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Optimum design for graded honeycomb as energy absorber device in elevator cabin

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Abstract

In this study graded honeycomb structure has been introduced as energy dissipation device. This type of energy absorbers can reduce casualty in case of an accident for moving vehicles. In this study energy absorbers design for elevator cabin is considered. A Numerical investigation conducted and results for total absorbed energy, load and acceleration subject to occupant has been presented. Numerical investigation has been done by ABAQUS software. Moreover, Optimization process using Matlab also implemented for graded honeycomb structures. In the optimization process structural mass to total absorbed energy is minimized. Considering high probability for free fall in elevator, protective devices such as energy absorbers plays an important role for reducing casualties. Therefore, considering EN 81-1 standard, energy absorber for free fall of elevator cabin designed in this paper. Designed energy absorber based on grade honeycomb can satisfy standard requirements. The result of this study can be used as a reference for future designs for energy dissipation devices.

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Keywords: graded honeycomb, in plane loading, energy absorption, elevator cabin fall, optimization, Genetic algorithm.

1. Introduction

Fast industrialization of societies and increasing speed of transportation means has consequences which one of them is increasing causality because of accidents. Therefore, one of vital research fields are improving energy absorption capacity for vehicles and protective structures[1,2]. Different mechanisms can be utilized to dissipate kinetic energy such as plastic deformations in deformable solids, elastic deformation and energy dissipation based on change in pressure for fluids. The main objective for all energy absorption devises are dissipation kinetic energy in certain time

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period. Different cross section has been studied to act as energy absorber in various applications[3 –8]Also many different type of materials such as composites[9–11] and metals [12–17]have been focus of investigation in recent years. Energy dissipation devices can be compressed in axial[18 –20]or lateral [21,22]directions.

One of natural energy absorbers are human and animal bones structure. Porous structure of in bones helps to reduce compression stress in joints [23]. Porous materials also in the form of industrial foams shows good energy absorption characteristics and many researchers use them as additional component to main energy absorption structure [24 –28] [29]. It is possible to optimize inner structure of bones considering main and shear stress [30]. Researchers inspired by natural examples and design optimum energy absorber with highest energy absorption capacity. One of these examples for natural structures are banana peel. In the banana structure peel has protective role for inner soft core. Banana peel structure is very ear to honeycomb structure (Fig.1) and it can be modeled as honey comb structure [31]. Among all different configuration and geometry of energy absorption devices honey combs show promising characteristic to dissipate impact force [32–35].

Galehdari et al[36]presented analytical model to predict absorbed energy for graded honey combs considering the lower-bound theorem, an analytical equation for plateau stress is represented, taking power hardening model. For validation experimental tests also conducted. Moreover experimental tests has been simulated by ABAQUS. Ajdari et al [37]using detailed finite element method studied both regular and irregular arrangement for two dimensional honeycombs under in plane dynamic crushing condition. Different mode of deformations and absorbed energy is estimated in this study. Ruan et al[38] studied numerically and theoretically hexagonal honeycomb in plane loading. Factors such as cell wall thickness and impact velocity which influence the localized deformation and plateau stress studied. Finally they introduce theoretical formula for plateau stress at high impact velocities is in terms of the cell wall thickness and velocity. Papka et al [39] studied experimentally and numerically the load –displacement response of both hexagonal-cell aluminum honeycombs and circular polycarbonate honeycombs under in-plane uniaxial loading. They also validate experimental result with numerical ones. Other study about velocity sensitivity of aluminum honeycomb under high-speed axial impact in velocity range from 20 to 80 m/s also conducted by Wang et al [40]. One of main subject of research for many researchers is finding optimum design by numerical methods for energy absorption devices [41–43]. Based on previous studies about honeycomb structures they have high potential for energy absorption and can be used as energy absorption devices in elevator cabins. In this investigation design of this type of structures for elevator cabins is presented.

