Design of Octagonal Energy Absorbing Members Subjected to Dynamic Load: Enhancement of Crashworthiness

Abdulbasit Abdullah, Khairul Salleh Mohamed Sahari, Samer F, Jamal O. Sameer

Abstract— This study examines the crashworthiness performance of the octagonal thin wall tube, based on numerical simulation. The purpose is to find the optimal design with the lowest weight and best crashworthiness parameters in order to protect the passengers' life. Octagonal members with various trigger mechanisms (circular, square and elliptical triggers) with different distributions from the free end of tube were compared with aluminium alloy (AA6060) tube of 5 mm wall thickness, filled with hollow aluminium foam. The filled aluminium tube has given better results by enhancing the energy absorption by 7.1%, CFE by 29.4% and peak force 16% in case axial loading.

Index Terms— dynamic compression, thin wall, energy absorption, direct and oblique loading, aluminum foam

I. INTRODUCTION

Lately, with the rapid development of the automotive industry, the number of the vehicles on the roads has been noticeably increased, which simultaneously causes a higher number of the road traffic accidents.Nowadays there is a high demand of the private and public transportation, as the modern lifestyle requires ability of fast mobility from the members of the society. The higher number of vehicles on the roads causes an increased number of road accidents. Being one of the major risk factors to the human life, these accidents have to be faced and safety measures have to be found, in order to decrease the negative impacts of the crash. All above mentioned factors justify a more detailed study of the energy absorption capabilities of the vehicles [1]. In order to find effective methods of enhancing the energy absorption capacities of the vehicles, improvements in the security level of the vehicle structures have to be achieved. A number of researchers [2]. [3] focused their attention on the mechanical behavior of the simple and multi-cell, aluminum-made, thin-walled members with polygon sections.

Manuscript Received on December 2014.
Abdulbasit Abdullah, Department of Mechanical Engineer, University of Tenaga Nassional, College of Engineering, Kuala Lumpur, Malaysia.
Khairul Salleh Mohamed Sahari, Assoc. Prof. Dr. Head of Department of Mechanical Engineering, University of Tenaga Nasional, College of Engineering, Kuala Lumpur, Malaysia,
Samer F, Department of Mechanical Engineer, Alanbar University, Engineering, Alanbar-Ramadi, Iraq.
Jamal O. Sameer, Department of Mechanical Engineer, University of Tenaga Nasional, College of Engineering, Kuala Lumpur, Malaysia.

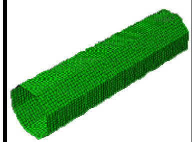
Sections of various shapes have been examined, and subjected to quasi-static loading, like octagonal, hexagonal, square, and triangular. Once the experimentations had been finished, a comparison between their results and the results of the numerical simulations has been done. Based on the outcomes, the members with multi-cell divisions present the best energy absorption capability. The research of [4] is based on thin-walled members with octagonal, hexagonal, and square divisions, and with origami patterns. The simulations of crash have been done by using the FE code Abaqus/Explicit. Their study reveals that by using the patterns on the tubes, a more constant crush process, and lower initial peaks can be obtained. Various other researches have been made on the thin wall tubes. The study of [8], on octagonal and the square thin-wall members under dynamic direct load, revealed that the octagonal members show decreased permanent displacement and increased mean direct load.The numerical and experimental study of [6], on the steel polygons with octagonal, square, and rhomboid divisions reveals that the 135° central angle doesn't result a significantly higher crush resistance of the angle of 90°.The theoretical and numerical research of [5], was based on the performance of the octagonal diviosions. The crashworthiness analyses of [7], are based on finite element models of thin-walled members with octagonal cross sections. The simplified models are designed based on the crash features of the thin-walled octagonal members subjected to direct load. The direct crash resistance of the members is calculated using the universal energy parameters. The nonlinear spring components were subjected to direct load in order to examine the buckling behavior.The study of [8] analyzes, based on the FEA, the usage of origami patterns on the thin-walled members with octagonal, hexagonal, and square divisions, subjected to direct load.The experimental studies of [10] examine the outcome of using various tube fillers when the tubes are subjected to quasi-static direct load. Comparing square and octagonal members, better results have been given by the octagonal ones.The scope of the study of [11], was to find the best dimensions for the tapered member with inner stiffener subjected to direct or oblique load. Having equal weight and peaks of crush load, the optimal tapered tube gives 29.3% lower crush displacement than the tube with octagonal division.The purpose of the research of [9] was to obtain columns with the highest possible energy absorption capacities in case of curved and straight hexagonal and octagonal members.

Published By:
Blue Eyes Intelligence Engineering
& Sciences Publication Pvt. Ltd.

II. DESIGN METHODOLOGY

The study examines the behaviour of the cross sectional, thin wall, octangoal aluminium alloy profile. The profile is long 350 mm, thick 2, 3, 4 and 5 mm, with perimeter of 300 mm. As a first step, we survey the crashworthiness properties of the cross sectional profiles, and this is followed by the research of their improvement possibilities and the choice of the optimal design. Octangoral profile of various weights is filled with hollow aluminium foam of 540 kg/m3 density, and is subjected to direct and oblique (30 degrees) impact load. The simulation is based on an impact mass of the 25% of the total weight of the vehicle; with an initial speed of 60 km/h. Table 1 illustrates the various profiles.

Table 1: Geometry and dimensions of tubes used in this current study

Profile	Specimen ID	Perimeter mm	Length mm	Major Dimension mm	Thickness	Shape
Octagonal	OCT-30 0	300	35 0	37.5 X 8	2 3 4 5	

A. *Force max and Peak load*

The value of the peak load value stands for the power which starts the deformation process [12]. The goal is to reduce the peak load, as this is the factor which cause injury of the vehicle passengers [14].The maximum force means the supreme impact, and the deformation that the members of the passenger car can absorb, maintaining the passenger cabin safe. The goal to achieve is to have vehicle members able to absorb the low-energy and low-velocity mass loads without constant deformation of the structure [16].

