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Design of a crush zone system for a railway passenger car to improve crashworthiness

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ABSTRACT

In the present study, after comprehensive examination of previous studies on railway vehicle crashworthiness strategies, an innovative crush zone system to improve crashworthiness of railway passenger car is developed. This system absorbs 3 MJ kinetic energy in a progressive manner. In order to evaluate the crashworthiness features of the crush zone, in the first step, energy absorber components (thin walled tubes and honeycomb structures) are simulated by using explicit finite element method. To validate the FE models, honeycomb structures are analyzed by conducting quasi-static tests. The good agreement between experimental measurements and FE analysis results indicates that the FE model is accurate. The primary energy absorbers are simulated by dynamic explicit finite elements method. By extracting the force-deformation curve of this structure, energy absorption and progressive crushing behavior is examined.

1. Introduction

There is a general trend throughout the world to improve the crashworthiness of railway vehicles by passive safety strategies. In the first published paper in 1987, design loads of railway passenger vehicles in Europe and the United States are investigated. By examining UK accident statistics, alternative ways to improve the structural crashworthiness are proposed. General design concept of A. Scholes's proposal is to localize the deformation to the vehicle ends and absorb collision energy in a controlled manner to protect the main passenger space. To reach this goal, crush zone systems are designed and examined in Europe and United States ad these zones are designed to collapse in a controlled manner during collision and absorb several million joules of kinetic energy. Also by minimizing vertical and lateral wagon motion, overriding and derailment are prevented [1-3].

In the last two decays, many projects started to improve crashworthiness features of railway vehicles in Europe, United stated, Japan and India. For instance, in 1994 full scale wagon crush test were done in UK, and full scale crush tests on TGV wagons were completed between 1994-1996 by French railway company (SNCF). In 1997, SAFETRAIN project started in Europe. In 2006, Indian railway company carried out a full scale crush test on GS and SLR passenger cars to evaluate their new designed crush zone performance on passenger railway cars. Starting in 2000, FRA in the United States conducted 6 full scale crush tests on passenger cars with and without crash zone systems in 3 different test scenarios: single wagon impact into a rigid wall, two wagons impact into a rigid wall and train to train impact [4-7].

Several CEM cab and coach car designs exist in service in the world such as TGV, TGV Duplex, the XTER, the modified Mark I, the Talgo XX1, the AgC, the Itino, the TRAXX locomotive, the Pendilino, and the TER2N. These designs share three common features. The first is the use of a push-back coupling mechanism that allows the ends of vehicles to come together and transmit load over a large area. The second is an anti-climbing mechanism that restricts vertical motions between colliding vehicles. The third common feature is some means of absorbing large amounts of energy at each crush zone interface [5-9].

2. Crush Zone Design

The requirement for the total energy absorption in the design stage is 3 MJ in about 0.88 m of crush. Limitations are determined on the standard acceleration/declaration rates to be limited to around 5g, but not more than 8g. This limitation is related to a general level which an unrestrained passenger can be expected to survive. In order to refuse of sudden accelerations in collision event and provide a smooth crush, honeycomb structures are used in the back of buffer beam. The honeycomb box is designed to be activated in prescribed impact force range and absorb certain kinetic energy of wagon.

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Based on accident statistics and normal traveling velocity in Turkey, the railway crush zone has been designed for an identical passenger car having an average mass of 32 tons and traveling velocity of 50 km/h. Figure 1 shows the ideal crush zone deformation-load curve, in the first step, coupling mechanism will be activated; then when the load reaches to the value of 2.5 MN, shear bolts of buffer beam shall be cut and honeycomb box energy absorber component shall be activated. At the final step, when the load reaches to 4.9 MN, sliding sill shearing bolts shall be cut and primary energy absorbers will be activated that absorbs about 2.7 MJ energy. At the end of these mechanisms, the car will stop.

