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Design and analysis of a graded honeycomb shock absorber for a helicopter seat during a crash condition

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ABSTRACT

An important issue that should be considered in accidents, including car, helicopter, and airplane crashes, is the safety of the human occupants. In this research, graded honeycomb structure is primarily introduced as a shock absorber. The amount of energy absorption, and the applied force and acceleration to the passengers have been calculated through numerical simulation in ABAQUS software. In order to validate numerical results, a low velocity experiment has been conducted on the test sample. Results represent an acceptable agreement with empirical ones. Given the occurrence of an emergency helicopter crash, it is of great importance to avoid any potential injurious loading to the passenger. Hence, according to Joint Aviation Regulations part 27, an optimised seat shock absorber has been designed and analysed for a helicopter in crash condition by the means of Genetic and Sequential Quadratic Programming algorithms. The designed shock absorber, which is of graded honeycomb structure, has satisfied complete standard specifications while being applied in the helicopter crash simulation. The applied design and simulation approach could be extended for any other types of shock absorbers.

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Graded honeycomb structure (GHS); shock absorber; banana peel; low velocity experimental test; helicopter crash; passenger; genetic algorithm; sequential quadratic programming algorithm

1. Introduction

Given the importance of passenger safety during transportation, energy absorbers have increasingly attracted the attention of researchers. Helicopter and airplane crashes; mine and under-armoured vehicle explosions; car accidents; and elevator impacts are examples of incidents during which passengers should be protected to the utmost feasible level. Based upon the small ratio of mass to the energy absorption for honeycomb structure, this structure is increasingly applied for shock absorbing purposes. Calladine and English [2] studied the effect of strain rate and inertia on the amount of energy absorption by the shock absorbers through analytical approach. Furthermore, Li et al. [13] studied the influence of geometry and the cell wall thickness of the honeycomb structure on its elastic properties. In this paper, the elastic properties of the structure have been scrutinised by using an analytical method. Jorg and Becker [11] introduced the effective stress–strain equation for a sandwich panel with two-dimensional (2D) cellular cores by the means of analytical equations. In this research, different materials and cell geometries were utilised. Liaghat and Sarailou [14] have performed the optimisation of honeycomb structure under compression loading. In this paper, the optimisation has been conducted by means of an optimisation algorithm in MATLAB® [15] and analytical solution. Optimisation for various geometries has been scrutinised as well.

Numerous research efforts have been conducted for the purpose of studying in-plane mechanical properties of honeycomb structure. Sayed et al. [21] have calculated the Young's modulus and Poisson's ratio of this structure under bending and 2D loading through deformation analysis. Hayes and Gibson [10] have performed the 2D and three-dimensional (3D) analysis of honeycomb structure under quasi-static loading, followed by the comparison of analytical and empirical results. Papka and Kyriakides [19,20] validated the behaviour of hexagonal honeycomb structure under axial and biaxial in-plane loadings through experiments and compared their results to those of numerical methods. Silva et al. [22] has analysed the influence of arranging honeycomb cells on the elastic behaviour and the strength of honeycomb structure with the purpose of analysing irregular honeycomb structure. The results by Hayes concerning the description of deformation modes and the failure mechanism of honeycomb structure are the most accurate available results. Wang [25] has studied the

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influence of geometry on the behaviour of paper honeycomb structure under impact loading. The length and cross section of the honeycomb are of great importance in terms of the amount of energy absorption. Increasing the relative density in this structure results in the increase of energy absorption. The thickness of cells has a fluctuating effect on its cushioning properties. For instance, these properties would decrease by increasing cells' thickness. Furthermore, creating stripes in the plane has an impressive influence on the compressive resistance. Using analytical simulations, the geometry of the structure has been optimised in terms of the energy absorption value. Wang and Yu [26] have introduced a mathematical model for describing the equation of energy absorption of the honeycomb structure in different weather conditions, and the derived equation has been represented. This model includes a step function representing the amount of the energy absorption for four different regions: (1) linear elastic, (2) yield, (c) flat and smooth, and (4) compression. Each of the above-mentioned regions can be described by a simple formula representing the equation of the energy absorption as a function of cells' thickness, material properties of the structure, and weather conditions. The comparison of analytical and empirical results indicates an appropriate concordance. The represented model can be further applied for optimising honeycomb structure with the utmost energy absorption. This analytical method has been used for studying the behaviour of banana peel structure.

