Damage Mitigation of a Steel Column Subjected to Automobile Collision Using a Honeycomb Panel

Hyungoo Kang[1] and Jinkoo Kim[2]

Abstract: This study investigates the performance and design of an aluminum honeycomb panel attached to the face of a steel column for reducing local damage caused by vehicle collision. The dynamic plateau stress of the honeycomb panel is obtained from impact analyses and is used for design of the panel to mitigate vehicle impact. To verify the impact energy-absorption capability of the honeycomb panel designed with the proposed method, a vehicle collision analysis is carried out using a finite-element model of an 8-t truck. According to the finite-element analysis results, the honeycomb panel can be effective in decreasing the damage of the column by absorbing part of the impact energy. Based on the analysis results it is concluded that the proposed design method reflecting the dynamic characteristics of the honeycomb panel can be useful for preliminary design of protective system for columns. **DOI: 10.1061/(ASCE)CF.1943-5509.0001394.**

Author keywords: Honeycomb panel; Vehicle collision; Finite-element analysis; Impact analysis.

Introduction

Vehicle impact on a column can cause severe damage to the column and may lead to progressive collapse of the whole structure (Kang and Kim 2015). One key aspect that makes a structure resistant to collapse is its ability to absorb impact energy without failure. To this end, various energy-absorbing materials or devices are used to protect the column by mitigating impact energy. Especially, honeycomb structures are used in various engineering applications such as in the automobile industry, aircraft, high-speed trains, and so on because of their high energy-absorption capacity and high strength to weight ratio. For use as an energy-absorption device, the strength characteristics of metallic honeycombs should be evaluated because the capacity of the honeycomb depends upon its geometrical configuration.

In order to determine proper configurations of honeycomb structure according to design needs, crush strength properties must be evaluated. In case of a moving deformable barrier, Shkolnikov (2002) investigated the mathematical model of a honeycomb as an energy absorber using physical vehicle impact tests. In that study, numerical modeling of the honeycomb model was obtained from the results of the constitutive model and drop silo tests. Lee and O'Toole (2004) analyzed the internal strain energy of top and bottom sandwich panels subjected to blast load using finite-element analysis and concluded that honeycomb sandwich panels could be used for energy absorption of explosive loads. Jackson et al. (2012) developed the design method for a deployable energy absorber based on a full-scale crash test of an MD-500 helicopter. In their study, the calibration process was performed using four material models available in LS-Dyna software (Mat-63, Mat-26, Mat-181, and Mat 142), and the most similar material model to the actual experimental results was selected.

Caccese et al. (2013) carried out optimal design of honeycomb material used to mitigate impact force and presented a simplified analysis technique using a genetic algorithm to select a minimum honeycomb depth to achieve a desired acceleration level. Han et al. (2016) investigated the behavior of the metallic honeycomb sandwich panels with folded thin metallic sheets to construct a novel core type for lightweight sandwich structures. They showed that it was possible to further improve the mechanical properties of conventional honeycomb-cored sandwich constructions with low relative densities.

As can be seen from these previous studies, a honeycomb structure has sufficient strength as an energy-absorbing device. However, out-of-plane impact properties of the general metallic cushioning materials depend on their configuration parameters and impact velocities (Deqiang et al. 2010). In order to estimate the energy-absorption capacity of the honeycomb cell, parameterization analysis of the plateau stress needs to be performed, taking into account the weight and velocity of the collision object. Based on this process, this study investigates the performance of an aluminum honeycomb panel attached to a steel column for reducing the local damage of the column caused by automobile impact loads.

Behavior of a Honeycomb Panel under Compression

Fig. 1 presents the typical hexagonal honeycomb structure and the stress-strain relation of a honeycomb crushed in out-of-plane direction. When a honeycomb panel is made by welding individual hexagonal honeycomb cells with wall thickness t as shown in Fig. 1, the thickness of the walls shared by two honeycomb cells is $2t$. As is well-known, the out-of-plane direction is the strongest one among the three principle directions of the honeycomb structure. The Young's modulus of the honeycomb along the out-of-plane direction under compression simply reflects that of the material of the cell wall, E_{hc}. Eq. (1) describes the Young's modulus of conventional hexagonal honeycomb ($b = h = l$ and $\alpha = 30°$) for a load in the out-of-plane direction

[1]Senior Researcher, Korea Radioactive Waste Agency, Decommissioning Waste R&D Team, 168, Gajeong-Ro, Yuseong-Gu, Daejeon 34129, Republic of Korea. Email: exult84@gmail.com

[2]Professor, Dept. of Civil and Architectural Engineering, Sungkyunkwan Univ., Chunchun-Dong, Jangan-Gu, Suwon 440-749, Republic of Korea (corresponding author). Email: jkim12@skku.edu

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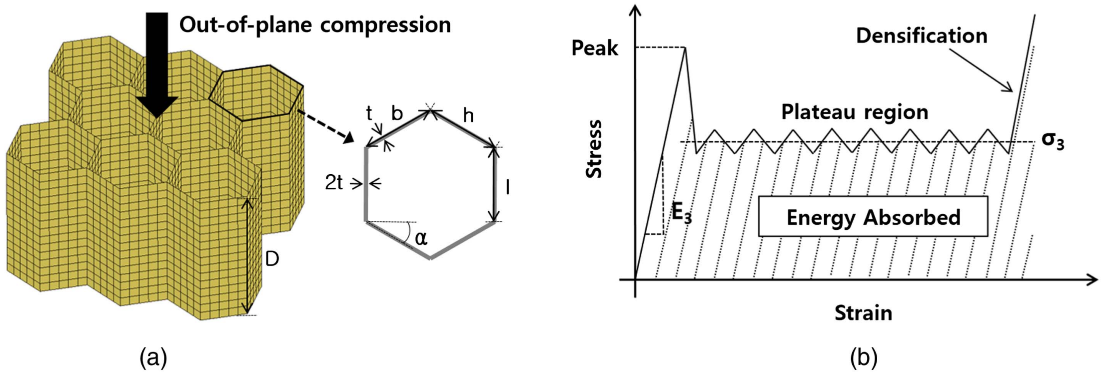

Fig. 1. Mechanical properties of a honeycomb cell: (a) geometry of hexagonal honeycomb cell; and (b) out-of-plane stress-strain behavior of honeycomb under compression.

$$E = \frac{2}{\cos\alpha(1+\sin\alpha)} \times \frac{t_{hc}}{l_{hc}} \times E_{hc} \tag{1}$$

where t_{hc} = thickness of the honeycomb cell wall; l_{hc} = length of the honeycomb cell wall; and α = angle between the two honeycomb cell walls minus 90°.

When the honeycomb structure subjected to the compressive force passes through the linear-elastic region, the honeycomb cell wall undergoes plastic deformation, and brittle fracture may occur depending on the characteristics of the material. In this case, the buckling load of the honeycomb cell can be calculated as Eq. (2) according to the moment of inertia of the cell wall (Zhang and Ashby 1992), and the elastic collapse stress can be calculated as Eq. (3)

$$P_{\text{crit}} = \frac{KE_{hc}}{(1-\nu_s^2)} \times \frac{t_{hc}^3}{l_{hc}} \tag{2}$$

$$\sigma = \frac{8P_{\text{crit}} + 2P_{\text{crit}}}{2\cos\alpha(1+\sin\alpha)l_{hc}^2} = \frac{5KE_{hc}}{(1-\nu_{hc}^2)\cos\alpha(1+\sin\alpha)} \times \frac{t_{hc}^3}{l_{hc}^3} \tag{3}$$

where ν_{hc} = Poisson's ratio of the honeycomb material; and the constant K = end constrain factor. If the vertical edges are simply supported and the height of the honeycomb cell D is large compared with the length of the honeycomb cell wall l_{hc}, then $K = 2.0$. If $D > 3l_{hc}$, the influence of D can be neglected (Gibson and Ashby 1999).

