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Study of High Velocity Impact Loading in Honeycomb Sandwich Panels Reinforced with Polymer Foam: Numerical Approach

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ABSTRACT

The employment of lightweight structures is one of the most important goals in various industries. The lightweight sandwich panel is an excellent energy absorber and also a perfect way for decreasing the risk of impact. In this paper, a numerical study of high-velocity impact on honeycomb sandwich panels reinforced with polymer foam was performed. The results of the numerical simulation are compared with experimental findings. The numerical modelling of high velocity penetration process was carried out using nonlinear explicit finite element code, LS-DYNA. The aluminum honeycomb structure, unfilled honeycomb sandwich panel, and the sandwich panels filled with three types of polyurethane foam (foam1: 56.94, foam2: 108.65 and foam3: 137.13 kg/m^3) were investigated to demonstrate damage modes, ballistic limit velocity, absorbed energy, and specific energy absorption (SEA) capacity. The numerical ballistic limit velocity of sandwich panels filled with three types of foam were more than the bare honeycomb core and unfilled sandwich panel. In addition, the numerical results show that the sandwich panel filled with the highest density foam could increase the strength of sandwich panel and the numerical specific energy absorption of this structure is 23% more than unfilled. Finally, the numerical results were in good agreement with experimental findings.

Keywords: Sandwich panel, Numerical simulation, Honeycomb structure, Polyurethane foam, Ballistic limit velocity, Absorbed energy

1. Introduction

Sandwich structure generally consists of two thin stiff skins and a lightweight thick core. Based on the specific operation requirement, different types of core shapes [1,2] and core material [3-5] have been used in sandwich structures. Among them, the honeycomb core that consists of very thin foils in the form of hexagonal cells perpendicular to the facings is the most common one [6]. Honeycomb sandwich structures have very low weight, high stiffness, and strength [7] which make them applicable in many industrial fields such as high-speed ground and air vehicles, shipbuilding and so on [8,9].

Due to the widespread usage of these structures, many researchers have carried out experimental investigations to realize the mechanical response of honeycomb sandwich panels composed of different skins and core materials under various loadings [10-12]. One of the most important loading condition is impact loading, which usually occurs at high velocities. Since honeycomb sandwich structures are extensively utilized in the aerospace engineering and there is always the possibility of sudden high-speed impacts (74-116 m/s) such as birds, hailstones or pebbles strike, the understanding of their mechanical behavior attracts a great deal of attention [13,14]. On the other hand, the experimental study of high velocity loading on the sandwich panels is totally time-consuming and expensive, therefore, it is essential to use accurate numerical modelling to predict their behavior under different conditions.

Up to now, numerical modeling on the individual honeycomb structure and the core of the sandwich structure were carried out. For example, Feli and Pour [15] proposed a method for modeling the penetration of composite sandwich panels with honeycomb core under high velocity impact. The residual velocity, penetration time, velocity-time history of the bullet, and energy absorption by the sandwich panel were estimated by analytical simulations. The results showed an adequate consistency with experimental and numerical results. Li et al. [16] studied the energy absorption properties of hexagonal metal honeycombs. They used the response surface method (RSM) for size optimization of the metal honeycomb energy absorber and found this method is very effective in solving crashworthiness design optimization problems. Also, they carried out the parametric studies using LS-DYNA and the influences of foil thickness and cell length on the metal honeycombs' crash performances were investigated. Buitrago et al. [17] studied the penetration of composite sandwich structures under high velocity impact. They modelled aluminum honeycomb core sandwich panels with carbon/epoxy sheets by a three-dimensional finite element model performed in ABAQUS/Explicit. The finite element confirmed models were verified by comparing numerical and experimental residual velocity, ballistic limit, and contact time. The effect of the skins and core on the performance of the sandwich panel subjected to ballistic test was estimated using this model and the influence of the failure mechanisms on the energy absorption from the projectile kinetic energy was distinguished.

High stiffness and durability at the given minimum weight make sandwich panels remarkable for using as components of aerospace vehicles, modern aircrafts, boats, building constructions and other applications where weight saving plays an important role [18]. This essential advantage of the sandwich structures is owing to the fact that a lightweight core separates two thin, stiff and strong face sheets. This separation enhances the structure resistance under compressive loadings. The core must offer the structure stiffness in the transverse direction for the purpose of avoiding the sliding of face sheets over each other [19].