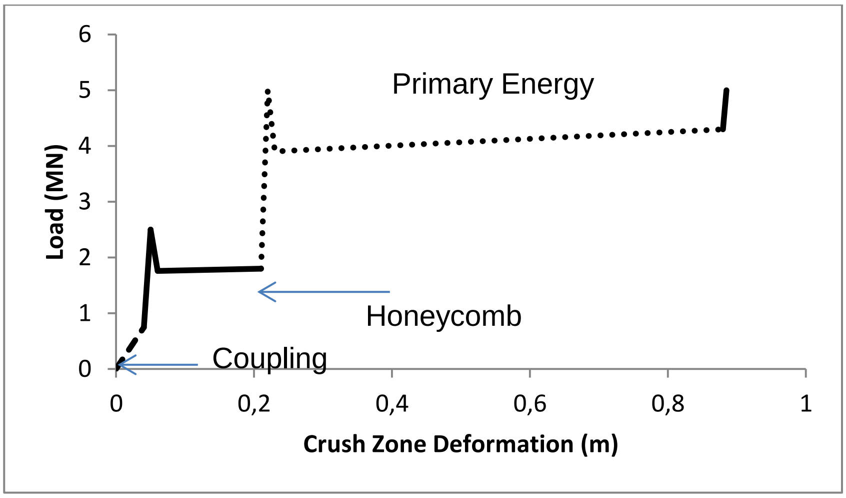

Figure 1. Ideal crush zone deformation-load curve.

One of the principles of crashworthiness design is to absorb collision energies of wagons. The crash scenario that is considered in this study to design crush zone system is as follows: one wagon impacts into a rigid standing wall, the collision energy is abstained by using the law of conservation of impulse

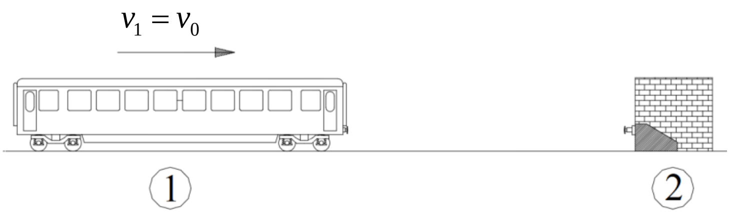

Figure 2. One wagon impacts into a rigid standing wall.

$$v_1 = v_0, \quad v_2 = 0, \;\; m_1 = m, \;\; m_2 \to \infty, \tag{1}$$

$$E_c = \frac{1}{2} m_1 V_0^2 \tag{2}$$

The deformation of the wagon is given by the following expression:

$$l = \frac{V_0^2}{2a_a} \tag{3}$$

The total energy absorbed by the vehicle structure can be expressed:

$$E_c = F_{av} l = m a_{Av} l \tag{4}$$

F_{av} is the magnitude of the average force acting on the vehicle during crash [8].

3. Crush Zone Components

The crush zone that has been developed for a passenger car include the coupling system, honeycomb box, the sliding sill mechanism, the fixed sill mechanism, primary energy absorber, roof energy absorbers, integrated end frame and anti-climber.

Honeycomb box: as showed in Figure 3 (d), the honeycomb box composed of the buffer beam, the buffer support, the shear bolts and the aluminum honeycomb block. Two honeycomb boxes are placed on the wagon, one on each side of the centerline of the car. When the impact force push the coupling to back and the ends of coupled cars come into contact through the anti-climbers, at the defined load, the shearing bolts between buffer beam and support plate will be cut and buffer beam will crush the the honeycomb block. The energy absorption capacity of the honeycomb box is 0.3 MJ within 17 cm stroke.

The sliding sill mechanism: The sliding sill provides a guide for the end crush zone of the car. It consists of U shaped beams sliding within guide channels. It is attached to the fixed sill through the shear bolts. After the honeycomb box is activated, the load at the end frame rises, and the sliding sill shear bolts fracture when the impact load reaches to a defined load; e.g., see Figure 3 (b).

Primary energy absorber: Two primary energy absorbers are on the car, one on each side of the centerline of the car and consist of 6 pyramidal thin walled tubes having different lengths. Each absorber is welded to and supported by the car structure that is not intended to deform. The load on the outboard ends of the primary energy absorbers is applied by the back of the support plate. An initial gap exists between the outboard end of the absorber and the back of the buffer beam before activation of the crush zone (so that the absorbers do not carry operational loads). The energy absorption capacity of primary energy absorber is 2.7 MJ within 65 cm stroke; e.g., see Figure 3 (c).

Roof energy absorbers: Two roof absorber assemblies are on the car located one on each side. They consist of tubes that riveted to each other and contain aluminum honeycomb cartridges. When the load at the roof absorber reaches to a defined load, the rivets shear and the tube pushes back against the honeycomb pieces. The energy absorption capacity of the roof energy absorbers is 0.02 MJ and the stroke is 25 cm, e.g., see Figure 3 (e).

