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Design and Analysis of Graded Open-cell Aluminum Foam Shock Absorber for Helicopter Seats During Emergency Landing Conditions

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Abstract

Ensuring the safety of passengers as much as possible is essential in automobile and airplane accidents. In this study, an open-cell aluminum foam was introduced as an energy absorber. Analytical equations of absorbed energy were extracted. The analytical results had acceptable agreement with numerical and empirical ones. Based on the graded nature of natural impact absorbers, graded designed was used for the helicopter seat impact absorber. Optimization methods including genetic algorithm and sequential quadratic programming algorithm were used to create an optimum graded impact absorber. Satisfying standard requirements of the JAR-27 air standard was used as a design goal for impact absorber. The designed impact absorber was then modeled in ABAQUS software to calculate the absorbed energy, acceleration, and the force applied to the passenger and HIC for the protected passenger. According to the results, the graded foam satisfies all requirements for helicopters during emergency landing. The derived analytical equations can be used to study the energy absorption of other foams.

Nomenclature

E^*	Young modulus of the foam	G^*	Shear modulus of the foam
I	2nd moment of area	E_s	Young modulus of base metal
ρ^*	foam density	ρ_s	Base metal density
ρ_{rel}	Relative density of the foam	v^*	Poisson ratio of the foam
σ^*_{pl}	Plateau stress resulting from plastic collapse mechanism	σ_{ys}	Yield strength of base metal
		ε_D	Densification strain

1. Introduction

In the helicopter and airplane accidents, automobile collisions or elevator crashes, passengers must be protected as much as possible. Due to high energy absorption of foam structures, using these types of materials as impact absorbers has become widespread [1].

Pinnoji et al. [2] replaced the thermoplastic filling of motorcycle helmets with a metal foam and investigated the behaviors of these helmets in various experiments; the Head Injury Criteria (HIC) were also calculated for helmets using metal foam and ABS (Acrylonitrile Butadiene Styrene) and it was concluded that using metal foam leads to better results. Cacchione et al. [3] investigated the energy absorption capacity of carbon foam to use in helicopter seats under emer-

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gency landing conditions. Zheng et al. [4] designed an impact absorber structure for the lower part of airplane fuselage using polymer foams. According to their results, an impact absorber structure with symmetrical design leads to better results. Galehdari and Khodarahmi [5] designed a suitable impact absorber with a honeycomb structure to use in helicopter seats during emergency landings. It was shown that the graded structures have better performance compared to normal structures. Reyes and Børvik [6] utilized sandwich panels with polystyrene foam and polypropylene cores. In this research the energy absorption of these structures under low velocity impact was studied. Based on their results, lower density foam causes more energy absorption. Sawei et al. [7] studied Cubic Cell, Gibson-Ashby, Tetrakaidecahedron, Kelvin, Voronoi, Three-Dimensional Random Spheres simulation models for metal foams and stated all the merits and faults of each model. Lopatnikov et al. [8] investigated the energy absorption and deformation of an aluminum foam under impact loading with four different velocities. They presented a relationship between the absorbed energy and foam density. Fischer [9] studied the relation between the relative density of open-cell foams and their specific energy absorption efficiency.

Metal foam is a spongy structure created by porous metals which shows properties vastly different from bulk metals. Metal foams have different physical, mechanical, and electrical properties compared to bulk metals and can have several applications including absorbers for impact loading, vibration, and sound. One of the most common applications of metal foams is in automobile and aerospace industries which is due to the high energy absorption capacity of these foams under compressive stress. Based on the definition, a metal foam is a metal structure with uniformly distributed gas-filled pores. If the pores in a metal foam are not connected, the foam is known as a closed-cell structure and if these pores are connected, then the foam is an open-cell foam [1]. Cellular materials are widely used in nature like honeycomb cells in wood and cork structures. Plant stems are made from an outer skin core, with the core having a cellular structure. The inner core of the human skull also has a cellular structure [10]. In the current study, an arranged foam was used. As a result, the structure proposed by Gibson – Ashby model for open-cell foam was selected for a single cell. Then, the foam was created by repeating this cell in horizontal and vertical directions. This cell and its repetitions are shown in Fig. 1.

2. Banana Peel; A Natural Impact Absorber

Banana peel can be considered a natural impact absorber. Fig. 2 shows a cross-section of banana peel [5].

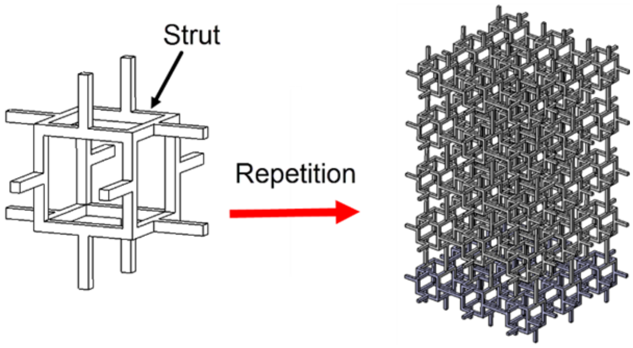

Fig. 1. Repetition of Gibson-Ashby cell and formation of open-cell foam.

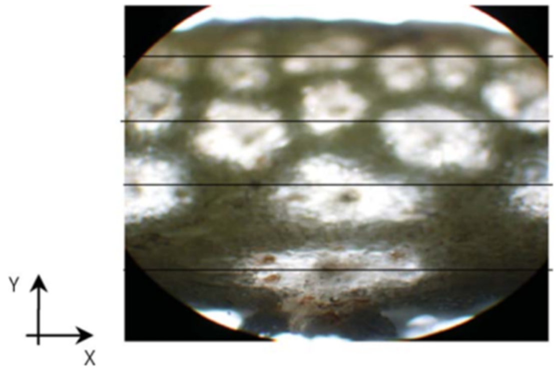

Fig. 2. Banana peel cross-section [5].