If the net section stress exceeds the fracture strength σ of the cell wall, the cell wall will fracture. This defines the fracture collapse strength of the honeycomb. Wierzbicki (1983) has given a plastic collapse stress for hexagonal honeycombs with a set of doubled walls in each cell as shown in Eq. (4)

$$\sigma = C \times \sigma_{ys}\left(\frac{t_{hc}}{l_{hc}}\right)^d \tag{4}$$

where σ_{ys} = yield strength of the honeycomb material; and coefficients C and d generally used under static loading conditions = 6.6 and 1.67, respectively (Deqiang et al. 2010). In this study, the plateau stress obtained from static analysis is named static plateau stress (SPS) in comparison with the plateau stress obtained by dynamic impact analyses in the following section.

Parametric Study of a Honeycomb Panel Subjected to Impact Loads

Finite-Element Modeling of Honeycomb Panels

Impact analyses of an aluminum honeycomb structure were conducted using the finite-element (FE) simulation software LS-DYNA (2006), which performs nonlinear dynamic analysis using explicit time integration. The honeycomb is modeled with *MAT PLASTIC KINEMATIC (MAT 003); the rigid material is modeled with *MAT RIGID (MAT 020); and the shell element for honeycomb structure is modeled as an element with five integration points using the Belyschko-Tsay element.

In order to define the limit state of the element at excessive deformation, the maximum effective strain of aluminum is set to 0.3, and the elements with deformation exceeding the maximum value are deleted, as done by Ashab et al. (2015). In addition, in case of a load applied at a high speed in a short period of time, such as an impact, high strain-rate effects are accounted for using the Cowper-Symonds model (Cowper and Symonds 1957) as follows:

$$\sigma_y = \left[1 + \left(\frac{\dot{\varepsilon}}{D}\right)^{1/p}\right] \times \sigma_0 \tag{5}$$

where $\dot{\varepsilon}$ = strain rate during dynamic crushing; and the values of D and p for aluminum = 6,500 s^{-1} and 4, respectively (Altenhof and Ames 2002). Table 1 presents the material properties of the aluminum for honeycomb structures.

In the analysis process of a complex structure, the joining condition and contact condition between each element should be defined because the defined contact conditions can lead to different results. In this study the contact condition between

Table 1. Material properties of aluminum

Property	Value
Mass density (t/mm^3)	2.74×10^{-9}
Young's modulus (MPa)	6.90×10^4
Poisson's ratio	0.3
Yield stress (MPa)	292
Strain-rate parameters	
D	6,500
p	4

the rigid impactor and the honeycomb structure is defined by the *CONTACT AUTOMATIC SINGLE SURFACE and *CONTACT AUTOMATIC SURFACE TO SURFACE keywords. The single surface contact is the general type of contact for which the program automatically searches all of the external surfaces within a model to determine if penetration has occurred. The surface to surface contact is a general algorithm commonly used for elements that have large contact areas and when the contact surfaces are known. The contact algorithm is also used when the surface of one body penetrates the surface of another. In this case the contact force can be calculated using the following equation:

$$\text{Contact force} = f_s \times k \times d_p \tag{6}$$

where f_s = penalty scale factor; k = proportional constant; and d_p = distance of the penetration at node. In materials that undergo extremely large deformations, elements may become distorted, and the volume of the element may be calculated as negative. To prevent this phenomenon, the *CONTACT INTERIOR keyword is used to prevent the occurrence of a negative volume due to large deformation in the honeycomb structure.

Parametric Study of the Honeycomb Structure

To validate the accuracy of the FE analysis method, numerical analysis results of a honeycomb panel subjected to an impact load are compared with the experimental data obtained by Xu et al. (2012). The test results of Specimen 4.5-1/8-5052-001 N (t = 0.0254 mm, l = 1.8331 mm, and D = 12.5 mm), numerically modeled in Fig. 2(a), were selected for comparison. The loading condition is quasi-static, and the strain rate is $\dot{\varepsilon} = 10^{-3}\text{s}^{-1}$. Figs. 2(b and c) show the out-of-plane compression test and analysis results of the honeycomb structure, respectively. It is observed that the maximum forces obtained from the test and the FE analysis are 2.5 and 2.4 kN, respectively, and the plateau forces are 1.6 kN in both test and analysis. Even though the overall time histories of the impact force are not exactly the same, the test and the analysis results match quite well in terms of the maximum and plateau forces.

A series of parametric analysis of the plateau stress was performed using the hexagonal honeycomb shown in Fig. 3 for various weights and velocities of the impactor. For the parametric study, the length of the honeycomb cell wall ($b = h = l$) was varied to 30, 40, and 50 mm, and the thickness of the cell wall (t) was varied to 0.1, 0.5, and 1.0 mm. When the height of the honeycomb cell (D) is changed, the overall performance of the honeycomb is affected; however, when the height of the honeycomb cell increases more than three times the cell wall length ($D > 3l$), the constant K in Eq. (3) and height D become independent, and the influence of D can be neglected (Fan et al. 2016). Therefore, in this study, the height D is maintained at a ratio of three times the length of the honeycomb cell.