B. *Energy Absorption*

The energy absorption ability can be calculated based on the load-displacement reaction.Energy absorption (Ea) is calculated from the load-displacement curve:

$$EA = \int_{0}^{\delta b} P.d\delta$$

P stands for the direct crush load, d for the present crush distance, while and dmax for the possible crush distance.The crushing load is calculated as below:

$$EA = \int_{0}^{\delta b} P.d\delta = P_m(\delta b - \delta i)$$

The constant bend, caused by the collapse loading, has the effect to enhance the crushing process. P_{max} stands for the permanent crushing load of the energy absorber .A way to calculate the energy absorption ability is based on the rectangular zone of a load-deflection curve. Figure 1 presents the features of the perfect energy absorber subjected to axial load.

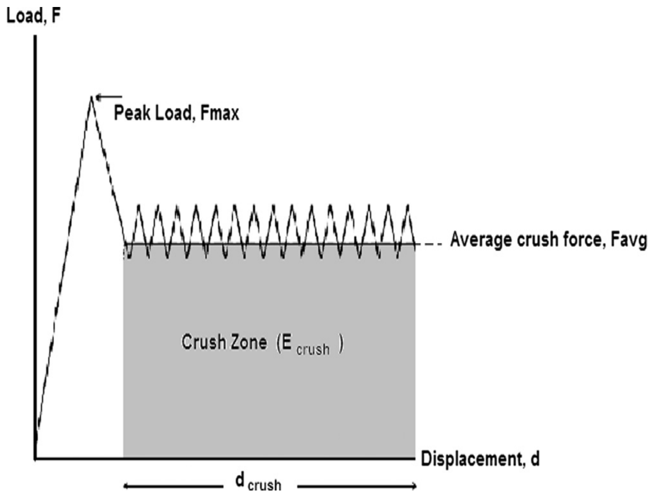


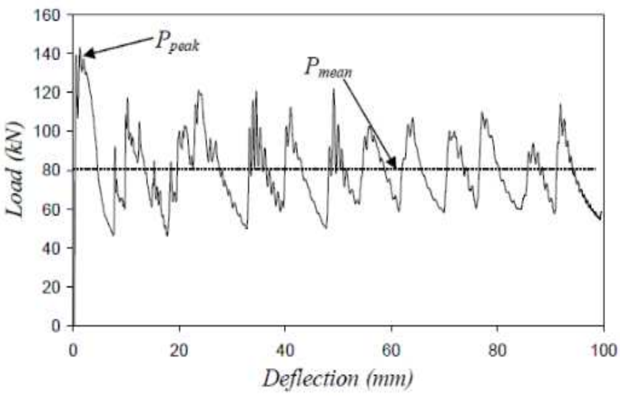
Fig. 1. Force displacement characteristics [22]

C. *Crush Force Efficiency, CFE*

The crush force efficiency (CFE) stands for the average crush force divided by the peak
crush force:

$$CFE = \frac{P_{mean}}{P_{peak}}$$

The goal to achieve is to obtain the highest possible rate of crush force efficiency (CFE) in order to ensure the safety of the passengers in the vehicle. Closest the CFE is to 100%, better energy absorption capacities and crashworthiness



properties the vehicle has [14], Figure 2 by [15] illustrates the load-deflection curve of a thin-walled member subjected to direct quasi-static loading.

Fig. 2. Load-deflection curve foe axial crush.[15]

III. DYNAMIC ANALYSIS

The results of the present research are based on the software of ABAQUS/Explicit version 6.10, as finite element method. The software was used to reduplicate and illustrate the energy absorption capacities of the frontal longitudinal members of

Published By:
Blue Eyes Intelligence Engineering
& Sciences Publication Pvt. Ltd.

the vehicle, when subjected to oblique and direct dynamic load.The advantage of ABAQUS is its ability of simulating computational fluid dynamics (CFD) processes and electrical/standard models. These simulations, in comparison with the implicit techniques, are more efficient from a financial point of view and require shorter time. The simulations are able to reproduce identical circumstances of high velocity and impact load [17] to the implicit methods and allow observing the impacts of the load on the energy absorber members.

A. Finite Element Modeling

The present study examines –with the help of the software ABAQUS as non-linear FE- the effects of the crash on the filled and not filled thin-walled octagonal members. The members were planned 5 integration spots along the thickness direction and with 4 node shell continuum (S4R) elements, suitable when the wall thickness in under 10% of the length**.** S4R is a three-dimensional doubly curved four node shell constituent. All nodes had three-rotation degrees of freedom and three displacements. R3D4 was the base to design the two fix plates. One of them was fixed with the movement freedom in the direction of the compactor load. The second one was fixed without any movement possibility. The hollow aluminum foam was planned with 4-nodded continuum elements and reduced integration performances together with time control. With the help of the rigidity-based time control, the artificial minor or deserted energy distortion modes could be avoided. Volumetric locking was evaded by the reduction, integration.Taking as a reference the results of a mesh convergence study, the size of 5 mm was chosen for the foam elements and shells. The mesh convergence is responsible to guarantee sufficient mesh density and to catch the deformation procedure. The interference among the components was designed based on a ‘‘general contact algorithm”. Based on the algorithm system, the friction coefficient rate among the contact surfaces (fixed and moving plate and the octagonal table) is a permanent value of 0.2 according [1,17]. Based on the data provided by the New Car Assessment Program (NCAP) by the National Highway Traffic Safety Administration (NHTSA), the impact body was modeled as fixed plate (rigid body) with the possibility of linear motion. The weight of the impact body was determined as 275 kg, which is the 25% of the average weight of a passenger car. The velocity of the compactor body was specified as 16.7 m/s (60 km/h). The frontal part of the vehicle had two longitudinal tubes in order to provide crashworthiness. The energy absorption capacity of the two longitudinal tubes is lower than the 50% of the total weight of the vehicle [20]. The software gives an analyze of the variables of the design and provides the possibility of having a test with high efficiency models. Figure (3) illustrates the element types.

