DYNAMIC SIMULATION OF ALUMINUM RECTANGULAR TUBES UNDER DIRECT AND OBLIQUE IMPACT LOAD: APPLICATION TO VEHICLE CRASHWORTHINESS DESIGN

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Abstract

This paper describes the behavior of thin wall rectangular tube cross- sections subjected to dynamic compression loading. Computational investigation studies the reaction of the tube of different diameters and thicknesses, subjected to direct and oblique loading. The rectangular tube, with ellipse, circular and square shape triggers and others filled with different weight of hollow foam are the subject of the investigation. The purpose of the examination was to compare the tubes of various parameters, chosen with the multi criteria decision making (MCDM) procedure, in order to find the design based on the best multi criteria, and performance indicators. The studied performance parameters were the energy absorption capacity in case of direct and oblique loading, the peak force, and the crush force efficiency. The material chosen for the current study was the aluminum alloy AA6060, due to its lighter weight, with the purpose of examining its ability to be used in energy absorption application. The enhancement of energy absorption by 76.4%, improve the CFE by 44% in direct load, while in case of oblique load we have an enhancement of the energy absorption by 73.4%, and an improvement of the CFE by 22.5%with hollow aluminum foam. Choosing thinner tube and using less aluminum foam is to keep the weight of the final design as low as possible, and at the same time to enhance its energy absorber capacities and the CFE.

Keywords: *dynamic compression, thin wall, energy absorption, direct and oblique loading, crush force efficiency*

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1. INTRODUCTION

With the fast-growing vehicle number all around the world, the number of road accidents increases proportionally [1]. One of the major objectives of the automobile industry is to develop safer car structures, and ensure a higher protection level of the passengers in case of collision. This can be obtained by increasing the capability of the structure members of the car to absorb impact energy and correspondingly by decreasing the impact energy arriving to the passengers. This structure is called frontal longitudinal, and is shown in Figure 1.

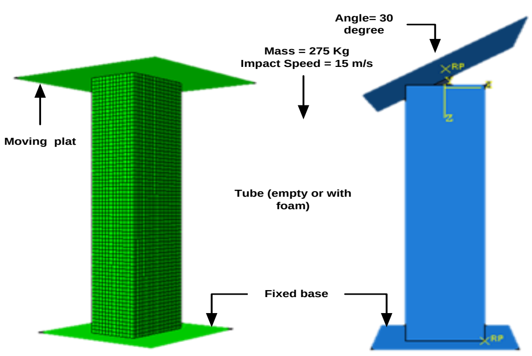


Fig -1: Design of frontal longitudinal members

In case of collision, the energy absorber members of the car are exposed to oblique and direct impact force. While the effects of the direct impact load on the rectangular thin walled tube and the enhancement of its energy absorption capacities were object of numerous researches [2], the effects of the oblique load on the same tube are still to be further investigated [4]. Generally known studies on the oblique impact belong to [5], [6-8], and [9]. A significant feature of the oblique loading that has to be taken into consideration is that the longitudinal, the frontal part of the vehicle, due to the bending mode caused by the collapse, has lower energy absorption capabilities in this case than in the case of the direct load [10]. The usage of metallic or polymeric foams in order to increase the energy absorption has been considered by several researchers [11- 17]. This helps to obtain an increase in the energy absorption capacity level how also a reduction of the energy absorption stroke, and is considered efficient, particularly in the design of compact cars. Recently the finite element analysis (FEA) obtained by computer simulation has become more popular compared to experimental work. In the impact studies the use of FEA gives a wider knowledge and understanding of the behavior and the deformation of energy absorber tubes exposed to impact loading [18, 19], and at the same time it diminishes the research time compared to the experimental work. The finite element analysis will be the method used also in the current study in designing the optimal energy absorbent by using the rectangular thin wall tube.

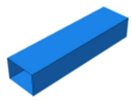
2- DESIGN METHODOLOGY

The cross sectional profile of the designed thin wall structure tube was rectangular. It was modeled using aluminum alloy (AA6060) as a material, was long, 350mm, thick 2mm and 3.4mm, and had three different perimeters of 200 mm, 300 mm and 400 mm.

The purpose of the research was to investigate the crash performance of the different cross sectional profiles, followed by the enhancement of the crashworthiness capacities of the selected best design.

The aluminum foam –used in the current study as foam filler material- had a density of 540kg/m3. The rectangular tube, filled by hollow aluminum foam of different weight, has been simulated by both direct and oblique (30 degrees) impact load. In the simulation process the impact mass had 25% of the total weight of the vehicle, and had the initial velocity of 54km/hr. Table 1 shows the details of profiles. The next section will include a detailed discussion of the simulation.

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

Profile	Specimen ID	Perimeter mm	Length mm	Major Dimension mm	Thickness	Shape
Rectangular	R-200 R-300 R-400	300	350	60×40 90×60 112×74	2	

2.1 Force Max and Peak Load

The value of the highest force of an element stands for the maximum impact that the members of the vehicle are able to absorb. The peak load causes substantial permanent distortion, but without endangering the safety of the passengers of the vehicle. While the peak load will cause deformation, is necessary for the vehicle to be able to face low-speed and low-energy impacts without permanent distortion of the structure [33].

2.2 Energy Absorption

The load-displacement curve shows how the energy absorption performance can be calculated, how also the design of the energy absorption (EA) as in Figure 2.

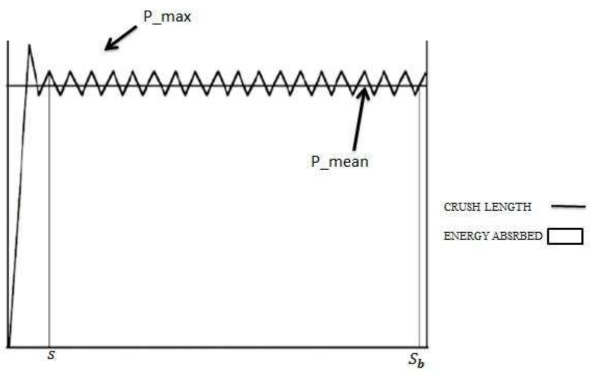


Fig -2: Force displacement characteristics [18]

$$EA= \int_0^{\delta b} p.\, d\delta \qquad (1)$$

With P being the direct crushing load, $d\delta$ is the length of crushing the sample, like in the equation (1).

$$EA= \int_0^{\delta b} p.\, d\delta = P_m\,(\delta b - \delta i) \qquad (2)$$

Where P_m is the main crushing load, δi is the original length of the crushing specimen. The perfect energy absorption would reach a supreme force and preserve it persists for the whole length of the distortion.