A highly applicable structure in shock absorbers is honeycomb structure, with the major application as the cell core in two-layered structures. The behaviour of this structure depends upon the speed of the impact and the structure geometry. Deqiang et al. [4] have represented a finite element (FE) model for studying the behaviour of this structure under impact loading in LS-DYNA® software. One of the measured parameters is the out-of-plane plateau stress. The speed of the impact was changed within the range of 3−350 m/s. According to results of this simulation, when the geometrical characteristics of the structure remain constant, the changes of out-of-plane plateau stress would be conic in relation with the impact speed. For a specific speed, the relation between the out-of-plane planar stress and the ratio of cell thickness to its edge length would be a power function. Moreover, Karagiozova and Yu [12] studied the fracture behaviour of honeycomb structure under in-plane compressive loading. This study was conducted to analyse the effect of geometry topology of this structure on its strength. In this research, three different aspects have been scrutinised: a numerical simulation, the empirical results concerning the plastic deformation of the structure, and the corresponding fracture stress under equal biaxial loading. Under this loading, a bending deformation without tension has occurred. Chen and Pugno [3] have studied in-plane elastic characteristics of nano-honeycomb through an analytical approach. In this paper, the influence of two geometric parameters on the elastic characteristics of the structure has been studied. These two parameters are the ratio of stiffness to density, and the ratio of strength to the density of the structure. Given the represented theory, there is the feasibility of producing nano-honeycomb structure. Additionally, Nakamato et al. [18] have investigated the behaviour of honeycomb structure filled with a solid material under in-plane impact loading by using FE method. In this research, the effect of filler's characteristics on deformation, average stress, locking strain, and the energy absorption of the structure has been studied. By increasing the volume of the filler material, the average stress and the locking strain have increased. The amount of energy absorption as a function of the volume of the filler material has been represented within an analytical equation. In order to study the possibility of using soft aluminium honeycombs, Tanaka et al. [24] studied this structure under in-plane impact loading. In this paper, the functionality of this structure as a shock absorber in cars for rescuing humans involved in accidents was considered. Moreover, conducting an empirical test, the characteristics of this type of deformation was analysed in the structure by the means of high velocity cameras. Given the ratio of strength to mass and the high specific energy of these structures, they have been increasingly used in aerospace industries. Veltin [1] studied the behaviour of regular honeycomb structure under in-plane impact loading through a simulation in ABAQUS software. In this study, by keeping the angle constant, the effect of cells' geometry on energy absorption was investigated. This structure has also been applied as the shock absorber under helicopter seats. Sandwich structures which are considered as shock absorbers usually contain honeycombs in their core parts. In order to increase the energy absorption capacity, honeycomb structure with graded mechanical characteristics can be applied. Stromsoe [23] has studied the application of honeycomb in sandwich structures using numerical simulations in ABAQUS software and applying graded structure for achieving highest levels of energy absorption. In this paper, an in-plane loading has been applied on the sandwich structure. Galehdari et al. [8] have studied a graded honeycomb structure (GHS) under quasi-static loading using numerical and experimental method. In this study, the results of some analytical equations were validated with experimental ones.

When designing shock absorbers for the protection of human occupants, reaction force must be considered as well as the amount of energy absorption. In this research, the design of shock absorbers under helicopter seats has been investigated. During a helicopter crash, impact energy would be applied to the seat first and then to the passenger. Hence, the amount of absorbed kinetic energy and the transferred force must be regarded during the design of shock absorbers. In-plane loading, in contrast with out-of-plane loading, applies a lower force to the protected part, considering specific energy absorption. On the other hand, grading the structure ensures that the application of reaction force to the protected part would happen with a lower rate and within a longer duration. Given the above-mentioned features for GHS, a shock absorber for helicopter crashes has been designed and analysed based upon Joint Aviation Regulations part 27 (JAR27) standard.

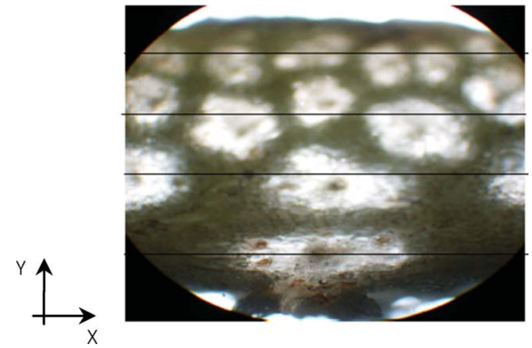


Figure 1. Cross section of a banana peel [17].

2. Banana peel; a compact energy absorber

The structures replicated from honeycombs are one of the prime candidates for reducing the impact in automobile, aerospace, and packaging industries. Figure 1 shows the cross sections of a banana peel. In technical terms, such kinds of materials are called functionally graded materials (FGM). In FGM, the composition and structure gradually vary with depth, resulting in corresponding changes in the properties of the material. Observations show that such types of FGM structures are highly adaptive to all boundary and loading conditions defined by their environment. For example, the interior structure of a bone has an optimised shape with respect to the direction of principal stress and the magnitude of shear stress [16]. In a banana peel, one of the main objectives is to protect the internal soft core from external impacts.

The arrangement of cells can be broken down into layers in the X-direction and the grading (variation in size) in the Y-direction (see Figure 1). Four layers can be easily identified. These layers are indicated by horizontal lines. The first layer from top is composed of closely packed cells. The second layer is composed of bigger cells with more spacing among them. This variation in the pattern continues until the last layer, where cells are widely dispersed. The graded structure shows that the stiffness changes with thickness. According to structural mechanics, two different-sized cross sections with the same shape factor have different degrees of stiffness. The larger the cross section is, the lower the stiffness. If a foreign object hits a banana peel from the top (or a banana falls on another object), the inner cells collapse first to protect the soft-core. This 'collapse mechanism' would flow up and layers would continue crushing until the whole structure is compromised. It is evident that such a 'collapse mechanism' allows structures to reduce the kinetic energy of the object over a finite period of time, and the overall effect is a reduction in the impact load. The presence of fluid in the cells enhances the integrity and total energy absorption capability of the structure. The widely dispersed (biggest) cells in the bottom layer do not communicate structurally with one another. This information leads to the hypothesis that the structure in a banana peel acts as a compact energy absorber. However, the banana peel structure is too complex to handle and, therefore, a few assumptions are made to redefine the structure without excessive deviation from the peel structure. It is assumed that the material is homogenous along the thickness, and that the cells have constant shape factor and are arranged in a uniform order. Figure 2 shows the modified peel structure which may be called the GHS. The global response of the structure can be achieved by summing up the local response of individual rows. Due to the complexity of the peel structure and the emphasis on extracting the cellular structure for investigating in-plane crushing behaviour, the following assumptions are made to model the banana peel structure: (1) the cell wall material is homogenous,

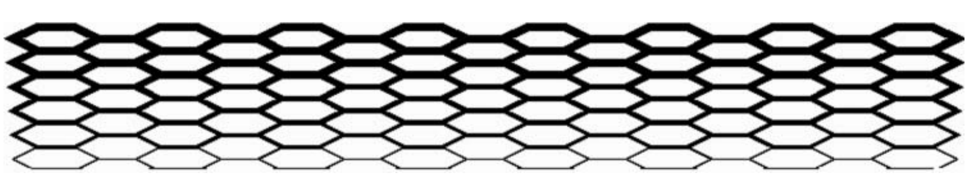


Figure 2. The modified peel structure [17].