Various core materials and core formations have been suggested to date. The most frequently applied core materials are honeycomb and foams [20,21]. The foam cores are ideally used when the waterproof, sound and heat insulation qualities of cores are essential. Moreover, the foam cores are the least expensive among core materials and can provide some benefits in sandwich manufacturing. The honeycomb cores have a higher stiffness to weight ratio compared to foam core materials. However, the weakest point of such cores is the small bonding area of honeycomb cells to the face sheets. Manufacturing defects, in-service conditions or mechanical loading. The filling of honeycomb cells with foam can be considered as a deterrent to debonding which leads to the production of new types of sandwich cores taking the advantages of both honeycomb and foam cores. In addition, the modification of the dynamic properties of the honeycomb sandwiches is another important advantages of the increased bonding area of foam-filled honeycomb cells [19].

After a comprehensive literature review, it has been concluded that to the best knowledge of the authors, no research has been carried out on the numerical analysis of high velocity impact on honeycomb sandwich panels filled with polymer foam. In this paper, honeycomb panels filled with different polyurethane foams were modeled. The aluminum honeycomb structure, the unfilled honeycomb sandwich panel, and the three types of polyurethane foam filled sandwich panel were simulated by LS-DYNA software. The analysis of the high velocity impact loading on specimens by flat ended projectiles was carried out. Also, the process of damage, the effect of foam on ballistic limit velocity, absorbed energy, and specific energy absorption (SEA) for all specimens were demonstrated and compared with the results of experimental study.

2. Materials

The sandwich structure consists of two skins and a core. In this research, sandwich structures are made from aluminum skins, unfilled or polyurethane foam filled honeycomb core and epoxy resin for bonding the skins to the core. Physical and mechanical properties of aluminum skins, honeycomb structure, three types of foam (foam 1, foam 2 and foam 3) and epoxy resin were explained in the previous study [21].

The honeycomb structure was made of 5052-H38 aluminum using corrugation process. The properties of 5052-H38 aluminum are given in Table 1. The aluminum skin was grade 1200 with 0.5 mm thickness supplied by Arak Aluminum Company. Tensile test has been done on this aluminum according to the ASTM E8M-04 standard (Figure 1). The mechanical properties obtained from these test are shown in Table 1. Commercially available closed-cell polyurethane foams (SKC501, SCC500) were used in the current study. Two types of experiments were performed to determine the physical and mechanical properties of polyurethane foams.

Polyurethane foam consisted of two organic units including Isocyanate and Polyol groups. Two types of foam (SKC501, SCC500) with different Polyol groups were used to make three type of foams by mixing them with various percentages of Isocyanate group to create PU foam 1 (Weak), PU foam 2 (medium) and PU foam 3 (strong) foams. The mixing ratios and commercial foam codes are presented in Table 1.

The apparent density of each type of polyurethane foams which were made to fill the honeycomb panels were 56.94, 108.65 and 137.13 kg/m^3, respectively. Foam densities were determined based on ASTM D1622 standard. At first, a foam filled cast with dimensions of 20 × 20 × 7 cm^3 was prepared and three test specimens of each foam type were cut to dimensions of 30×30×30 mm^3. Then the samples were weighted by a scale with a precision of 0.00001 and the standard deviation for each foam was calculated using equation 1 and its results were reported in Table 2.

$$s = \sqrt{\frac{\sum X^2 - nX^2}{n-1}} \tag{1}$$

Where s is the estimated standard deviation, X is the value of a single observation, n is the number of observations.

Static compressive tests were carried out according to ASTM D1621 standard using a Universal Testing Machine (model WDW-300E) at displacement rate of 2 mm/min. The size of specimens was 30×30×30 mm^3 (Figure 2) [22]. Five samples were prepared for each density and tested. Figure 2 shows the stress-strain curves for three types of foams. The compressive stress of the foam (σ_c) and compressive modulus of the foam (E_c) are summarized in Table 1.

3. Numerical analysis

In this study, the finite element analysis (FEA) was carried out by the nonlinear explicit finite element software, LS-DYNA. The model consists of two parts; the projectile, and the target with its components. The projectile was a rigid and flat-ended cylinder with 15 mm length, 10 mm diameter, 8.5 g mass and 60 RC hardness and it was modeled with 8 node solid164 elements. The aluminum skins were 75×75 mm^2 with 0.5 mm thickness (Figure 3a) and they were modeled with 4 node shell163 elements. The modelled honeycomb structure is 75×75×19.15 mm^3 and the geometry of a cell is demonstrated in Figure 3. For modelling the honeycomb geometric shape, one cell was simulated according to the dimensions stated in Figure 3. Then, the coordinates of cell were copied to X and Y axes to create the whole honeycomb structure (Figure 3b, 3c). The honeycomb structure was also modeled with 4 node shell163 elements. Honeycomb structure was modeled with 213 cells. After studying the mesh sensitivity, each cell wall was meshed with an element size of 1.19 mm (Figure 4).