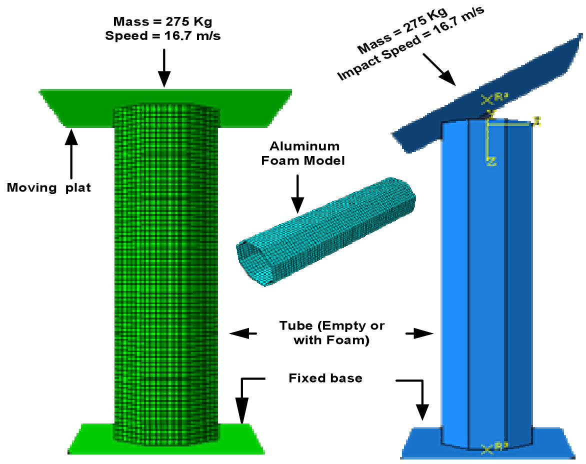


Fig. 3. Design of frontal longitudinal members

Table 2: Mechanical properties of aluminium alloy material [23]

ρ_f (kg/m3)	σp (MPa)	E (GPa)	ν (ratio)
2700	235	68.3	0.3

Table 3: Specific specifications of the aluminium foam material [21]

ρ_f (kg/m3)	σ_P(MPa)	α	α_2 (MPa)	β	γ	E_D
540	12.56	2.12	1544	3.68	1	1.6206

B. Interaction, Boundary Conditions and Loading

The current study is based on the octagonal tube, stabilized from the one end to the rigid plate by tying constraint, allowing only linear motion in the direction of the displacement. The nodes on the octagonal tube were allowed to have rotational motion. The rigid plates were planned as rigid contact surfaces, and as such, they enabled the contact simulation. One of them allowed to the compactor body only the axial movement. One of the reference points of the applied mass was in the middle of one moving plate, while the other was at the end of the tube in order to record reactions. The dynamic load was simulated in the middle of one of the moving plates with specified velocity and mass compactor. The dynamic load and time period were specified by the software. The time period was chosen based on the control and element structure, and mesh dimension.The extended time interval requires high CPU competency and longer time period to show the results. A self-contact between the walls has to be defined for the tube walls and the aluminum foam. Among rigid plates and tube, parts of the telescope or any part of the telescope and a tube, a surface to surface contact is needed. The interaction option can be considered completed when the contact surface and the friction coefficient "penalty" are specified. The rigid plate and the octagonal tube from both sides were fixed in a way to act as one body during the simulation. The octagonal tube had deformation length, while crushing. The mesh size was defined at 5mm [18], [19].

Published By:
Blue Eyes Intelligence Engineering
& Sciences Publication Pvt. Ltd.

IV. RESULT AND DISCUSSION

Tables 4 and 5 illustrate the results of the examination. The tables are followed by detailed explanation.

Table 4: Show the results of crashworthiness for octagonal tube for various thickness has parameter 300 mm (Axial loading)

Indicators	Axial Load OCT- 300		
	P max (KN)	CFE	Energy (KN)
2 mm	74.5	0.544	7.7
3 mm	126	0.583	14.1
4 mm	175	0.611	20.5
5 mm	234	0.627	28.2

Table 5: Show the results of crashworthiness for octagonal tube for various thickness has parameter 300 mm (30 degrees loading)

Indicators	Oblique Load OCT- 300		
	P max (KN)	CFE	Energy (KN)
2 mm	41.6	0.627	5
3 mm	79.5	0.642	9.6
4 mm	109	0.75	15.4
5 mm	157	0.748	21.9

A. Force displacement feature of different thickness profile.

The force displacement diagrams in Figures 4 and 5 illustrate the effects of direct and oblique loading on the 300 mm perimeter tube of various geometric profiles, while Tables 4 and 5 show the results given by the same type of 300 mm perimeter profile of various thicknesses (2, 3, 4, and 5 mm respectively). The energy absorption values are considerably higher in case of direct load, which is given by the fact that the progressive crush caused by the oblique load combines the axial compression with the bending mode. Based on the results of the force-displacement it can be concluded that the folding process is not influenced by the different parameters, when the octagonal profile is subjected to axial or oblique load.

