

An Investigation on Dynamic Response of Truncated Thick Walled Cones with Edge Ring under Axial Compressive Impact Load

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Abstract
In this study, steel based combined truncated conical cones with end caps having different cone angles (20°, 25° and 30°) were investigated as an energy absorber under axial impact loading. The structure was modelled between two rigid plates and crushed by a striker mass of 1000kg with 6 different impact velocities (5m/s, 10 m/s, 15 m/s, 20 m/s, 25 m/s, 30 m/s). Structures have been modelled in ABAQUS CAE environment and simulations have been performed in explicit solver package. According to simulation results, peak reaction force F_p, mean reaction force F_m, absorbed energy E_A, specific energy absorption (SEA) and crush force efficiency (CFE) values were calculated in order to compare the results of crash performance of the models.
Keywords: finite element method, energy absorbing, truncated cone, crashworthiness.

Nomenclature

σ_y	Yield strength	R_1	Small diameter of the cone
E	Modulus of elasticity	R_2	Large diameter of the cone
F_m	Mean reaction force	t	Thickness
F_p	Peak reaction force	E_A	Absorbed energy during crash
K	Tangent modulus	β	Base cone angle
ρ	Mass density	v	Poisson's ratio

1. Introduction

Emerging technology in the transportation industry brings along a vital importance of vehicle and occupant safety. Therefore, vehicles are required to have the ability to absorb the impact energy and be survivable for passengers in case of an accident. In this context, designers face with a challenge to design energy absorbers to increase time required by vehicle to stop and dissipate more energy to protect occupants.

Energy absorbers dissipate energy both in reversible and irreversible way such as elastic strain energy and plastic deformation energy. Collapsible energy absorbers aim to convert the majority of the kinetic energy of impact into plastic deformation in an irreversible manner. [1] Commonly used geometrical shapes of energy absorbers in most studies are cylindrical tube [2, 3], square tube [4, 5] and truncated