(2) the shape of the cells and their aspect ratios do not vary with thickness, and (3) the effect of fluid inside the cells is neglected. In addition, the cell arrangement depicted in Figure 1 more or less stays the same in Z-direction, which allows an in-plane 2D analysis feasible. The quasi-static and low dynamic analysis for impact velocities up to 20 m/s show that the peel graded structure has superior energy absorbing characteristics for a broader range of impact velocities, while considered in a restricted space as compared to regular honeycombs of constant wall thickness. A balanced response between the structural integrity and attenuation of reaction load is observed. At microscopic scale, the compacted cell shape keeps the impact effects low. A theoretical model was presented that correctly captured the crushing response of peel structure at static and low dynamic loadings [17].

3. Specific energy absorption by a graded honeycomb structure

Galehdari et al. [7] primarily derived the fully plastic moment equation for obtaining the analytical equation describing the specific energy absorption of the GHS. It is noteworthy that this equation was derived based on a power hardening material model. Accordingly, the fully plastic moment on the cross section of the honeycomb cells' wall would be described by Equation 1

$$M_p = 2b\int_0^{\frac{d}{2}} y\sigma dy \tag{1}$$

Considering the power hardening material model and by substituting $\sigma = K\varepsilon^n$ and $\varepsilon = \frac{2y}{d}\ \varepsilon_{\max}$ in Equation (1), the corresponding fully plastic moment can be obtained

$$M_{pu} = \frac{b\sigma_u d^2}{2(n+2)} \tag{2}$$

where σ_u is the ultimate strength of the material of structure cell. An upper bound on the load acting on the wall is given by

$$P = \sigma_p\ (c + l\sin\phi)b \tag{3}$$

A lower bound on a collapse load is calculated by equating the internal negative moment on the cell wall to the external positive moment (Equation 4)

$$2\ M_p = P(l-d)\sin\phi \tag{4}$$

Substituting Equations (2) and (3) into Equation (4), the power hardening model plateau stress is derived as

$$\sigma_p = \left(\frac{\sigma_u}{n+2}\right)\frac{d^2}{(c+l\sin\phi)(l-d)\sin\phi} \tag{5}$$

The corresponding locking strain based on relative density can be calculated as [9]

$$\rho^* = \frac{\left(\frac{d}{l}\right)\left(\frac{c}{l}+2\right)}{2\left(\sin\phi + \frac{c}{l}\right)\cos\phi}\rho_s \tag{6}$$

It is noteworthy that ρ^* is the cell density of the honeycomb structure, and ρ_s is the density of the material used to fabricate the honeycomb structure. The porosity, which in fact is the pore volume, is $1 - \frac{\rho^*}{\rho_s}$. This value is approximately equal to the locking strain ε_d as [9]

$$\varepsilon_d = 1 - \frac{\rho^*}{\rho_s} = 1 - \frac{\left(\frac{d}{l}\right)\left(\frac{c}{l}+2\right)}{2\left(\sin(\phi) + \frac{c}{l}\right)\cos(\phi)} \tag{7}$$

Parameter ε_d is the strain corresponding to the end of deformation in each row. In the following equation, it is assumed that the energy absorption is equal to the strain energy of the entire structure. The equation of strain energy is

$$U = \int\left(\int \sigma d\varepsilon\right) dV \tag{8}$$

where σ is the stress tensor applied to the structure, ε is the strain tensor, and V is the volume of the structure. In order to calculate the strain energy per unit volume, the surface area below the stress–strain curve is found. Since the thickness of the cell walls is changing, the equations presented above can only capture the response of individual rows. Each row of the structure would confront deformation in plateau stress and locking strain. Therefore, considering $\sigma = \sigma_p$ and $\varepsilon = \varepsilon_d$ (the equal values of stress to plateau stress and strain to locking strain), strain energy per unit volume would be

$$u = \sum_{i=1}^{6} \sigma_{p_i}\ \varepsilon_{d_i} \tag{9}$$

Based on the equations, the strain energy for the entire structure can be obtained as

$$U = \int\left(\int \sigma_p d\varepsilon_d\right) dV = AL\sum_{i\ =\ 1}^{6} \sigma_{p_i}\frac{\varepsilon_{d_i}}{6} \tag{10}$$

where σ_{p_i} is the plateau stress of each row for the material model with power hardening, ε_{d_i} is the corresponding locking strain for each row, A is the cross section of the structure perpendicular to the longitudinal direction, and L is the structure height in longitudinal direction (direction of collision). According to the dimensions of honeycomb, the energy equation (Equation (10)) can be rewritten as

$$U = 2bl\cos(\phi)(15c + 16l\sin(\phi)) \sum_{i=1}^{6} \sigma_{p_i}\varepsilon_{d_i} \qquad (11)$$

where σ_{p_i} is calculated for power hardening model as

$$\sigma_{\mathrm{p}_i} = \left(\frac{\sigma_{\mathrm{u}}}{n+2}\right) \frac{d_i^{\,2}}{(c + l\sin(\phi))(l - d_i)\sin(\phi)} \qquad (12)$$

The cell volume and mass are

$$V_c = db(4l + 2c) \qquad (13)$$

$$m_c = \rho_s \, db(4l + 2c) \qquad (14)$$

The structure has 6 rows and 15 cells in each row. Since the thickness of each row is different in this structure, the mass of the entire structure is

$$m = \rho_s \, b(32l + 23c) \sum_{i=1}^{6} d_i \qquad (15)$$

An important parameter in the design of energy absorbers is the specific energy of the structure which is

$$e = \frac{U}{m} = \frac{2bl\cos\phi(15c + 16l\sin\phi)\sum_{i=1}^{6}\sigma_{p_i}\varepsilon_{d_i}}{\rho_s b(32l + 23c)\sum_{i=1}^{6} d_i} \qquad (16)$$

Galehdari et al. [7] have compared the results of the equations listed above to those of numerical methods using ABAQUS software. This comparison is demonstrated in Figure 3.