For modelling the sandwich panel, the honeycomb was simulated as described above (Figure 5a). The geometry of the foam was simulated according to the dimensions of a honeycomb cell. As shown in exploded view in Figure 5b, the components of the foam filled sandwich panel were modeled, separately. The polyurethane foam was modeled with 8 node solid164 elements.

Material model 20 (*MAT_RIGID) was chosen for projectile. Aluminum skins and aluminum honeycomb structure were modeled with material model 3 (*MAT_PLASTIC_KINEMATIC), and polyurethane foam with material model 63 (*MAT_CRUSHABLE_FOAM).

Plastic kinematic model is based on Cowper-Symonds strain rate hardening model and isotropic hardening effect is considered. Strain rate is accounted for using this model which scales the yield stress by the strain rate dependent factor which is given by equation 2.

$$\sigma_Y = \left[1 + \left(\frac{\dot{\varepsilon}}{C}\right)^{\frac{1}{p}}\right]\left(\sigma_0 + \beta E_p \varepsilon_p^{eff}\right) \qquad (2)$$

where σ_0 is the initial yield stress, $\dot{\varepsilon}$ is the strain rate, β is hardening parameter, C and P are the Cowper-Symonds strain rate parameters [23], ε_p^{eff} is the effective plastic strain, and E_p is the plastic hardening modulus which is given by equation 3.

$$E_p = \frac{E_{tan}E}{E-E_{tan}} \tag{3}$$

Where E_{tan} is tangent modulus. The coefficients of the Cowper- Symonds equation for aluminum skins and honeycomb structure are given in Table 3 [21,24].

The best candidate for modeling polyurethane foam is crushable foam model. The input data necessitated for this material model are included five parameters such as density of material, modulus of elasticity, Poisson's ratio, stress-strain curve, tensile stress cutoff, and damping coefficient. These parameters for polyurethane foams are given in Table 3 [21]. There is a value between 0.05 and 0.5 for the viscous damping coefficient (DAMP) in the LS-DAYNA software. According to [25], the effect of DAMP is reliant on the mesh density and its effect reduces with improved mesh density, approaching 'zero effect'. Also, DAMP is not related to material properties of polyurethane foam which has been mentioned in ref. [25]. In addition, the use of a very low damping coefficient for the solution stability has been advised. Thus, 0.1 for the DAMP which was also suggested in [26] was used. The material model 0 (*MAT_ADD_EROSION) was used for failure mode parameter and added to crushable foam model.

The crushable foam model is obtained from the equation 4. In this equation, the elastic modulus is considered constant and the stress is updated assuming elastic behavior:

$$\sigma_{ij}^{n+1} = \sigma_{ij}^{n} + E\dot{\varepsilon}_{ij}^{n+\frac{1}{2}}\Delta t^{n+\frac{1}{2}} \tag{4}$$

Where $\dot{\varepsilon}_{ij}$ is the strain rate, E is the elastic modulus, and t is time.

The material model 0 (*MAT_ADD_EROSION) was attached to crushable foam model for the purpose of removing failed foam elements after creating K File in LS-DYNA software and editing this file. The erosion criteria were maximum principal and shear strain. The principal stress and shear strain were exploited from stress-strain curves related to each foam shown in Figure 2 [21]. In this paper, the mechanical behavior and structure of the foam were considered as isotropic material and casting-able (its shape becomes the same as its container and here it means that the shape of foam is the same as the shape of hexagonal structure), respectively. In addition, the mechanical behavior of foam in a three-dimensional field of stress was considered based on the generalization of the one-dimensional destruction model.

Various contact algorithms were used to model the perforation process precisely. Contact automatic single surface algorithm was employed between each part of the sandwich panel. Contact eroding surface to surface was used between the projectile and each target sections. Contact tied shell edge to surface was employed between two shells of honeycomb cell walls. Contact tied surface to surface was applied between honeycomb structure and polyurethane foam. Contact tiebreak surface to surface was used between aluminum skins and honeycomb structure. Contact tiebreak equation is based on the failure stresses (normal and shear) as indicated in equation 5. The damage was initiated by the criterion with the out-of-plane shear stress (τ) and normal stress (σ) components:

$$\left[\frac{\sigma}{\sigma_{fail}}\right]^2 + \left[\frac{\tau}{\tau_{fail}}\right]^2 \geq 1 \tag{5}$$

Where σ_{fail} and τ_{fail} are tensile and shear strength of the adhesive material, respectively [27]. The value of 24.38 MPa and 12.34 MPa was considered for normal stress and shear stress, respectively [28].