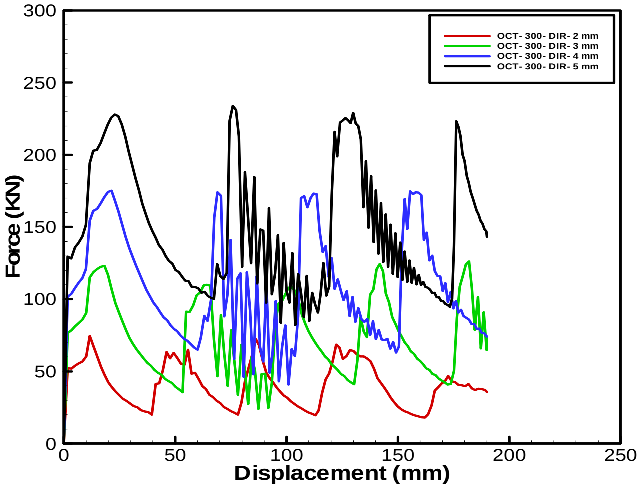


Fig. 4. Force VS displacement for OCT-300 in case direct load

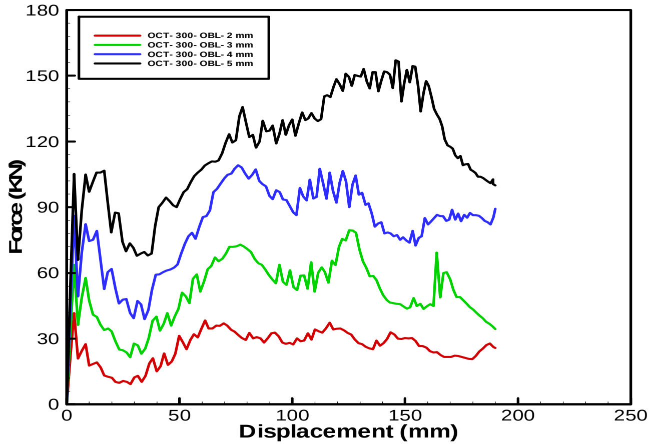


Fig. 5. Force VS displacement for OCT-300 in case oblique load

B. Energy Absorption

Figures 6 and 7 illustrate the energy absorption capability of the octagonal tube of various wall thicknesses and with deformation length of 190 mm under different loading. The time factor has not been taken in consideration. Based on the results, the energy absorption capability of the profile increases proportionally with the thickness of the wall, in case of all load types. Tables 4 and 5 illustrate the effects of oblique loading of 30 degrees on the 300 mm perimeter tube of different thicknesses. Based on the results of the octagonal profile subjected to axial or oblique load has given energy absorption values reduced by 15 – 55 %. The factors to be considered in the choice of the wall thickness are the energy absorption capacity, the CFE, the weight, and the fabrication procedure.

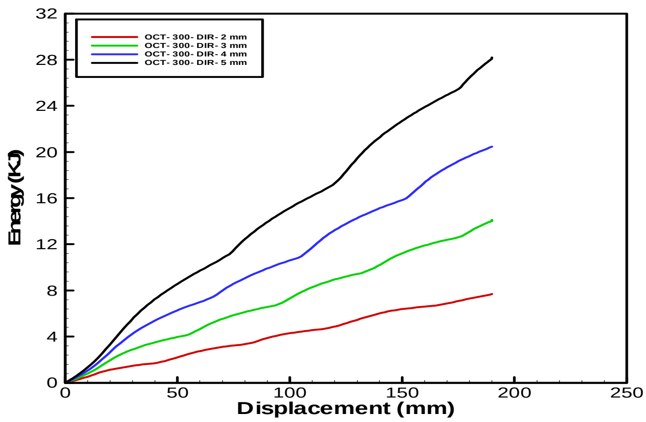


Fig. 6. Energy VS displacement of OCT-300 in case of direct load

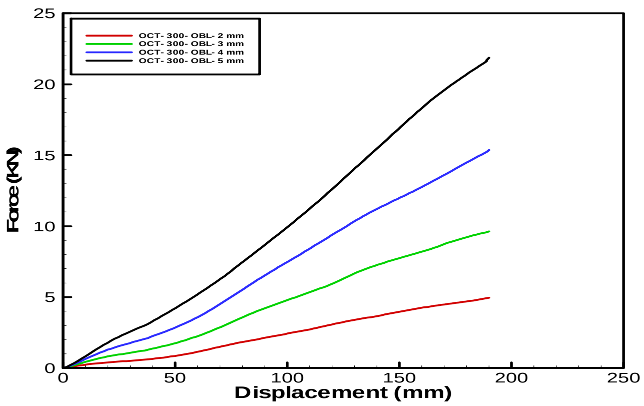


Fig. 7. Energy VS displacement of OCT-300 in case of oblique load

Published By:
Blue Eyes Intelligence Engineering
& Sciences Publication Pvt. Ltd.

C. Choise of the optimal profile

The current research utilizes the multi criteria decision-making (MCDM) procedure, based on the complex proportional assessment method (COPRAS). Based on the comparison of the filled and non-filled octagonal profiles of 300 mm diameter, the best results in CFE and energy absorption have been given by the 5 mm thick octagonal profile filled with hollow aluminium foam.

D. Effect of hollow foam on the energy absorption, peak force and CFE

Further research has been done on the 300 mm perimeter octagonal tube with various wall thicknesses (2 mm, 3 mm, 4 mm, and 5 mm), and weights of the hollow aluminium foam filling (F1/ 0.87 kg, F2/ 0.62 kg , F3/ 0.49 kg, F4/ 0.25 kg, and F5/ 0.15 kg). Figures 8, 9, 10, and 11, show the increased CFE and energy absorption values with the usage if F5 (0.15 kg) weight of hollow aluminium foam combined with various wall thicknesses. Tables 7-8 show the results given by the profile with 5 mm wall thickness and 190 mm deformation length filled by hollow aluminium foam type (F5), which in comparison with the non-filled, 2 mm thick steel octangular profile results an enhancement in energy absorption from 28 KJ to 30 KJ, and in CFE from 0.51 to 0.65, how also a decrease of peak force from 282 KN to 237 KN in case of direct load. The same conditions in case of oblique load have a result of increasing the energy absorption from 21.2 KJ to 22 KJ and CFE from 0.71 to 0.75, while decreasing the peak force form 160 KJ to 157 KJ.