Integrated end frame: The integrated end frame was designed to remain sufficiently stiff in transmitting the impact load to the energy absorbers, ensuring the proper functioning of the crush zone elements. The integrating end frame can appropriately trigger and allow crushing of energy absorbers when the coupling and anti-climber share the impact loads or when the lead path is only through the coupling or the anti-climber, e.g., see Figure 3 (e).

Anti-climber: In order to prevent the climbing of two interacting vehicles during a collision, a ribbed anti-climber element is mounted on the end of the under frame over the coupling. No energy absorption is associated with the anti-climber element; e.g., see Figure 3 (d).

4. Component Design and Simulation

Honeycomb energy absorber simulation and test: Aluminum honeycomb structures are characterized by high weight/strength ratios, high energy absorption capacity, long stroke, stable deformation on out-of-plane direction and low weight. Because of these characteristics, honeycombs structures are widely used in airplane, automotive industry and railway cars in order to increase the crashworthiness of vehicles. Crashworthiness parameters of Aluminum honeycomb structures depend on the cell specifications such as expanding angel, foil thickness, length/thickness ratio and impacting velocity. These structures are produced by manufacturers in different cell sizes and specifications [11-12]. In order to select the structure with high crashworthiness parameters, we conducted a compressive study experimentally and numerically. In the honeycomb box, 2 aluminum honeycomb blocks with 200× 300× 1000 mm dimension are positioned. These blocks absorb 0.3 MJ energy in 17 cm stroke.

$$E_T \approx (P_{crush})(stroke) \tag{5}$$

$$P_{crush} = \sigma_{crush} \times A \tag{6}$$

E_T : Total absorbed energy,

P_{crush}: The mean crush load,

Stroke: crush distance

σ_{crush}: The crushing stress

A: the area of crushing zone

As the energy absorption requirement is 0.15 MJ for each block, by substituting this value and the dimensions into Equation 6 we find σ_{crush} = 2.95 MPa.

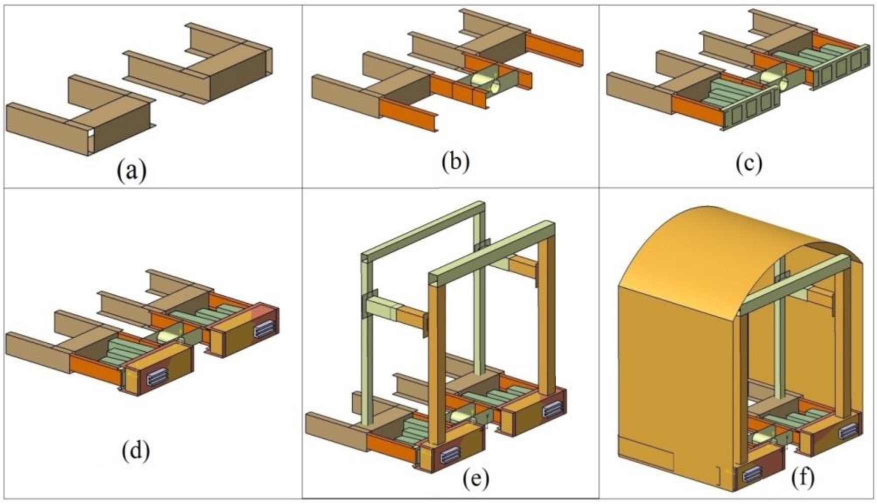

Figure 3. Crush zone component and integration sequence for the passenger wagon.

The material chosen to meet this requirement was determined from the material selection tables provided by Hexcel company for their 5052 Aluminum Alloy Hexagonal honeycomb. The suitable materials are 1/8-5052- 6.1 with 3.1 MPa crushing stress and 1/4-5052-6.0 with 3.0 MPa crushing stress. In order to study the crushing behavior of the honeycomb, specimen with 30 mm width, 30 mm length and 20 mm height are modeled by using Altair Hyperworks. In order to verify the numerical results, quasi-static tests are conducted. There is a good agreement between numerical and experimental results, e.g., see Figures 4, 5 and 6.