2.3 Crush Force Efficiency, CFE

The crush force efficiency (CFE) can be defined as the main crushing force (P mean) divided by the peak crushing load (P peak) as below.

$$CFE = \frac{P_{mean}}{P_{peak}} \qquad (3)$$

The CFE is one of the main factors to estimate the performance of energy absorbing structures [21]. The value of crash force efficiency can be calculated by dividing the average value of force with the load-displacement curve on the peak force at the moment of the crush [23]. The low value of crush force efficiency means high peak force, and this leads to decreased passenger safety which needs to be evaded. The CFE indicates how effective is the vehicle structure, in case of a crush [24]. The value of CFE will increase with the proportionally diminishing value of the peak load. This decrease can be attained by the usage of the trigger mechanism, which will proportionally increase the CFE value [25]. We can achieve a higher level of crush force efficiency also by increasing the thickness of the tube wall.

3. DYNAMIC ANALYSIS

The modern software of finite element codes ABAQUS/Explicit version 6.10 has been used in this current study to model the behavior of the energy absorber members of the frontal part of the vehicle exposed to direct, and oblique dynamic load.

The software program is appropriate for the simulation of different types of procedures of computational fluid dynamics (CFD) or standard/ electrical model. Explicit simulation methods have lower cost, and time investment, compared with the implicit methods. ABAQUS, is appropriate to simulate the same circumstances on a lower cost than an implicit method would require, such as high speed dynamic, and impact load [4]. This makes it suitable to examine the effects of the direct and oblique loads how also the mechanism of absorber energy.

3.1 Finite Element Modeling

In current research, the software ABACUS as non-linear FE designs the rectangular tube with and without filling and examines the reaction of the impact on them. The thin wall rectangular tube was designed with 4 node shell continuum (S4R) elements and with 5 integration spots alongside the thickness direction. The hollow aluminum foam was designed with 4-noded continuum elements and decreased integration performances combined with time control. The role of rigidity-based time control was to evade artificial small or neglected energy distortion modes, while the role of reduction integration was to avoid the volumetric locking. Based on the results of a mesh convergence study, the shells and foam elements had the size of 5 mm. A mesh convergence has the role to provide a sufficient mesh density with the purpose of capturing the deformation process. The simulation of contact interaction between the components was designed as ‘‘general contact algorithm” in order to evade the penetration of the wall of the rectangular tube. This study relies on the principle of the algorithm system to calculate the less intense with simulation time. The friction coefficient value between the contact surfaces (the rectangular tube, the moving plate and the fixed plate) is a constant value equal to 0.2 according to the algorithm system [25.26]. The impact body was modeled as a fixed plate (rigid body) with possible translation displacement. The compact car had mass 1100 kg, the mass of impact body was assumed to be 25% (275 kg) of the total mass of the compact car and the speed of the compactor body was 15 m/s (54 km/h). Both values were taken from the specification of the New Car Assessment Program (NCAP) by the National Highway Traffic Safety Administration (NHTSA). Every design of longitudinal crashworthiness in the car contained two longitudinal members in the frontal part of the car. The maximum ability that can be absorbed by two longitudinal members is much less than the 50% of the total mass of the compactor car [33]. The advantage of using ABAQUS, that contains extensive information from the elements, is that gives the possibility of facilitating the application process by the analysis of each design variables, and thus helps to solve all the problems facing the designer to set up a test with high efficiency models. The figure (3) shows the type of elements.

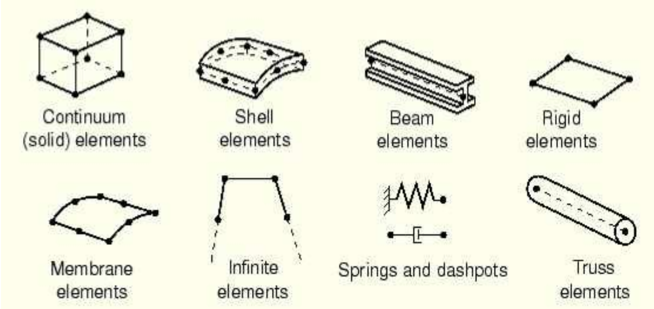


Fig -3: The generally used group of elements of the program (ABAQUS Version 6.6 documentation)

The thin-walled tube was modeled by the shell element, appropriate when the tube thickness is smaller than 10% of the length. S4R gave the base for the design of the absorber elements. S4R is a three-dimensional doubly curved four node shell component with. Every node has three-rotation degrees of freedom and three- displacements. R3D4 gave the base in designing the two rigid bodies. One of the rigid bodies was stabilized to have the liberty of movement only in the direction of the compactor load. The second rigid body was stabilized not allowing any movement to it.

3.2 Interaction, Boundary Conditions and Loading

In the experiment the rectangular profile from one end was entirely fixed to a rigid body (plate) by tied constraint, which allows only a linear movement along the displacement direction. The nodes had the liberty of rotational movement on the rectangular profile based on the edge of the rigid body. The rigid bodies as plates were designed as rigid contact surfaces in order to facilitate the contact simulation. One of the rigid bodies was fixed in a way to avoid any possible movement except of the axial direction of the compactor body. The applied mass was in one of the reference points in the center of one moving plate. The other reference point was placed in the edge of the tube and the fixed plate to record the reaction on the design. The dynamic load is simulated in the center of one moving plate with defined speed and mass compactor. The program defines the necessary step time with the appropriate dynamic load, explicit action, and time period. The time interval is dependent on the mesh dimension, and on the control and element structure. The extended time interval needs more time to visualize the outcome and require high CPU competency. The interaction obtainable in ABAQUS/Explicit has been completed in each part of the structure. A self-contact among the walls must be specified in case of the tube walls and the aluminum foam, while between every rigid plate and the tube, the parts of the telescope or any part of the telescope and a tube, a surface to surface contact is required. Once the contact surface is defined and the friction coefficient "penalty" is specified, the interaction option will be completed. The connection method between the rigid body and the end of the rectangular tube from both sides was constraint in tied to act as one body during the simulation. The rectangular tube had deformation length during the crush. The mesh size was specified at 5mm size [31], [32].