Nomenclature

GHS	Graded Honeycomb Structure	U	Strain energy
A	Cross section area of GHS	σ_y	Yield stress
b	Depth of GHS cell	ρ^*	Density of honeycomb structure
c	Cell horizontal wall length	ρ_s	Density of honeycomb structure material
d	Cell wall thickness	ε_d	Locking strain
e	Specific absorbed energy	ϕ	Cell wall angle
l	Cell inclined wall length	σ_p	Plateau stress
m	GHS mass		
m_c	Mass of each cell		

2. Specific absorbed energy for graded honeycomb

Honeycomb structures transform in-plane kinetic energy into strain energy by crushing the rows which is equal to plastic hinge plastic energy. The most important parameters characterizing cellular material energy absorption properties are the plastic collapse stress generally known as the plateau stress and the relative density. The plateau stress has been determined using the upper and lower bound theorems. According to the upper bound theorem, an external load computed on the basis of an assumed mechanism, in which the forces are in equilibrium, is always greater than or equal to the true collapse load. On the other hand, the lower bound theorem states that an external load computed on the basis of an assumed distribution of internal forces, in which the forces are bounded by limit values and the forces are in equilibrium, is less than or equal to the true collapse load [44]. Based on this assumption, plastic hinge moment and plateau stress for elastic-perfectly plastic material obtained. The elastic-perfectly plastic plateau stress can be derived :

$$\sigma_P = \left(\frac{\sigma_y}{2}\right)\frac{d^2}{(c + l\sin\phi)(l - d)\sin\phi} \tag{1}$$

It is noteworthy that the relative density is the ratio of structure cell density to the density of the material of the honeycomb structure. In the above mentioned equation, ρ_s is the density of the material of honeycomb structure. The porosity, which in fact is the pore volume, is $1-(\rho^*/\rho_s)$. This value is approximately equal to the locking strain ε_d as [44]:

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

It is worthy to mention that ε_d is the corresponding locking strain for each row. In order to obtain formula for specific absorbed energy, it is assumed that absorbed energy for whole honey comb structure is equal to strain energy for whole structure. Considering geometrical parameters of structures energy equation can be rewrite as:

$$U = 2bl\cos\phi(17c + 18l\sin\phi)\sum_{i=1}^{20}\sigma_{P_i}\varepsilon_{d_i} \tag{3}$$

This structure has 20 rows and each rows contains 17 cell. Considering that there is variation in the thickness of each row, total mass for whole structure can be calculated as:

$$m = \rho_s b(36l + 26c)\sum_{i=1}^{20} d_i \tag{4}$$

Finally, specific absorbed energy for structure studies in this paper will be:

$$e = \frac{U}{m} = \frac{2bl\cos\phi(17c + 18l\sin\phi)\sum_{i=1}^{20}\sigma_{P_i}\varepsilon_{d_i}}{\rho_s b(36l + 26c)\sum_{i=1}^{20} d_i} \tag{5}$$

Galehdari et al [45] compare results obtained from above equation by numerical methods. Fig. 2. Depict outcome of this compression.

3. Design and discussion energy absorber device for elevators cabin

3.1. Design Requirements

Considering En 81-1 [46], standard following factors should be considered for designing energy absorber device for elevator cabin:

A. For conservative design energy absorption device should absorbed kinetic energy.

B. Weight of cabin and occupants should be assumed 1500 kilogram and impact velocity is 1.15 m/s (15% of nominal velocity). Elevator cabin should crush in axial direction to energy absorber devices and kinetic energy applied should be 991.875 J.

C. Negative acceleration for more than 2.5g should not continue more than 0.04 second.

D. Reaction force from cabin to occupant feet should be less than 6200 N.

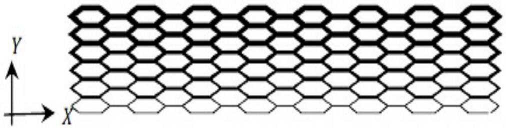

Fig.1. improved model for banana peel.