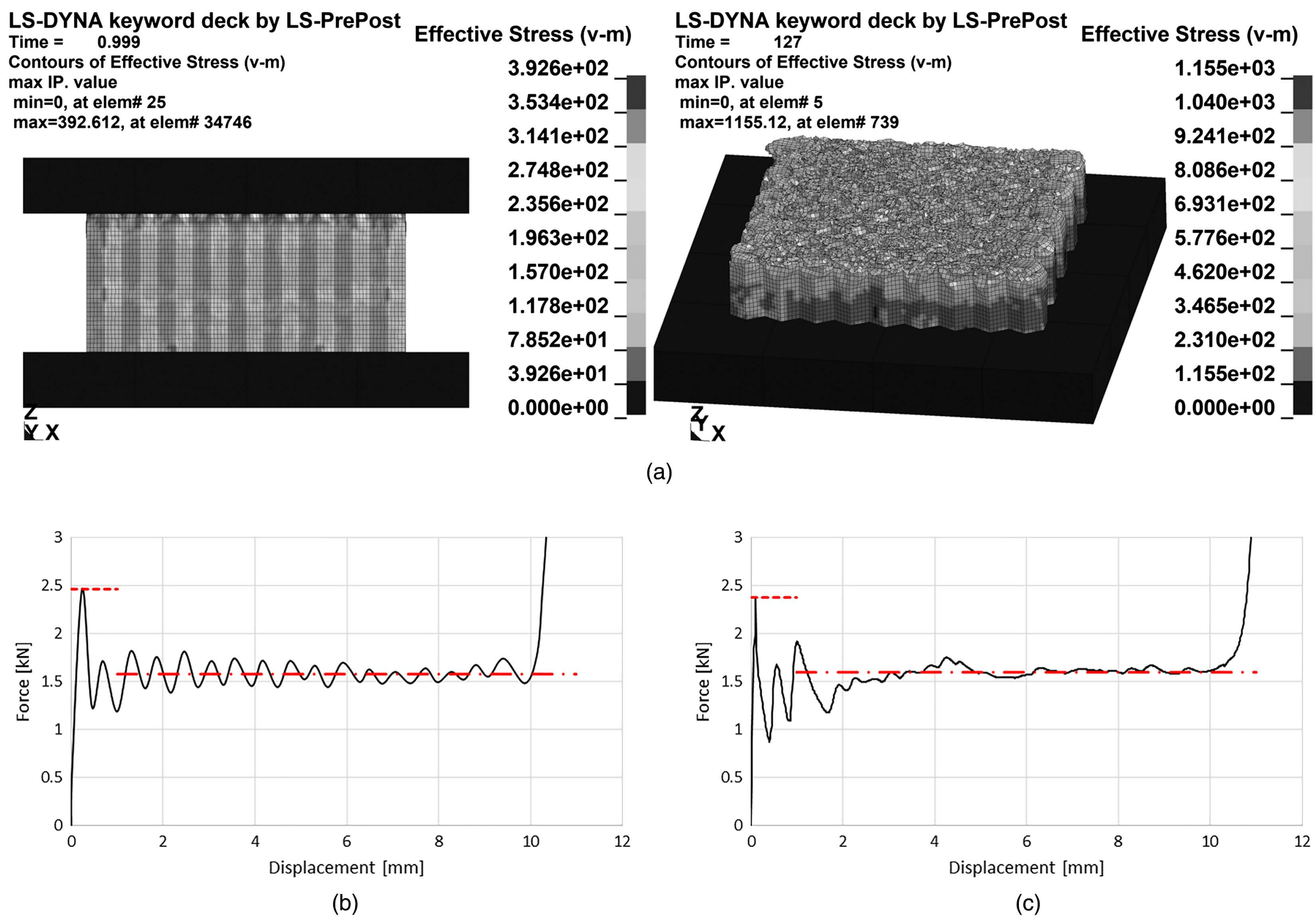

Fig. 2. Comparison of the impact test results obtained by Xu et al. (2012) and numerical analysis results: (a) LS-DYNA FE numerical model of the test specimen; (b) experimental result (Xu et al. 2012); and (c) FE analysis result of the test specimen.

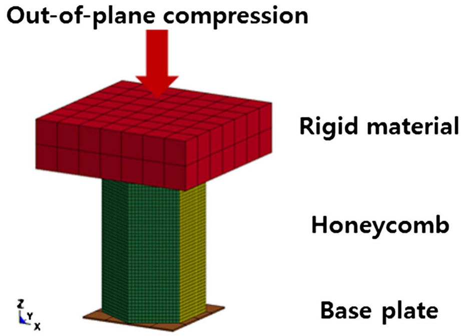

Fig. 3. Impact simulation of the honeycomb structure.

In the honeycomb cell subjected to out-of-plane compressive force, the initial stress is rapidly increased due to the stiffness of the cell wall. After the honeycomb cell is completely deformed, the cell wall is collapsed and the energy-absorption ability is lost. Because the width of the initial peak is quite narrow, the average stress of the plateau region becomes a dominant factor for calculating the energy-absorption capacity of the honeycomb structure.

Fig. 4 shows the deformed shape and vertical stress distribution of the honeycomb cell subjected to the statically applied out-of-plane compressive force. Fig. 5 shows the resultant stress-strain history obtained from the finite-element analysis and from Eq. (4). It can be observed that as the cell wall length becomes shortened or the thickness increased, SPS increases. It is also observed that in case the t/l ratio is low, the analysis results and the prediction by Eq. (4) become similar, and if the ratio is increased, the difference becomes larger, as indicated in Table 2.

Fig. 6 depicts the analysis results and their best-fit line. According to Ashab et al. (2016), the parameters C and d in Eq. (4) are about 2.9 and 1.4, respectively. In this study, the C and d obtained from static analyses are estimated to be 2.12 and 1.47, respectively, based on the best-fit line presented in Fig. 6. Also according to Xu et al. (2012), plateau stress was about 2.3 when t/l was about 0.014 under static conditions similar to those in this study.

To obtain the parameters C and d under dynamic load, 36 cases of impact analyses were carried out with various thicknesses of honeycomb cell wall (0.1, 0.5, and 1.0 mm) and length b, h, and l (30, 40, and 50 mm) at an impact velocity of 50 and 100 km/h. The mass of the impactor is 10 and 20 kg. Fig. 7 and Table 3 provide the coefficients C and d obtained under dynamic impact loading condition. The results show that the coefficient C obtained from the dynamic analysis is reduced to 23%–49% of the values obtained under static load depending on the mass and speed of the impactor. The coefficient is reduced more significantly as the mass and speed increase. Also, the coefficient d is reduced to 88%–76% of the static value depending on the mass and speed of the impactor. As with the coefficient C, more reduction is observed when the mass and speed of the impactor increase. These observations show that the stress state and consequently the absorbed energy under impact load may be quite different from those obtained from static analysis. Also, it seems to be possible to estimate the plateau stress of the honeycomb under dynamic load depending on the impact velocity and mass of the impactor.

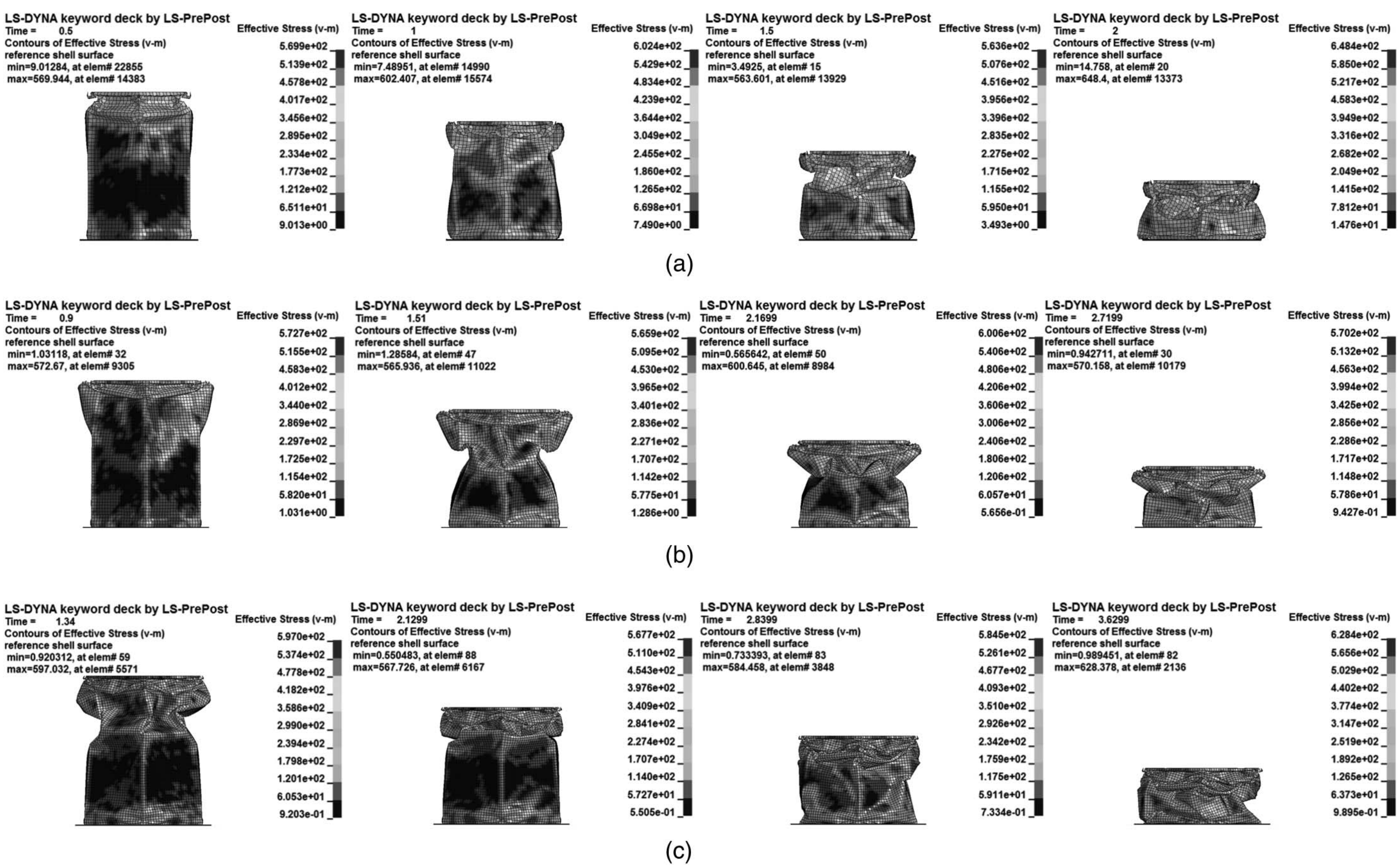

Fig. 4. Stress distribution of the honeycomb cell under compression with $t = 0.1$ mm: (a) $b = 30$ mm; (b) $b = 40$ mm; and (c) $b = 50$ mm.