4. RESULTS AND DISCUSSION

Tables 2-3 show the results of examination conducted. Detailed description is provided in the following subsections.

Table 2: Result of crashworthiness parameters for rectangular tube for three different parameters (direct loading)

Indictors	Direct Load R- 200			Direct Load R- 300			Direct Load R- 400		
	P max	CFE	Energy	P max	CFE	Energy	P max	CFE	Energy
1.5 mm	27.8	0.57	3.2	41.7	0.433	3.64	55.9	0.35	4
2.0 mm	43.8	0.59	5.22	58.4	0.43	5.07	78	0.42	6.67
3.0 mm	84.4	0.61	10.4	105	0.478	10.1	133	0.43	11.5
3.4 mm	99.7	0.64	12.9	126	0.5	12.68	159	0.4	13
4.0 mm	123	0.69	13.5	164	0.527	17.57	198	0.44	17.8

Table 3: Result of crashworthiness parameters for rectangular tube for three different parameters (oblique loading)

Indictors	Oblique Load R- 200			Oblique Load R- 300			Oblique Load R- 400		
	P max	CFE	Energy	P max	CFE	Energy	P max	CFE	Energy
1.5 mm	24.3	0.56	2.7	33	0.38	2.6	40.5	0.31	2.54
2.0 mm	32.4	0.68	4.5	45.7	0.47	4.3	53.8	0.38	4.1
3.0 mm	60	0.7	8.5	67.7	0.65	8.82	82	0.52	8.7
3.4 mm	70	0.78	10.8	76.5	0.7	10.9	95	0.57	10.6
4.0 mm	88.6	o.81	14.1	99.6	0.73	14.7	115	0.63	14.5

4.1 Force Displacement Features of Different Perimeter and Thickness Profile

Figures 4, 5 and 6 present the force displacement diagrams of tested profiles. Every drawing shows the load answer of various rectangular geometric profiles subject to direct and oblique impact. The profiles consist of three classes of perimeters (200 mm, 300 mm and 400 mm) and five classes of thickness (1.5mm, 2 mm, 3 mm, 3.4 mm and 4 mm respectively). As it is shown in the figures, the absorbed energy in case of oblique loading is noticeably minor than in case of direct loading. This is because the oblique impact has two compounds of analyses. One of these forces has axial compression and the other one is in bending mode. The bending mode is a result of the progressive crush caused by the oblique impact. The result of the force-displacement shows that the different parameters don't have any role in the folding process during the crush of the rectangular tube in oblique and direct load. Both actions have quite similar results in the progressive collapse.

