Enhancement of Energy Absorption for Crashworthiness Application: Octagonal-Shape Longitudinal Members

Samer F, Abdulbasit Abdullah, 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 mild steel A36 tube of 2 mm wall thickness, filled with hollow aluminium foam. The filled steel tube has given better results by enhancing the energy absorption by 15% and 36.8 in case of direct and oblique impact respectively .While The better result has given by enhancing CFE by 30%and 9.9% in case of direct and oblique impact respectively.

Index Terms— direct and oblique impact load, thin wall, energy absorption, CFE, Trigger and 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. Sections of various shapes have been examined, and subjected to quasi-static loading, like octagonal, hexagonal, square, and triangular.

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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.

II. DESIGNMETHODOLOGY

The study examines the behavior of the cross sectional, thin wall, octagonal mild steel profile. The profile is long 350mm,

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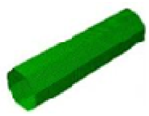
thick, 1, 1.3, 1.5, 1.7and 2 mm, with perimeter of 300 mm. As a first step, we survey the crashworthiness properties of the cross sectional profiles, and this is followed by the research of their improvement possibilities and the choice of the optimal design. The octagonal profile of various weights is filled with hollow aluminium foam of 540kg/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 60km/hr. Table 1 illustrates the various profiles.

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

Profile	Name of sample	Perimeter (mm)	Length (mm)	Thickness (mm)	Dimension (mm)	Shape
Octagonal	OCT -300	300	350	1 1.3 1.5 1.7 2	37.5x8	

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 causes 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 dδ 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.

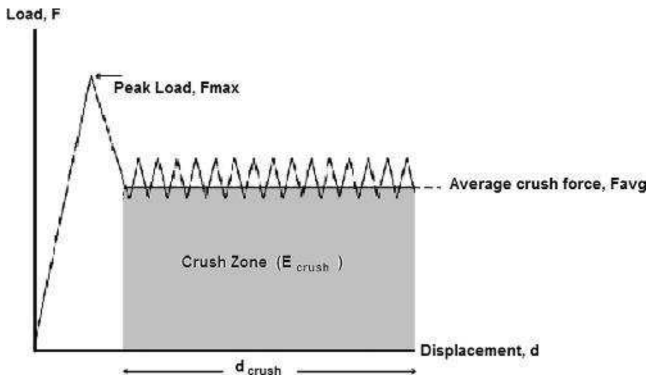


Fig. 1. Force displacement characteristics [22]

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], Figure2by [15] illustrates the load-deflection curve of a thin-walled member subjected to direct quasi-static loading.

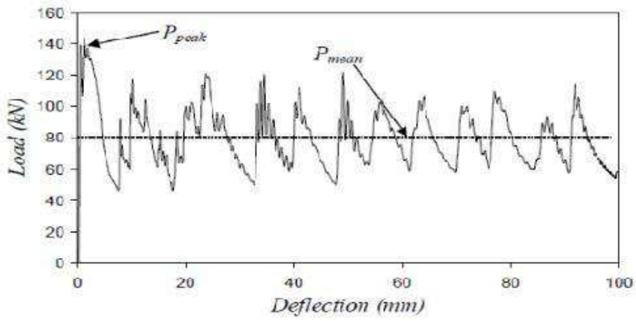


Fig. 2. Load-deflection curve foe axial crush. [15]

III. DYNAMICANALYSIS

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 the vehicle, when subjected to oblique and direct dynamic load. The advantage of ABACUS 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.

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A. Finite Element Modeling

The present study examines –with the help of the software ABACUS 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.

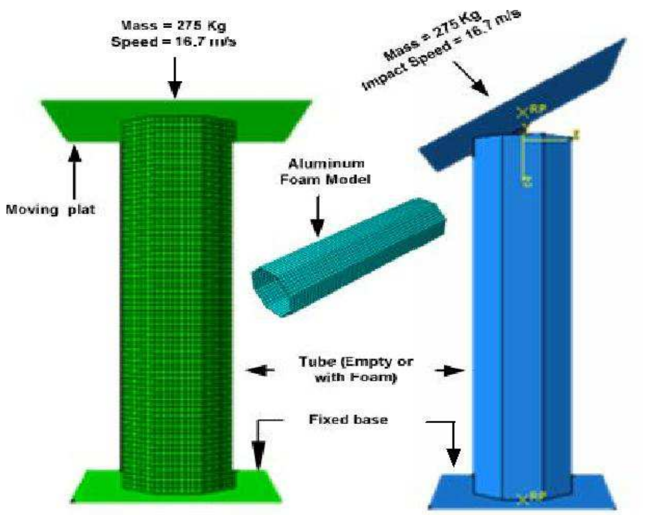

Fig. 3. Design of frontal longitudinal members

Table 2: Mechanical properties of mild steel material [17].