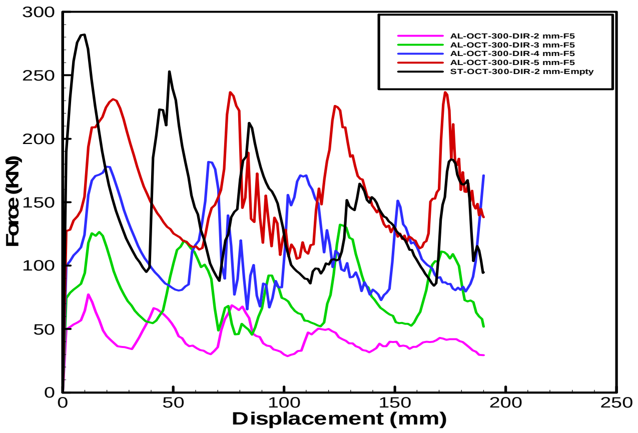


Fig. 8. Force VS displacement of OCT-300 with hollow aluminium foam F5, and different wall thickness of tube, in case of direct load

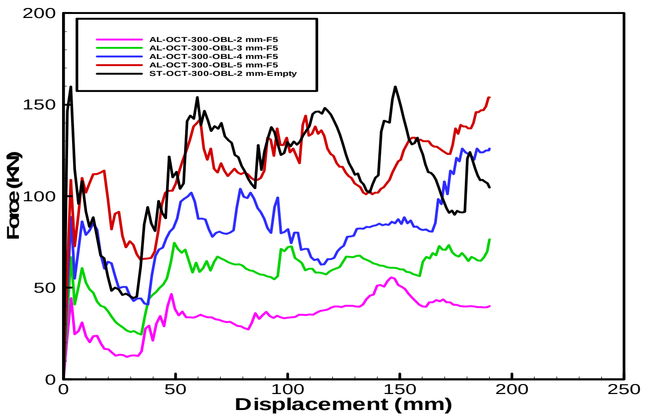


Fig. 9. Force VS displacement of OCT-300 with hollow aluminium foam F5, and different wall thickness of tube, in case of oblique load

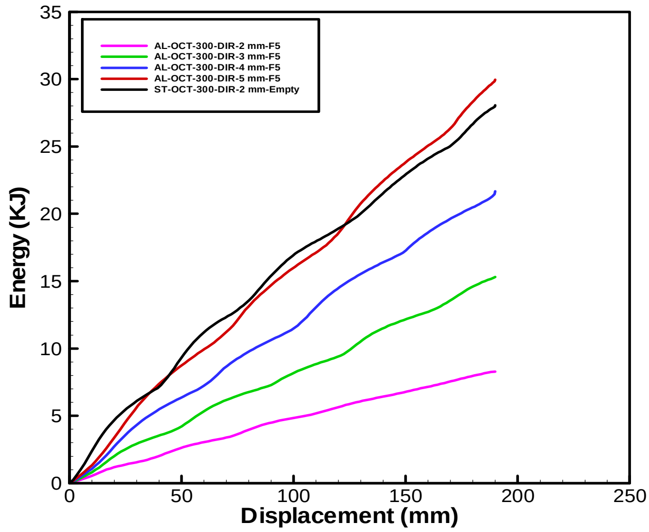


Fig. 10. Energy VS displacement of OCT-300 with hollow aluminium foam F5, and different wall thickness of tube, in case of direct load

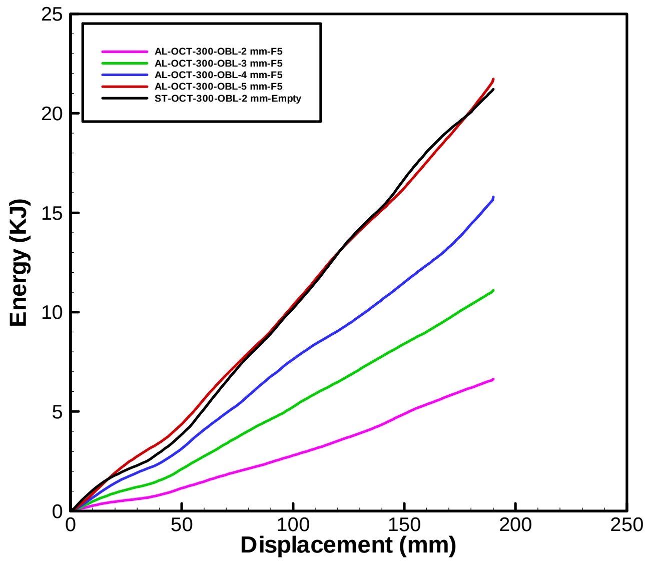


Fig. 11. Energy VS displacement of OCT-300 with hollow aluminium foam F5, and different wall thickness of tube, in case of oblique load

E. Effect of trigger mechanism

Comparing the various shapes (ellipse, circular, and square) of triggers, the best results have been given by the ellipse shape trigger of first type of distribution with one hole in longitudinal side. As shown in [22] and [24] the best reduction is 10%. Comparing the positions of 10, 20, 40, 60, 80, 100, and 120 mm distance from the tube end with the reduction is 10%, the distance of 80 mm showed the best values. As shown in Tables 6, comparing the 5 mm thick aluminium profile with trigger and the 2 mm thick steel profile without trigger, the usage of the triggered aluminium profile increased the energy absorption by 3.5%, and the CFE by 29.4%, while decreased the peak force by 18.8%. Figures 12, and 13, illustrate the force and energy displacement function with the trigger mechanism.

Published By:
Blue Eyes Intelligence Engineering
& Sciences Publication Pvt. Ltd.

Table 6: CFE, peak force, and energy absorption capacity of OCT-300 tube with three different triggers at the position 80 mm from free end, subjected to axial load

The Profile-OCT-300 - Aluminuim Tube has Thickness 5 mm VS Steel Tube has Thickness 2 mm			
Criteria / Type of Trigger	Peak force (KN)	CFE	Energy (KJ)
Steel Tube Without trigger	282	0.51	28
The AL - Tube has Circular trigger with 1 hole	235	0.643	28.7
The AL - Tube has Circular trigger with 2 holes	233	0.635	28.3
The AL - Tube has Circular trigger with 4 holes.	231	0.635	28.1
The AL - Tube has Circular trigger with 8 holes.	223	0.657	28
The AL - Tube has Elliptical trigger with 1 hole	229	0.662	29
The AL - Tube has Elliptical trigger with 2 holes	232	0.645	28.5
The AL - Tube has Elliptical trigger with 4 holes	231	0.635	28.1
The AL - Tube has Elliptical trigger with 8 holes	227	0.643	28
The AL - Tube has Square trigger with 1 hole	235	0.639	28.7
The AL - Tube has Square trigger with 2 holes	234	0.631	28.4
The AL - Tube has Square trigger with 4 holes	234	0.63	28.2
The AL - Tube has Square trigger with 8 holes	228	0.641	27.9