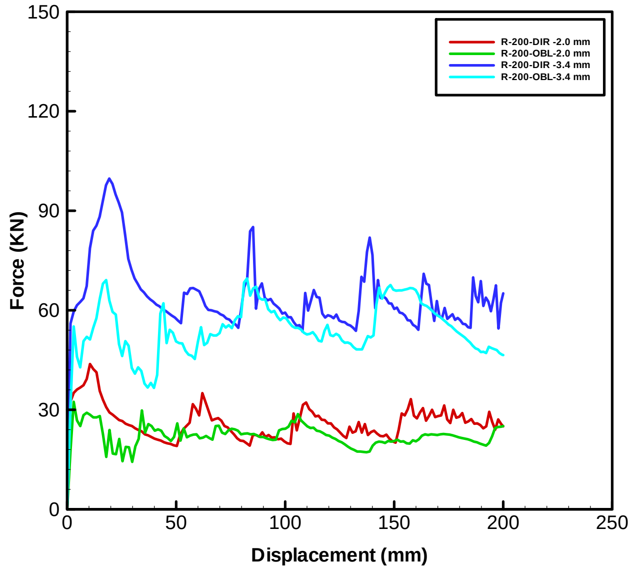


Fig -4 Force VS displacement for R-200 in direct and oblique load

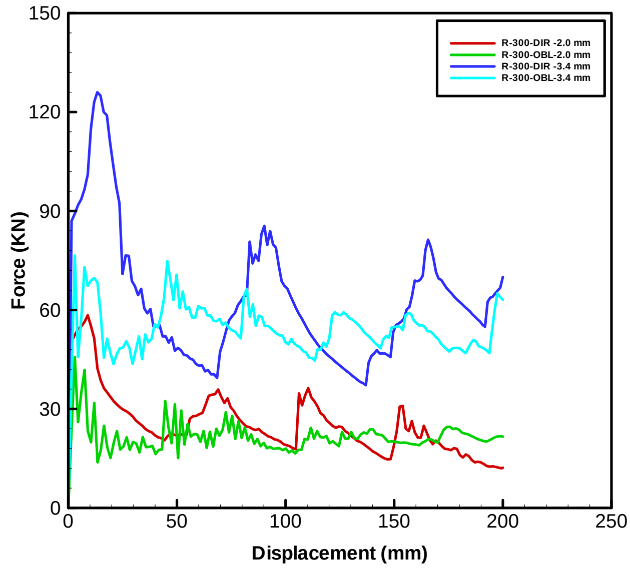


Fig -5 Force VS displacement for R-300 in direct and oblique load

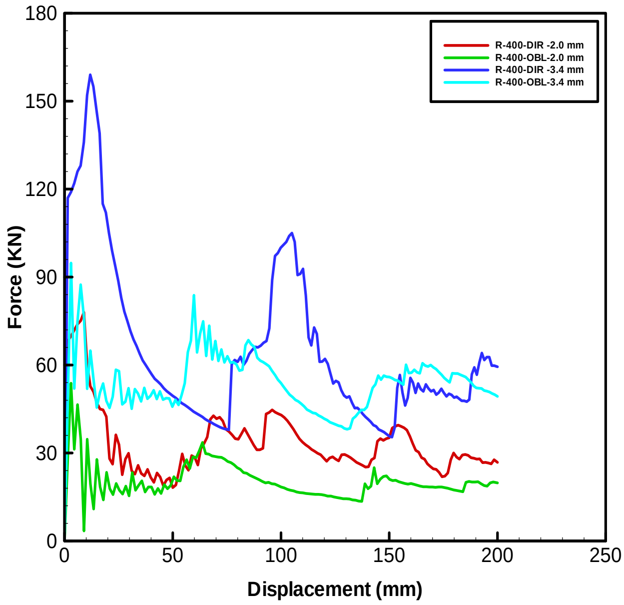


Fig -6 Force VS displacement for R-400 in direct and oblique load

4.2 Energy Absorption

Figures 7, 8 and 9 demonstrate the simulation of the energy absorption related to the deformation length of the rectangular tube without focusing on the time factor. Based on the test results, we can conclude that the energy absorption of the rectangular profile geometry R-200 is less than of any other profile in each of the impact conditions, while the best energy absorber is the rectangular profile with 300 mm in diameter. Figures 12 and 13 give the details of the energy absorption capacities of various rectangular profiles subject to different impact loads. It is clearly shown, that in case of an increased perimeter 200 mm, 300 mm to 400 mm, energy absorption capacity of the profiles were raised. The best specific energy absorption (SEA) capacity was presented by the rectangular profile of 300 mm perimeter. Tables 2 and 3 demonstrate the energy absorption in case of the oblique load of 30 degrees with various tube thicknesses and perimeters. Generally we can conclude that profiles exposed to oblique load showed reduced energy absorption, with an overall difference of an average 15 – 55 %. The optimal perimeter and thickness of the geometry profile has to be selected based on the CFE, energy absorption capacities, weight, and fabrication process

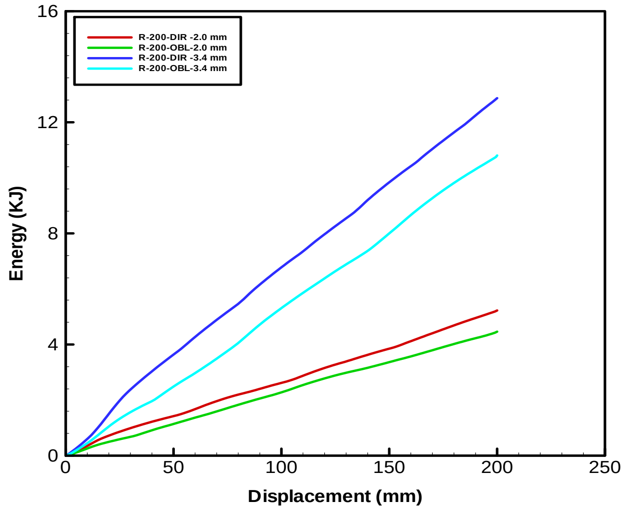


Fig -7 Energy VS displacement for R-200 in direct and oblique load

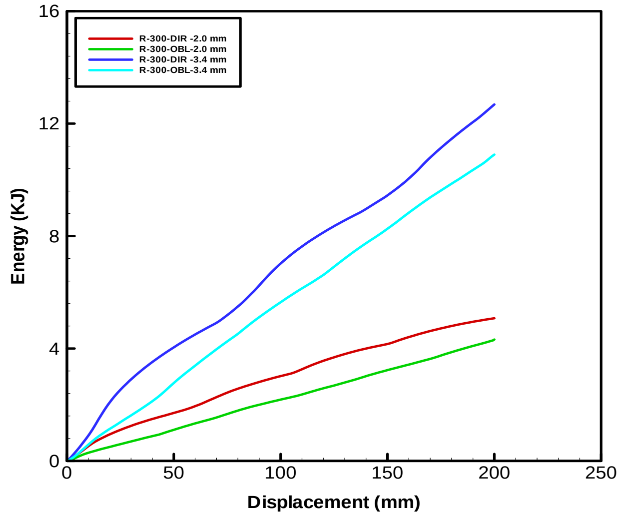


Fig -8 Energy VS displacement for R-300 in direct and oblique load

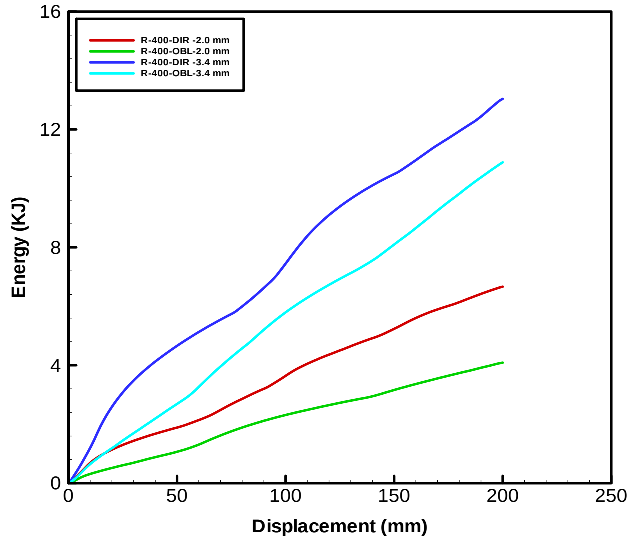


Fig -9 Energy VS displacement for R-400 in direct and oblique load

4.3 Selection of Best Profile

In this current study the multi criteria decision making (MCDM) procedure is based on the complex proportional assessment method (COPRAS). One of the advantages of this method is the easy handling of it. Among the non-filled tubes, the highest energy absorbing capacity has been shown by the 300 mm in diameter rectangular profile. The second-best result has been given by the 400 mm in diameter rectangular profile. Comparing filled and non-filled tubes, the best result was given by the rectangular profiles of 300 mm in diameter with hollow aluminum foam have shown the best results, will give the best value of energy absorption and crush force efficiency.