Parameter	Value	Description
A	146.7 MPa	Material
B	896.9 MPa	Material
N	0.32	Strain power
C	0.033	Material
M	0.323	Temperature
ε^o	1.0 s−1	Reference
ρ	7850 kg/m3	Density
Tm	1773 K	Melting
Cp	486	Specific heat

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

ρ_f	σ_P(MPa	α	α_2	β	γ	E_D
540	12.56	2.1	1544	3.6	1	1.620

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, 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].

IV. RESULT ANDDISCUSSION

Tables 4 and 5 show the results of the investigation. The detailed description follows in the next subsections.

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

Indicators	Axial Load OCT- 300		
	P max (KN)	CFE	Energy (KN)
1 mm	126	0.41	9.8
1.3mm	168	0.436	14.1
1.5 mm	198	0.474	18.1
1.7mm	231	0.469	21.2
2 mm	282	0.51	28

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)
1mm	72.2	0.538	7.3
1.3 mm	98.2	0.651	12
1.5mm	114	0.619	13.2
1.7mm	134	0.648	16.2
2 mm	160	0.71	21.2

A. Force displacement feature of different thickness profile.

Figures 4and 5 show the force displacement diagrams of the profile with 300 mm of the perimeter, and the reaction of the different geometric profiles by the direct and oblique load. Table 4 and 5 illustrate one type of perimeter 300 mm and five different thicknesses (1, 1.3,1.5,1.7 and 2 mm respectively). Based on the results, the quantity of the absorbed energy is significantly higher in case if direct load. This is caused by the fact that the oblique load has the force of the axial compression and also of the bending mode, resulted by the progressive crush. The result of the force-displacement demonstrates that the various parameters don't have effect on the folding process during the crush of the octagonal tube, subjected to oblique and direct load. Both actions have the same kind of results during the progressive collapse.

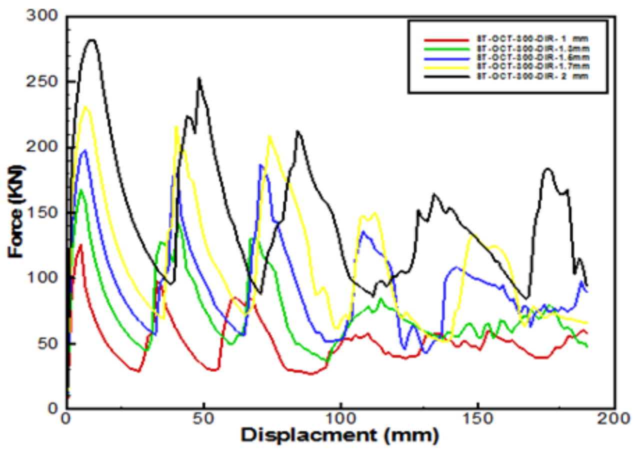

Fig. 4. Force VS displacement for OCT-300 in case direct load

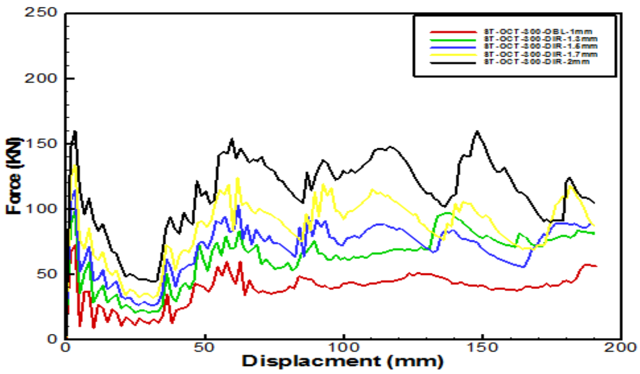

Fig. 5. Force VS displacement for OCT-300 in case oblique load

B. Energy Absorption

Figures 6 and 7 show the energy absorption capacity of the octagonal profile with 190 mm of deformation length, and with different wall thickness, subjected to various impact loads, and without concentrating on the time factor. As shown by the figures, in every impact condition, by increasing the wall thickness proportionally increases the energy absorption capacity of the tube. Tables 4 and 5 show the energy absorption capacity of profiles with various thicknesses and perimeter 300 mm, in case of the direct and oblique load of 30 degrees. Based on the results, the tubes subjected to oblique load, had reduced energy absorption with a difference of 15 – 55 %. The optimal thickness of the tube need to be chosen based on the CFE, the energy absorption capabilities, fabrication process, and weight.