Figure 3 shows that the analytically predicted energy absorption is acceptably close to numerical results, which shows that the analytical equations can be utilised in finding the energy absorption of the structure. The maximum difference between numerical and analytical energy absorption is 6.4%. Hence, in this research the derived equation for specific energy absorption is used in the optimisation problem.

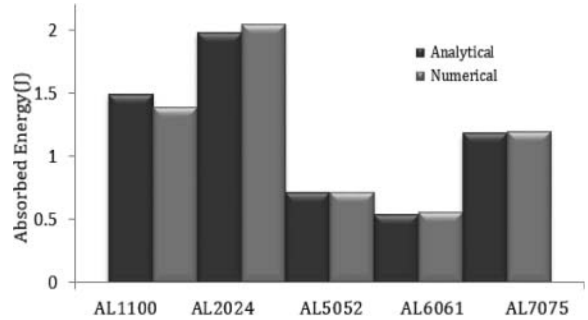


Figure 3. Comparing of analytical and numerical energy absorption of GHS with different types of aluminum [7].

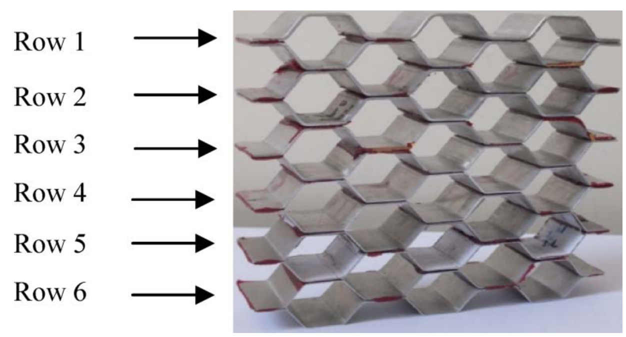


Figure 4. Test sample of 6-row GHS.

4. Experimental test

In order to validate the numerical simulation method in ABAQUS software, an experimental test was conducted. In this test, a falling weight of low velocity was dropped on a GHS. The experimental model is a 6061-O aluminium GHS. This structure has six rows with different thicknesses which are glued to each other by adhesive film. The thickness of the first to sixth rows is 1.6, 1.27, 1.016, 0.8125, 0.635, and 0.508 mm, respectively. For this model, c = 15 mm, l = 12 mm, b = 28.5 mm, and the height and width are both 130 mm. This structure is demonstrated in Figure 4.

To determine the characteristics of this alloy for a more precise analysis, the materials of all of six layers have been analysed under a tension test using a load test machine. The mechanical characteristics of each row are listed in Table 1. It is noteworthy that the density and Poisson's ratio of this aluminium are 2700 kg.m^{-3} and

Table 1. Mechanical properties of different thickness of AL6061-O plate.

Thickness (mm)	E (GPa)	n	K (MPa)	e_f %	Sut (MPa)	Sy (MPa)
1.6	68.28	0.213	202.77	23.76	131.39	51.59
1.27	66.98	0.245	242.66	25.142	141	51.92
1.016	62.5	0.291	220.8	25.168	131	50.7
0.8125	63.51	0.229	205.6	30.72	141	50
0.635	64.3	0.247	228	27.06	134	48.15
0.508	66.81	0.303	217.27	31.092	124	53

0.33, respectively. The mechanical properties are used to define the material properties in FE simulation. A low velocity impact test was performed by a drop hammer test device. In this test, 99 Joule kinetic energy was applied to the GHS. A system of 9776.6 gr was dropped from a 1.2 metre height. The acceleration of the block was recorded during the impact to the structure, and energy absorption was derived from the accelerometer data. Based on these results, a force-displacement diagram was sketched.

4.1. Numerical simulation

Based on the material and geometric characteristics of the structure, the mass and the velocity of the weight, and loading, and boundary conditions, the experimental test on the sample was simulated in ABAQUS software. The FE model made of aluminium 6061-O of GHS is demonstrated in Figure 5. The dropped weight and the structure base are modelled by plates A and B, respectively. Hourglass controlled, 8-node, reduced-integration linear brick elements (C3D8R) are used to mesh the structure, and rigid bilinear quadrilateral elements (R3D4) are used to mesh plate A and plate B, respectively. The boundary conditions are defined by constraining the discrete rigid plate A to move only in the Y-axis and by fixing all the rotational and translational degrees of freedom of the discrete rigid plate, B. Interaction properties are imposed using a general contact condition for each row, and surface-to-surface kinematic contact conditions between the upper surface of the structure and the rigid plate A. A penalty contact condition with friction tangential behaviour is applied between the lower surface of the structure and the rigid plate B. The coefficient of friction is considered equal to 0.6. Using the measured material properties, the plastic behaviour of AL-6061O is defined using a power