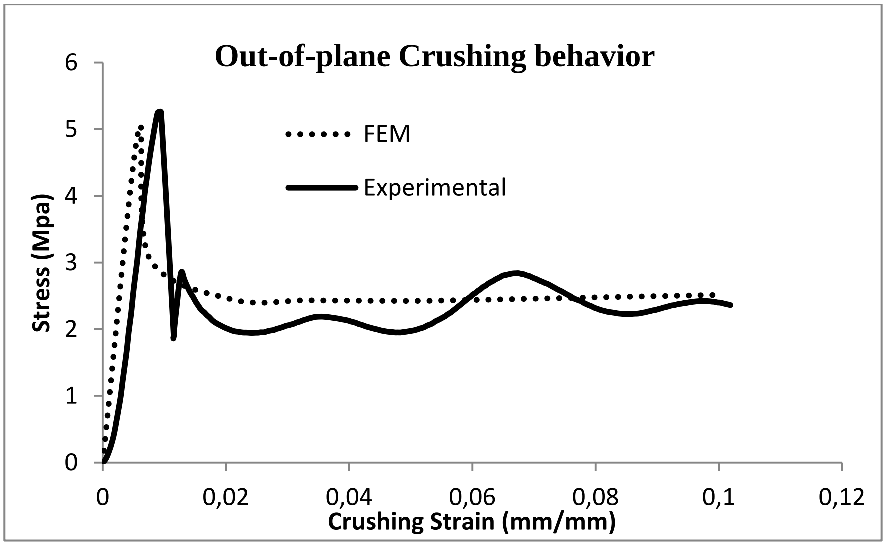

Figure 4. Crushing strain-stress for out-of-plane crushing behavior of the honeycomb structure.

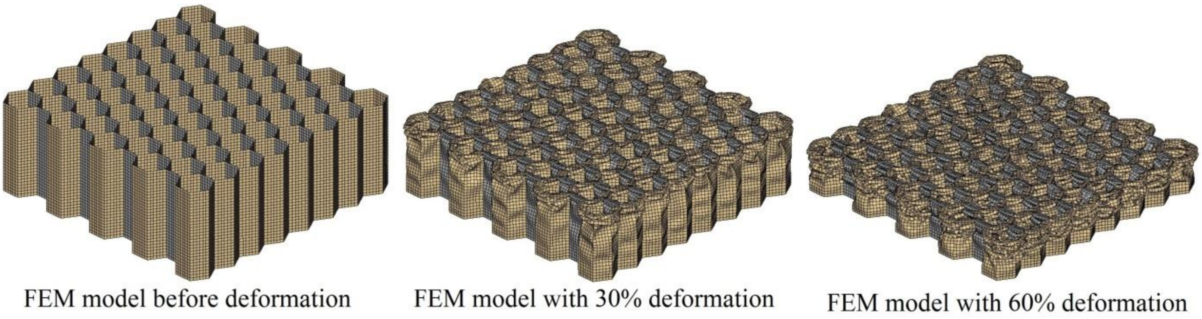

Figure 5. FE deformation stages of aluminum alloy honeycomb specimens in out-of-plane direction.

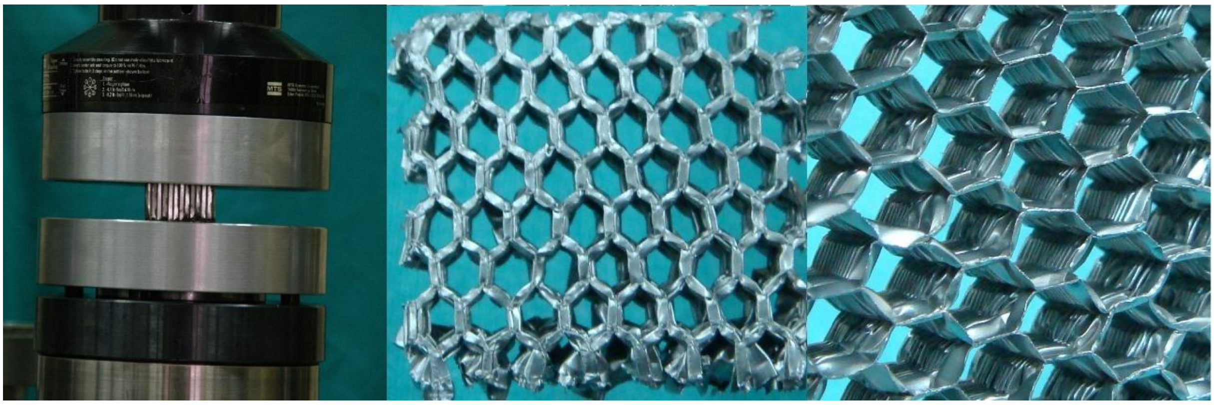

Figure 6. Quasi-static crush test of aluminum alloy honeycomb specimens in out-of-plane direction.