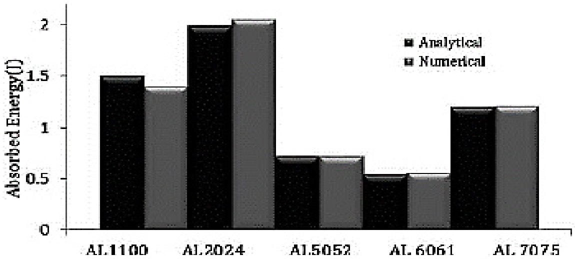

Fig. 2. Comparing of analytical and numerical absorbed energies of graded honey comb structures with different types of aluminum for [45].

3.2. Optimization

One of important parameter for energy absorbers are specific absorbed energy. In designing energy absorbers, it is preferable to have maximum absorbed energy by choosing optimum geometrical design or materials. Considering theoretical equations in previous section Genetic algorithm method is used for optimization. Matlab software is used for computational calculations. Objective function for this study is ratio of mass of energy absorber device to absorbed energy. All other design variables are provided in Table 1. The lower and upper boundaries of the variables are defined with ***lb*** and ***ub*** vectors.

lb = [0.015 0.012 0.628 0.00049 0.00048 0.00047 0.00046 0.00045 0.00044 0.00043 0.00042 0.00041 0.0004 0.00039 0.00038 0.00037 0.00036 0.00035 0.00034 0.00033 0.00032 0.00031 0.0003]

ub = [0.025 0.02 1.3 0.0005 0.00049 0.00048 0.00047 0.00046 0.00045 0.00044 0.00043 0.00042 0.00041 0.0004 0.00039 0.00038 0.00037 0.00036 0.00035 0.00034 0.00033 0.00032 0.00031]

Table 1. Design variables in optimizing the elevators cabin shock absorber.

Design variable	c	l	ϕ	d_1
Definition	Cell's horizontal wall length	Cell's inclined wall length	Cell's wall angle	Cell's wall thickness

In order to consider bending momentum first constrain defined. For response load applied to protected occupant is defined as second constrain. Moreover kinetic energy is third constrain.

$$\frac{d}{l} < 0.25$$
$$\sigma_P A < 6200\,N$$
$$U = 991.875J$$

In this method initial population consider 30 and maximum generated population limited to 100. And iterative population without change in optimum point defined 15. Results from optimizations are as follow.

Optimized value by GA = [0.015049 0.014765 0.654738 4.9407E-4 4.8584E-4 4.7060E-4 4.6628E-4 4.5065E-4 4.4645E-4 4.3327E-4 4.2808E-4 4.1518E-4 4.0778E- 4 3.9475E-4 3.8691E-4 3.7639E-4 3.6098E-4 3.5213E-4 3.4128E-4 3.3341E-4 3.2006E-4 3.1690E-4 3.0650E-4]

3.3. Numerical simulation

In order to investigate results in optimization section, obtained geometrical parameters simulated using ABAQUS software. Also aimed to satisfy standard requirement load displacement diagram, acceleration and kinetic energy verses time sketched. Geometry of single cell and row of cells presented in Fig. 3.

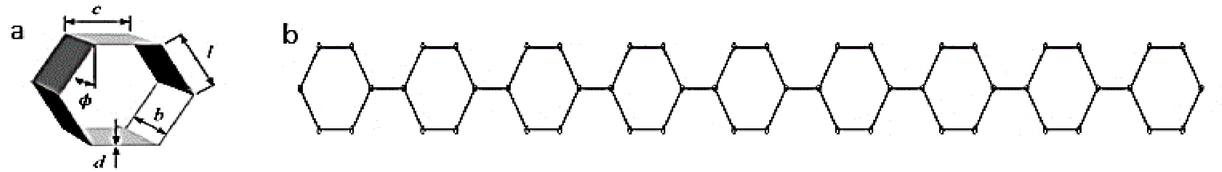

Fig. 3. Geometry of (a) one single cell; (b) one row of honeycomb.