conical tube [6] also known as frustum. Although most of the studies are focused on cylindrical and rectangular tubes, crashworthiness of conical frustums has been studied by many substantial authors.
Mamalis and Johnson [7] investigated aluminum alloy tubes and truncated circular cones under quasi-static compression. Authors compared the behavior of the circular cylinders and truncated circular cones and observed characteristic modes of collapse. Mamalis et al. [8] investigated circular tubes and truncated circular cones of low carbon steel using two semi-apical angled frusta (5° and 10°), at elevated strain rates. Non-symmetric diamond buckle patterns were observed as modes of collapse. Crumpled conical tubes and truncated conical cones were studied by Mamalis et al. [9] under axial compression for the concertina mode of deformation. In an attempt to show the effect of geometric parameters, numerical results are obtained and a theoretical model has been developed. Aljawi and Alghamdi [10] examined deformation modes of frusta as an impact absorber with using explicit code of Abaqus software. They also obtained a good agreement between finite element results and experimental results. Inversion of truncated conical cone as an energy absorber was investigated by Aljawi and Alghamdi [11] with respect to deformation modes both experimentally and numerically with using Abaqus FE code. Numerical and experimental results gave a good agreement with each other. Singace et al. [12], El-Sobky et al. [13] investigated the energy absorption performance of right circular frusta under dynamic axial load. Quasi-static tests in the range of velocities 2-5 m/s have been performed by using a drop hammer and then the results were compared. Optimum geometric parameters have been identified in order to obtain maximum energy absorbing capacity. Alghamdi et al. [14] investigated modes of deformation of capped end frusta between two parallel plates with different semi-apex angles and thicknesses. Experiment results have been compared with Abaqus finite element results and obtained a good agreement. Karbhari and Chaoling [15] have been carried out series of investigations into the crush of circular frusta under axial and off-axis loads. They have used various types of materials as well as loading angles. As a result of the study, one can say that using hybrid structures can lead to good levels of energy absorption. Easwara et al. [16] performed various crush experiments on a large number of spherical domes and conical frusta with various sizes. Experiments have been performed using a drop hammer with various impact velocities and comparisons have been made between modes of collapses and load-deformation curves. Gupta et al. [17] carried out on experimental and numerical studies on conical aluminum frusta of various thicknesses and semi-apical angles under axial compression. Experimental and numerical results have been compared with influence of rolling and stationary plastic hinges. Gupta and Venkatesh [18] investigated energy absorbing capacity and modes of collapse of aluminum conical shells with various semi-apical angles by impact axial compression experiments. The Impact response and static responses of the specimens have been compared. Ahmad and Thambiratnam [19, 20] investigated the difference between foam filled conical tubes and empty conical tubes with respect to crush and energy absorption performances under quasi-static axial loading. Ahmad et al. [21] also continued the study on foam-filled conical tubes under oblique impact loading. As a result of the study, it has been found that foam-filled tube has better energy absorbing capacity in comparison with empty tubes. Crush behavior of the straight and conical structures with various cross-section geometry, wall thickness and semi-apical angle have been investigated by Guler et al. [22] under axial impact loading. Consequently, circular absorber with a semi apical angle of 12.5° and a wall thickness of 2mm has been found as the most efficient combination. Ghamarian and Zarei [23] investigated the crash response of the end-capped cylindrical and conical tubes with various geometrical parameters. An Explicit FE code Abaqus/explicit have been used and compared with experimental results in order to find the influence of the geometrical parameters. Ghamarian et al. [24] also investigated the crash behavior of empty and poly-urethane foam filled conical tubes. They performed numerical analyses to simulate quasi-static tests and achieved a good match between numerical study and experiments. They also compared cylindrical and conical tubes. They indicated that empty cylindrical tube has 18.4% less specific energy absorption of the empty conical one. Tarlochan et al. [25] performed dynamic numerical analyses on several cross sectional geometries with both oblique and direct loading. An impact speed of 15 m/s and impact mass of 275kg was used in simulations. The results pointed out that a hexagonal tube has better crush performance on both oblique and direct loading. They also investigated the effect of foam filling and trigger mechanism on the crush performance of the structures. Lin et al. [26] studied on crushing force and energy absorption of the foam-filled conical tubes with fiber-reinforced layer by means of analytical method. They introduced a validated finite element method to simulate the collapse and be able to make comparison between numerical and analytical models. Baykasoglu et al. [27] also investigated and compared the crush behavior of different aluminum and steel based thin-walled structures under axial impact loading with square and circular cross sections, different semi-apical angles and thicknesses. This study pointed out that circular energy absorber is more efficient than the square absorber for all wall thickness values. Kathiresan et al. [28]

carried out the crashworthiness of the thin walled E-glass fibre/epoxy resin reinforced composite conical frusta under quasi-static axial compression. Structures having semi apical angle ranging from 15° to 27° have been investigated with the influence of the buckling modes of collapse and crush performances. Zhang Y. et al. [29] explored the multiobjective optimization for the hollow and foam-filled conical tubes under different and multiple oblique impact conditions. Specific energy absorption and crash force efficiency parameters have been investigated with respect to critical design parameters. Tanlak and Sonmez [30] performed an optimization study on the shape of a crash-box to maximize its crashworthiness. Design optimization made by authors on several geometric shapes and also low-velocity impact and high velocity impact situations were taken into consideration. Zhang X. and Zhang H [31] investigated the energy absorption characteristics of conical tubes with graded thickness under oblique impact load. Numerical analyses have been performed and validated by the experiments with influences of load angle, structure layout, strain rate effect, inertia effect and forming effects.

The main purpose of this study is to determine the energy absorbing capacity of capped-end truncated cones with relatively low base cone angles and edge ring. Unlike most of the studies, more shallow structures needed to be examined in detail for using as impact energy absorbers. Instead of a thin-walled structure encountered commonly in the literature, a thick-walled structure is used to compensate the shallow characteristic of the geometry. In this manner, three models with various geometric parameters have been modelled and studied numerically with influence of crashworthiness parameters such as load-deflection curves, crash force efficiencies and specific absorbed energy values.