In FEM simulations for honeycomb block models, the honeycomb orthotropic (28) material model is used. Total 12 material parameters are required to model the honeycomb as follows: the Young's modulus in 3 directions, shear modulus in 3 direction and yield stress as a function of strain in 6 directions.

Primary energy absorber simulation: The primary energy absorption tubes are the key component of crush zone system. In order to design the thin-walled tube with high crashworthiness parameters, a comparative study on the crashworthiness parameters of tubes having circular, square, ellipse cross-section, pyramidal and conical shapes was conducted [10]. In the light of finite element simulation results and theoretical studies, it is found that pyramidal tube has the highest crashworthiness parameters with a stable crush behavior. In order to simulate the tube crush behavior, HyperCrash and RADIOSS solver are used. Tubes are modeled as a 3D deformable shell and shell elements of type S4R are used to model the tubes. The tubes have the lengths of 850 mm, 750 mm and 650 mm while the thickness is 6.5 mm. Material properties assigned to the model are those of the mild steel A572- 50. The steel has a yield strength of σ_y =304MPa, Young's modulus of E=207GPa and Poisson ratio of 0.3. For simulation of the dynamic axial crushing, the fixed plate is constrained in all degrees of freedom. The support is tied to the end of the tube, and the mass 16000 kg is attached to the reference point of support. Both the tube and support are impacted to the fixed sill plate with 13.8 m/s downward initial velocity.

In order to evaluate the crush zone performance, the first step is the verification of energy absorber element response. In finite element simulation, as showed in Figures 8, 9 and 10, we model 3 tubes with different lengths in pyramidal shape. In practice, it has advantages to include reinforcements, diaphragm and localized deformation to reduce the initial collapse load and to ensure that crush occurs in a similar manner for various collision speeds and load combination. Figure 8 shows the energy absorption component impact load-crush response for a 13.8 m/s velocity with a mass of 16000 kg. There are 3 high load pick, which correspond approximately to the plastic buckling load of each tube. Except these peak loads, the crush load is quite uniform until the stroke of 0.65 m during crush. Figure 9 shows the energy absorption crushing pattern at different instants. It is clear that the elements have uniform plastic deformation.

Force-displacement curve is a useful tool to evaluate the energy absorption efficiency of an energy absorber. It enables us to determine the peak load, mean load and crushing length of the structure. From the typical force-displacement curve, i.e., Figure. 8, it is observed that initially the tube behaves elastically and as the load increases at the steady state, then initial peak load is followed by a rapid decrease in load. Thereafter, the post-buckling phase is developed with secondary peaks and trough directly related to the formation of subsequent buckling during the crushing process. The secondary and third peak load belongs to the second and third tubes respectively.

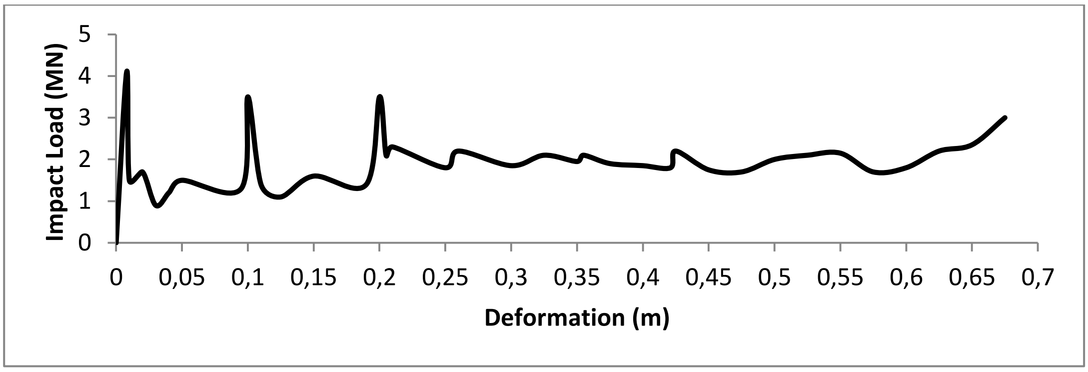

Figure 8. Crushing behavior of primary energy absorber component in crush zone.