Based on Fig. 2, the porosity of the peel changes across its width. These types of materials are known as Functionally Graded Materials (FGM). The graded structure of an impact absorber decreases the force applied to the protected structures and increases the time for the energy absorption process [5].

Based on the idea of banana peel, the foam shown in figure 1 also has graded strut thicknesses where the strut thickness of foam changes from one row to the next and this thickness decreases from top to bottom. Fig. 3 shows the front view of a graded open-cell foam.

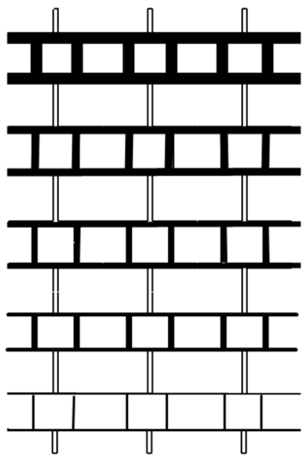

Fig. 3. Graded open-cell foam with changes in strut thickness.

To create an open-cell foam with this structure, two assumptions are required:

1. The material used in the foam is isotropic.
2. Despite changes in strut thickness from one row to the next, cell dimension in rows remains constant.

Although graded structure can have a positive effect on energy absorption capacity, creating graded structures have infinite possibilities and the main question will be selecting the structure with the best possible absorption capacity.

Therefore, the problem is an optimization problem where the objective function is the energy absorption of the structure with inputs including the geometrical dimensions of cells. To determine the objective function in this problem, it is necessary to use analytical equations for energy absorption based on cell dimensions.

3. Extracting Analytical Equations for Energy Absorption

Fig. 4 shows the stress-strain graph of a cellular structure. This graph includes a flat part and the stress related to this part is known as plateau stress. In this area, the severe strain is created in the structure of the absorber without a significant increase in the stress applied to the support. The strain at the end of the flat segment is known as densification strain. According to this figure, integrating the stress function based on strain in the plateau stress region can be used to calculate the amount of energy absorbed by the structure. In other words, the product of the densification strain in plateau stress is the area under the graph which is the amount of energy absorbed by the structure. Assuming that plateau stress is due to the plastic collapse in the cell, for a Gibson - Ashby cell, plateau stress and densification strain are calculated using Eqs. (1) and (2) [1].

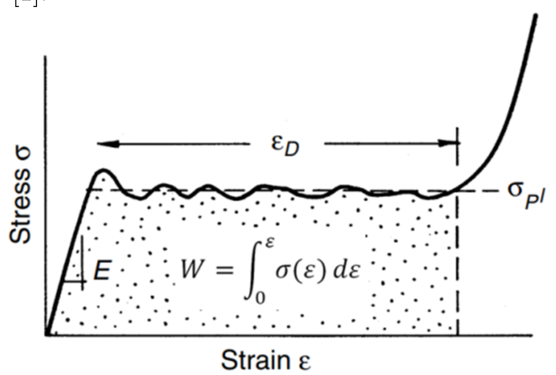

Fig. 4. Stress-strain graph of a cellular structure.

$$\sigma^*_{pl} = 0.3\sigma_{ys}\left(\frac{\rho^*}{\rho_s}\right)^{\frac{3}{2}} \tag{1}$$

$$\varepsilon_D = 1 - 1.4\left(\frac{\rho^*}{\rho_s}\right) \tag{2}$$

where σ^*_{pl} is the plateau stress caused by plastic collapse, ε_D is the densification strain, σ_{ys} is the yield stress of base metal, ρ^* is the foam density and ρ_s is the density of the base metal. By considering a constant value for plateau stress in stress-strain graph based on the assumption of elastic perfectly plastic material behavior required for Gibson-Ashby model, it is possible to consider the shaded area in figure 4 as a rectangular whose area is equal to the amount of the absorbed energy and is calculated using Eq. (3):

$$E = \left(0.3\sigma_{ys}\left(\frac{\rho^*}{\rho_s}\right)^{\frac{3}{2}}\right)\left(1 - 1.4\left(\frac{\rho^*}{\rho_s}\right)\right)V \tag{3}$$

In this equation, V is the volume of the foam.

However, if the aim is to write the equations for the absorbed energy based on cell dimensions of an open-cell foam, first, we have to calculate foam density based on its cell dimensions. To this end, cell dimensions were designated as shown in Fig. 5 while struts in the other two directions were designated as b and c.

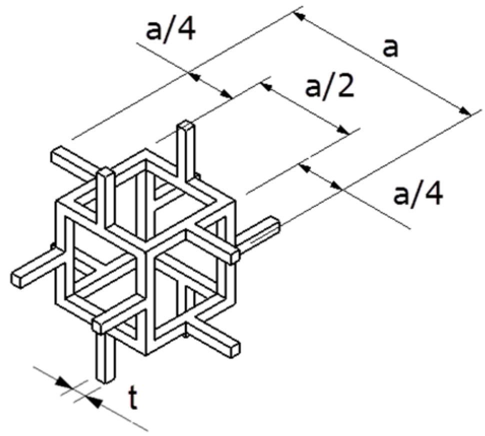

Fig. 5. Cell dimensions for a Gibson-Ashby cell.