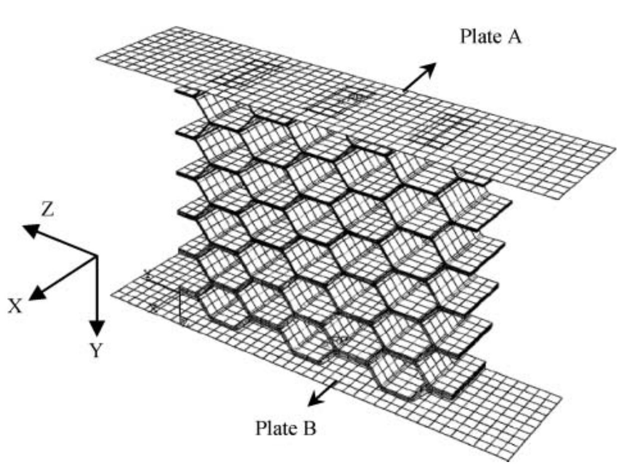


Figure 5. FE model of GHS.

hardening model for each row individually. The FE problem is solved by dynamic/explicit solver.

In this simulation, the reaction force-deformation diagram of the structure obtained from numerical solution is compared to the experimental results.

5. Design and analysis of the shock absorber

5.1. Introducing JAR27 aviation standard

Receiving verification for newly designed and manufactured helicopters and airplanes is a highly necessary and inevitable step in their design process. For this very purpose, a rotorcraft must satisfy all of the relevant articles of the JAR standard. A collection of these rules has been codified by International Civil Aviation Organization. Article 562 of this standard addresses the emergency crash condition of helicopters. Different sections of this article are reprinted, as follows [5].

The rotorcraft, although it may be damaged in an emergency crash landing, must be designed to reasonably protect each occupant when:

a. Each seat type design or other seating device approved for crew or passenger occupancy during takeoff and landing must successfully complete dynamic tests or be demonstrated by rational analysis based on dynamic tests of a similar type seat in accordance with the following criteria. The tests must be conducted with an occupant, simulated by a 170-pound anthropomorphic test dummy, as defined by 49 CFR 572, subpart B, or its equivalent, sitting in the normal upright position.
b. A change in downward velocity of not less than 30 feet per second when the seat or other seating device is oriented in its nominal position with respect to the rotorcraft's reference system, the rotorcraft's longitudinal axis is canted upward 60° with respect to the impact velocity vector, and the rotorcraft's lateral axis is perpendicular to a vertical plane containing the impact velocity vector and the rotorcraft's longitudinal axis. Peak floor deceleration must occur in not more than 0.031 s after impact and must reach a minimum of 30g's.

The conditions of performing emergency landing simulation test are presented in Figure 6 [5].

5.2. Design of seat shock absorber

According to JAR 27.562 standard, the following information should be considered in designing a shock absorber.

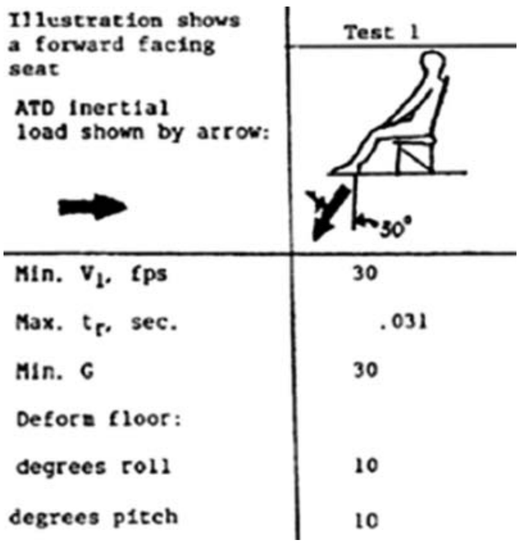


Figure 6. Conditions of emergency landing simulation test [5].

a. For the purpose of a conservative design, the shock absorber is supposed to absorb the entire applied kinetic energy (the energy absorption by the seat's legs is neglected).
b. The passenger weight is 77 kg and the impact speed is 9.1 m/s. Given the conditions of crash simulation test, the helicopter should crash to the ground with a 30° angle with horizontal axis, and the applied energy to the shock absorber should be 2391.15 J.

5.2.1. *Optimised design*

In designing shock absorbers, it is required to select the structure's geometry and material in a way that the specific energy of the structure always remains maximum. Accordingly, the structural optimisation was performed regarding the analytical equations derived previously. During the design process, the selected shock absorber thickness is similar to that of the empirical test sample. Therefore, the thicknesses of 1.6, 1.27, 1.016, 0.8125, 0.635, and 0.508 mm have been used for each row of the structure.

The specific energy is required to be maximised by changing geometric parameters. The structure experiencing the crash is a plate of the mass of 77 kg with the initial velocity of 7.88 m/s. The purpose of optimisation in this step is minimising the ratio of the mass to the amount of energy absorption of the structure. To design the shock absorber, optimisation has been performed using Genetic and Sequential Quadratic Programming (SQP) algorithms in MATLAB®. For both of these algorithms, the objective function is the ratio of the mass to the energy absorption of the structure as shown in Equation 17

$$e = \frac{U}{m} = \frac{2bl\cos\phi(15c + 16l\sin\phi)\sum_{i=1}^{6} K_i \sigma_{p_i} \varepsilon_{d_i}}{\rho b(32l + 23c)\sum_{i=1}^{6} d_i} \tag{17}$$

In the equation mentioned above, K_i is the number of rows in each of the six determined thicknesses. In order to apply the influence of bending moment on the structure's deformation, the condition of $\frac{d}{l} \leq 0.25$ should be met. According to the standards, the reaction force exerted to the passenger pelvis must be less than 6674 N. Moreover, the structure should be capable of absorbing a minimum of 2391.15 J of kinetic energy.