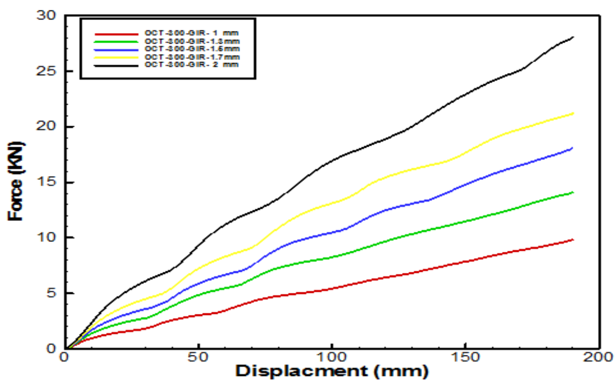

Fig. 6. Energy VS displacement of OCT-300 in case of direct load

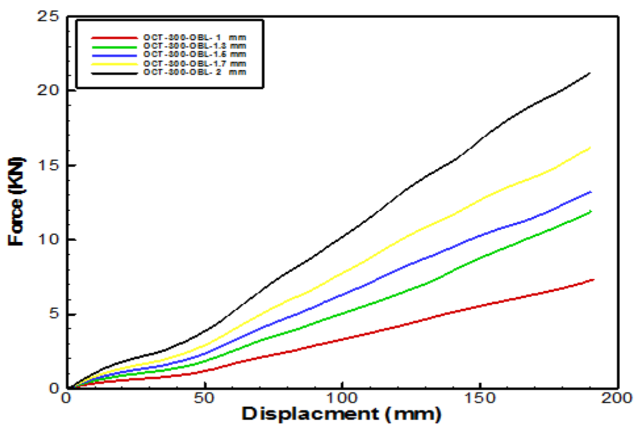

Fig. 7. Energy VS displacement of OCT-300 in case of oblique load

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C. Choise of the optimal profile

In this study the multi criteria decision making (MCDM) procedure is based on the complex proportional assessment method (COPRAS), which has the positive side of being convenient to handle.

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

The octagonal profile of 300 mm perimeter has been chosen for further examination regarding the wall thickness of the tube (1 mm, 1.3 mm, 1.5 mm,1.7 and 1.8 mm), and the weight of the hollow aluminium foam filling F1(0.62 Kg), F2 (0.49 Kg),F3 (0.25Kg), andF4 (0.15 Kg) . Figures8,9,10 and 11, illustrates how the use of weight at F2 (0.49 kg) of hollow aluminium foam, and different wall thickness increases the CFE and energy absorption. Thinner tube has been chosen in order to keep the final design as low as possible, while increasing the absorber capability and the CFE. As illustrated in Tables 7-8, in case of 1.8 mm wall thickness and 190 mm deformation length with hollow aluminium foam type (F2) . In case compare between steel has thickness 2 mm without foam and octagonal profile has thickness 1.8 mm with foam weight type (F2=0.49 kg) the an enhances in energy absorber and CFE from 28 KJ to 32.2 KJ and 0.51 to 0.663 respectively in case of direct impact load, while the oblique loading give the results by an increase of energy from21.2 KJ to 29 KJand CFE from 0.71 to 0.78 respectively.

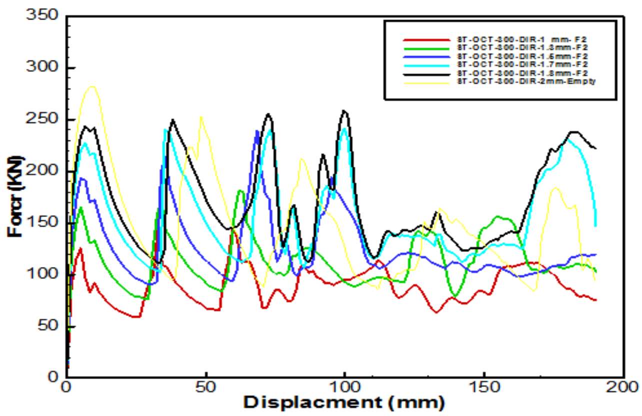

Fig. 8. Force VS displacement of OCT-300 with hollow aluminium foam F5, and different wall thickness of tube, in case of direct load

