Japanese)
- (3) Yasuki MOTOZAWA, Takahiro KAMEI: New Conception of Occupant Deceleration Control in Crash, HONDA R&D Technical Review, Vol.11, No.1, p.129-134(1999) (Written in Japanese)

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