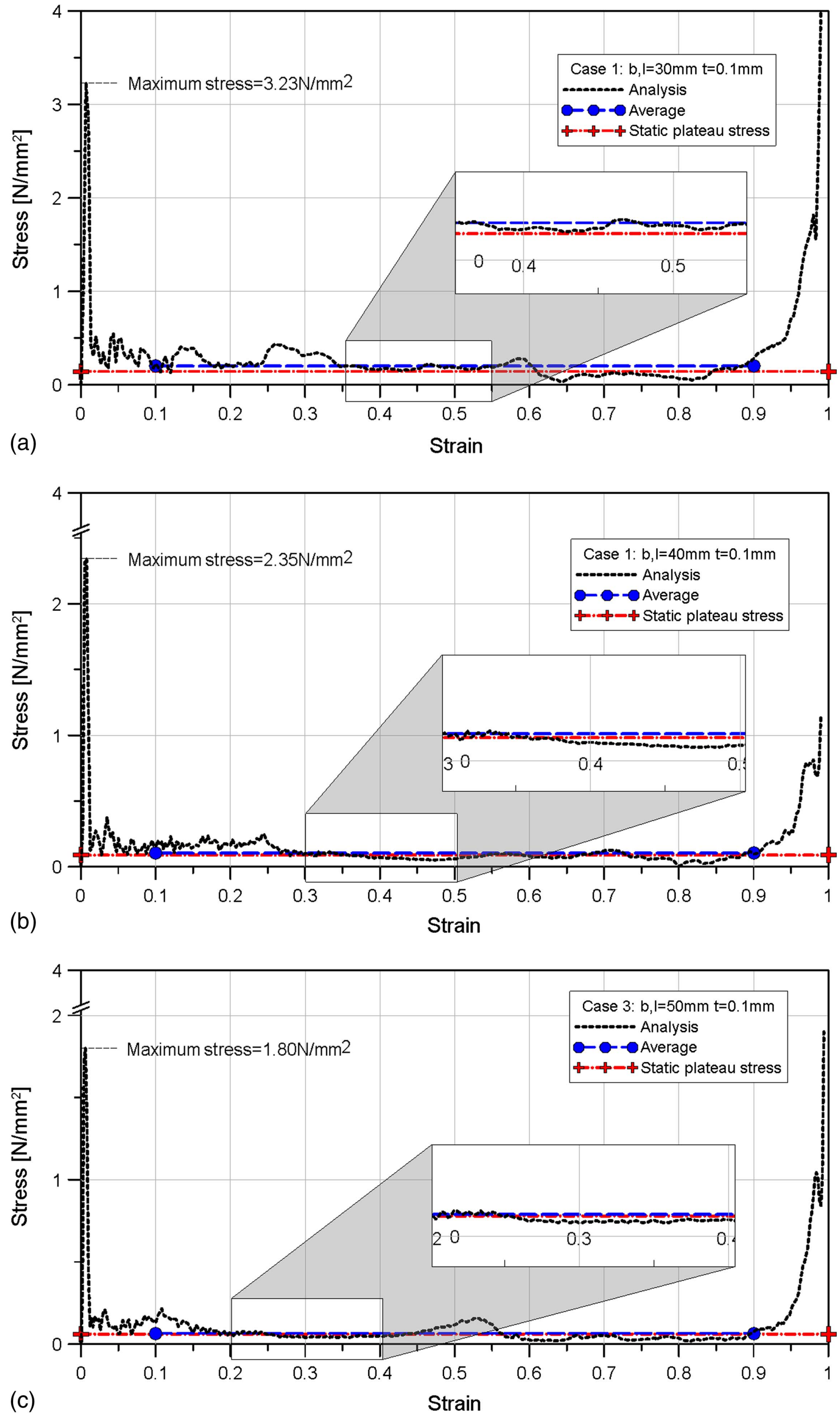

Fig. 5. Stress-strain relationship of a single honeycomb cell under compression with $t = 0.1$ mm: (a) $b = 30$ mm; (b) $b = 40$ mm; and (c) $b = 50$ mm.

Table 2. Analysis result of SPS

	t/l ratio								
Plateau stress (N/mm^2)	0.002	0.003	0.007	0.010	0.013	0.017	0.020	0.025	0.033
Calculated result	0.06	0.09	0.46	0.89	1.30	2.10	2.84	4.12	6.65
Analysis result	0.07	0.10	0.14	1.23	1.71	2.50	2.11	2.19	2.79

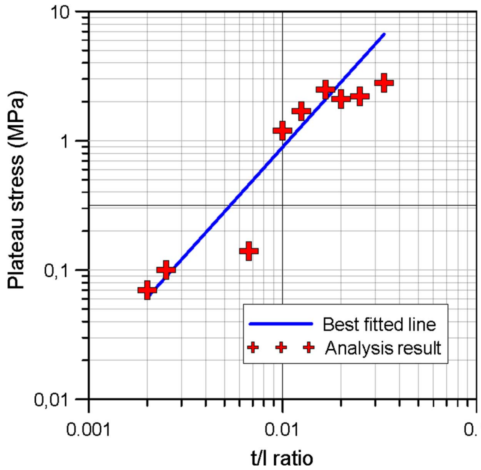

Fig. 6. Relationship of t/l ratio and static plateau stress.

Table 3. Analysis result of DPS

State	Loading condition		$C_{dynamic}$	$d_{dynamic}$
	Mass (kg)	Velocity (km/h)		
Static	Original value of Eq. (4)		6.6	1.67
Dynamic	10	50	3.23	1.47
	20		2.56	1.41
	10	100	1.96	1.34
	20		1.51	1.27

Impact Analysis of the Honeycomb Panel

In this section, impact analyses of a honeycomb panel are performed to obtain the force-displacement relationship and absorbed energy of the panel. The mass and speed of the impactor are set to be 1,000 kg and 50 km/h, respectively. The wall thickness of the unit cells forming the honeycomb panel is 0.1 mm, the cell wall length b, h, and l are 30 mm, and the height D is 90 mm. The total number of honeycomb cells in the panel is 116, and the overall length and width of the panel are 960 and 311 mm, respectively, as shown in Fig. 8.