In this case, the volume occupied with cells is:

$$V_{\text{members}} = t^2(3a + 3b + 3c - 16t) \tag{4}$$

And mass of supports is:

$$m = \rho_s \times V = \rho_s t^2(3a + 3b + 3c - 16t) \tag{5}$$

Therefore, foam density is calculated using:

$$\rho^* = \frac{\rho_s t^2(3a + 3b + 3c - 16t)}{abc} \tag{6}$$

Finally, the amount of the energy absorbed by a single cell is calculated using its dimensions as shown in Eq. (7):

$$E = \left(\left(\frac{0.3\sigma_{ys}}{\rho_s^{1.5}}\right)\left(\frac{\rho_s t^2(3a + 3b + 3c - 16t)}{abc}\right)^{1.5} - \left(\frac{0.3\sigma_{ys}}{\rho_s^{1.5}}\right)\left(\frac{1.4}{\rho_s}\right)\left(\frac{\rho_s t^2(3a + 3b + 3c - 16t)}{abc}\right)\right. \tag{7}$$

Eq. (3) is rewritten in terms of the absorbed energy of the unit mass of the structure below:

$$E = \frac{\left(0.3\sigma_{ys}\left(\frac{\rho^*}{\rho_s}\right)^{\frac{3}{2}}\right)\left(1-1.4\left(\frac{\rho^*}{\rho_s}\right)\right)V}{m} \tag{8}$$

In the Eqs. (4) and (5) the mass and the volume of the cell were mentioned. Thus the absorbed energy for the unit mass of the structure is:

$$E = \frac{\left(0.3\sigma_{ys}\left(\frac{\rho^*}{\rho_s}\right)^{\frac{\varepsilon}{2}}\right)\left(1-1.4\left(\frac{\rho^*}{\rho_s}\right)\right)(t^2(3a+3b+3c-16t))}{\rho_s t^2(3a+3b+3c-16t)}$$

$$= \frac{\left(0.3\sigma_{ys}\left(\frac{\rho^*}{\rho_s}\right)^{\frac{\varepsilon}{2}}\right)\left(1-1.4\left(\frac{\rho^*}{\rho_s}\right)\right)}{\rho_s} \tag{9}$$

For example, using Eq. (7), a cell with dimensions of $a = b = c = 0.04$m, $t = 0.002$m and yield stress of 1.53×10^8Pa is capable of absorbing 8.35J of energy.

To compare the results obtained from analytical equations with numerical simulation results, ABAQUS software was used. To this end, a single Gibson-Ashby cell with dimensions of $a = b = c = 0.04$m and $t = 0.002$m was modeled in the software and was investigated under impact loading. Geometrical modeling of Gibson − Ashby cell along with a striker plate at the top and a support plate at the bottom was carried out in CATIA environment and the model was then imported into ABAQUS software. Model dimensions were in metric units. After introducing material properties in the software, a concentration mass of 5Kg was modeled on the striker plate. After assembly of the parts, the Step environment was used to define an explicit dynamic step with period of 0.02s. In the loading and support condition section, an initial velocity of 1.82m/s was applied to the mass on the striker plate. The type of the cell elements were solid while the type of the rigid plate elements were shell. The lower rigid plate was fully fixed and the end of cell struts had a symmetrical support conditions, because this cell was part of a continuous foam structure. The finite element model of cell and rigid plates is shown in Fig. 6.

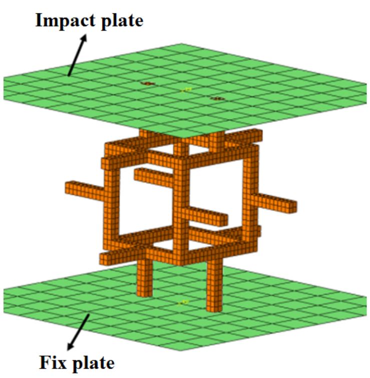

Fig. 6. Finite element model of a single cell along with striker and support rigid plates.

To introduce material properties into the software, the results of a simple tensile strength test for an Al6061 sample were extracted from reference [11]. Since in Gibson − Ashby model, material stress behavior is considered to be elastic perfectly plastic, the strain related to yield stress is zero. Table 1 shows the plastic properties defined in the ABAQUS environment.

Table 1
Plastic properties of Al6061.

Stress (MPa)	Strain
153	0
153	0.181478

The simulation results can be compared to the results of the analytical equations only when the kinetic energy of the striker defined in the software is fully absorbed by the structure. In other words, at the end of the simulation, the kinetic energy of the striker should be equal to zero. According to Eq. (10), the kinetic energy of modeled striker in the software is equal to:

$$K = \frac{1}{2}mv^2 = \frac{1}{2} \times 5 \times 1.82^2 = 8.35(\text{J}) \tag{10}$$

After the end of the analysis, the KE time history is presented in Fig. 7. It can be seen that kinetic energy was entirely absorbed by the structure.

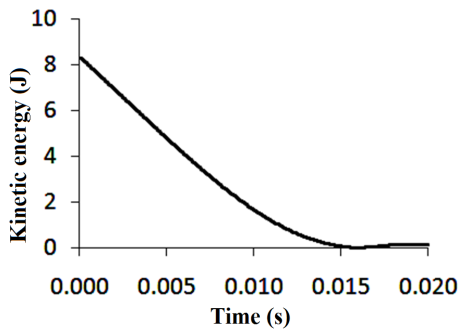

Fig. 7. Absorbed kinetic energy time history by a single cell.

Furthermore, Fig. 8 shows the position of the striker at the last time step which indicates full absorption of kinetic energy. On the other hand, based on the deformation of the cell, it can be seen that the maximum bending capacity of the cell was also reached. Furthermore, Time history output of the software shows that this kinetic energy was absorbed only due to the bending capacity of the cell without compression of two vertical struts after bending of the horizontal struts which had touched each other. In other words, observing the changes in the shape of the cell under impact loading showed that impact had stopped before bent struts can touch each other.

Therefore, the results of numerical simulation are fully in agreement with the results obtained from the analytical equations.