In the numerical simulation of graded structure this values considered for geometry. $c=15mm$, $l=14mm$, $\phi=38^{\circ}$ in order to make structure graded thickness of walls reduced from firs row to twentieth row and values for this wall thickness are *0.49, 0.48, 0.47, 0.46, 0.45, 0.44, 0.43, 0.42, 0.41, 0.4, 0.39, 0.38, 0.37, 0.36, 0.35, 0.34, 0.33, 0.32, 0.31, 0.3 mm* respectively.

Based on design proposed whole structure is 20 rows. Material for all rows are polypropylene, which its material property is presented in Table 2.

Table 2. Material property of polypropylene used in numerical simulation.

Density (Kg/m^3)	Young modulus (MPa)	Poisson ratio	Yield stress (MPa)
905	2.8	0.38	45.45

Considering geometrical and material property of problem simulation is done ABAQUS software. Mass and impact velocity, loading condition and constrains also defined in ABAQUS according to problem definition. In order to simulate impact of mass with protective structure upper and lower plated considered rigid. For modelling honeycomb structure shell elements with 4 nodes used (S4R) and for upper and lower rigid plate surface element with 4 nodes (R3D4) used. Simulated model with upper and lower rigid plates are presented in Fig. 4.

Contact between rigid plate A and structure is “kinematic” type. For contact between rigid plate B and structure “penalty” type with friction is defined. Also movement of rigid plate A is limited to only Y direction and rigid plate B is fixed and can’t be moved. In addition, upper plate weight is 1500 kilogram and it is moving with 1.15 m/s speed and crush to honeycomb structure. Simulated model for energy absorber is presented in Fig. 5.

4. Discussion

The acceleration diagram, reaction force applied to the passenger, and the kinetic energy as a function of time are plotted in Fig. 6. According to standard item C for design requirement says negative acceleration more than 2.5g should not be continued more than 0.04s. Considering diagram 6- a variation of acceleration in this time period is 0.74 which is far less than 2.5g. There item C is satisfied. According to item D for design requirement “Reaction force from cabin to occupant feet should be less than 6200 N”. Considering figure 6-b maximum response load is 4950 N which is lower than limited define value. Therefore item D is also satisfied. In Fig. 6-C totally 991.875 J energy is applied to structure, which means items A and B in design requirement for elevator is satisfied.

All above discussion shows design of energy absorber of elevator cabins is satisfying all design requirements. In other words, this energy absorber is appropriate to be used in elevator cabins and it has capacity to protect occupant. In conclusion considering constrains from standard, it is possible to use this energy absorber for cabin with 1500 kg weight (totally for occupant and cabin) which is moving 1 meter per second.

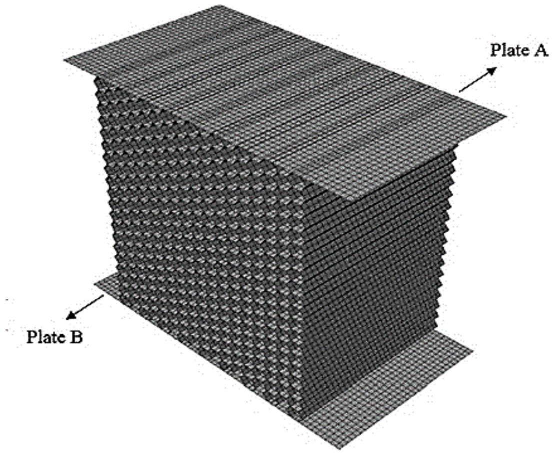

Fig. 4. Finite element model for graded honeycomb structure with rigid upper and lower plates.