2. Numerical Models and Simulation

Numerical models and simulations have been performed by using finite element software ABAQUS. [32] Simulations were made with the Explicit module of the software. Models were designed for 3 different angles (20°, 25°, and 30°) in order to compare the crashworthiness performance of the structures with the influence of base cone angle. The basic sketch of the absorbing structure is given in Figure 1 with dimension parameters. Dimension values are given in Table 1.

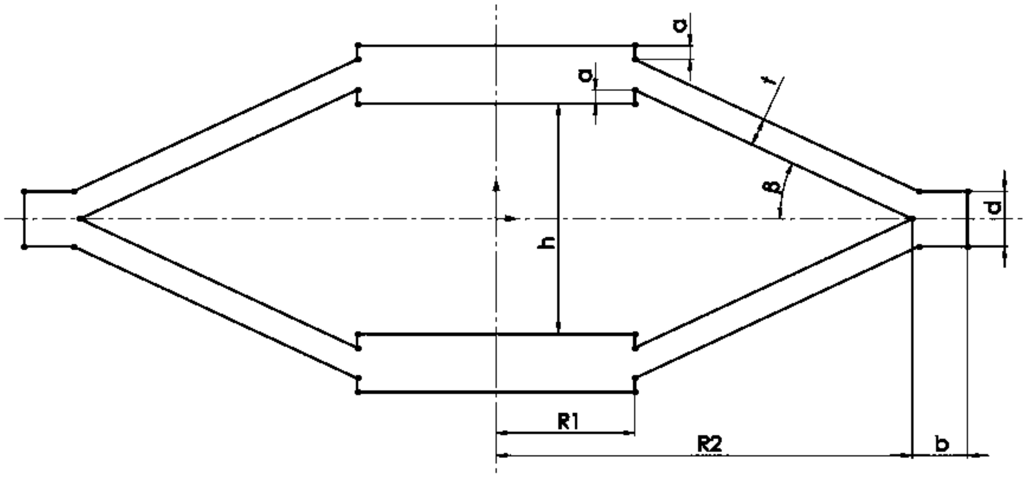

Figure 1. Geometry and dimension parameters of the absorber structure.

Table 1. Dimension values of the absorber structure for each model with different angles

Dimension Parameter	**β** [deg]	**h** [mm]	**t** [mm]	**b** [mm]	**d** [mm]	**a** [mm]	**R_1** [mm]	**R_2** [mm]
Model 1 (β =20°)	20°	62.8	10	20	20	5	50	150
Model 2 (β =25°)	25°	83.26	10	20	20	5	50	150
Model 3 (β =30°)	30°	105.48	10	20	20	5	50	150

The absorber was positioned between two rigid plates (top plate and bottom plate) for simulating the crash box and also controlling the deformation of the structure. The absorber and the two rigid plates are connected together with tie constraint. The bottom plate was fully constrained in all directions. Another rigid plate (striking plate) was used to simulate the striking mass crushing to the structure. The top plate and the striking plate were allowed to translate only along y-axis. Assembly of all plates and absorber structure is plotted in Figure 2.a. A gap of 10mm between the absorber structure and the striking mass was modelled in simulations in order to examine the effect of first contact more conveniently.

In further studies, it is planned to use of several conical absorbers combined together as a crash box of a railway vehicle. In this manner, a mass of 1000kg was chosen for the striker mass as a scaled mass of the vehicle and defined as isotropic mass to the striking plate. Impact velocity of 5 m/s, 10 m/s, 15 m/s, 20 m/s, 25 m/s, 30 m/s were defined respectively to the striking plate in simulations. Impact velocities are chosen within the limits of velocities used in the European standards.

All rigid plates were modelled with total amount of 10362 shell elements including 10048 of 4 node 3D bilinear rigid quadrilateral (R3D4) elements and 314 of 3 node rigid triangular elements with the combined element size of 5mm and 2mm. Absorber structure was modelled with 8 node hexahedral solid elements with reduced integration and hourglass control (C3D8R). A mesh refinement study was performed by using finite element analysis software ABAQUS in order to find the optimum element size for meshing. An element size of 2mm was chosen from the mesh refinement study with respect to the mean load, internal energy and time cost of the simulations. Number of hexahedral elements used for modelling absorber structures with β= 20°, 25°, 30° are 252918, 304872 and 314628 respectively. Cross-sectional view of the meshed structure and simulation assembly is given in Figure 2.