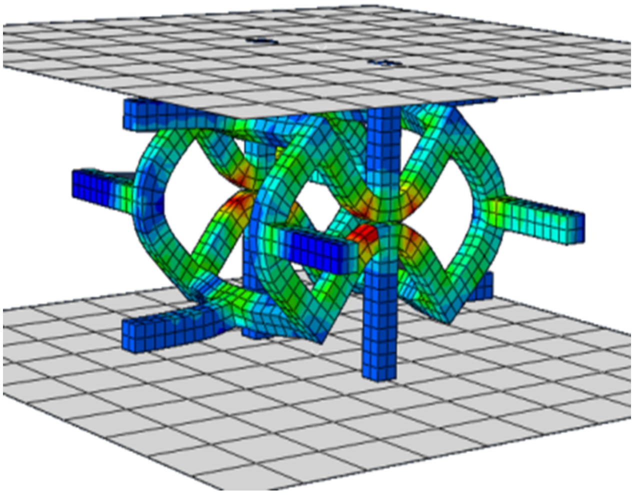

Fig. 8. Deformation of a single cell under impact loading.

In order to ensure that the results are independent of finite element meshing, the analysis was repeated in three steps and with three different meshing sizes. The reaction force applied to the support plate was extracted in each simulation. The reaction force in the second simulation was different from the force in the first simulation but there was no significant difference between reaction forces of the second and third simulations. This indicates a convergence in results after the second simulation as shown in Fig. 9.

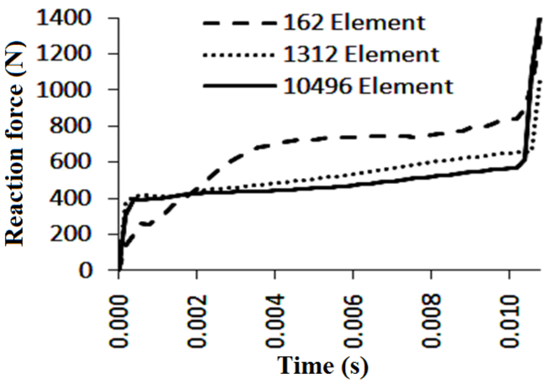

Fig. 9. Comparison between reaction forces transferred to the support plate in three different simulations.

In order to ensure the accuracy of the results, numerical simulation results presented in reference [12] were used. The important fact in this reference is that a five-cell Gibson -Ashby cell was modeled under different loadings and results were recorded. The geometrical properties of this foam and loadings are also exactly known. Fig. 10 shows the open-cell foam investigated in this reference.

The dimensions of the cube modeled in this reference are $1 \times 1 \times 1$in. Furthermore, strut thickness is equal to 0.015in, elasticity modules are 16500000 (psi) and Poisson's ratio is 0.29. The loading of the model is also carried out by applying a total force of 2lb at four nodes of the upper cell. Based on the results, this foam is compressed by 0.00005565in [12]. This foam was modeled in the ABAQUS environment with metric dimensions. Fig. 11 shows the finite element model of this foam in ABAQUS software.

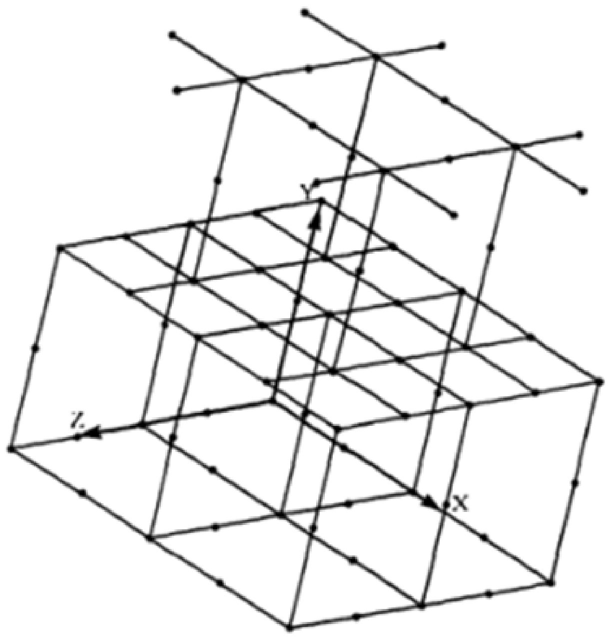

Fig. 10. A five-cell foam modeled with beam element [8].

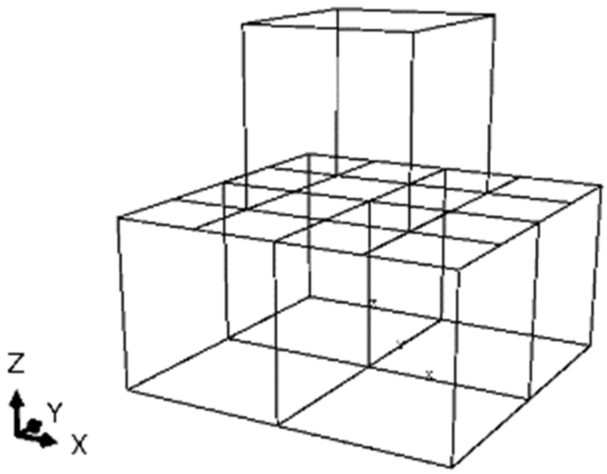

Fig. 11. Finite element model of five-cell foam in ABAQUS environment.

Boundary conditions and loading were also set as mentioned above. After the analysis, as shown in Fig. 12, the maximum displacement was 1.975×10^{-6}m. The model displacement in reference [12] is 1.4135×10^{-6}m. Therefore, displacement in the simulated model is 0.562×10^{-6}m higher than reference [8] which indicates a 39% difference. This difference can't be attributed to error for two reasons. First is that the solution method used in both cases is finite element method and second is that this difference can be due to the difference between some elements of two models or improvement in solution algorithms for finite element. Therefore, the results obtained from ABAQUS software have an acceptable agreement with the results presented in reference [12].