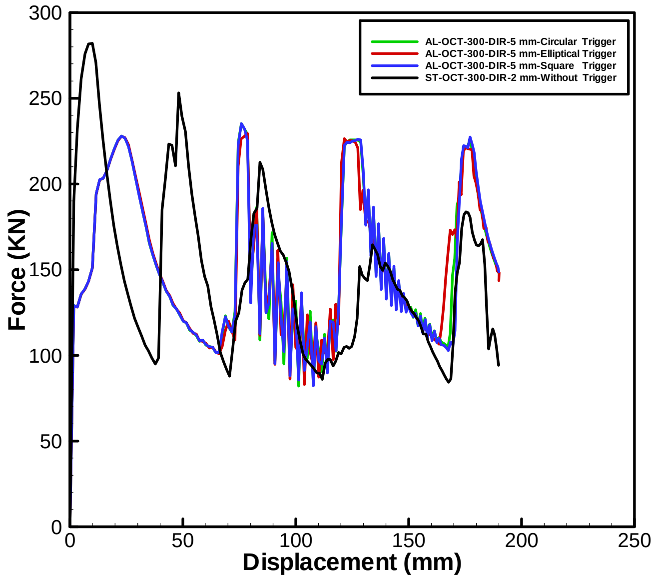


Fig. 12. Force VS displacement of OCT-300 with the various triggers in case of direct load

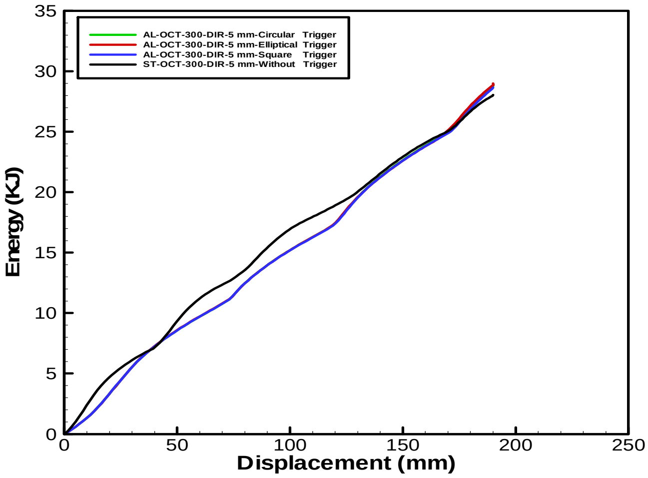
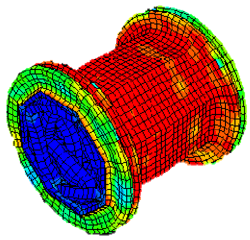


Fig. 13. Energy VS displacement of OCT-300 with the various triggers in case of direct load

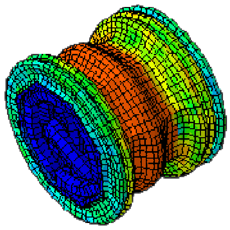
V. CONCLUSION

The present research studies the effects of the direct, and impact load on the octagonal, aluminium alloy profile with various wall thicknesses. In order to find the profile with the best CFE and energy absorption values, first the optimal wall thickness had to be chosen. This was followed by finding the optimal weight of the aluminium foam filling from as one option, and to examine the effects of the trigger mechanism on the energy absorption capacity and the CFE of the profile, as a second option. The dynamic simulation was based on axial and oblique load (30 degrees) of 16.7 m/s impact speed and on 275 kg compact mass, which is the 25% of the total weight (1100 kg) of the passenger car. Various factors, like manufacturing practicality, cost, crash performance indicators and energy absorption values have been taken in consideration in order to choose the optimal profile. The best results have been given by the 5 mm thick octagonal profile with hollow foam filling of F5 (0.15 kg) type. The profile subjected to direct load resulted 0.66 of CFE, 30 KJ of energy absorption, and 237 KN of peak force values, while when subjected to oblique load, the resulted values were 0.75, 22 KJ, and 157 KN respectively. Among the triggered profiles, the best result were shown by the 5 mm thick octagonal profile with ellipse shape trigger of 80 mm trigger position and 10 % reduction. In case of direct load the energy absorption was increased by 7.1% (29 KJ), the CFE by 29.4%, (0.66), while the peak force decreased by 15.9% (229 KN). In case of oblique load, the values were 3.8%, increase of energy absorption, and 4.3% of CFE, while 2% decrease of peak force. Based on the dynamic simulations, the optimal results have been given by the 5 mm thick octagonal profile enforced either by the filling of hollow aluminium foam type F5 (0.15 kg), or by ellipse shape triggers on the longitudinal side of the profile. Both types of octagonal profile can be considered potential to be used in crashworthiness applications.

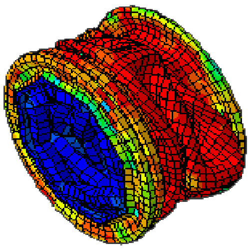
Published By:
Blue Eyes Intelligence Engineering & Sciences Publication Pvt. Ltd.