The area of the stress-strain curve estimated by the impact analysis of the honeycomb panel can be defined as the energy-absorption capacity of the honeycomb structure. By calculating the dynamic plateau stress (DPS) and modulus of elasticity under a given load condition, the energy-absorption capacity can be calculated using the linearized stress-strain relationship shown in Fig. 9. The plateau stress is calculated using Eq. (4) with the coefficients C and d obtained from the parametric study. In Fig. 9, the maximum and the densification point obtained from the impact

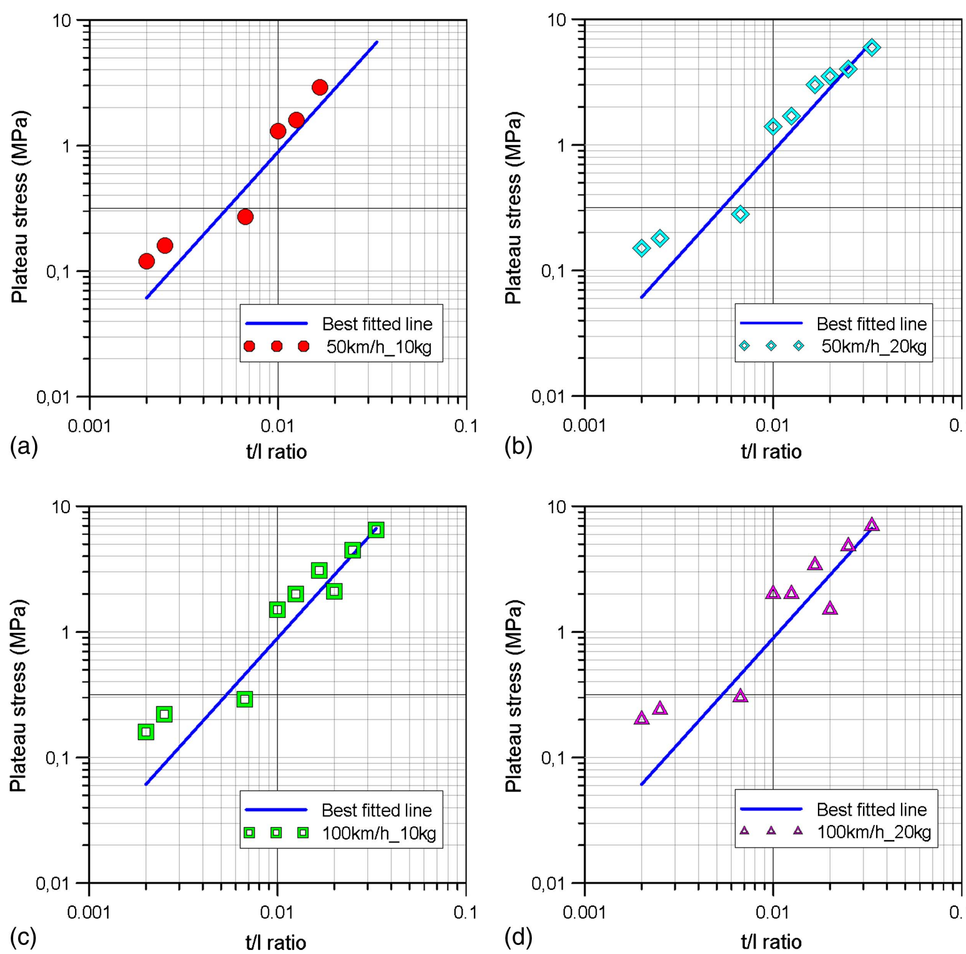

Fig. 7. Relationship of t/l ratio and dynamic plateau stress: (a) impact velocity = 13 m/s and mass of impactor = 10 kg; (b) impact velocity = 13 m/s and mass of impactor = 20 kg; (c) impact velocity = 27 m/s and mass of impactor = 10 kg; and (d) impact velocity = 27 m/s and mass of impactor = 20 kg.

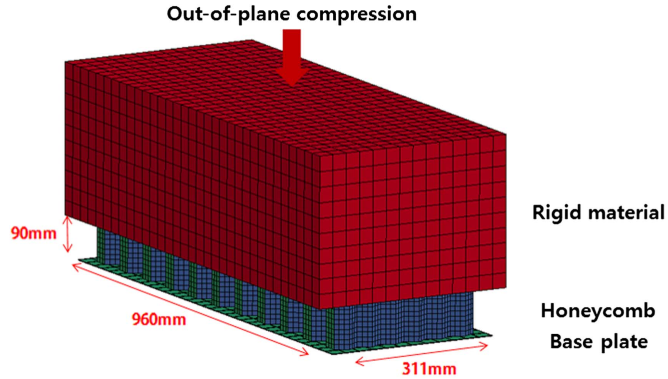

Fig. 8. Out-of-plane impact analysis of the honeycomb panel.

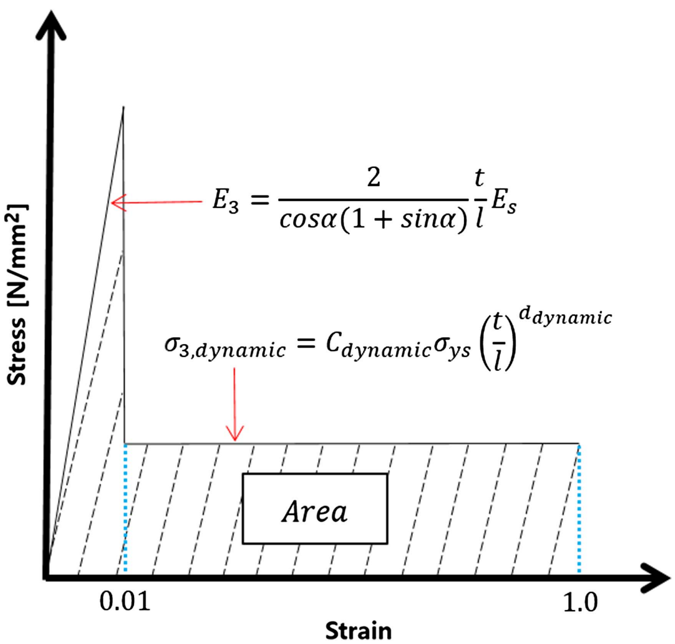

Fig. 9. Linearized stress-strain curve of honeycomb panel.

analyses are approximately 0.01 and 0.9, respectively, which are used in the following study.

Fig. 10 depicts the stress-strain relationship of the honeycomb panels with three different heights subjected to the impact force. Also shown are the linearized curve using the plateau stress coefficients obtained from both the static and dynamic analyses. The elastic moduli are calculated using Eq. (1), which are 354, 265, and 212 MPa for the panels with height of 9, 12, and 15 cm, respectively. The coefficients C and d in Eq. (4) are calculated as 0.58 and 1.06, respectively, as a result of linear interpolation of the impact analysis results in Table 3. The DPS values of the three panels calculated using the estimated coefficients C and d are 0.41, 0.30, and 0.24 $\mathrm{N/mm^2}$, respectively, and they are converted to the linearized forces of 110,156, 81,237, and 64,146 N, respectively. It can be observed that the linearized force obtained using DPS is much closer to the mean value of the finite-element analysis result than the force obtained using the SPS.

Table 4 presents the area under the force-displacement curve, which represents the absorbed energy, of three different honeycomb panels obtained from impact analysis. The difference of the area under the curve obtained from the impact analysis, and the linearized one is 36%, 21%, and 14%, respectively, for honeycomb panels with $b = h = l = 30$, 40, and 50 mm. This shows that the absorbed energy estimated by the linearized force based on the dynamic plateau stress coefficients becomes closer to the impact analysis result as the size of the honeycomb cell increases from 30 to 50 mm.

Design of a Honeycomb Panel for Absorbing Target Impact Energy

In the case of vehicle terrorism, the lower part of a first-story column is most vulnerable because it is directly exposed to the impact. Damage to the first-story column caused by vehicle impact may lead to progressive collapse of the whole structure (Kang and Kim 2017). The purpose of this section is to develop a simple design process of honeycomb panels to be installed on the surface of first-story columns to reduce the damage and consequently to prevent overall progressive collapse.