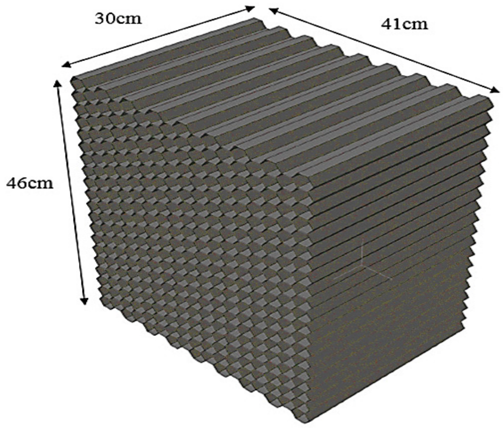

Fig. 5. Geometrical model for elevator cabin energy absorber.

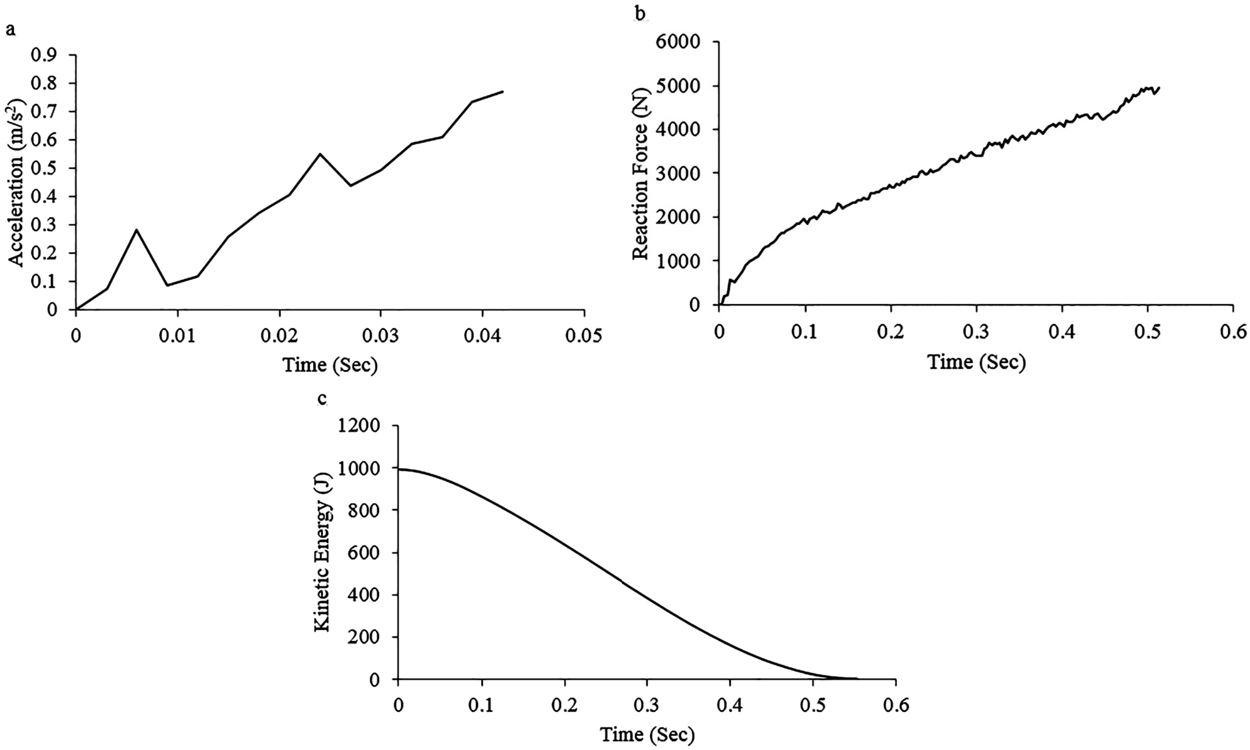

Fig. 6. Diagram (a) applied acceleration to the passenger; (b) applied reaction force to the pass enger; (c) kinetic energy as a function of time.