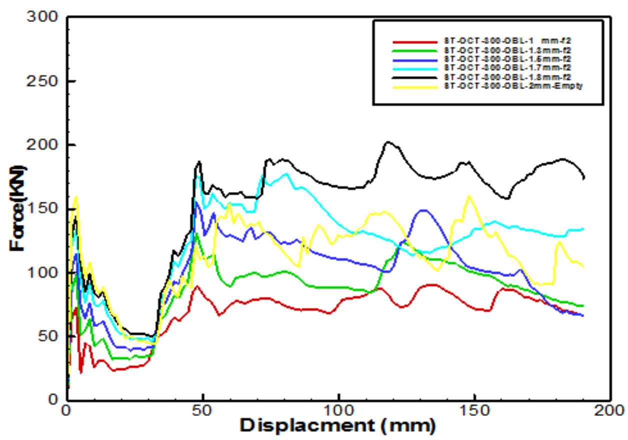

Fig. 9. Force VS displacement of OCT-300 with hollow aluminium foam F5, and different wall thickness of tube, in case of oblique load

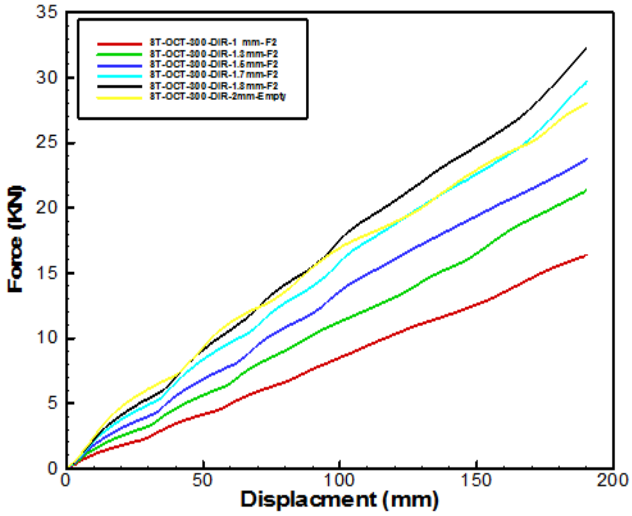

Fig. 10. Energy VS displacement of OCT-300 with hollow *aluminium foam F5, and different wall thickness of tube, in case of direct load*

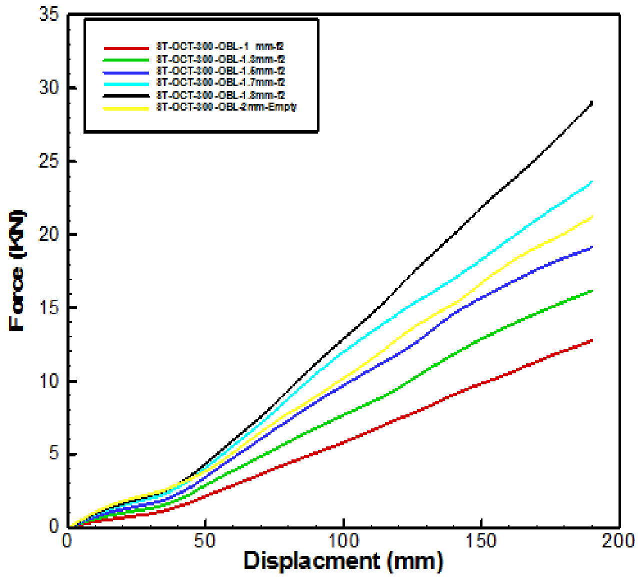

Fig. 11. Energy VS displacement of OCT-300 with hollow aluminium foam F5, and different wall thickness of tube, in case of oblique load.

E. Effect of trigger mechanism

This study examines the effects of circular, ellipse, and square shaped triggers. The best results were given by the trigger with ellipse shape, and with the first type of distribution with one hole in longitudinal side. [22], [23] and [24] show the best choice of the reductions is 10%, this reduction has carried on this study, the best result was given by the reduction of 10%, while from the positions of10,20,40, 60 , 80 , 100 and 120 mm distance from the tube end, the best result was given by the position of 40 mm. As shown in Tables 6, the trigger mechanism had the best results in the case of the octagonal profile with 2 mm wall thickness instated steel tube has thickness 2 mm without a trigger. The usage of the trigger mechanism increased the CFE by 5.7%and the energy absorption by 5 %. Figures 12, and 13, show the force and energy displacement function with the trigger mechanism.