Following the testing conditions described in [21], the boundary conditions of experiments were simulated in numerical modeling and the boundary conditions of all structures were clamped as shown in Figures 3 and 5. All of the nodes in the face sheets and core edges in the models were clamped.

4. Numerical results and validation

4. 1. Process of damage

The results of experimental investigations reported in [21] and the numerical analyses were studied separately for each specimen. In experimental study, the number of test repetition and the range of initial velocity for each specimen are presented in Table 4.

According to Figure 6, the numerical analyses of perforation in the bare honeycomb core are similar to those observed in experiments. After the projectile, colliding with front side of the honeycomb structure, stress waves are created and began to damage the structure. At velocities higher than the ballistic limit velocity, the projectile passed through the target, compressed the honeycomb core and finally caused to cut and crumple the cells surrounding projectile.

In the numerical analyses of the unfilled honeycomb sandwich panel, initially, the projectile perforated aluminum skin and formed a plug on it. Then, a local debonding happened between the aluminum skin and core due to the projectile high velocity. Subsequently, the projectile along with the plug and the damaged parts of core exited from the rear aluminum skin and formed petals (Figure 7).

The projectile could have two forms of deviation; either deviates from its path before entering to target in an oblique direction or deviates from its path due to the existence of honeycomb structure after entering the core. The cause of this deviation is the position of the projectile when it penetrates the core. If the projectile collided at the connection point of the cells, it deviates from its path in the core. If the projectile exits from the target without any deviation, the symmetrical petal would be created in the unfilled honeycomb core sandwich structure. But if the deviation of the projectile at the time of exit from the unfilled sandwich structure happened, an asymmetric petal would be created. Figure 8 shows that the asymmetric petal shape of the unfilled sandwich panel in both experimental and numerical analyses were similar to each other.

Figure 9, shows the cut out view of the sandwich panel filled with foam3 (experimentally and numerically). The destruction steps of the foam filled sandwich structure resembled unfilled ones with the difference that the foam increased the strength of the core. The destruction of the core led to a large local debonding between the core and the back skin, which was completely visible in both experimental and numerical methods.

Also, in Table 5, a quantitative comparison of the damage process between experimental and numerical states for all structures were shown.

4. 2. Analyses of ballistic limit velocity

Honeycomb structure was modeled and the simulation result of velocity-time curve of the projectile was obtained for an input velocity of 50 m/s (Figure 10). The input velocities of projectile were 65, 70, 80, 90 m/s for unfilled sandwich panel and foam1 to foam3 filled sandwich panels, respectively. The simulations results of their velocity-time curves of the projectile are shown in Figure 10. Finally, the numerical ballistic limit velocities were calculated using the equation 6 [29,30] for all specimens in which v_{bn}, v_{out}, v_{in}, α, M, and m are the numerical ballistic limit velocity, output velocity, input velocity, the coefficient (it was considered 1), mass of projectile, and mass of material expelled from the target, respectively, and the results for each specimen are given in Table 6.

$$v_{bn} = \alpha\sqrt{v_{in}^2 - v_{out}^2}\ , \quad \alpha = \frac{M}{M+m} \tag{6}$$

The experimental ballistic limit velocity was obtained from equation 7 [31], in which v_{be}, v_{max} and v_{min} were the experimental ballistic limit velocity, maximum velocity in which full penetration does not occur, and minimum velocity in which full penetration occurs, respectively, and the results for each specimen are given in Table 6 [21].

$$v_{be} = \frac{v_{max}+v_{min}}{2} \tag{7}$$

According to Table 6, the ballistic limit velocities of the numerical findings were in good agreement with experimental data. The numerical ballistic limit velocity of the unfilled sandwich panel is 63.11 m/s whereas it is 45.38 m/s for the bare honeycomb core. These findings showed a significant enhancement in the ballistic limit velocity of sandwich panels versus the bare honeycomb core. Actually, the ballistic limit velocity was remarkably enhanced due to the interaction between the honeycomb core and aluminum skins. The interaction mechanism could be explained that the upper skin spreads the stress wave over the structure and decreases the projectile initial velocity slightly; after entering the projectile in to the core, the bottom skin has reinforced the honeycomb core and made it more resistant to projectile perforation. So, the sandwich structure has benefits in proportion to the other usual structures in high velocity impact loading conditions.

In addition, it was found that at lower velocities than the ballistic limit velocity, the projectile trapped into the target and entirely perforated at higher velocities; then it exited from the structure.