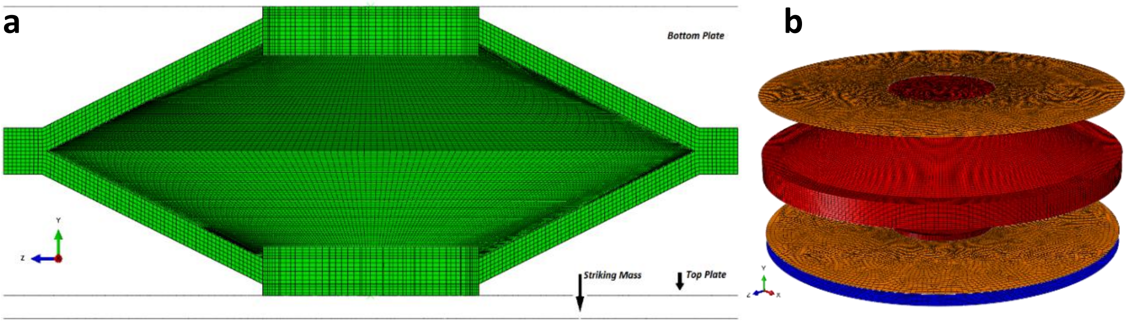

Figure 2. (a) Cross-sectional view of the mesh of the absorber; (b) Mesh of the FE model

Artificial strain energy and internal energy values taken from post-processor of the software were compared in order to ensure the mesh quality of the model. Artificial strain energy values were less than %5 of the internal energy of the absorber structure, proving that the model has relatively good mesh quality. [33] Comparison values between artificial strain energy and internal energy of the models are given in Table 2.

Table 2. Percentage of artificial strain energy (ALLAE) to internal energy (ALLIE) of the absorber structure.

	5 m/s [%]	**10 m/s** [%]	**15 m/s** [%]	**20 m/s** [%]	**25 m/s** [%]	**30 m/s** [%]
Model 1 (β =20°)	1.18	1.06	1.11	1.25	1.44	1.67
Model 2 (β =25°)	0.26	0.52	0.81	1.05	1.22	1.37
Model 3 (β =30°)	0.23	0.59	0.74	1.17	1.35	1.55

Interactions between parts were defined by using self-contact and surface to surface contact algorithms. Self-contact algorithm of the software was used to define self-contact of absorber structure with penalty contact having friction coefficient of 0.2 as tangential contact behavior and hard contact as normal contact behavior. Surface to surface contact was defined between all plates and energy absorber structure as penalty contact with finite sliding in order to make sure that no penetration occurs.

Material employed in the models were considered as steel with bilinear material behavior. Properties of the material used in numerical analysis were; yield strength (σ_y) of 235 MPa, modulus of elasticity (E) of 200GPa, Poisson's ratio (υ) of 0.3, mass density (ρ) of 7850 kg/m^3 and tangent modulus (K) of 20 MPa. With respect to the density and the geometry of the structures, mass of the absorbers for different angles β=20°, β=25° and β=30° were calculated as 16.21 kg, 16.65 kg and 17.24 kg, respectively.

3. Results and Discussion

Some parameters needed to be described in order to characterize the crashworthiness performance of the energy absorbing structures. To understand the crash response of the structure, the following parameters were obtained.

3.1 Force-displacement curves

All reaction forces were obtained from FE analysis results with respect to the reference point on the bottom plate. Force-displacement curves are plotted in Figures 3-5 to acquire a comparison between geometries of the models and impact velocities of the striking plate. Displacement values in the figures include the additional gap of 10mm at the interference of striking plate and absorber. After 10 mm of displacement, first contact between the striking plate and the absorber structure occur. As the impact occurs instantaneously, velocity of the contact surface of the absorber increases immediately and causes transient shock waves with positive and negative acceleration values. For negative acceleration values, direction of the reaction force changed and this caused negative reaction force values on the graph. In a small time increment, oscillations are damped pretty good and initial peak reaction force occurred. Same situation happened with the second contact between striking mass and the conical surface of the absorber after some deformation during the crash simulation. Remaining part of the crush simulation exhibit decreasing reaction force behavior as expected. The referred figures indicate that first peak loads of the simulations were affected inconsiderably by the impact velocity. However, both mean reaction forces and peak reaction forces are strictly dependent on the β angle of the geometry.