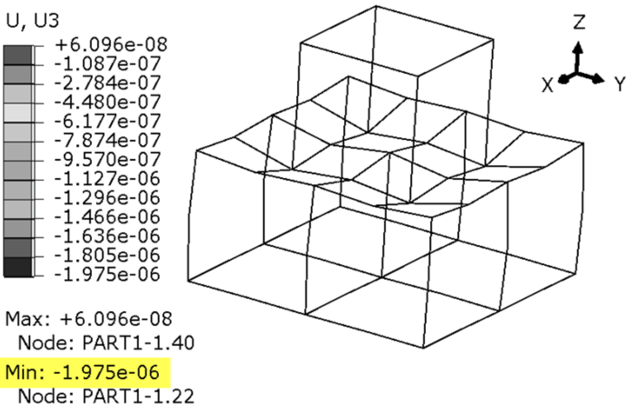

Fig. 12. Deformation of foam under loading.

4. Design of Helicopter Seat Impact Absorber (First model)

Based on the JAR27.562 standard, the following assumptions were made for the design of the impact absorber. First, based on the conservative design, it was assumed that the impact absorber absorbs all of the impact's kinetic energy (thus the energy absorbed by the seat's supports is ignored). The Second assumption was that the passenger weights 77Kg and the impact velocity was 9.1m/s. Based on the conditions of the crash test shown in Fig. 13, the helicopter hits the ground at the angle of 30 degrees from the vertical line. Therefore, kinetic energy is calculated using Eq. (11):

$$K = \frac{1}{2}m(v\cos\theta)^2 = \frac{1}{2} \times 77 \times (9.1\cos 30)^2 = 2391.15\text{J} \tag{11}$$

Based on standard requirements, the force applied to passenger's hips should be less than 6674(N). Furthermore, the structure should be capable of absorbing at least 2391.15J of energy [13].

To design the helicopter seat absorber, a foam with a length of 600mm and width of 240mm was considered. In this area, it is possible to place 10 cells with 60mm struts in the length and 4 similar cells in a width of the structure. Therefore, the surface area of the foam includes a total of 40 cells as shown in Fig. 14.

It was assumed that for every cell shown in Fig. 14, a column including 10 other graded cells exists through the height of the structure (Fig. 15) which creates a graded foam. To prevent the complexity of analytical calculations and numerical simulations, only one of the columns was considered and the energy absorbed by a single column of graded cells was optimized. One such column was presented in Fig. 15. The final optimized foam was built by repeating this column of the optimized cells.

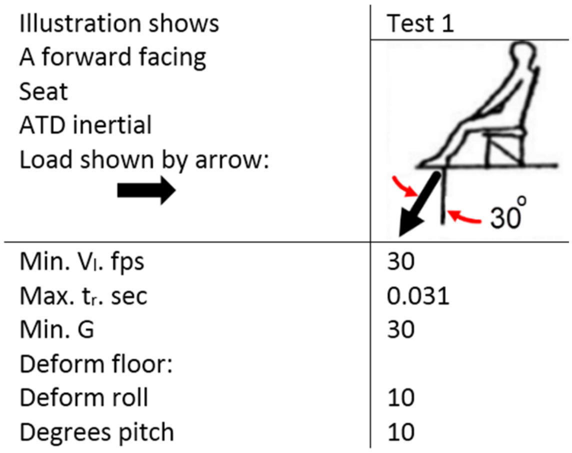

Fig. 13. The conditions of the emergency landing simulation [13].

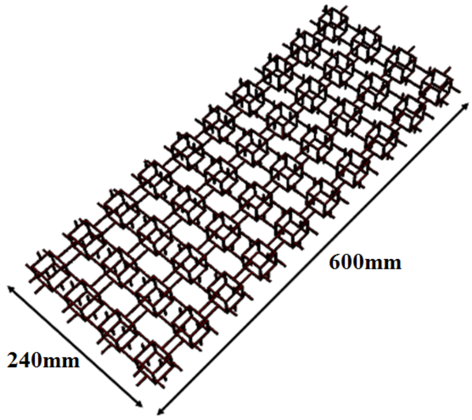

Fig. 14. Placement of 40 cells in the surface area.

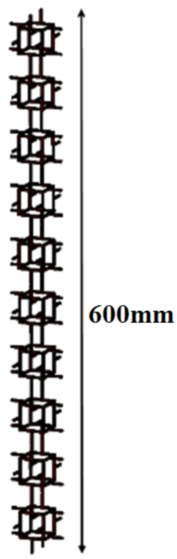

Fig. 15. A single foam column made from 10 graded cells.

The goal of optimization was to design to an ideal impact absorber. An ideal impact absorber is one in which the ratio of absorbed energy to mass is the maximum possible amount. Dividing the amount of the absorbed energy to the mass of the absorber leads to the Specific Energy Absorption (SEA) criterion calculated by Eq. (12) which indicates the energy absorption capacity of the absorber in a unit of mass [14].

$$\mathrm{SEA} = \frac{EA}{m} \tag{12}$$

Higher SEA numbers show a higher absorption capacity of the structure. During the optimization of behavior for a graded impact absorber, first, it is necessary to determine the specific Energy Absorption function of the structure.