4.4 Effect of Hollow Foam on the Energy Absorption

The rectangular profile of 300 mm perimeter was selected to examine its characteristics based on the wall thickness of the profile and the weight of the hollow aluminum foam filling. 2 mm, 2.4mm, 2.8 mm, 3.2 mm and 3.4 mm respectively have been studied as the thickness of the rectangular profile. The hollow foam has different weight A (0.955kg), B (0.9074kg), C (0.841kg), D (0.756kg), and E (0.652kg). Figure 10, 11, 12, 13, 19 and 20 shows that the usage of foam E (0.652kg) and wall thickness 3.4 mm improves the CFE and energy absorption capacities. It raises the energy from 12.7 KJ to 22.4 KJ and 0.5 to 0.72 respectively in direct load for the same deformation length. Show the result in Table 6 and Table 7. The purpose of choosing thinner tube and using less aluminum foam is to keep the weight of the final design as low as possible, and in the same time to enhance its energy absorber capacities and the CFE. When used thinner tube with increases the aluminum foam the design in oblique load not enough good to improve the absorption energy, but in case axial loading it is gave a good result.

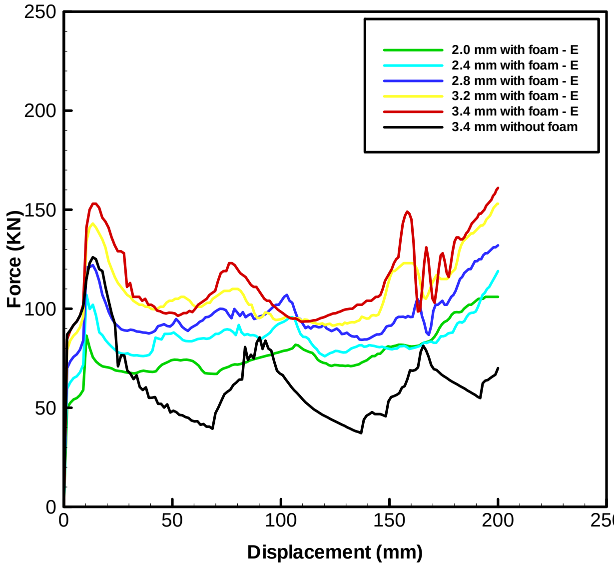


Fig -10 Force VS displacement for R-300 with hollow aluminum foam type E in direct load

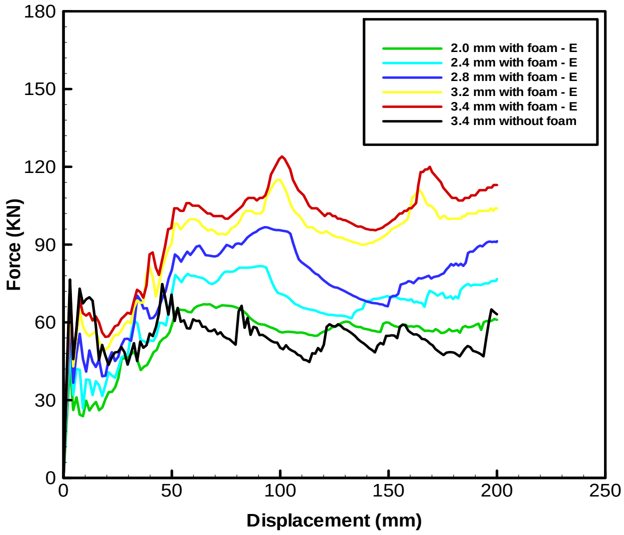


Fig -11 Force VS displacement for R-300 with hollow aluminum foam type E in oblique load

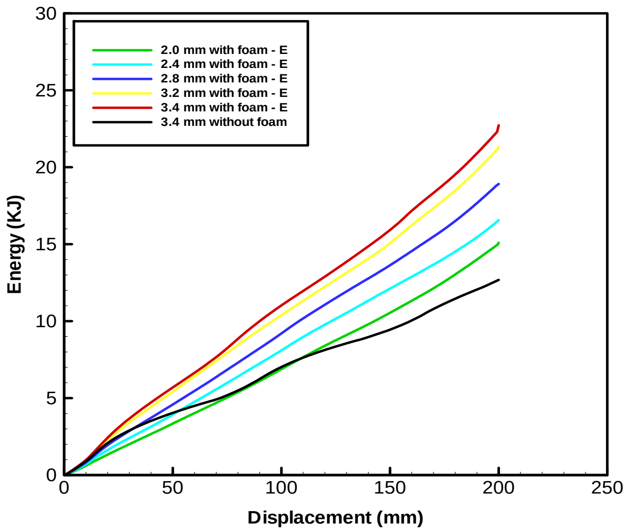


Fig -12 Energy VS displacement for R-300 with hollow aluminum foam type E in direct load

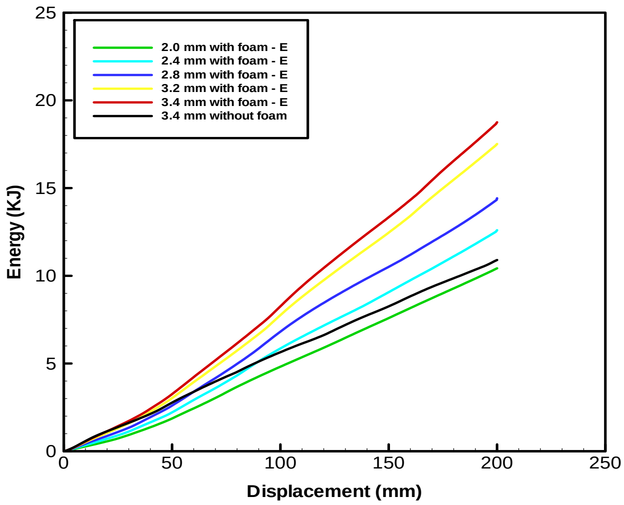


Fig -13 Energy VS displacement for R-300 with hollow aluminum foam type E in oblique load

4.5 Effect on Trigger Mechanism

The trigger shapes investigated in the current study were the ellipse, circular and square. The ellipse trigger at the third distribution with three holes in each long side has the best result. 10% reduction during the simulation gave the best result from the five options of 5, 7.5 10, 15 and 20%. 50 mm from the end of the tube of the trigger position with the ellipse geometric at the third distribution gave the best result from the different triggers and positions (10, 20, 30, 40, 50, 60 and 70 mm). The rectangular tube with 2 mm thickness gave the best result when used the trigger mechanism as the result in Table 4 and Table 5. With the use of the trigger mechanism the CFE is raised by 16.3% and the energy absorption by 14.6%. Figure 14, 15, 16, 17 and 18 shows the force and energy displacement function with the trigger mechanism.