Hence, the constraints of the problem for both of the algorithms are defined in Equation 18

$$\begin{gathered} \frac{d}{l} < 0.25 \\ \sigma_p A < 6674 \text{ N} \\ U = 2391.15 \text{ J} \end{gathered} \tag{18}$$

Design variables are provided in Table 2. The upper and lower boundaries of the variables are equal to vectors ***ub*** and ***lb***, respectively.

$$\begin{aligned} \boldsymbol{lb} = [&0.015, 0.012, 0.628, 0.000245, 0.00025, 0.000255, \\ &0.00026, 0.000265, 0.00027, 1, 2, 3, 4, 7, 13] \end{aligned}$$

$$\begin{aligned} \boldsymbol{ub} = [&0.025, 0.02, 0.7, 0.00025, 0.000255, 0.00026, \\ &0.000265, 0.00027, 0.000277, 20, 20, 20, 20, 20, 20] \end{aligned}$$

According to the above-mentioned design variables and Equation 18, the optimisation problem can be defined as Equation 19

$$\left[\min \frac{m(\boldsymbol{X})}{U(\boldsymbol{X})}\right] \text{ Objective function} \tag{19}$$

Table 2. Design variables in optimising the helicopter seat shock absorber.

Geometric parameter	Design variable	Geometric parameter	Design variable
x(1)	c Cell's horizontal wall length	x(10)	K_1
x(2)	l Cell's inclined wall length	x(11)	K_2
x(3)	φ Cell's wall angle	x(12)	K_3
x(4)	d_1 Cell's wall thickness	x(13)	K_4 Number of rows with d_i thickness
x(5)	d_2	x(14)	K_5
x(6)	d_3	x(15)	K_6
x(7)	d_4		
x(8)	d_5		
x(9)	d_6		

s.t.

$$2bl\cos\phi(15c+16l\sin\phi)\ \sum_{i=1}^{6}K_i\sigma_{p_i}\varepsilon_{d_i}=2391.15\ \mathrm{J}$$

$$\sigma_{p_i}=\left(\frac{\sigma_u}{n+2}\right)\frac{d_i^{\,2}}{(c+l\sin(\phi))(l-d_i)\sin(\phi)}$$

$$\times\ b(15c+16l\sin(\phi))\le 6674\ \mathrm{N\ and}\qquad \text{Constraints}$$

$$\frac{d(i)}{l}<0.25,\ \ \text{for } i=1,2,3,4,5,6$$

$$\boldsymbol{lb}\le\boldsymbol{X}\le\boldsymbol{ub}\ \text{Bounds}$$

The optimisation is conducted by two Genetic and SQP algorithms and their results are compared to each other. The SQP algorithm is used to evaluate the optimisation results of Genetic algorithm.

In Genetic algorithm, 30 generations have been considered with two stopping criteria applied: a maximum number of 100 generations, and a maximum number of 15 for continuous generations without a change in optimum point. A constraint and function tolerance of 10^{-9} is also applied. In SQP method, the constraint and function tolerance equal to 10^{-6} and the start point equal to lower limit (lb) has been considered.

5.3. Numerical simulation

In order to evaluate the optimisation results, the designed model has been simulated in ABAQUS software. For the purpose of meeting the requirements of the standard, the diagrams of reaction force, acceleration, and kinetic energy should be plotted as a function of time. According to the results of the optimisation, the geometry of each row is demonstrated in Figure 7.

Regarding the conducted design, the entire structure consists of 30 rows. The material of all of the rows is AL6061-0. The material properties of each row with its specific thickness are considered based on the properties mentioned in Table 1. A scheme of the designed shock absorber is represented in Figure 8.

For meshing the structure, the four-node shell element of S4R, and for rigid plates A and B, the bilinear and four-node element of R3D4 has been used. Applying the impact loading to the structure has been realised by the upper plate A with a mass of 77 kg and a velocity of 7.88 m/s, while the interaction of the shock absorber with the passenger has been simulated by the lower plate B. The interaction of plate A and the structure is a kinematic surface-to-surface contact, and the interaction of plate B and the structure is defined as a frictionless penalty contact. Defining boundary conditions on plate A, motion is only allowed in the Y-direction (impact direction) while all degrees of freedom are fixed for plate B.

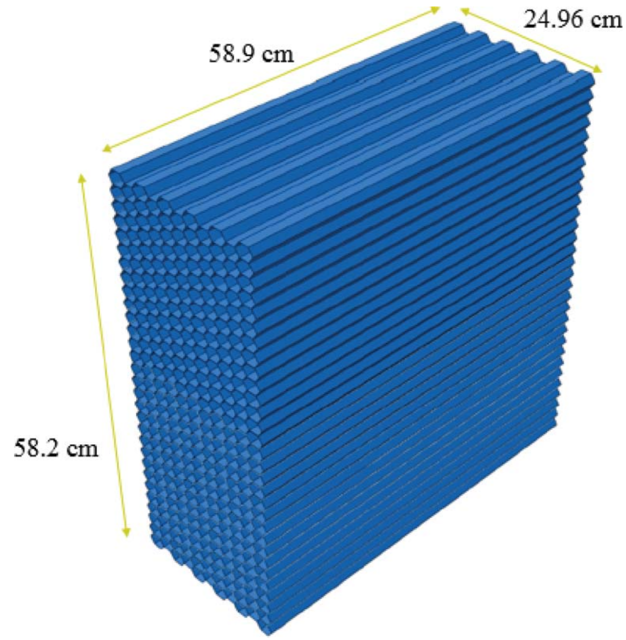


Figure 8. Geometrical model of the helicopter seat shock absorber.