The first step for design is to determine the target impact energy to be reduced. The kinetic energy of an 8-t truck provided by the NTRC (2017) moving at the speed of 100 km is estimated to be about 3,080,000 J. When a vehicle collides with a structure, a large amount of energy is absorbed by the deformation of the vehicle. In addition, because the size of the vehicle colliding with the structure is much larger than the size of the honeycomb panel installed on the column at the collision point, the energy absorbed by the vehicle is generally significantly larger than the energy absorbed by the honeycomb panel. Based on a series of computation on similar cases, it is assumed that the energy to be absorbed by the honeycomb panel is 15% of the kinetic energy generated by the vehicle, which is 450,000 J (Nm).

Next, a honeycomb panel having the predetermined target energy-absorption capability was designed. Based on the assumption that all honeycomb cells in the panel are uniformly and completely squeezed due to the impact, the absorbed impact energy of the honeycomb panel can be derived using Eq. (7) as follows:

$$J = \frac{E_{hc} \times \varepsilon_{\mathrm{peak}}^2 \times A_{hc} \times h_{hc}}{2} + \left[\left(h_{hc} - (h_{hc} \times \varepsilon_{\mathrm{peak}}) \times \left(\left(C_d \times \sigma_{ys}\left(\frac{t_{hc}}{l_{hc}}\right)^{d_d}\right) \times A_{hc}\right)\right]\right. \tag{7}$$

where $\varepsilon_{\mathrm{peak}}$ = strain at the maximum stress point; and h_{hc} and A_{hc} = height and horizontal area of the honeycomb panel, respectively. The coefficients C_d and d_d are obtained from the dynamic plateau stress.

The dimensions of the honeycomb panel are determined as follows using the target absorbed impact energy and Eq. (6): the wall thickness of the unit cell constituting the honeycomb panel is 1.0 mm; the cell wall lengths b, h, and l are 30 mm; and the height D (thickness of the honeycomb panel) is 180 mm. The overall length and width of the panel are 960 and 311 mm, respectively. The total number of honeycomb cells included in a given panel is 116. For this panel, the DPS at the impact condition defined previously is 1.64 $\mathrm{N/mm^2}$, and the energy-absorption capacity of the designed honeycomb panel is estimated to be 453,178 Nm if the panel is uniformly deformed up to the densification point.

However, the deformation in each honeycomb cell is not uniform during automobile collision but varies depending on the size of the impactor and location of impact (i.e., contact point of the

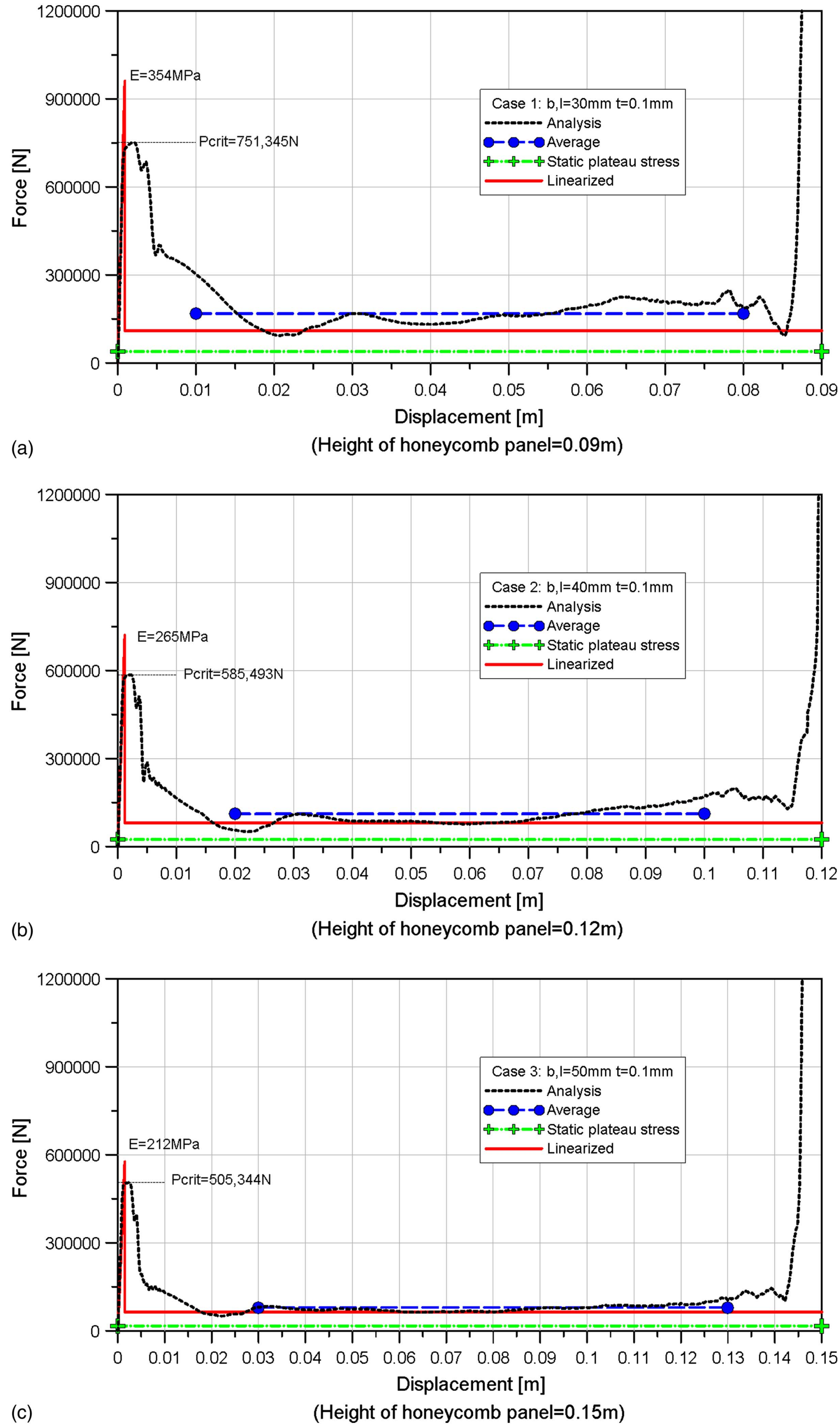

Fig. 10. Stress-strain relationship of the honeycomb panel under compression with $t = 0.1$ mm: (a) $b = 30$ mm; (b) $b = 40$ mm; and (c) $b = 50$ mm.

Table 4. Analysis result of honeycomb panel

Case	Dimensions			Absorbed energy (Nm)		Dynamic plateau stress (N/mm^2)
	b (mm)	l (mm)	t (mm)	Analysis	Linearized	
1	30	30	0.1	15,985	10,247	0.41
2	40	40	0.1	12,688	10,083	0.30
3	50	50	0.1	11,565	9,957	0.24

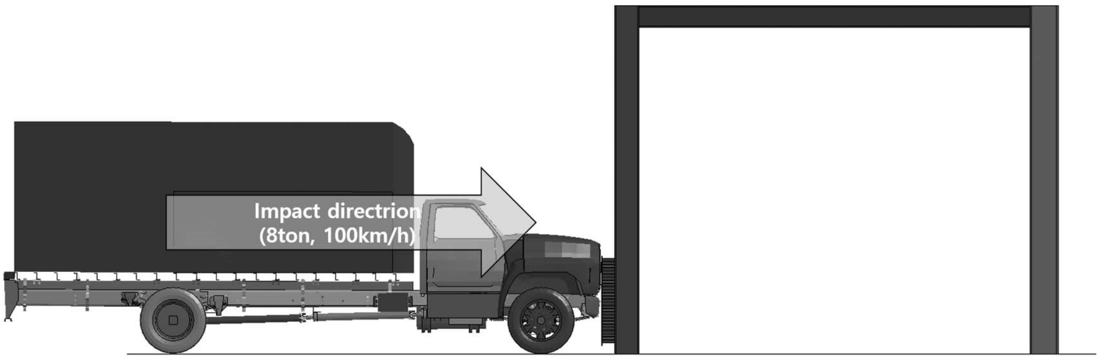

Fig. 11. Truck collision analysis of a simple frame.