Table 4: Energy absorption, peak force and CFE of profile R-300 with three different trigger under direct load

Profile-R-300 – Thickness 2 mm			
Criteria / **Type of Trigger**	**Peak force (KN)**	**CFE**	**Energy (KJ)**
Without trigger	58.4	0.43	5.07
Circular trigger first dist.	57.6	0.46	5.31
Circular trigger second dist.	58	0.45	5.22
Circular trigger third dist.	58.2	0.455	5.34
Elliptical trigger first dist.	57.7	0.45	5.24
Elliptical trigger second dist.	58	0.44	5.11
Elliptical trigger third dist.	58.2	0.5	5.81
Square trigger first dist.	57.3	0.455	5.25
Square trigger second dist.	57.9	0.46	5.36
Square trigger third dist.	58.1	0.464	5.5

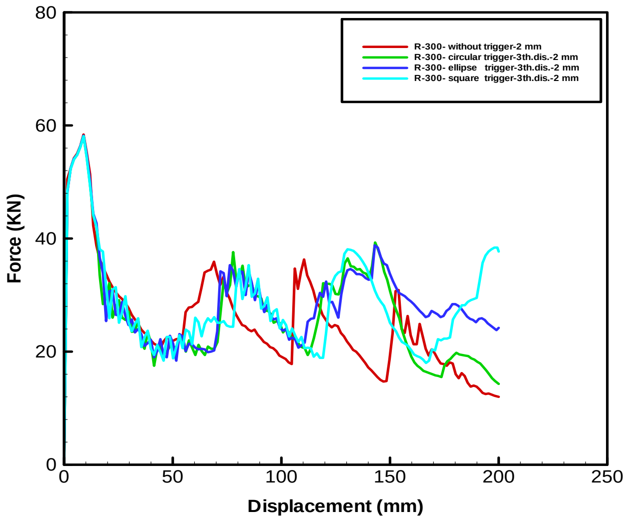


Fig -14 Force VS displacement for R-300 with the ellipse trigger in direct load

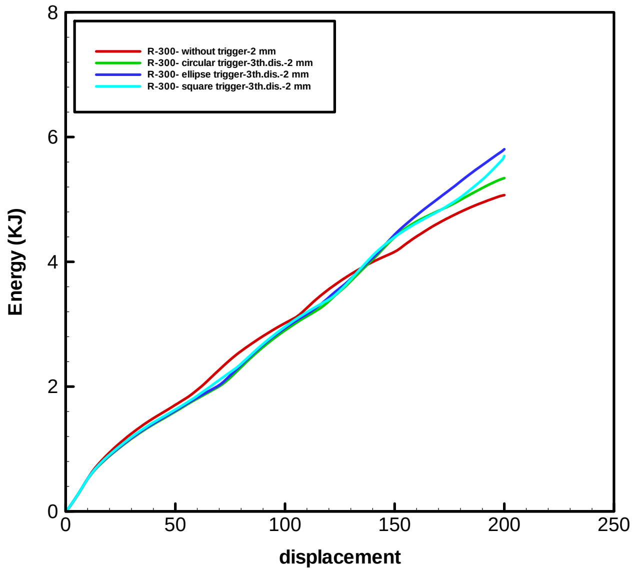


Fig -15 Energy VS displacement for R-300 with the ellipse trigger in direct load

Table 5: Energy absorption, peak force and CFE of profile R-300 with five different trigger reductions under direct load

Profile-R-300 – Thickness 2 mm			
Criteria / **Trigger of type reduction**	**Peak force (KN)**	**CFE**	**Energy (KJ)**
Without trigger	58.4	0.43	5.07
5% elliptical	58.3	0.45	5.26
7.5% elliptical	58.2	0.45	5.41
10% elliptical	58.2	0.5	5.81
15% elliptical	57.9	0.46	5.35
20% elliptical	57.4	0.43	5.05

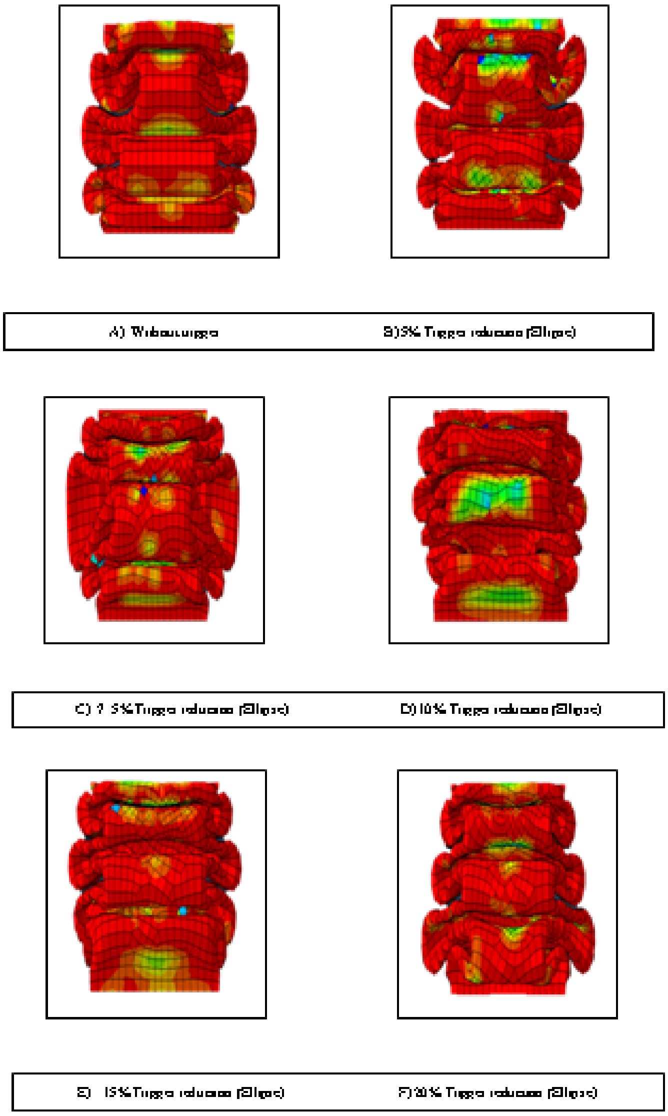


Fig -16 Deformation rectangular extrusion tube R-300 of different trigger reduction for aluminium alloy