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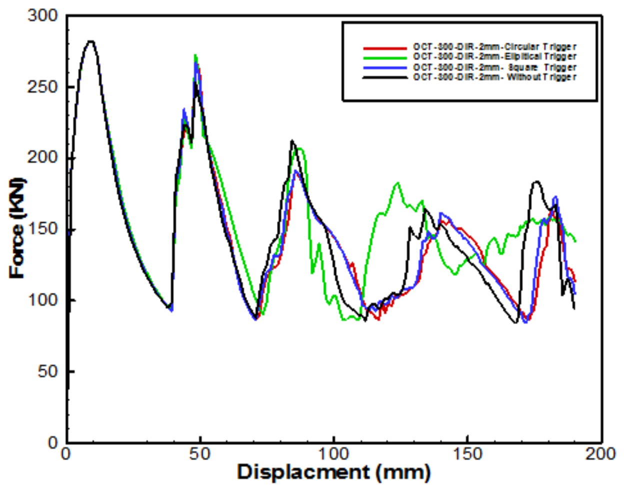

Fig. 12. Force VS displacement of OCT-300 with the various triggers in case of direct load

Table 6: Peak force, CFE and Energy absorption of profile OCT-300 with three different triggers under direct load at a position 40 mm from free end.

Criteria / Type of Trigger	Peak force (KN)	CFE	Energy (KJ)
Steel Tube Without trigger	282	0.51	28
The ST - Tube has Circular trigger with 1 hole	282	0.5	27.6
The ST - Tube has Circular trigger with 2 holes	282	0.504	28.6
The ST- Tube has Circular trigger with 4 holes.	282	0.523	27.4
The ST- Tube has Circular trigger with 8 holes.	282	0.496	28
The ST - Tube has Elliptical trigger with 1 hole	282	0.539	29.4
The ST-Tube has Elliptical trigger with 2 holes	232	0.529	29
The ST-Tube has Elliptical trigger with 4 holes	282	0.519	28.5
The ST-Tube has Elliptical trigger with 8 holes	281	0.494	27.1
The ST - Tube has Square trigger with 1 hole	282	0.504	27.7
The ST - Tube has Square trigger with 2 holes	282	0.506	27.9
The ST - Tube has Square trigger with 4 holes	282	0.529	29
The ST - Tube has Square trigger with 8 holes	282	0.493	27.9

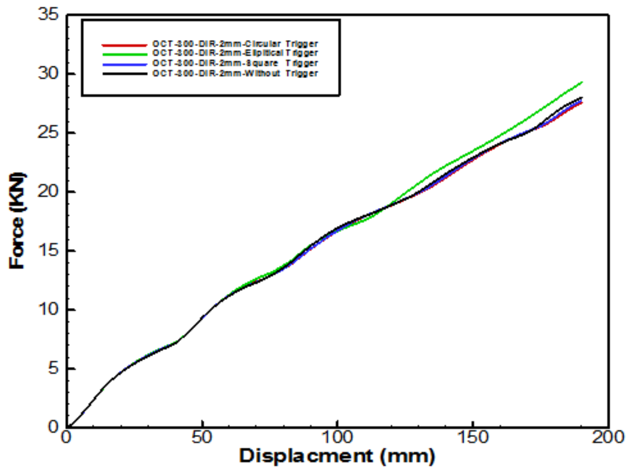

Fig. 13. Energy VS displacement of OCT-300 with the various triggers in case of direct load