If a column of cells shown in Fig. 15 is made from 10 cells, using Eq. (3), the energy absorption capacity of the first cell (U_1) is calculated using Eq. (13):

$$U_1 = E_1 V_{\mathrm{foam}} \tag{13}$$

Furthermore, the mass of the first cell is equal to:

$$m_1 = V_{\mathrm{members}} \times \rho_s \tag{14}$$

where ρ_s is the density of the base metal. Similar equations can be written for second to tenth cells, resulting in U_2 to U_{10} and m_2 to m_{10}. Therefore, the total energy absorbed by this foam is calculate using Eq. (15):

$$u = U_1 + U_2 + \cdots + U_{10} \tag{15}$$

And foam mass is calculated using the following equation:

$$m = m_1 + m_2 + \cdots + m_{10} \tag{16}$$

According to the definition of specific energy absorption, the value of $\frac{u}{m}$ should be maximized. In other words, if the cost function is the inverse of this value, then the value for $\frac{m}{u}$ should be minimized. This was done by MATLAB optimization toolbar. It was assumed that cell dimensions and strut thicknesses have a starting and ending band. Upper and lower limits in each of the cells in a column are written as follows:

lb = [0.06 0.06 0.06 0.001 0.06 0.06 0.06 0.001 0.06 0.06 0.06 0.001 0.06 0.06 0.06 0.001 0.06 0.06 0.06 0.001 0.06 0.06 0.06 0.0015 0.06 0.06 0.06 0.0015 0.06 0.06 0.06 0.0015 0.06 0.06 0.06 0.0015 0.06 0.06 0.06 0.0015]

ub = [0.06 0.06 0.06 0.0015 0.06 0.06 0.06 0.0015 0.06 0.06 0.06 0.0015 0.06 0.06 0.06 0.0015 0.06 0.06 0.06 0.0015 0.06 0.06 0.06 0.002 0.06 0.06 0.06 0.002 0.06 0.06 0.06 0.002 0.06 0.06 0.06 0.002 0.06 0.06 0.06 0.002]

In order to define the limits of the problem, it is necessary to investigate the densification mechanism of a Gibson – Ashby cell. The relative density of each foam is determined by dividing the foam density with the density of the based metal used in the foam as shown in Eq. (17) [1]:

$$\rho_{\mathrm{rel}} = \frac{\rho^*}{\rho_s} \tag{17}$$

In an open-cell foam, if the relative density of the foam is higher than 0.3, then the foam will have short and squat struts and material yields through tensile or compression yield instead of bending. In other words, under these conditions, it is better to consider the foam as a filled structure instead of a porous foam. Furthermore, if the relative density is too low, the elastic collapse will dominate over the plastic collapse which is also undesirable. Therefore, it is necessary to set two limits for the plastic behaviors of the foam.

1. Relative density of the foam should be lower than 0.3, as stated in inequality (18) [1]:

$$\left(\frac{\rho^*}{\rho_s}\right) < 0.3 \tag{18}$$

2. Relative density of the foam should not be lower than a certain limit, satisfying inequality (19) [1] so that the structure experiences the plastic behavior.

$$\sigma_{pl}^* < \sigma_{el}^* \tag{19}$$

This inequality can be rewritten based on Eq. (1) and using plateau stress equation (when elastic collapse mechanism is active) as shown in Eq. (20) [1]:

$$\left(\frac{\rho^*}{\rho_s}\right) > 36 \left(\frac{\sigma_{ys}}{E_s}\right)^2 \tag{20}$$

The boundaries shown in Eqs. (16) and (17) were used for the optimization problem of foam energy absorption. After optimization, based on the answer provided by MATLAB, the strut thickness of five cells closer to the support was set to 1.5mm while the strut thickness of the next five cells was set to 2mm. This setup was then investigated using MATLAB code which showed it can absorb 63.9J of energy.

Since 40 columns with these characteristics were repeated in the final foam, the final structure was capable of absorbing 40 times the amount of energy:

$$63.9 \times 40 = 2556\mathrm{J}$$

This amount of energy absorption is acceptable according to the standard because it is higher than the threshold value of 2391J. To control the acceleration applied to the passenger, this foam was modeled in the ABAQUS environment with 10 cells. Fig. 16 shows the finite element model of this foam.

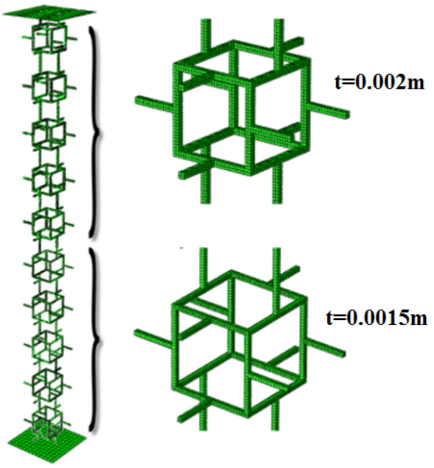

Fig. 16. Finite element model of a 10-cell column.

The deformed shape of the foam is shown in Fig. 17.

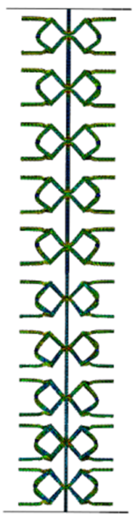

Fig. 17. Deformed shape of the foam after loading.

As expected, based on MATLAB program, the kinetic energy of the striker was fully absorbed. Fig. 18 shows the changes in kinetic energy of the striker.

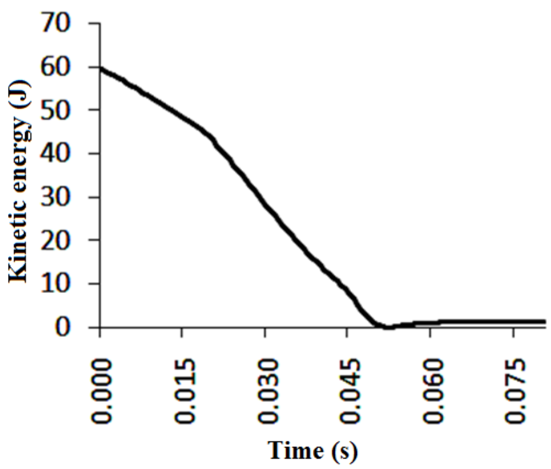

Fig. 18. Kinetic energy time history of the striker by the foam.

The deceleration of the striker is shown in Fig. 19.