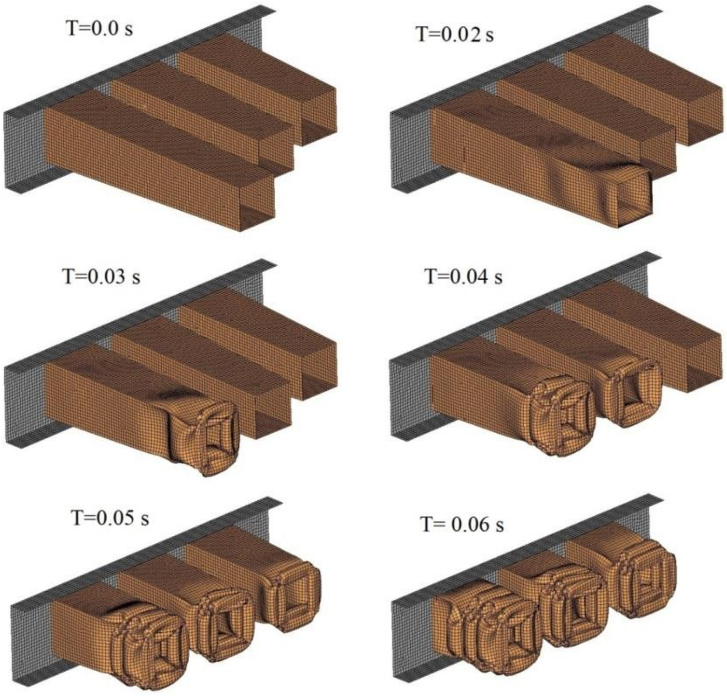

Figure 9. Plastic deformation of primary energy absorber component at different instants.

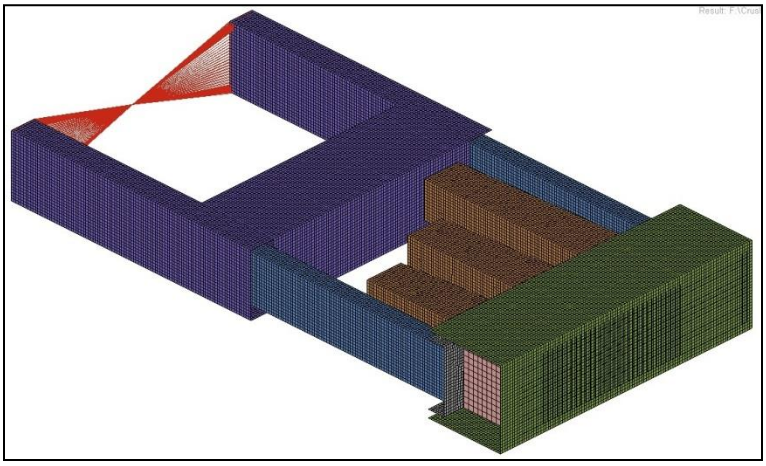

Figure 10. Finite element simulation of half of crush zone.

5. Conclusion

An innovative crush zone system for passenger cars traveling in Turkey railway is suggested which is appropriate with car geometry, mass and traveling conditions and that can easily be integrated to the both used space at the end of the car. Quasi-static experiments of the honeycomb yield good agreement with finite element simulation results verifying the accuracy of FE simulations. Finite element simulation of primary energy absorber and honeycomb box results showed that energy absorbing capacity of this component is sufficient for crush zone energy absorption requirements but in order to decrease the initial peak loads in thin walled tube crush-load curve, holes and reinforcement must be produced in the tubes in practice.

In finite element analysis, because of symmetry in crush zone, only half of it is modeled. In order to analyze the acceleration/declaration parameter and velocity of passengers inside the wagon, it is necessary that after integration of crush zone to the end of car, integrated model is simulated by the finite element method. Various honeycomb structures are analyzed to evaluate the energy absorption characteristics and numerical results are verified by experiments. Also, primary energy absorber components are simulated and their impact force-deformation characteristics are examined. It is concluded that the crush zone mechanism studied in this paper satisfy the design requirements on crush energy absorption and passenger acceleration provided that the associated parameters are properly tuned up.

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