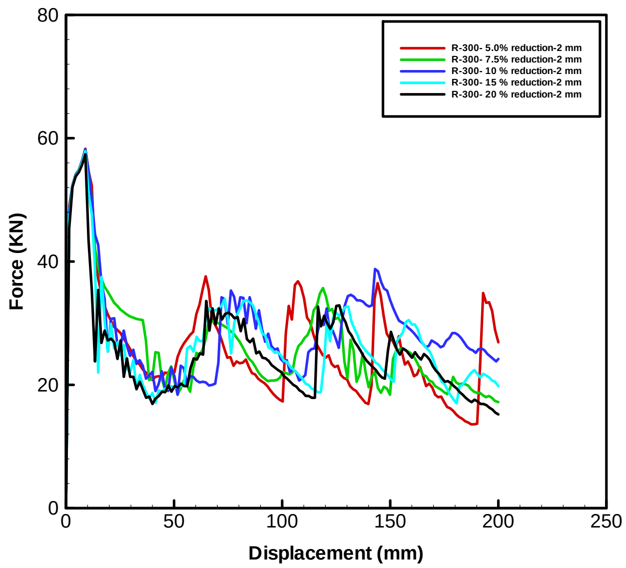


Fig -17 Force VS displacement for R-300 with different reduction of the ellipse trigger in direct load

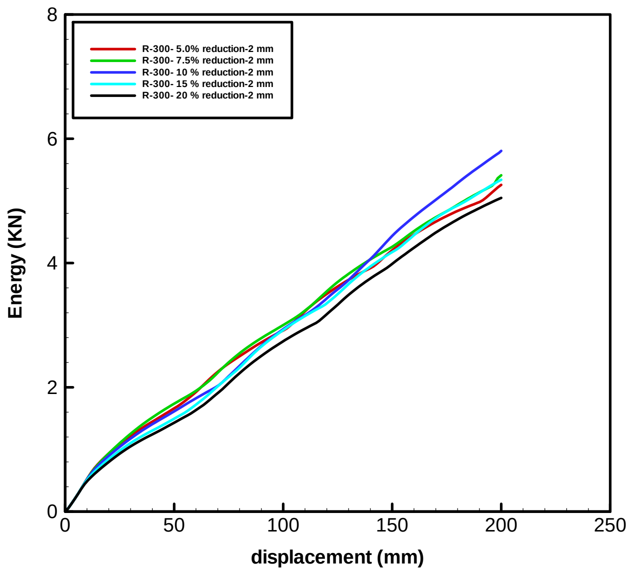


Fig -18 Force VS displacement for R-300 with different reduction of the ellipse trigger in direct load

5. CONCLUSIONS

This study investigated the different impact loads (oblique and direct) and the effects of the crush on the rectangular profile of ductile material of aluminum alloy (AA6060). The examination of the various rectangular profiles with different parameters was followed by selecting the profile with the best results. The selected profile was further examined, testing the profile with different hollow aluminum foam fillings and the choosing the tube with the filling which has given the best results in case of oblique and direct loading. As a last step the result given by the usage of the trigger mechanism was investigated. The dynamic simulation was conducted with the compact mass of 25% of the total weight (1100 kg) of the vehicle, and 15 m/s of impact velocity with direct, and oblique angles of the impact load on the rectangular tube. Based on the crash performance indicators, cost and manufacturing feasibility, the best results of energy absorption were given by the rectangular profile with wall thickness 3.4 mm and hollow foam type E (0.652kg). The 3.4 mm thick rectangular profile had 12.7 KJ of energy absorption and CFE of 0.5 in case of direct loading, while 10.9 KJ of energy absorption and CFE of 0.71 in case of oblique loading. The crash performance indicators improved, by adding the hollow foam type E (0.652kg) filling. The energy absorption of the 3.4 mm thick rectangular profile was 22.4 KJ and the CFE was 0.72 in case of direct loading, while these values were 18.9 KJ and 0.87 respectively in case of oblique loading. The trigger mechanism with 2 mm thick rectangular profile helped to increase the energy absorption from 5.07 to 5.81 KJ and enhanced the CFE from 0.43 to 0.5.10% reduction and 50mm trigger position are best selected. To summarize, the enhancement of energy absorption by 76.4%, improve the CFE by 44% in direct load, while in case of oblique load we have an enhancement of the energy absorption by 73.4%, and an improvement of the CFE by 22.5%. As the best result has been given by the 3.4 mm thick rectangular profile filled with hollow aluminum foam type E (0.652 kg) and with an ellipse shape triggers on the longitudinal side of the tube, this can be identified as a potential energy absorber candidate for crashworthiness applications.