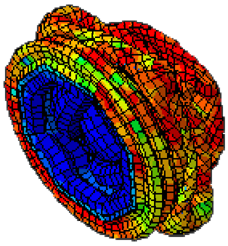
Type (F5) T= 5 mm



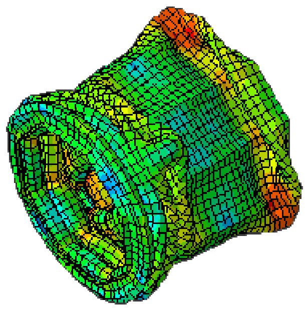
Type (F5) T= 4 mm



Type (F5) T= 3

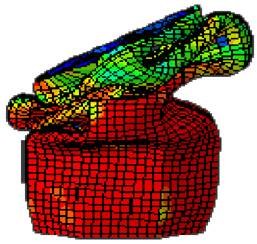


Type (F5) T= 2 mm

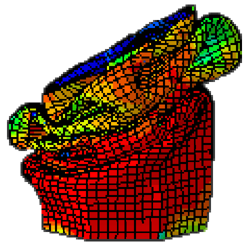


Empty steel tube, T= 2 mm

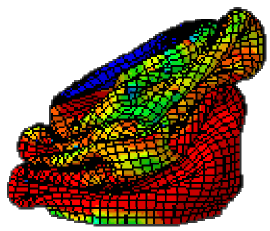
Fig. 16. Crashing deformation of the longitudinal members, using different hollow aluminium foams, in case of steel material, and under direct load



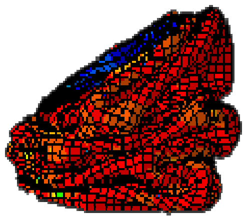
Type (F5) T= 5 mm



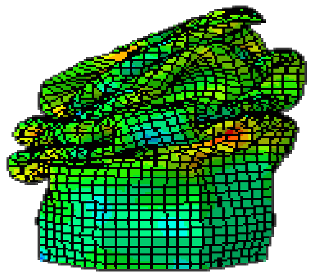
Type (F5) T= 4 mm



Type (F5) T= 3



Type (F5) T= 2 mm



Empty steel tube, T= 2 mm

Fig.15. Crashing deformation of the longitudinal members, using different hollow aluminium foams, in case of steel material, and under oblique load

Published By:
Blue Eyes Intelligence Engineering
& Sciences Publication Pvt. Ltd.

Table 7: Show the influence of using various aluminum foam weight and various tube thickness for OCT-300 subjected to axial impact loading at length deformation of 190 mm

Foam weight (Kg/mm2)		F1 (0.87 Kg)			F2 (0.62 Kg)			F3 (0.49 Kg)			F4 (0.25 Kg)			F5 (0.15 Kg)		
Thickness	Criteria	P max (KN)	CFE	Energy (KJ)	P max (KN)	CFE	Energy (KJ)	P max (KN)	CFE	Energy (KJ)	P max (KN)	CFE	Energy (KJ)	P max (KN)	CFE	Energy (KJ)
2 mm		217	0.6	22.9	159	0.59	17.3	124	0.59	13.7	82	0.564	8.8	77.3	0.56	8.3
3 mm		X	X	X	192	0.67	23.7	162	0.68	21	137	0.603	15.9	132	0.60	15.3
4 mm		X	X	X	X	X	X	206	0.74	28.7	193	0.61	22.7	182	0.61	21.6
5 mm		X	X	X	X	X	X	X	X	X	X	X	X	237	0.66	30
Empty Steel Tube thickness = 2 mm, Weight = 1.638 Kg														282	0.51	28

Note: X represents the design (tube + foam) which is above the intended weight.

Table 8: Show the influence of using various aluminum foam weight and various tube thickness for OCT-300 subjected to oblique impact loading at length deformation of 190 mm

Foam weight (Kg/mm2)		F1 (0.87 Kg)			F2 (0.62 Kg)			F3 (0.49 Kg)			F4 (0.25 Kg)			F5 (0.15 Kg)		
Thickness	Criteria	P max (KN)	CFE	Energy (KJ)	P max (KN)	CFE	Energy (KJ)	P max (KN)	CFE	Energy (KJ)	P max (KN)	CFE	Energy (KJ)	P max (KN)	CFE	Energy (KJ)
2 mm		124	0.72	16.2	89.6	0.71	11.9	75.4	0.73	10.4	55	0.659	7	55.7	0.62	6.6
3 mm		X	X	X	126	0.74	17.5	116	0.71	15.4	90.7	0.679	11.7	76.4	0.76	11.1
4 mm		X	X	X	X	X	X	145	0.76	20.8	130	0.677	16.8	126	0.66	15.8
5 mm		X	X	X	X	X	X	X	X	X	X	X	X	157	0.74	22
Empty Steel Tube thickness = 2 mm, Weight = 1.638 Kg														160	0.71	21.2

Note: X represents the design (tube + foam) which is above the intended weight.

VI. ACKNOWLEDGMENT

The authors would like to thank Universiti Tenaga Nasional (UNITEN) for their generous support of the research projects that forms the basis for the current research.

REFERENCES

[1] Ahmad, Z. (2009). Impact and energy absorption of empty and foam-filled conical tubes. Queensland University Of Technology Australia, December 2009.

[2] Z. Fan, G. Lu And K. Liu, 2011, Quasi-Static Axial Compression Of Thin-Walled Tubes Different Cross-Sectional Shapes.Engineering Structures.

[3] Alavi Nia, A., & Parsapour, M. (2014). Comparative analysis of energy absorption capacity of simple and multi-cell thin-walled tubes with triangular, square, hexagonal and octagonal sections. Thin-Walled Structures, 74, 155-165.

[4] Song, J., Chen, Y., & Lu, G. (2012). Axial crushing of thin-walled structures with origami patterns. Thin-Walled Structures, 54, 65-71.

[5] Mamalis, A. G., Manolakos, D. E., Baldoukas, A. K., & Viegelahn, G. L. (1991). Energy dissipation and associated failure modes when axially loading polygonal thin-walled cylinders. Thin-Walled Structures, 12(1), 17-34.