Figure 11, shows the results of experimental and numerical analyses related to the front and back faces of unfilled honeycomb sandwich panel when the projectile penetration was not complete. In fact, the projectile penetrated into the front skin and formed the plug on it; then, it continued through the core, and finally trapped in the target because of its low kinetic energy. As shown in Figure 11, the trapped projectile in the target resulted in debonding the core from the rear skin, creating tension on the rear skin and making the first crack on the rear skin in rolling direction. The destruction mode, the deviation of the trapped projectile, the tension of the rear skin, and the shape of the first crack of the rear skin, are the same in both experimental and numerical studies.

The numerical ballistic limit velocities of sandwich panels filled with three types of foam1, foam2, and foam3 are 66, 70.93 and 82 m/s, respectively. Apparently, the ballistic limit velocities of foam filled sandwich panels are more than unfilled ones. Comparing the three types of foam filled sandwich panels with unfilled one indicates that the numerical ballistic limit velocity of the first, second, and third type of foam filled sandwich panels are 4.6%, 12.4%, and 30% more than unfilled one sandwich panel, respectively. Furthermore, the difference between experimental and numerical ballistic limit velocity results of the unfilled sandwich structure, the first, second and third type of foam filled sandwich panels are 13%, 14%, 15%, and 16%, respectively, which these results are absolutely remarkable in terms of impact loading range. These differences could be attributed to the complex interaction of the damage development and the failure modes (such as the core crush, the polyurethane foam density, the failure of the skins, the local and global debonding between the aluminum skin and core, etc.) in overall mechanical behavior of the physical specimens. According to some research, these results are significant due to multi-material structures and have also been reported in previous studies for example; Barvik et al. [32] were investigated the penetration of steel plates by three types of projectile noses including flat, conical and hemispherical shapes under experimental and numerical study. They obtained 6% difference between the experimental and the numerical ballistic limit velocity results related to steel plate by flat-ended projectile. Also, this difference was 10% when they have used compatible meshing. Deka et al. [33] carried out some studies about composite laminates such as investigating their damage evolution and energy absorption under the ballistic impact. The difference between the experimental and the numerical ballistic limit velocity results related to 12 layer and 8 layer of composite laminate were 16% and 8%, respectively. So, the results of FEA modeling of the ballistic impact using LS-DYNA software are in good agreement with the experimental results.

4. 3. Analyses of the absorbed energy

Using the numerical ballistic limit velocity and the projectile mass, the numerical absorbed ballistic energy was calculated from the kinetic energy of projectile [34] ($E = \frac{Mv_{bn}^2}{2}$).

In Figure 12, the numerical absorbed energy at ballistic limit velocity of the bare honeycomb core, unfilled and foam filled sandwich panels and their experimental values [21] are shown. The amount of numerical absorbed energy of unfilled sandwich panel and the bare honeycomb core are 16.92 and 8.75 J, respectively. The findings demonstrate that the numerical absorbed energy at the ballistic limit velocity of the unfilled sandwich panel is higher than absorbed energy of the bare honeycomb core. Finally, the aluminum skins enhance the strength of the honeycomb structure, and the interaction between them increases the amount of numerical absorbed energy.

The numerical energy absorption of sandwich panels filled with three type of foams is foam1 18.51, foam2 21.38, and foam3 28.58 J which are 9.4%, 26.36 %, and 69% higher than the unfilled sandwich panel, respectively. These results show that the foam filled sandwich structures have preferable ballistic performance than the unfilled specimen. Foam with higher density has a higher numerical energy absorption. The enhancement in the numerical energy absorption results from the interaction between the honeycomb core and polyurethane foam as well as interaction between the skins and foam. As a result; a considerable improvement in stiffness and resistance to ballistic impact of foam filled sandwich panels is observed.

Also, the effect of foam on energy and displacement of projectile as well as total energy, the sandwich panel filled with foam3 (the best energy absorber in this study), the unfilled sandwich panel and honeycomb structure were investigated in the numerical analysis.

As shown in Figure 13, the projectile penetrates along Z axis and perpendicular to the target. Due to Figure 13, the energy-displacement curve of the sandwich panel filled with foam3 is the highest one, in fact, the projectile consumes more energy to penetrate the structure. The effect of foam using in the core results in the high resistance of foam3 filled sandwich versus the bare honeycomb structure and unfilled sandwich panel and this feature has caused the deviation of the projectile in the core of this structure.

The projectile perforation energy of the sandwich panel filled with foam3 is not only higher than the other specimens, but also the total energy of foam3 filled sandwich panel is the highest one. The total energy of system equals to sum of the projectile and target energies that each of them consists of the internal and kinetic energies. As, the projectile is a rigid body, its internal energy is zero (Table 7). So, subtraction of the projectile kinetic energy from total energy of the system equals to the target energy and this difference are shown in Figures 14, 15 and 16.