5. Conclusion

Correlation between experimental and numerical result indicate that numerical simulation is powerful tool for prediction of energy absorption characteristics of graded honeycomb structures. In this work in order to study fall of elevator cabin numerical simulation in ABAQUS software environment is implemented. Based on numerical result, response force felt by occupant variation of acceleration and all other requirement for design is satisfied. Then by using genetic algorithm for, specific absorbed energy, which is ratio of total absorbed energy to structure weight, is maximized. There designed structure can be used as energy dissipation device in elevator cabin. Also it can be concluded that graded honeycomb has lots of potential to be utilized as optimum energy absorber device and many application for vehicles and protective structures.

References

[1] C. Bisagni, Crashworthiness of helicopter subfloor structures, Int. J. Impact Eng. 27 (2002) 1067 –1082. doi:10.1016/S0734-743X(02)00015-5.
[2] R. Hedayati, S. Ziaei-Rad, A. Eyvazian, A.M. Hamouda, Bird strike analysis on a typical helicopter windshield with different lay-ups, J. Mech. Sci. Technol. 28 (2014). doi:10.1007/s12206-014-0125-3.
[3] G. Sun, T. Pang, G. Zheng, J. Song, Q. Li, On energy absorption of functionally graded tubes under transverse loading, Int. J. Mech. Sci. 115 (2016) 465–480.
[4] E. Acar, M.A. Guler, B. Gereker, M.E. Cerit, B. Bayram, Multi-objective crashworthiness optimization of tapered thin-walled tubes with axisymmetric indentations, Thin-Walled Struct. 49 (2011) 94 –105. doi:10.1016/j.tws.2010.08.010.
[5] M. Haghi Kashani, H. Shahsavari Alavijeh, H. Akbarshahi, M. Shakeri, Bitubular square tubes with different arrangements under quasi-static axial compression loading, Mater. Des. 51 (2013) 1095 –1103. doi:10.1016/j.matdes.2013.04.084.
[6] C. Kılıçaslan, Numerical crushing analysis of aluminum foam-filled corrugated single- and double-circular tubes subjected to axial impact loading, Thin-Walled Struct. 96 (2015) 82 –94. doi:10.1016/j.tws.2015.08.009.
[7] D. Meric, H. Gedikli, Thin-Walled Structures Energy absorption behavior of tailor-welded tapered tubes under axial impact loading using coupled FEM / SPH method, Thin Walled Struct. 104 (2016) 17 –33. doi:10.1016/j.tws.2016.03.002.
[8] A. Alavi Nia, S. Chahardoli, Optimizing the layout of nested three-tube structures in quasi-static axial collapse, Thin-Walled Struct. 107 (2016) 169–181. doi:10.1016/j.tws.2016.06.010.
[9] E. Mahdi, T.A. Sebaey, Crushing behavior of hybrid hexagonal/octagonal cellular composite system: Aramid/carbon hybrid composite, Mater. Des. 63 (2014) 6 –13. doi:10.1016/j.matdes.2014.06.001.