V. CONCLUSION

The current study examined the oblique and the direct impact loads, and the effects of the impact on the octagonal tube of ductile material of mild steel. The purpose of the research was to choose the best from the different octagonal profiles with various wall thickness. The next step was to observe the behaviour of the chosen profile, filled with aluminium foam of various weights, and to discover the best filling option in case of direct and impact load. Another examined option of increasing the energy absorption, and the CFE was the usage if the trigger mechanism. The dynamic simulation was conducted with the compact mass of 25% of the total weight (1100 kg) of the passenger car, with impact speed of 16.7 m/s, and with both direct, and oblique load (30 degrees) on the octagonal profile. Based on the crash performance indicators, cost and manufacturing practicality, the optimal result of energy absorption capacity was given by the octagonal profile, thick1.8 mm, and filled with hollow foam of typeF2(0.49kg), giving the values of 32.2KJ for energy absorption, 0.663 for CFE in case of direct load, while 29 KJand 0.78 in case of oblique load.The simulation with direct load with an exam the trigger mechanism with 2 mm thick octagonal tube enhanced the energy absorption capacities to 29.4 KJ and the CFE 0.66. The best result is given by the trigger of the ellipse shape with a 10 % reduction and 40 mm trigger position. To recapitulate, in case of oblique load the increase of energy absorption, CFE was 36.8%, 9.9 %, respectively. While in case of direct load the values were 15 %, and 30 %, respectively. The best result has been given by the 1.8 mm thick octagonal profile filled with hollow aluminium foam type F2 (0.49 kg), and with the profile with an ellipse shape triggers on the longitudinal side of the tube. These profiles can be recognized as a potential energy absorber candidate for crashworthiness applications.

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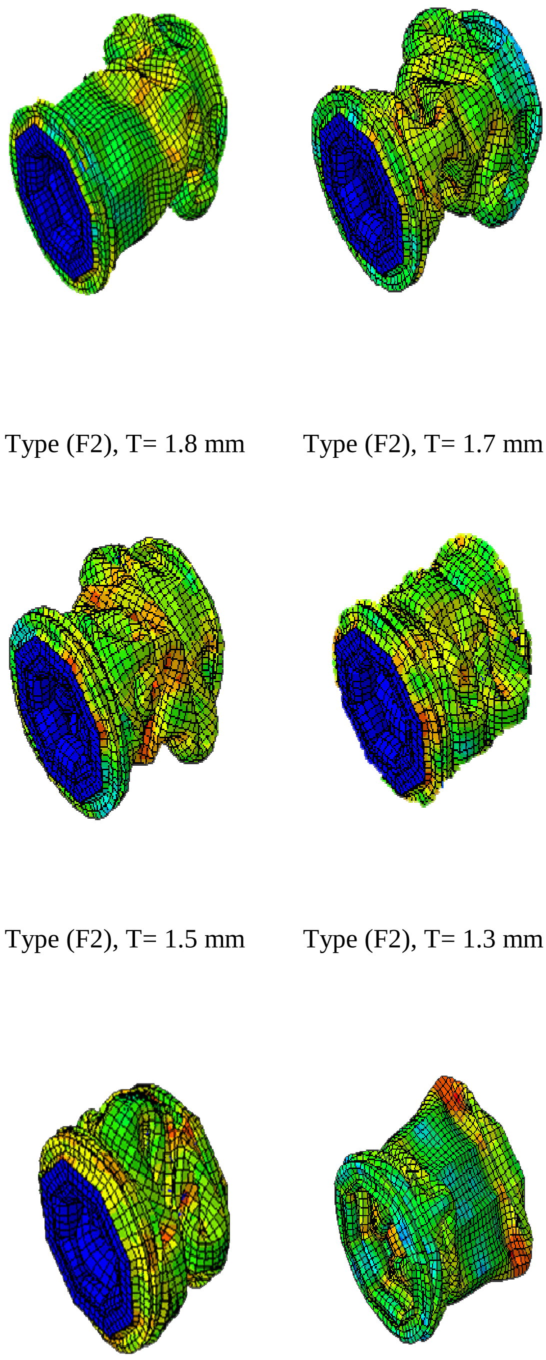
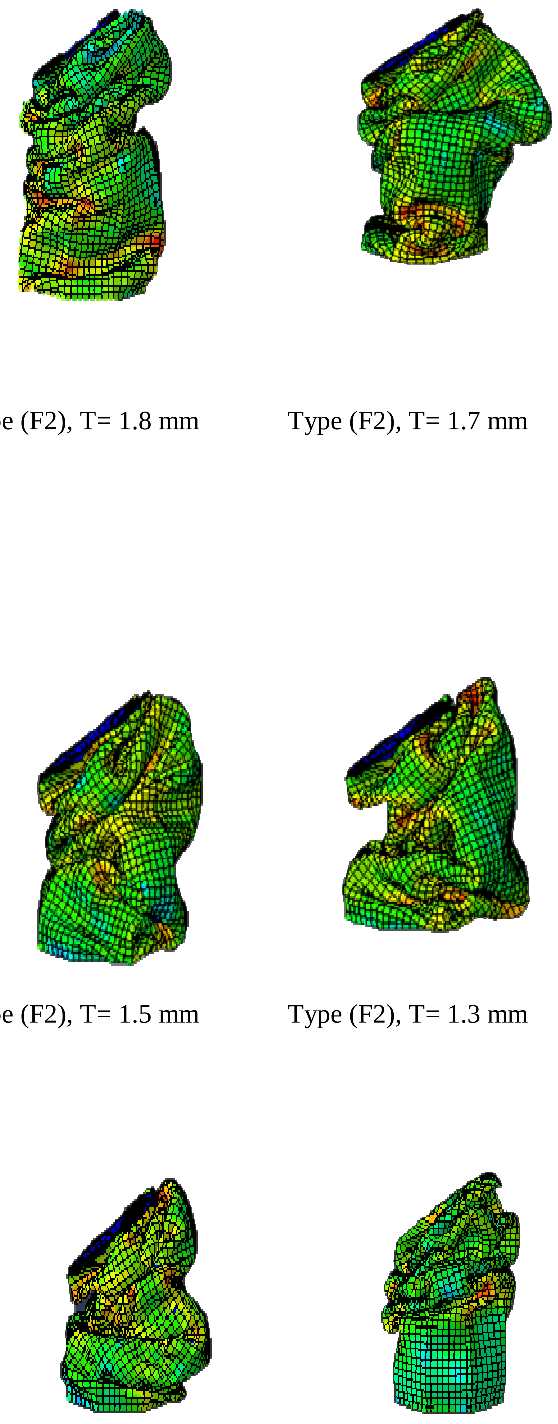