According to the curves shown in Figures 14, 15 and 16, the projectile has an initial velocity, thus it has an initial energy to penetrate into the target. As the projectile enters into the target, its energy is used to penetrate or in the other words its energy will be wasted, and the target absorbs the projectile energy. There are various energy absorption mechanisms in the sandwich structures that devote the energy of projectile at each stage of the penetration into the target. All of these energy absorption mechanisms for honeycomb structure, unfilled panel and foam3 filled panel are shown in Figure 17.

4. 4. The influence of polymer foam on specific absorbed energy

Owing to growing demand of aerospace, marine, automotive, and building industries for lightweight and high-strength structures; SEA analysis of structures has special significance. SEA is a better indicator of energy absorption capability of the structures, as it is independent of the weight of the structure. The interaction between foam and honeycomb core as well as the interaction between foam and skins caused the remarkable enhancement in energy absorption and strength of the sandwich panel.

By measuring the weight and the numerical ballistic absorbed energy of the specimens, the numerical specific energy absorption of each sandwich structure was calculated. The experimental [21] and numerical specific energy related to each structure are presented in Figure 18. SEA is 572 and 569 J/kg for the first and the second types of foam filled panels, respectively. These results are not favorable, as the numerical specific absorbed energy of unfilled structure (664 J/kg) is more than them, because of that the weight of sandwich structures are involved in the specific ballistic energy absorption. Focusing on SEA values show that although foam filling of panels improves their energy absorption capacity, it has no effect on the amount of SAE in sandwich panels filled with the foam1 and foam2, because heavier foam increases both the mass of the panel and the absorbed energy. So, SEA which is the ratio of these two parameters remains roughly unchanged and this result has also been reported in previous studies [35].

However, the foam3 filled the sandwich panel with 815 J/kg numerical specific energy absorption had the greatest ballistic impact efficiency. Though, the third foam filled sandwich panel had weight gain because of its filler material. So, its weight is reciprocally related to the specific absorbed energy, but the effect of weight gain was compensated by higher energy absorption, resulting in the highest SEA.

4.5. Investigation of existing errors between experimental and numerical modeling

Due to the results of experimental and numerical modeling, the results of the modeling section had an error in predicting the corresponding cases of their experimental ones. Although the error was in the acceptable range for modeling, but it is necessary to mention a few points about the causes of these errors for employing in future studies. The causes of the errors are as follows:

• The essence of modeling: It should be noted that the use of modeling software is inherently causing an error in the investigation. This error occurs because the calculations in the software are eliminated many non-linear equations to simplify, reduce the amount of computation, reduce the problem-solving time, etc. from equations. This is one of the main reasons for the error in all cases of using modeling software.

• Modeling Quality: The modeling quality is one of the points that can make the modeling error possible. This means that the similarity of the model made by the software to experimental mode, in terms of geometry and analytical quantities, is important.

• Material quality (in the experimental analysis): In modeling by software, it is assumed that the component which is being modeled, has the same quality at all points with no defects. While this assumption is not even true for high-precision components and usually have defects in experimental mode. This is yielding different results than those obtained in the experiments.

5. Conclusions

In this paper, the performance of the sandwich panels is compared to the bare honeycomb core, as well as with sandwich panels injected by polyurethane foam in the honeycomb core. Ballistic limit, damaged modes, energy absorption and specific energy absorption (SEA) of the various sandwich structure were investigated by FEA. The results of FEA simulations were compared with ballistic impact test results reported earlier [21]. The destruction shape from FEA simulation is nearly identical to the experimental results. The numerical ballistic limit velocity of the unfilled sandwich panel is 63.1 m/s whereas it is 45.4 m/s for the bare honeycomb core. These results demonstrated the interaction effect between the honeycomb structure and aluminum skins increases the ballistic limit velocity.

The numerical ballistic limit velocities of sandwich panels filled with three types of foam1, foam2, and foam3 are 66, 70.9, and 82 m/s, respectively. The ballistic limit velocities of foam filled sandwich panels are more than unfilled ones. Comparing the three types of foam filled sandwich panels with unfilled one indicates that the ballistic limit velocity of the foam1, foam2, and foam3 filled sandwich panels are 4.6%, 12.4%, and 30% more than unfilled one sandwich panel, respectively. Furthermore, the difference between experimental and numerical ballistic limit velocity of the foam1, foam2 and foam3 filled sandwich panels are 14%, 15%, and 16%, respectively which sound acceptable in dynamic loading range. Increasing of the foam density enhanced the structure strength and the ballistic limit velocity in both experimental and numerical analyses. Also, the sandwich panel filled with the foam3 material had more resistance against the projectile impact.