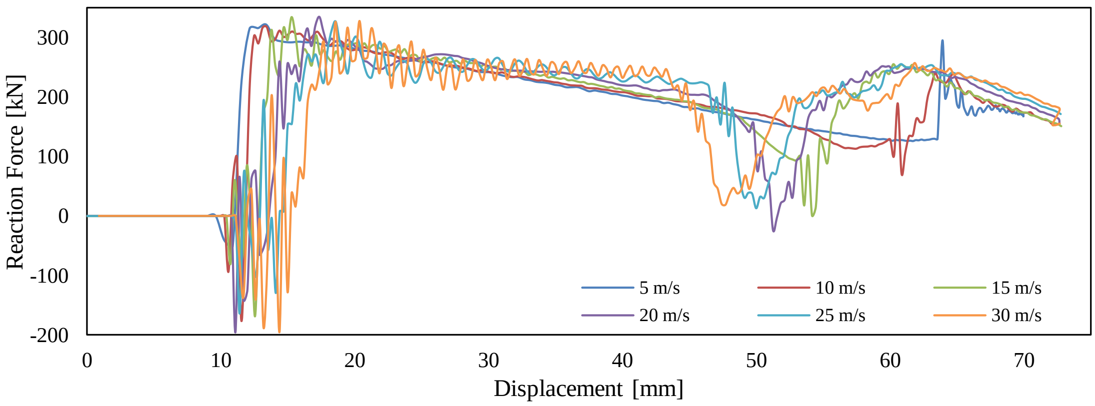

Figure 3. Force-Displacement plot of the model β=20°

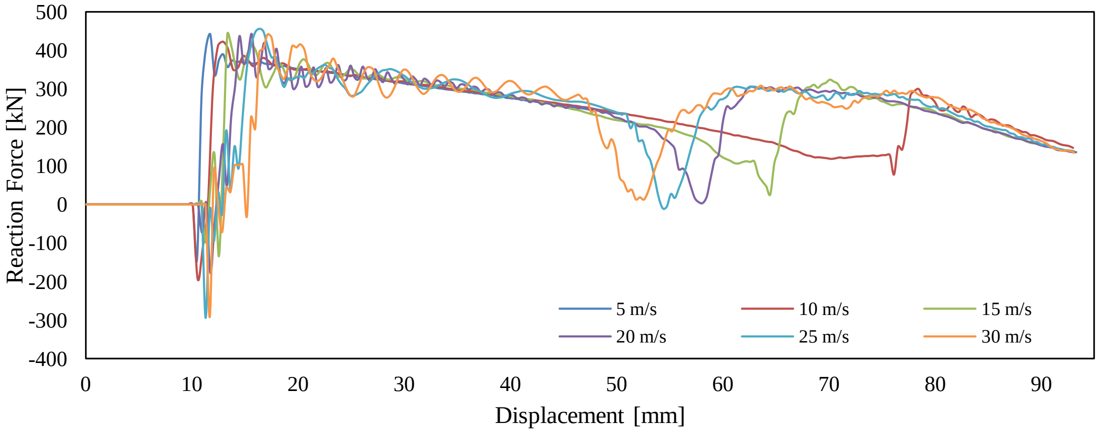

Figure 4. Force-Displacement plot of the model β=25°

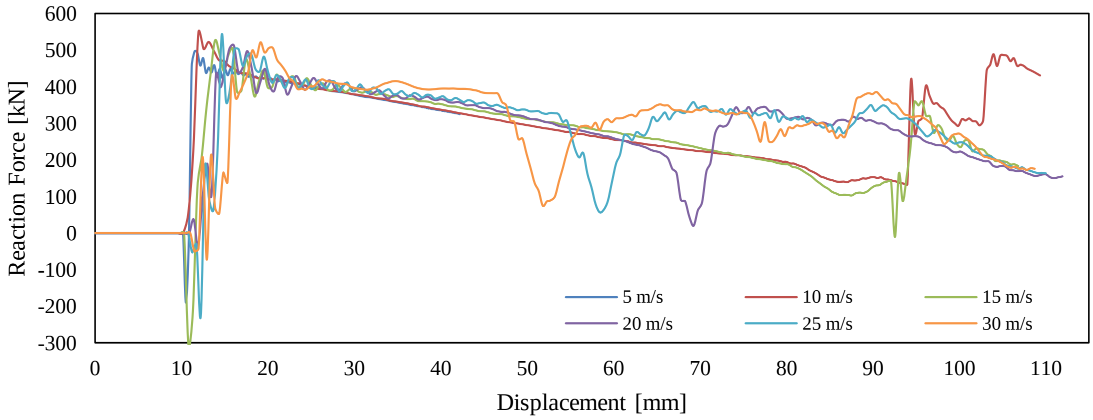

Figure 5. Force-Displacement plot of the model β=30°

3.2 Crash Force Efficiency (CFE)

The crash force efficiency is described as the ratio of the mean force to the peak force during the deformation caused by the impact. By describing this ratio, one can say if CFE is close to unity, that means there were no rapid changes on deceleration of the system. It is requested to have this ratio close to unity to approve the absorbing system has a better crash performance [34].

$$CFE = \frac{F_m}{F_p} \tag{1}$$

In the model, values for reaction forces in each time step were taken at post-processing stage of the study, in order to observe and calculate the mean and peak force values. After calculations, it has seen that the maximum average change of the peak reaction forces and crash force efficiencies are 9.6% and 23.3%, respectively. Hence, as seen in Figure 6, impact velocity has significantly low effect on both peak reaction forces and CFE values.