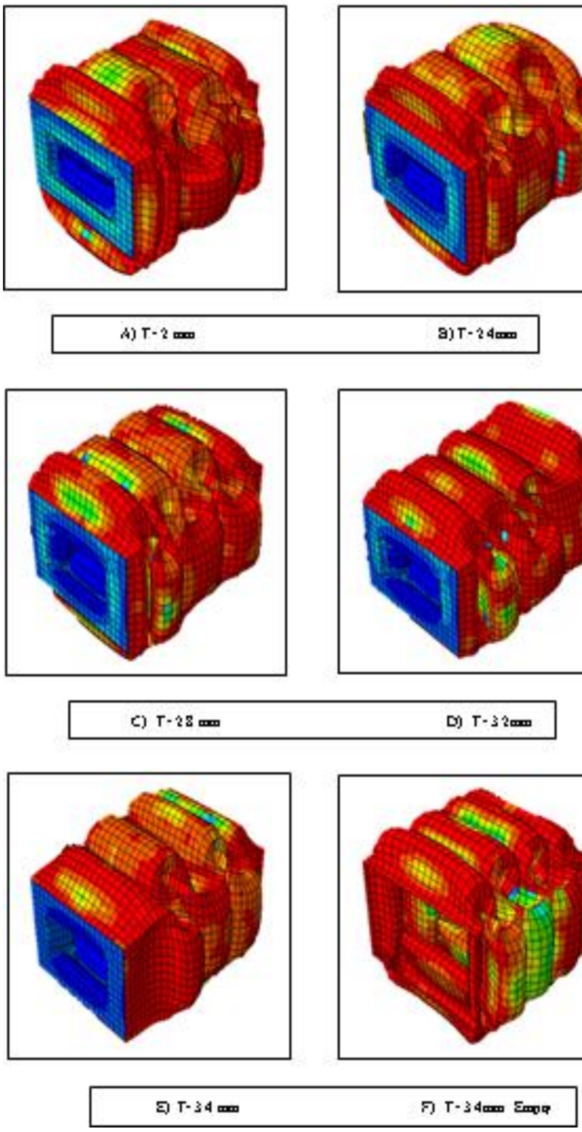


Fig -19 Crashing deformation longitudinal member, using different tube thicknesses with hollow aluminum foams E (0.652 Kg) for aluminum alloy under direct load

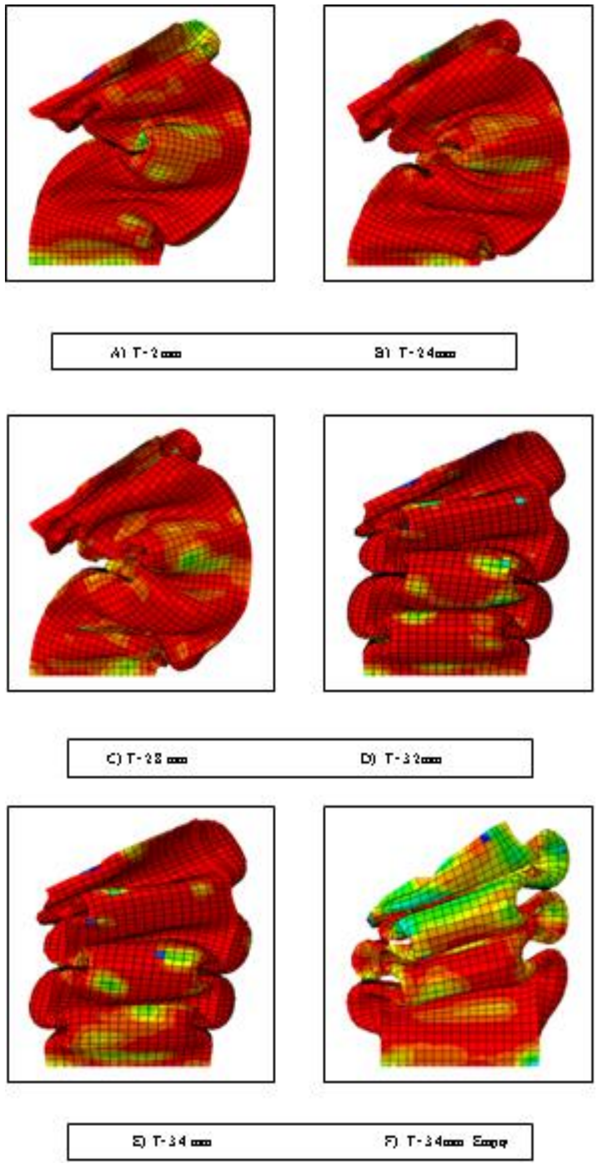


Fig -20 crashing deformation longitudinal member, using different tube thicknesses with hollow aluminum foams E (0.652 Kg) for aluminum alloy under Oblique load

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Table 6: Effect of using different aluminum foam thickness and different tube thickness for R-300 subjected to direct loading

Foam weight (Kg/mm2)	A (0.955 Kg)			B (0.9074 Kg)			C (0.841 Kg)			D (0.756 Kg)			E (0.652 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	250	0.57	25.4	232	0.59	24.5	144	0.77	20.9	128	0.79	19.2	106	0.73	15.1
2.4 mm	247	0.6	28	212	0.69	26.3	169	0.8	23.9	152	0.73	21.1	119	0.72	16.6
2.8 mm	X	X	X	X	X	X	155	0.84	24.7	166	0.75	23.5	132	0.74	18.9
3.2 mm	X	X	X	X	X	X	X	X	X	X	X	X	153	0.71	21.3
3.4 mm	X	X	X	X	X	X	X	X	X	X	X	X	161	0.72	22.4
Empty Tube thickness = 3.4 mm, Weight = 0.964 Kg													126	0.5	12.7

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

Table 7: Effect of using different aluminum foam thickness and different tube thickness for R-300 subjected to oblique loading

Foam weight (Kg/mm2)	A (0.955 Kg)			B (0.9074 Kg)			C (0.841 Kg)			D (0.756 Kg)			E (0.652 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	82.4	0.85	12.3	79.9	0.85	11.6	78.4	0.84	11.2	79.1	0.82	11.3	67	0.81	10.4
2.4 mm	96.6	0.86	14.4	97	0.84	14.2	96.3	0.83	14.3	93.7	0.80	14.3	81.7	0.80	12.6
2.8 mm	X	X	X	X	X	X	117	0.80	16.6	113	0.78	16.6	96.7	0.78	14.4
3.2 mm	X	X	X	X	X	X	X	X	X	X	X	X	115	0.80	17.5
3.4 mm	X	X	X	X	X	X	X	X	X	X	X	X	113	0.87	18.9
Empty Tube thickness = 3.4 mm, Weight = 0.964 Kg													76.5	0.71	10.9

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

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