Type (F2), T= 1.8 mm Type (F2), T= 1.7 mm

Type (F2), T= 1.5 mm Type (F2), T= 1.3 mm

Type (F2), T= 1 mm Empty= 2 mm

Fig. 14 Crashing deformation of the longitudinal members, using different hollow aluminium foams, in case of steel material, and under direct load



Type (F2), T= 1.8 mm Type (F2), T= 1.7 mm

Type (F2), T= 1.5 mm Type (F2), T= 1.3 mm

Type (F2), T= 1 mm Empty = 2 mm

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

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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 (0.62 Kg)			F2 (0.49 Kg)			F3 (0.25Kg)			F4 (0.1 Kg)		
Thickness	Criteria	max (KN)	CFE	Energy (KJ)	max (KN)	CFE	Energy (KJ)	max (KN)	CFE	Energy (KJ)	max (KN)	CFE	Energy (KJ)
1 mm		175	0.675	20.7	147	0.661	16.4	126	0.503	11.9	119	0.454	10.3
1.3 mm		218	0.662	25.9	181	0.628	21.3	165	0.537	17	165	0.48	15.2
1.5 mm		229	0.659	28.1	239	0.518	23.7	193	0.542	20.1	193	0.516	19.3
1.7 mm		274	0.669	33.9	242	0.65	29.7	227.1	0.661	24.3	227	0.526	23.2
1.8 mm		283	0.668	35.5	259	0.663	32.2	244	0.533	26.1	243	0.533	25.2
Empty Steel Tube thickness = 2 mm, Weight = 1.638 Kg					282	0.51	28						

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 (0.62 Kg)			F2 (0.49 Kg)			F3 (0.25 Kg)			F4 (0.15 Kg)		
Thickness	Criteria	max (KN)	CFE	Energy (KJ)	max (KN)	CFE	Energy (KJ)	max (KN)	CFE	Energy (KJ)	max (KN)	CFE	Energy (KJ)
1 mm		101	0.736	14	90	0.764	12.7	72.1	0.668	9.1	70.3	0.631	8.4
1.3 mm		140	0.685	17.9	131	0.665	16.2	107.4	0.661	13.1	101	0.641	12.3
1.5 mm		168	0.661	20.8	156	0.649	19.9	127.4	0.692	16.6	122	0.654	15.1
1.7 mm		210	0.676	26.4	172	0.709	23.6	157	0.668	20.2	141	0.686	18.2
1.8 mm		266	0.688	28.7	203	0.78	29	166	0.714	22.2	157	0.681	20.2
Empty Steel Tube thickness = 2 mm, Weight = 1.638 Kg					160	0.71	21.2						

VI. ACKNOWLEDGMENT

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