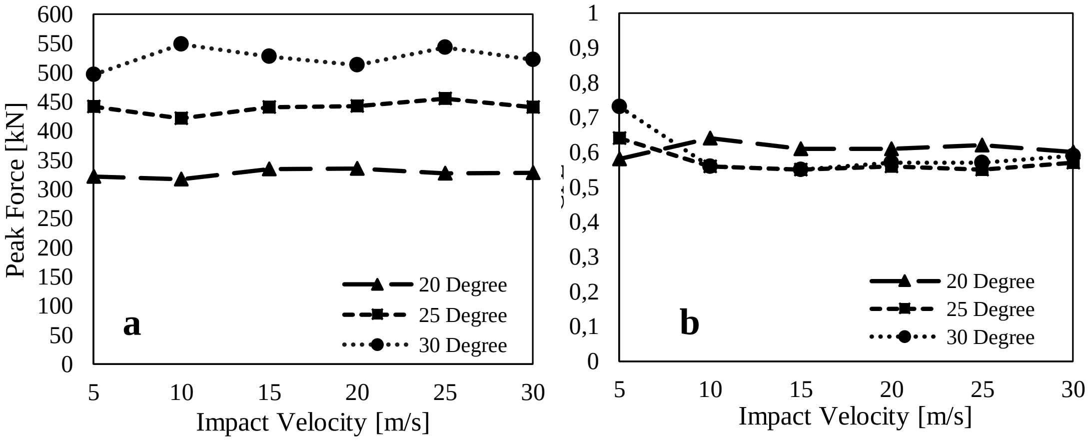

Figure 6. (a) Peak forces of the absorbing structure, (b) Crash force efficiency of the absorbing structure

3.3 Absorbed Energy

The total energy of the model is expressed as a composition of different type of energies in the ABAQUS software. Basically, the total energy of the system at the end of the impact is the sum of the absorbed energy and the kinetic energy left behind. Absorbed energy is determined as the sum of the total strain energy (ALLIE), viscous dissipation energy (ALLVD) and frictional dissipation energy (ALLFD) in the software. In Figure 7, absorbed energy values for each β angle and impact velocity are illustrated. The energy absorption characteristics of the structure significantly affected by the β angle and velocity. In Figure 8, absorbed energy of three models are plotted as a percentage of the total kinetic energy in the beginning.