6. Results and discussions

In the experimental test, the force–displacement response of the structure under impact load with low velocity is plotted in Figure 9, along with results from the numerical simulation. Furthermore, the deformed model of the structure and numerical simulation are compared for $t = 1.3$ ms and $t = 2.4$ ms in Figure 10.

Regarding Figures 9 and 10, numerical results exhibit an acceptable level of agreement with experimental results, in a way that the maximum error between the reaction force of experimental and numerical results is 5.4%. Hence, the numerical simulation method and the applied parameters in this simulation are validated.

Genetic algorithm with an initial population of 70 has been applied for the purpose of optimisation. Maximum produced generations of 100 and 15 continuous generations without a change of the optimum point have been

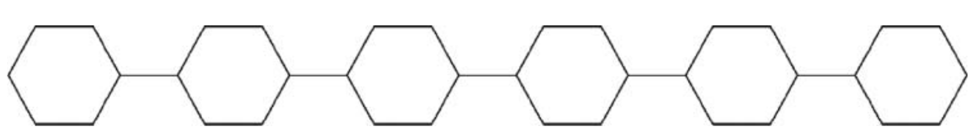


Figure 7. Geometry of one row of helicopter seat shock absorber.

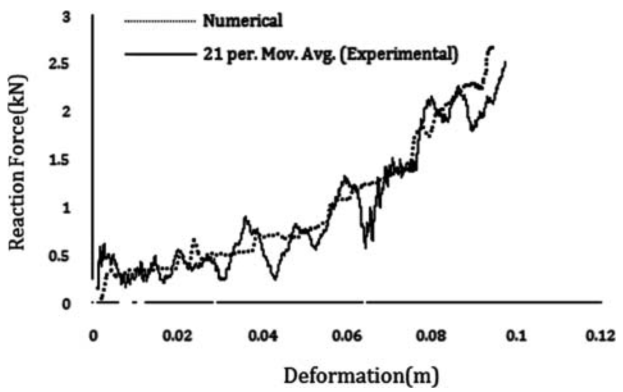


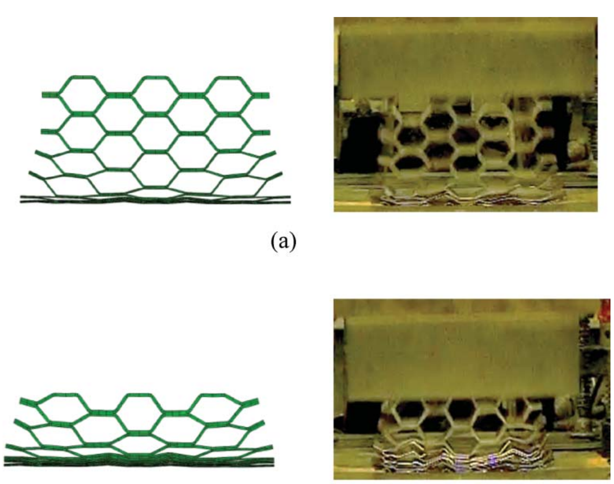
Figure 9. Force–displacement diagram of GHS for experimental test and numerical solution.

Table 3. Obtained design parameters from Genetic and SQP algorithms.

Design variable	Optimised value by GA	Optimised value by SQP
c	15 mm	18.04 mm
l	12 mm	12.17 mm
ϕ	30°	30.8°
d_1	0.25 mm	0.2408 mm
d_2	0.255 mm	0.2549 mm
d_3	0.26 mm	0.255 mm
d_4	0.265 mm	0.2646 mm
d_5	0.27 mm	0.2695 mm
d_6	0.277 mm	0.2751 mm
K_1	1	2.569
K_2	2	3.628
K_3	3	4.6
K_4	4	5.592
K_5	7	8.59
K_6	13	14.39

defined as the terminating criteria. The optimisation has stopped after the production of 90 generations while meeting the second termination threshold. The ratio of the mass to the energy absorption of the structure has been recorded as 0.0029 kg/J. The optimisation using SQP algorithm with a function and constraint tolerance of 10^{-6} and an starting point of lower boundary (lb) has stopped after 186 iterations with the function and constraint tolerance terminating criterion. The ratio of the mass to the energy absorption of the structure has been recorded as 0.00382 kg/J. Table 3 lists the design parameters from two optimisation algorithms.

Comparing the results from Genetic and SQP algorithms, since the ratio of the mass to the energy absorption of the structure has been higher in Genetic algorithm, the results of Genetic algorithm have been considered as the optimal.



(a)

(b)

Figure 10. A deformed schema of the GHS obtained from experimental test and numerical solution in (a) t = 1.3 and (b) t = 2.4 ms.

A change in the initial conditions of an optimisation problem can lead to different results from a specific algorithm. Therefore, following the selection of Genetic algorithm, the optimisation process has been replicated with various initial conditions in order to verify the results. The optimisation via Genetic algorithm has been conducted for six different initial conditions, as listed in Table 4. These initial conditions include the population size, the number of generations without a change in the optimum point (stall generations), the tolerance of objective function, and the constraint tolerance.

According to the table represented above, the change of initial condition has not significantly influenced the optimisation result, which is the ratio of the mass to the energy absorption. The maximum error of the ratio of the mass to the energy absorption is 0.7% for the first result; hence, it can be concluded that the selected result (first row of the table) is exclusively the best result.

It is noteworthy that increasing geometric parameters such as cell's horizontal and inclined wall length and cell's wall angle of a GHS causes a reduction in energy absorption. On the other hand, increasing the cell wall thickness raises the energy absorption [6].

The acceleration diagram, reaction force applied to the passenger, and the kinetic energy as a function of time are plotted in Figure 11.