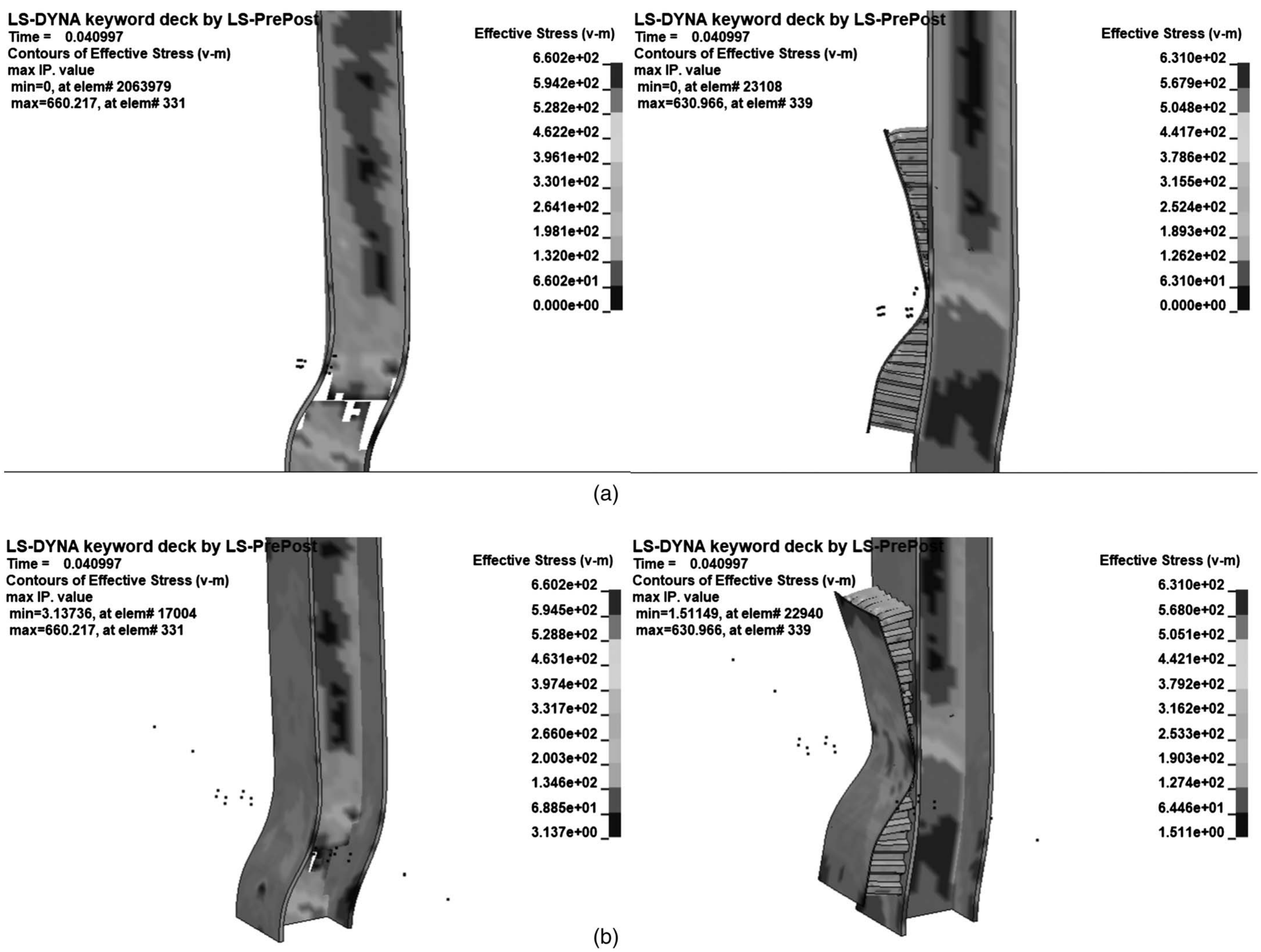

Fig. 12. Stress distribution at damaged column with honeycomb panel ($t = 0.4$ s): (a) side view; and (b) three-dimensional view.

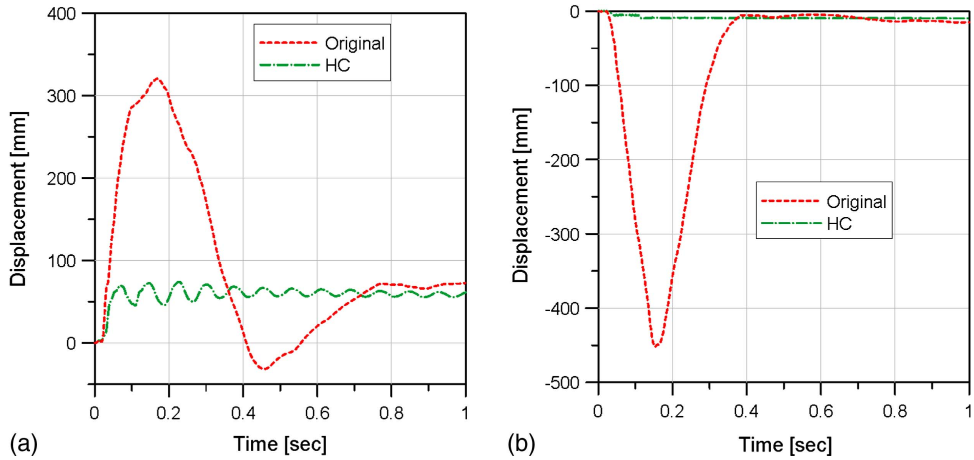

Fig. 13. Displacement time histories of the column due to the truck collision: (a) horizontal displacement; and (b) vertical displacement.

bumper). In this study, a 5-mm-thick steel plate is placed on the front surface of the honeycomb panel to induce more uniform damage distribution. According to the preliminary analysis results, however, the presence of the steel plate with a thickness in a practical range does not make a significant difference in the overall damaged configuration of the panel. Therefore, it is expected that only half of the impact energy absorbed in the impactor analysis conducted in the previous section will be absorbed in the automobile collision analysis. This is based on the assumption that the deformed shape (side view) of the panel subjected to vehicle collision is triangular, in which the mean deformation of the panel is only about half of the full densification level.

The honeycomb panel is installed on the surface of the single story frame column, as shown in Fig. 11, in a position where its center contacts with the bumper of the automobile. The height and span length of the single-story frame are assumed to be 5 and 6 m, respectively, and the contact condition and the analysis modeling used in the finite-element analysis are the same as done in previous section. The beam and column are modeled using A36 and A572 steel, respectively.

Fig. 12 shows the damaged shape and the stress distribution in the column before and after installation of the honeycomb panel. It can be observed that when the honeycomb panel is not installed, the web of the H-shaped column is completely severed. However, after the honeycomb panel is installed, the column remains stable, even though significant stress is concentrated in the web.

Fig. 13 shows the horizontal and vertical displacement time histories of the column caused by the truck collision. Fig. 13(a) shows the horizontal displacement at the contact with the vehicle, and Fig. 13(b) shows the vertical displacement at the top of the column subjected to the collision. It can be observed that both horizontal and vertical displacements are significantly reduced as a result of the installation of the honeycomb panel.