[10] M. Moeinifard, G. Liaghat, G. Rahimi, A. Talezadehlari, H. Hadavinia, Experimental investigation on the energy absorption and contact force of unstiffened and grid-stiffened composite cylindrical shells under lateral compression, Compos. Struct. 152 (2016) 626 –636.
[11] W. Tan, B.G. Falzon, M. Price, H. Liu, The role of material characterisation in the crush modelling of thermoplastic composite structures, Compos. Struct. 153 (2016) 914 –927. doi:10.1016/j.compstruct.2016.07.011.
[12] M.J. Rezvani, A. Jahan, Effect of initiator, design, and material on crashworthiness performance of thin-walled cylindrical tubes: A primary multi-criteria analysis in lightweight design, Thin-Walled Struct. 96 (2015) 169 – 182. doi:10.1016/j.tws.2015.07.026.
[13] H. Mirahmadi, M. Azimi, S.S. Mirjavadi, Calculation of stress intensity factor for functionally graded cylinders with two radial cracks using the weight function method, Theor. Appl. Fract. Mech. 85 (2016) 447 –456.
[14] D. Agrawal, S. Rawat, A.K. Upadhyay, Crashworthiness of Circular Tubes with Structurally Graded Corrugations, in: SAE Tech. Pap., SAE International, 2016. doi:10.4271/2016-28-0050.
[15] A. Niknejad, M. Moeinifard, Theoretical and experimental studies of the external inversion process in the circular metal tubes, Mater. Des. 40 (2012) 324 –330. doi:10.1016/j.matdes.2012.04.005.
[16] S. Pirmohammad, S.E. Marzdashti, Crushing behavior of new designed multi-cell members subjected to axial and oblique quasi-static loads, Thin-Walled Struct. 108 (2016) 291 –304.
[17] M.B. Azimi, M. Asgari, A new bi-tubular conical--circular structure for improving crushing behavior under axial and oblique impacts, Int. J. Mech. Sci. 105 (2016) 253 –265.
[18] F. Tarlochan, S. Ramesh, Composite sandwich structures with nested inserts for energy absorption application, Compos. Struct. 94 (2012) 904–916.
[19] A. Eyvazian, M. K. Habibi, A.M. Hamouda, R. Hedayati, Axial crushing behavior and energy absorption efficiency of corrugated tubes, Mater. Des. 54 (2014) 1028 –1038. doi:10.1016/j.matdes.2013.09.031.
[20] M.J. Rezvani, M.D. Nouri, Axial crumpling of aluminum frusta tubes with induced axisymmetric folding patterns, Arab. J. Sci. Eng. 39 (2014) 2179 –2190. doi:10.1007/s13369-013-0734-7.
[21] F. Tarlochan, F. Samer, A.M.S. Hamouda, S. Ramesh, K. Khalid, Design of thin wall structures for energy absorption applications: Enhancement of crashworthiness due to axial and oblique impact forces, Thin-Walled Struct. 71 (2013) 7–17. doi:10.1016/j.tws.2013.04.003.
[22] A. Eyvazian, I. Akbarzadeh, M. Shakeri, Experimental study of corrugated tubes under lateral loading, Proc.