The numerical energy absorption of sandwich panels filled with three type of foams are foam1 18.51 J, foam2 21.38 J, and foam3 28.58 J which are 9.4%, 26.36 %, and 69% higher than the unfilled sandwich panel, respectively. However, the specific energy absorption of the foam3 filled sandwich panel is significantly higher than the other types of foam filled sandwich panels and unfilled ones. The numerical specific energy of this structure is 23% more than unfilled.

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Figure 1. The stress-strain behaviors of aluminum plate from tensile test

Figure 2. The stress-strain behaviors of the three types of foam from static compressive test [22]

Figure 3. The models of (a) aluminum skin (b) honeycomb cell structure (c) honeycomb structure and (d) the boundary conditions (all dimensions are in mm)

Figure 4. Mesh convergence of the honeycomb structure

Figure 5. The exploded view of modeled (a) unfilled sandwich panel (b) foam filled sandwich panel and (c) the overall view of boundary conditions

Figure 6. (a) Numerical results of backside of honeycomb structure after the projectile exit from the specimen at an initial velocity of 50 m/s (b) backside of honeycomb structure after the projectile exit from the specimen at an initial velocity of 54 m/s in the experiment

Figure 7. (a-d) the penetration steps in unfilled sandwich structure at an initial velocity of 65 m/s

Figure 8. (a) The backside of unfilled sandwich panel after the projectile exit from the specimen at an initial velocity of 65 m/s in numerical analysis (b) the backside of unfilled sandwich panel after the projectile exit from the specimen at an initial velocity of 78.5 m/s in experiment

Figure 9. The cut in half foam3 filled sandwich panel (a) in numerical model at an initial velocity of 90 m/s (b) in tested specimen at an initial velocity of 99.5 m/s. The delamination of back skin is clearly visible in both experiment and numerical model

Figure 10. The velocity-time curves of the projectile during ballistic impact on various specimens

Figure 11. The projectile trapped in the unfilled sandwich structure (a, b) at an initial velocity of 60 m/s in the numerical analysis (c, d) at an initial velocity of 67 m/s in the experiment

Figure 12. Comparison of the energy absorption of different specimen in numerical and experimental methods

Figure 13. The energy-displacement curves of the projectile during ballistic impact on various specimens

Figure 14. The energy-time curves of the honeycomb structure

Figure 15. The energy-time curves of the unfilled sandwich panel

Figure 16. The energy-time curves of the sandwich panel filled with foam3

Figure 17. The mechanisms of the energy absorption in various specimens

Figure 18. Comparison of specific energy absorption (SEA) of different sandwich panel structure from numerical analyses and experiments

Nomenclature

C	Cowper-Symonds strain rate parameter
E	Elastic modulus
M	Mass of projectile
m	Mass of material expelled from the target
n	Number of observations
P	Cowper-Symonds strain rate parameter
s	Estimated standard deviation
t	Time
v_{bn}	Numerical ballistic limit velocity
v_{out}	Output velocity
v_{in}	Input velocity
v_{be}	Experimental ballistic limit velocity
v_{max}	Maximum velocity
v_{min}	Minimum velocity
X	Value of a single observation
E_c	Compressive modulus of the foam
E_p	plastic hardening modulus
E_{tan}	Tangent modulus
SEA	Specific energy absorption
$\dot{\varepsilon}$	Strain rate
ε_d	Densification strain
ε_u	Failure strain
ε_p^{eff}	Effective plastic strain
ν	Poisson ratio
ρ	Density
α	Coefficient
β	Hardening parameter
σ_0	Initial yield stress
σ_c	Compressive stress of the foam
σ_d	Densification stress
σ_u	Ultimate tensile strength
σ_y	Yield strength
τ_u	Ultimate shear strength

Table 1. The properties of material

Material	Properties					
Honeycomb (5052-H38)	ρ (kg/m^3)	E (GPa)	σ_y (MPa)	σ_u (MPa)	τ_u (MPa)	ν
	2680	70	255	290	165	0.3
Al plate	ρ (kg/m^3)	E (GPa)	σ_y (MPa)	σ_u (MPa)	ε_u	*
	2637	67	117	133	0.08	*
	ρ (kg/m^3)	E_c (kPa)	σ_c (kPa)	$\frac{\%\text{Isocyanate}}{\%\text{Polyol}}$	Code	*
PU foam 1	56.94	4	352	50 / 50	SKC501	*
PU foam 2	108.65	8	864	75 / 25	SCC500	*
PU foam 3	137.13	20	1553	75 / 25	SKC501	*

Table 2. Density of each type of polyurethane foam

Foam	Number of samples	Average density (kg/m^3)	Standard deviation (s)
PU foam 1	3	56.94074	2.028657
PU foam 2	3	108.6543	2.094506
PU foam 3	3	137.1333	1.867425