[6] Zhang, X., & Zhang, H. (2012). Experimental and numerical investigation on crush resistance of polygonal columns and angle elements. Thin-Walled Structures, 57, 25-36.

[7] Rossi, A., Fawaz, Z., & Behdinan, K. (2005). Numerical simulation of the axial collapse of thin-walled polygonal section tubes. Thin-walled structures, 43(10), 1646-1661.

[8] J. Marzbanrad, M. Ebrahimi-Fand M. Khosravi (2014) . Optimization of Crush Initiators on Steel Front Rail of Vehicle. International Journal of Automotive Engineering Vol. 4, Number 2.

[9] Tanlak, N., & Sonmez, F. O. (2014). Optimal shape design of thin-walled tubes under high-velocity axial impact loads. Thin-Walled Structures, 84, 302-312.

[10] Liu, Y., & Day, M. L. Simplified Modeling of Thin-Walled Tubes with Octagonal Cross Section–Axial Crushing. Proceedings of the World Congress on Engineering and Computer Science 2007 WCECS 2007, October 24-26, 2007, San Francisco, USA.

[11] Hosseini-Tehrani, P., Pirmohammad, S., & Golmohammadi, M. (2008). Study on the collapse of tapered tubes subjected to oblique loads. Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering,222(11), 2025-2039.

[12] Huw C Daves, Francois Godilon And Mervyn J Edwards, 2004, Assessment Of Car Compatibility Performance And The Development Of Improved Compatibility, Trl Limited, Crowthome.

[14] Shetty, S. K. (2006). Finite element study of energy absorption

characteristics of a hybrid structure-composite wrapped on a square metal tube (doctoral dissertation, wichita state university).

[14] Abdewi, E. F., Sulaiman, S., Hamouda, A. M. S., & Mahdi, E. (2008). Quasi-static axial and lateral crushing of radial corrugated composite tubes. Thin-Walled Structures, 46(3), 320-332.

[15] Guillow, S. R., Lu, G., & Grzebieta, R. H. (2001). Quasi-static axial compression of thin-walled circular aluminium tubes. International Journal of Mechanical Sciences, 43(9), 2103-2123.

[16] Sameer, J. O., Zaroog, O. S., Samer, F., & Abdullah, A. (2-014). Dynamic simulation of aluminum rectangular tubes under direct and oblique impact load: application to vehicle crashworthiness design. International Journal of Research in Engineering and Technology, 03 (11), 1-11.

[17] Nagel, G. (2005). Impact and energy absorption of straight and tapered rectangular tubes (Doctoral dissertation, Queensland University of Technology).

[18] Witteman, W. J. (1999). Improved vehicle crashworthiness design by control of the energy absorption for different collision situations: proefschrift. Technische Universiteit Eindhoven.

[19] Ahmad, Z., & Thambiratnam, D. P. (2009). Dynamic computer simulation and energy absorption of foam-filled conical tubes under axial impact loading.Computers & Structures, 87(3), 186-197.

[20] Duan, C. Z., Dou, T., Cai, Y. J., & Li, Y. Y. (2011). Finite element simulation and experiment of chip formation process during high speed machining of AISI 1045 hardened steel. AMAE International Journal on Production and Industrial Engineering, 2(1).

[21] Deshpande, V. S., & Fleck, N. A. (2000). Isotropic constitutive models for metallic foams. Journal of the Mechanics and Physics of Solids, 48(6), 1253-1283.

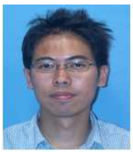
[22] Tarlochan, F., Samer, F., Hamouda, A. M. S., Ramesh, S., & Khalid, K. (2013). Design of thin wall structures for energy absorption applications: Enhancement of crashworthiness due to axial and oblique impact forces. Thin-Walled Structures, 71, 7-17.

[23] Sameer, J. O., Zaroog, O. S., Samer, F., & Abdullah, A. (2014). “A

Numerical Comparison between Aluminium Alloy and Mild Steel in Order to Enhance the Energy Absorption Capacity of the Thin Walled Tubes". International Journal of Advanced Engineering and Nano Technology. 2 (1), 1-12.

[24] Samer, F., Sameer, J. O., & Abdullah, A. (2014) "Design of Longitudinal Members To Vehicle: Enhances The Energy Absorption of Thin Walled Structures Under Dynamic Load". International Journal of Engineering and Advanced Technology . 4 (2), 21-33.

Abdulbasit Abdullah, was born in Iraq. He obtained his bachelors in General of Mechanical Engineering from Iraq. He is currently pursuing his Master of Mechanical Engineering in the field of applied Mechanical Engineering at Universiti Tenaga Nasional (UNITEN), Kajang, Selangor, Malaysia.



Khairul Salleh Mohamed Sahari, Assoc. Prof. Dr. He obtained his B. Eng (Mech.), Waseda University, 2000. He obtained his M.Eng, Kanazawa University, 2002 and PhD from Kanazawa University, 2006. He is Head of the Department of Mechanical Engineering, College of Engineering, Universiti Tenaga Nasional (UNITEN), Kajang, Selangor, Malaysia.

Samer F, Was born in Iraq. He obtained his Bachelors in Mechanical Engineering and Master in Mechanical Engineering from Iraq. He obtained his PHD in Mechanical Engineering from UNITEN, Malaysia. He is a lecturer at College of Engineering, Alanbar University, Iraq.

Jamal O. Sameer, was born in Iraq. He obtained his Bachelors in Mechanical Engineering from Iraq. He is currently pursuing his Master of Mechanical Engineering in the field of applied Mechanical Engineering at Universiti Tenaga Nasional (UNITEN), Kajang, Selangor, Malaysia.

Published By:
Blue Eyes Intelligence Engineering
& Sciences Publication Pvt. Ltd.