Inst. Mech. Eng. Part L J. Mater. Des. Appl. 226 (2012) 109–118. doi:10.1177/1464420712437307.
[23] Y. Miyamoto, W.A. Kaysser, B.H. Rabin, A. Kawasaki, R.G. Ford, Functionally graded materials: design, processing and applications, Springer Science & Business Media, 2013.
[24] S. Mohsenizadeh, R. Alipour, M. Shokri Rad, A. Farokhi Nejad, Z. Ahmad, Crashworthiness assessment of auxetic foam-filled tube under quasi-static axial loading, Mater. Des. 88 (2015) 258–268. doi:10.1016/j.matdes.2015.08.152.
[25] H. Mozafari, H. Molatefi, V. Crupi, G. Epasto, E. Guglielmino, In plane compressive response and crushing of foam filled aluminum honeycombs, J. Compos. Mater. 49 (2015) 3215–3228.
[26] S. Mohsenizadeh, R. Alipour, Z. Ahmad, A. Alias, Influence of auxetic foam in quasi-static axial crushing, Int. J. Mater. Res. 32 (2016) 146.111418. doi:10.3139/146.111418.
[27] H. Mozafari, S. Khatami, H. Molatefi, Out of plane crushing and local stiffness determination of proposed foam filled sandwich panel for Korean Tilting Train eXpress--Numerical study, Mater. Des. 66 (2015) 400–411.
[28] A. Niknejad, G.H. Liaghat, M. Moeinifard, Experimental Investigation of the Rubber-Filled Tubes in the Tube Inversion Process, in: Adv. Mater. Res., 2012: pp. 33–38.
[29] F. Tarlochan, S. Ramesh, S. Harpreet, Advanced composite sandwich structure design for energy absorption applications: blast protection and crashworthiness, Compos. Part B Eng. 43 (2012) 2198–2208.
[30] J.C. Koch, The laws of bone architecture, Am. J. Anat. 21 (1917) 177–298.
[31] A. Muhammad, Study of a compact energy absorber, Ph. D. Thesis. (2007).
[32] M.Z. Mahmoudabadi, M. Sadighi, A. Eyvazian, Theoretical and experimental crushing analysis of metal square honeycombs under quasi-static loading, in: ASME 2010 10th Bienn. Conf. Eng. Syst. Des. Anal. ESDA2010, 2010. doi:10.1115/ESDA2010-24739.
[33] V. Crupi, G. Epasto, E. Guglielmino, H. Mozafari, S. Najafian, Computed tomography-based reconstruction and finite element modelling of honeycomb sandwiches under low-velocity impacts, J. Sandw. Struct. Mater. (2014) 1099636214531515.
[34] H. Mozafari, S. Khatami, H. Molatefi, V. Crupi, G. Epasto, E. Guglielmino, Finite element analysis of foam-filled honeycomb structures under impact loading and crashworthiness design, Int. J. Crashworthiness. 21 (2016) 148–160.
[35] B. Yu, B. Han, P.B. Su, C.Y. Ni, Q.C. Zhang, T.J. Lu, Graded square honeycomb as sandwich core for enhanced mechanical performance, Mater. Des. 89 (2016) 642–652. doi:10.1016/j.matdes.2015.09.154.
[36] S.A. Galehdari, M. Kadkhodayan, S. Hadidi-Moud, Low velocity impact and quasi-static in-plane loading on a graded honeycomb structure; experimental, analytical and numerical study, Aerosp. Sci. Technol. 47 (2015) 425–433. doi:10.1016/j.ast.2015.10.010.
[37] A. Ajdari, H. Nayeb-Hashemi, A. Vaziri, Dynamic crushing and energy absorption of regular, irregular and functionally graded cellular structures, Int. J. Solids Struct. 48 (2011) 506–516.
[38] D. Ruan, G. Lu, B. Wang, T.X. Yu, In-plane dynamic crushing of honeycombs; a finite element study, Int. J. Impact Eng. 28 (2003) 161–182.
[39] S. Da Papka, S. Kyriakides, Experiments and full-scale numerical simulations of in-plane crushing of a honeycomb, Acta Mater. 46 (1998) 2765–2776.
[40] Z. Wang, H. Tian, Z. Lu, W. Zhou, High-speed axial impact of aluminum honeycomb--Experiments and simulations, Compos. Part B Eng. 56 (2014) 1–8.
[41] M. Kiani, I. Gandikota, M. Rais-Rohani, K. Motoyama, Design of lightweight magnesium car body structure under crash and vibration constraints, J. Magnes. Alloy. 2 (2014) 99–108. doi:10.1016/j.jma.2014.05.005.
[42] M. Kiani, A.R. Yildiz, A Comparative Study of Non-traditional Methods for Vehicle Crashworthiness and NVH Optimization, Arch. Comput. Methods Eng. (2015) 1–12. doi:10.1007/s11831-015-9155-y.
[43] N. Tanlak, F.O. Sonmez, M. Senaltun, Shape optimization of bumper beams under high-velocity impact loads, Eng. Struct. 95 (2015) 49–60. doi:10.1016/j.engstruct.2015.03.046.
[44] L.J. Gibson, M.F. Ashby, Cellular solids: structure and properties, Cambridge university press, 1999.
[45] S.A. Galehdari, M. Kadkhodayan, S. Hadidi-Moud, Analytical, experimental and numerical study of a graded honeycomb structure under in-plane impact load with low velocity, Int. J. Crashworthiness. 20 (2015) 387–400. doi:10.1080/13588265.2015.1018739.
[46] Part 1 (EN 81-1), Safety rules for the construction and installation of Electric lift, European Committee for Standardization, (1998).