Fig. 14 shows the actual and simplified deformation shape of the honeycomb panel after the truck collision. It can be observed that the overall configuration of the honeycomb panel after the collision is triangular in shape, with the maximum deformation at the contact point with the bumper. The mean deformation of all honeycomb cells is half of the full densification point. This implies that the amount of energy absorbed by the honeycomb panel during the truck collision will be only half of the energy previously predicted by the impactor analysis where all honeycomb cells are uniformly deformed.

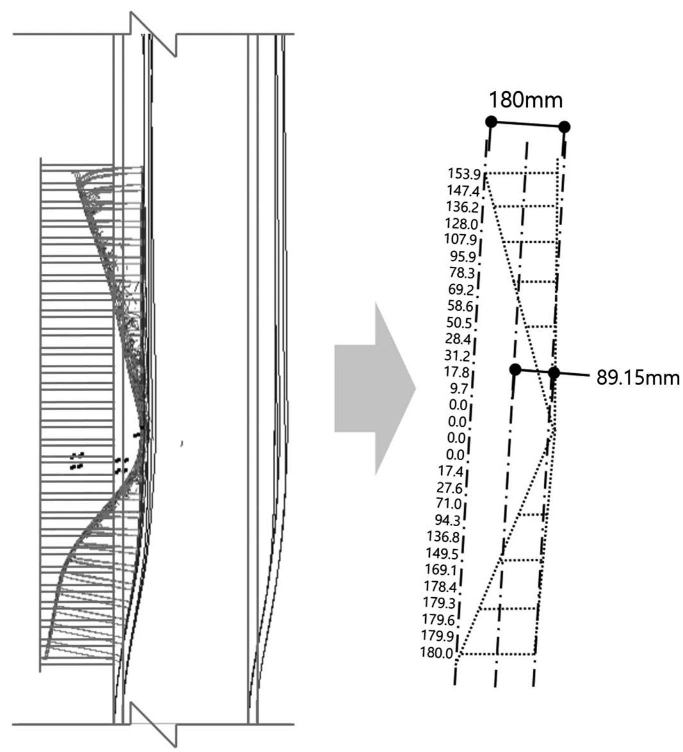

Fig. 14. Mean deformation of the honeycomb panel after the truck collision.

Fig. 15 shows the time history of the absorbed energy at each component before and after the installation of the honeycomb panel caused by the truck collision. It can be observed that in both cases, most of the energy is absorbed by the damaged truck. Also, a certain amount of kinetic energy is absorbed during the deformation process of the frame. However, after installing the honeycomb panel, the amount of energy absorbed by the structure is reduced almost to half, and the remaining half is absorbed by the honeycomb panel. Even though the energy absorbed by the panel is not large compared with the total impact energy, the panel is quite effective in damage mitigation of the column considering its small size in comparison with the size of the structure or the truck. It also can be noticed that the absorbed energy in the honeycomb panel

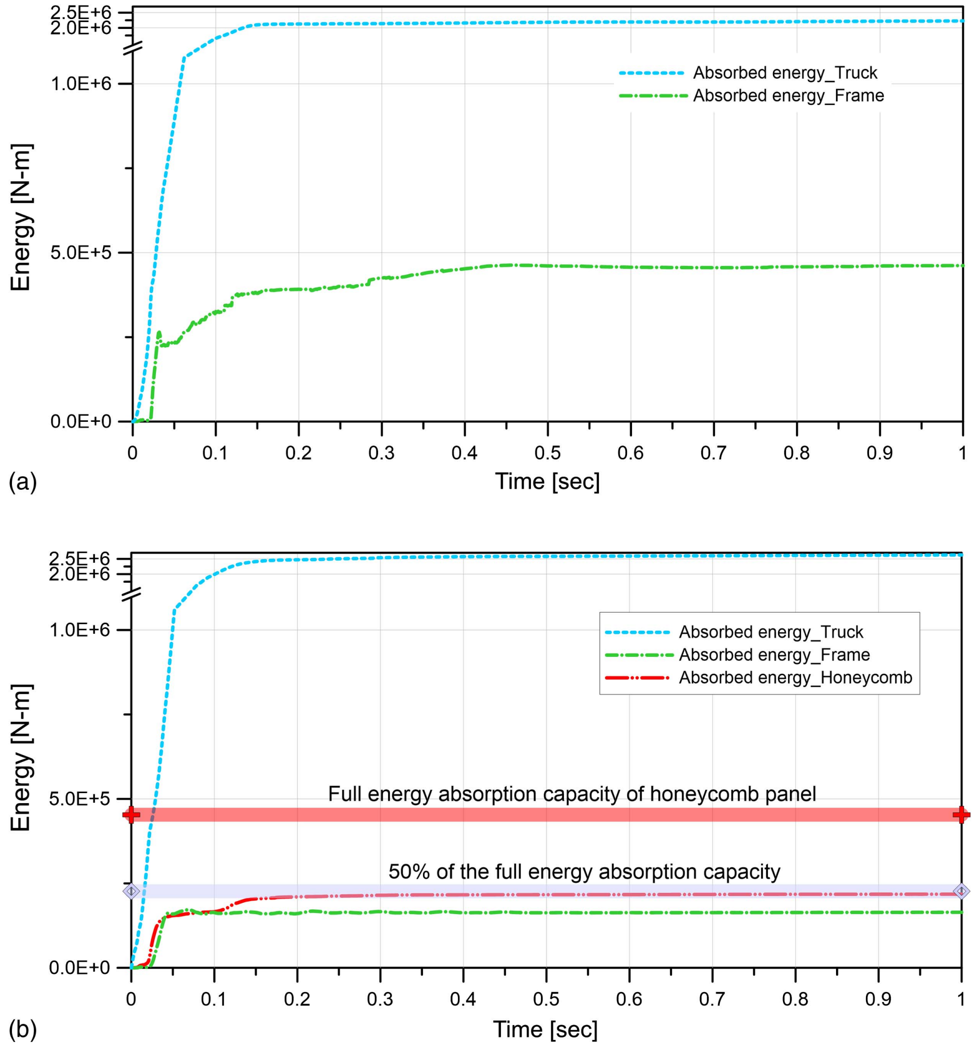

Fig. 15. Time history of the impact energy caused by the truck collision: (a) time history of the bare frame; and (b) time history of the installed frame.

during the collision is approximately half of the energy absorbed in the full densification state. This can be expected from the deformed shape of the honeycomb panel shown in Fig. 14. Therefore, it would be recommended that approximately half of the energy absorbed in the impact test be utilized for design of the honeycomb panel to protect a column against vehicle collision.

Conclusions

This study investigated the performance of an aluminum honeycomb panel attached to the face of a steel column for reducing local damage caused by automobile collision. A method for estimating the dynamic plateau stress of the honeycomb was proposed based on the mass and velocity of the vehicle. In addition, the stress-strain histories were linearized to easily estimate the amount of energy absorbable by the honeycomb panel. To verify the impact energy-absorption capability of the honeycomb panel designed with the proposed method, a vehicle collision analysis was carried out using an 8-t truck and a single-story steel frame.

The parametric study of a honeycomb structure showed that it was possible to estimate the plateau stress of the honeycomb structure subjected to an impact load using a simple formula as a function of impact velocity and mass of the impactor. The vehicle collision analysis results showed that the honeycomb panel applicable in size for practice could be effective in decreasing the displacement of the structure due to vehicle collision. It was also shown that the simple design process developed based on the dynamic plateau stress of the honeycomb panel could be applicable to protect a column from total damage.

In this study, the numerical analysis was validated by test data obtained from dynamic test of a small honeycomb panel. However, it should be pointed out that further study is still required for validation of the proposed design and simulation techniques by comparison with more realistic case studies, such as vehicle collision test on columns.

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