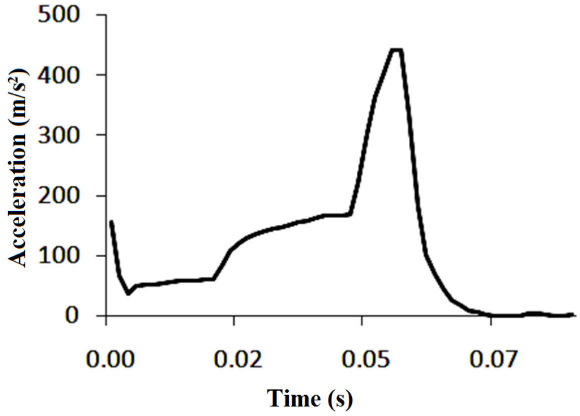

Fig. 19. Acceleration-time graph for the striker.

Maximum deceleration of the striker was 405m/s^2. This is equal to 41g which is higher than the standard value of 30g and therefore unacceptable.

$$\text{g factor} = \frac{405}{9.81} = 41g > 30g$$

5. Design of Helicopter Seat Impact Absorber (Second Model)

In the second model, the following upper and lower limits were assumed for the columns. Upper and lower limits of cell strut thickness were defined with the aim of increasing the graded nature of the foam.

lb = [0.16 0.16 0.16 0.001 0.16 0.16 0.16 0.001 0.16 0.16 0.16 0.0015 0.16 0.16 0.16 0.0015 0.05 0.05 0.05 0.002 0.05 0.05 0.05 0.002]

ub = [0.16 0.16 0.16 0.0015 0.16 0.16 0.16 0.0015 0.16 0.16 0.16 0.002 0.16 0.16 0.16 0.002 0.05 0.05 0.05 0.0025 0.05 0.05 0.05 0.0025]

The problem was then solved using two approaches of genetic algorithm and Sequential Quadratic Programming (SQP). The variables obtained from these two algorithms are presented in Table 2. If the design variables obtained from genetic algorithm are used as the final answer, each column of open-cell foam can absorb a total of 59.88J of energy but if the answer for the SQP algorithm is used, each foam column will have the energy absorption capacity of 43.21J. Therefore, the answer from genetic algorithm is used as the final answer to the problem.

After optimization, based on the answer provided by MATLAB, thickness of struts for first two cells closer to the support was 1.5mm, thickness for the two middle cells if 2mm and thickness for the two upper cells was 2.5mm. The design was simulated in ABAQUS environment and its finite element model is presented in Fig. 20.

Table 2
Design variables obtained using genetic algorithm and SQP.

Design variable	Initial value (m)	Optimised value by GA (m)	Optimised value by SQP (m)
a_1	0.16	0.16	0.16
b_1	0.16	0.16	0.16
c_1	0.16	0.16	0.16
t_1	0.001	0.0015	0.001
a_2	0.16	0.16	0.16
b_2	0.16	0.16	0.16
c_2	0.16	0.16	0.16
t_2	0.001	0.0015	0.001
a_3	0.16	0.16	0.16
b_3	0.16	0.16	0.16
c_3	0.16	0.16	0.16
t_3	0.0015	0.002	0.0015
a_4	0.16	0.16	0.16
b_4	0.16	0.16	0.16
c_4	0.16	0.16	0.16
t_4	0.0015	0.002	0.0015
a_5	0.16	0.16	0.16
b_5	0.16	0.16	0.16
c_5	0.16	0.16	0.16
t_5	0.002	0.0025	0.002
a_6	0.16	0.16	0.16
b_6	0.16	0.16	0.16
c_6	0.16	0.16	0.16
t_6	0.002	0.0025	0.002

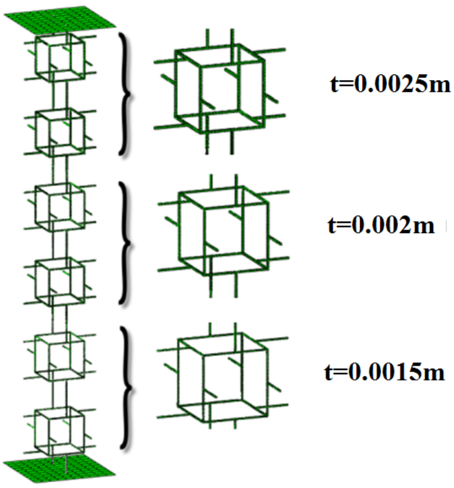

Fig. 20. Finite element model of open-cell foam with six cells.

Deformation shape of the foam at the final stage of densification is shown in Fig. 21.

Subsequently, compatibility with standard requirements was investigated. Fig. 22 shows the decrease in striker's kinetic energy until zero which indicates full absorption of kinetic energy by the foam.

As can be seen in Fig. 23, the reaction force of the support is not higher than 160N. Therefore, by multiplying this number by 40, the total force applied to the base is 6400N which is lower than the threshold of 6673N and therefore in compliance with the standard.

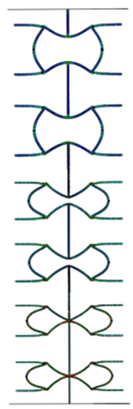

Fig. 21. Deformation of the foam under impact loading.

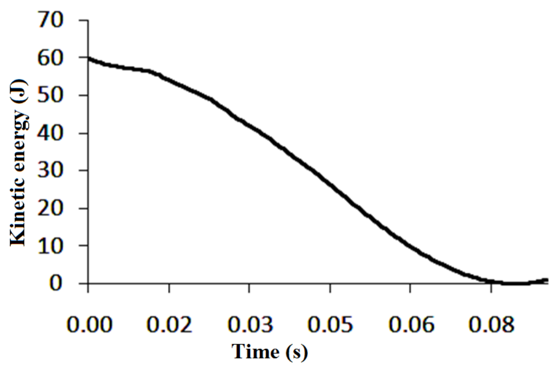

Fig. 22. Full absorption of kinetic energy of the striker by the foam.