Table 3. Parameters for rigid material, plastic kinematic model, and crushable foam model

Projectile	DENS (ρ) E NUXY (ν)	7800 kg/m^3 $210e9$ Pa 0.3	
Aluminum plate	DENS (ρ) E NUXY (ν) YIELD STRESS TANGENT MODULUS STRAIN RATE(C) STRAIN RATE(P) FAILURE STRAIN	2637 kg/m^3 $67e9$ Pa 0.3 $117e6$ Pa $1.449e9$ Pa 6500 s^{-1} [23] 4 [23] 0.08	
Honeycomb structure	DENS (ρ) E NUXY (ν) YIELD STRESS TANGENT MODULUS STRAIN RATE (C) STRAIN RATE (P) FAILURE STRAIN	2680 kg/m^3 $70e9$ Pa 0.3 $255e6$ Pa $1.33e9$ Pa [24] 6500 s^{-1} [23] 4 [23] 0.18	
Polyurethane foam	**Foam1**	RO (ρ) E PR (ν) LCID	56.94 kg/m^3 $0.42e6$ Pa 0.3 Strain-stress curve
	Foam2	RO (ρ) E PR (ν) LCID	108.65 kg/m^3 $7.8e6$ Pa 0.3 Strain-stress curve
	Foam3	RO (ρ) E PR (ν) LCID	137.13 kg/m^3 $19.8e6$ Pa 0.3 Strain-stress curve

Table 4. The number of test repetition and the range of initial velocity for each specimen

Specimens	The number of test repetition	the range of initial velocity (m/s)
Honeycomb structure	4	47-72
Sandwich panel with unfilled honeycomb core	3	67-85
Sandwich panel filled with foam1	5	70-93
Sandwich panel filled with foam2	5	81-110
Sandwich panel filled with foam3	5	99.5-105

Table 5. Results of damage process for different specimens

Specimens	Numerical data (mm)				Experimental data (mm)				difference between experimental and numerical (mm)			
	*A	**B	***C	****D	*A	**B	***C	****D	*A	**B	***C	****D
Honeycomb structure	7	d_1=10.99 d_2=16.10	-	-	7	d_1=11.90 d_2=14.96	-	-	0	d_1=0.91 d_2=1.14	-	-
Sandwich panel with unfilled honeycomb core	9	d_1=11.31 d_2=16.07	13.64	6.26	10	d_1=10.60 d_2=15.76	13.06	4.30	1	d_1=0.71 d_2=0.31	0.58	1.96
Sandwich panel filled with foam1	7	d_1=11.37 d_2=11.89	8.23	6.45	7	d_1=11.12 d_2=11.50	5.40	4.10	0	d_1=0.25 d_2=0.39	2.83	2.35
Sandwich panel filled with foam2	7	d_1=11.37 d_2=11.89	8.52	6.13	7	d_1=11.15 d_2=11.67	6.80	7.20	0	d_1=0.22 d_2=0.22	1.72	1.07
Sandwich panel filled with foam3	7	d_1=11.37 d_2=11.89	8.93	6.66	7	d_1=11.04 d_2=11.70	6.88	7.80	0	d_1=0.33 d_2=0.19	2.05	1.14

* A parameter = Number of destroyed cells

**B parameter = Diameter of damage area

***C parameter = Length of petal

**** D parameter = Length of debonding

Table 6. Results of ballistic limit velocity for different specimens

Specimens	Numerical data				Experimental data				% difference between experimental and numerical
	v_{in} (m/s)	v_{out} (m/s)	v_{bn} (m/s)	% change with respect to unfilled panel	v_{max} (m/s)	v_{min} (m/s)	v_{be} (m/s)	% change with respect to unfilled panel	
Honeycomb structure	50	21	45.38	-	54	47	50.5	-	10
Sandwich panel with unfilled honeycomb core	65	15.56	63.11	-	78.5	67	72.75	-	13
Sandwich panel filled with foam1	70	24.11	66	4.6	83	70	76.5	5.2	14
Sandwich panel filled with foam2	80	37	70.93	12.4	87	81	84	15.5	15
Sandwich panel filled with foam3	90	37.88	82	30	99.5	97	98.25	35.05	16

Table 7. The rules of energy calculation

Total energy of the system=Projectile energy+Target energy
Projectile energy=Internal energy+Kinetic energy , Internal energy=0
Target energy= Internal energy+Kinetic energy
Total energy of the system before impact= Total energy of the system after impact+Waste energy
Total energy of the system after impact=Energy of the projectile after impact+ Energy of the target after impact