Table 4. Optimisation results for different initial conditions.

Ratio of the mass to the energy absorption (kg/J)	Population size	Stall generations	Tolerance of objective function	Constraint tolerance	Number of generations
0.0029	30	15	10^{-9}	10^{-9}	70
0.00291	30	10	10^{-9}	10^{-9}	29
0.00292	40	10	10^{-9}	10^{-9}	74
0.00291	40	12	10^{-6}	10^{-6}	76
0.0029	45	20	10^{-6}	10^{-6}	171
0.00291	50	25	10^{-6}	10^{-6}	164

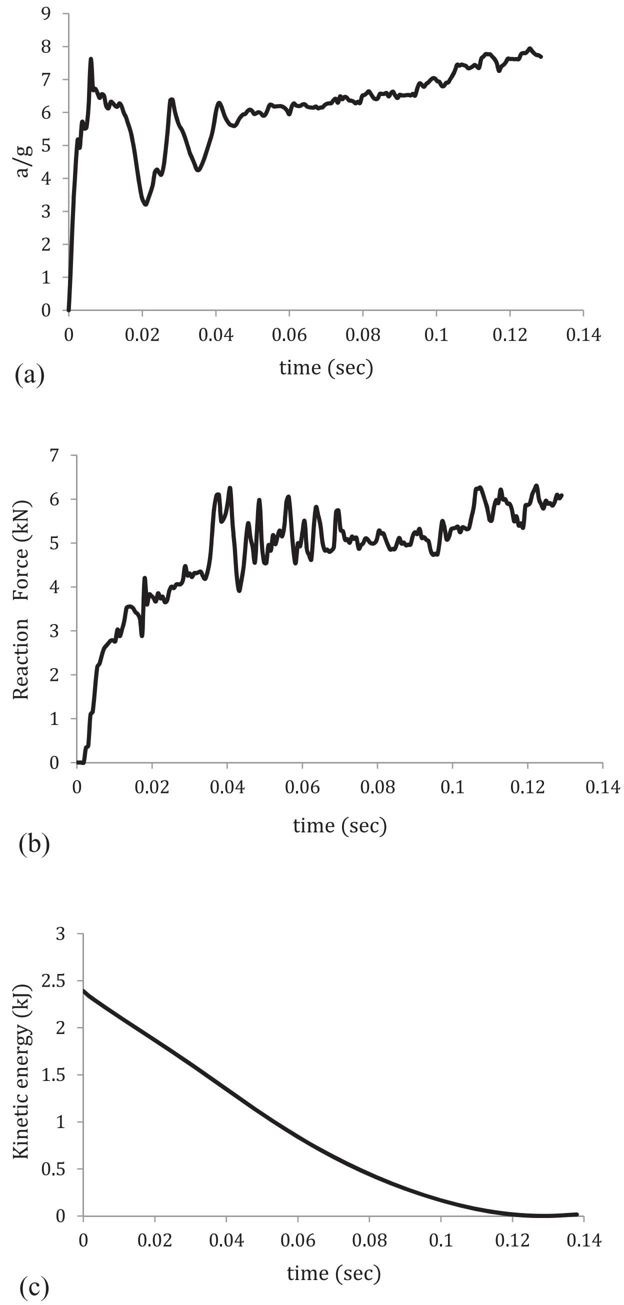


Figure 11. Diagram (a) applied acceleration to the passenger, (b) applied reaction force to the passenger and (c) kinetic energy as a function of time.

According to article (b) of JAR standard, the acceleration change applied to the passenger in a duration of 0.031 second should be less than 30 g. Based on Figure 11 (a), the acceleration changes in this time period are recorded as 7.62 g which is far less than 30 g. Hence, this part of the standard has been met by the designed absorber.

Given the above-mentioned article, the applied force to passenger's pelvis should not exceed 6674 N. According to Figure 11(b), the maximum reaction force is 6300 N which is lower than the allowance limit. Therefore, this section of the standard is also satisfied through the designed shock absorber.

Regarding Figure 11(c), the entire 2391.15 J amount of kinetic energy has been absorbed; thus, the design is appropriate in terms of the energy absorption as well.

Consequently, the designed shock absorber meets all of the articles of JAR27.562 standard for helicopter crash, and could be applied under a helicopter seat as the shock absorber in case of emergency landing conditions.

The energy absorption for the designed structure has been simulated in ABAQUS software according to the optimisation results. Given the results of the numerical simulation (Figure 11(c)), the structure is capable of absorbing the optimal amount of energy. Hence, the optimisation results have been again evaluated and verified.

7. Conclusions

The appropriate agreement between empirical results and the numerical simulation of energy absorption by GHS indicates the efficiency of numerical simulation method in ABAQUS software. Using Genetic algorithm and SQP approach, an optimum helicopter seat shock absorber has been designed in terms of energy absorption and transferred reaction force. The analysis of shock absorber in crash condition has been performed through numerical simulation by ABAQUS software. Regarding the obtained numerical results, the amount of applied reaction force to the passenger's pelvis and acceleration changes have satisfied all of the relevant articles of the standard. Hence, this shock absorber could be applied under helicopter seat for crash conditions. Moreover, numerical simulation and other optimisation methods can be applied in order to design miscellaneous shock absorbers. To exemplify the case, the shock absorber of an elevator for emergency impact conditions can be designed following the represented approaches. Satisfying different articles of JAR aviary standard implies the proper performance of GHS. On the other hand, it can be concluded that honeycomb structure retains a great functionality for in-plane directions in terms of energy absorbance and reducing the damages of a crash.

Disclosure statement

No potential conflict of interest was reported by the authors.

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