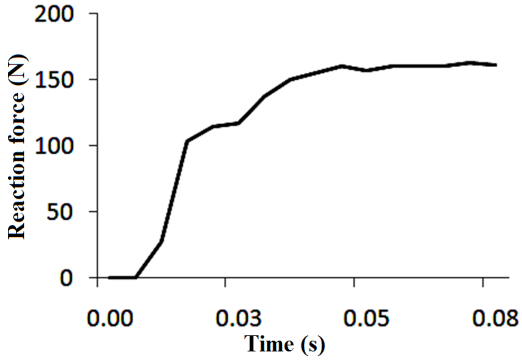

Fig. 23. Reaction force-time graph at the fix plate.

According to Fig. 24, maximum deceleration of the striker is $17\mathrm{m/s^2}$ which is equal to 18g. Since this value is lower than 30g, standard requirements have been met.

$$\text{g factor} = \frac{173}{9.81} = 18g < 30g$$

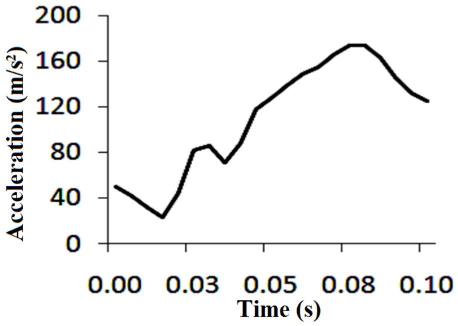

Fig. 24. Acceleration over time graph for the striker

To calculate the head injury criteria (HIC), MATLAB software was employed and the sum of the sine approach was used to obtain a curve with the known equation for the acceleration-time graph. The equation for this curve was determined by MATLAB as shown in Eq. (21). Since, according to the standard presented in reference [34], this value should be converted to g value, the entire function is divided into 9.81.

$$\begin{aligned} f(x) = \big[&a_1 \times \sin(b_1 \times x + c_1) + a_2 \times \sin(b_2 \times x + c_2) \\ &+ a_3 \times \sin(b_3 \times x + c_3) + a_4 \times \sin(b_4 \times x + c_4) \\ &+ a_5 \times \sin(b_5 \times x + c_5) + a_6 \times \sin(b_6 \times x + c_6) \\ &+ a_7 \times \sin(b_7 \times x + c_7)\big]/9.81 \end{aligned} \tag{21}$$

Constants a_1, b_1, c_1 to a_7, b_7, c_7 are shown in Table 3.

This function should be integrated over the time period of the impact based on Eq. (22) [12].

$$HIC = (t_2 - t_1)\left[\frac{1}{t_2 - t_1}\int_{t_1}^{t_2} a(t)dt\right]^{2.5} \tag{22}$$

where $a(t)$ is a function of $f(x)$ and x is a time variable. After calculating the integral value in MATLAB, the HIC value was calculated to be 43 which is acceptable according to the standard.

Table 3
Function constants obtained from MATLAB solution.

Constant	value	Constant	value
b_1	29.31	a_5	6.055
c_1	0.7578	b_5	219
a_2	342.5	c_5	0.8195
b_2	35.27	a_6	5.784
c_2	3.925	b_6	328.5
a_3	12.85	c_6	-0.498
b_3	251.1	a_7	3.333
c_3	1.538	b_7	429.4
a_4	3.728	c_7	2.095
b_4	97.78		
c_4	5.18		

This structure occupies a total volume of $0.36\mathrm{m^3}$. Another absorber with honeycomb structure made from aluminum was investigated under plane loadings reference [5]. This absorber is shown in Fig. 25.

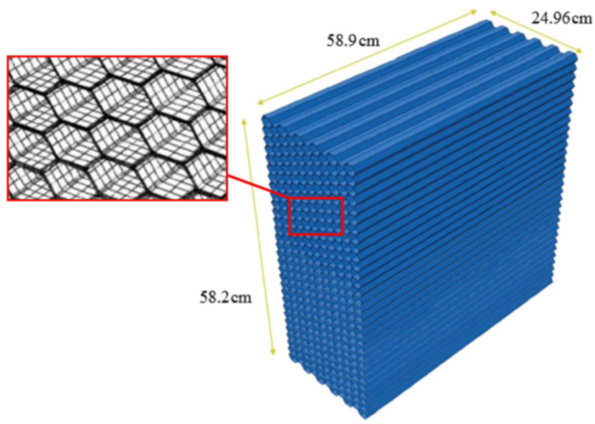

Fig. 25. Graded honeycomb Impact absorber model for helicopter seat [5].

This absorber has a total volume of $0.085\mathrm{m^3}$. Therefore, the volume of absorber with honeycomb structure is almost four times lower than the volume of absorber with Gibson-Ashby cell structure.

6. Conclusions

By considering the Gibson-Ashby model for an open-cell foam, the analytical equation of the absorbed energy based on cell dimensions was determined. In order to verify the analytical result, energy absorption of this foam was simulated in the ABAQUS software. The analytical results had an acceptable agreement with the numerical ones. On the other hand, the numerical simulation results for a five-cell open-cell foam under compression loading were verified by comparing with empirical results.

By repeating a single Gibson-Ashby cell and inspiring of a banana peel, an open-cell graded foam was designed. This structure had different thickness for each row. Using genetic and Sequential Quadratic Programming (SQP) algorithms, an optimum shock absorber for helicopter seats was designed in accordance by JAR 27 standard. The proposed absorber in this study was a graded foam structure made from 240 cells, each cell with dimensions of 160mm and strut thickness varied from 1.5 to 2.5mm. The energy absorption of the shock absorber was simulated in the ABAQUS environment. The results indicated that this structure can satisfy energy absorption standard requirements and can also decrease the force and deceleration of passengers based on standard requirements.

The proposed method in this study can also be used to measure the absorbed energy of closed-cell foams or open-cell foams with various fillings.

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