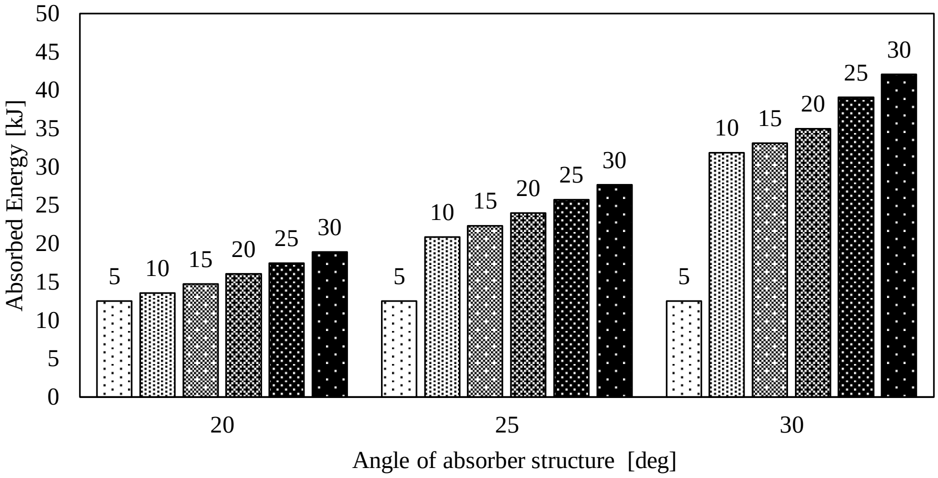

Figure 7. Comparison of absorbed energies between structures with different β angles and impact velocities

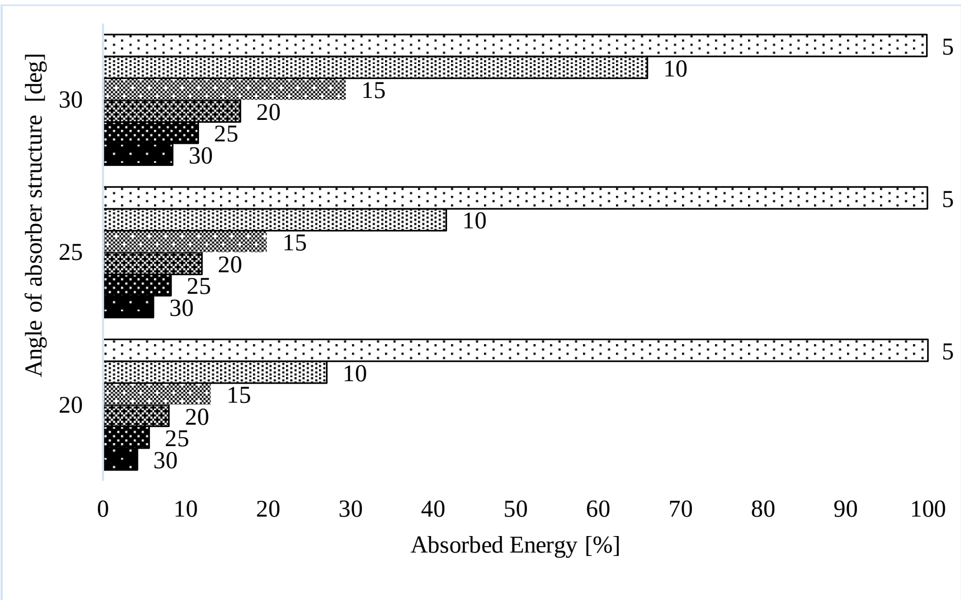

Figure 8. Percentage of absorbed energies for different β angles and impact velocities.

In all of three models with 5 m/s of impact velocity, all of the kinetic energies in the beginning were converted into plastic deformation energies and totally absorbed by the structure. Increasing velocity of the striking mass has a positive effect to the amount of energy absorbed, but the percentages of the total absorbed energy to the initial total kinetic energy decrease. It was observed that energy absorber structure became more inefficient with the increasing impact velocity. Structure exhibits more efficient behavior at higher β angles.

The variations of the reaction force and absorbed energy at an angle of 30° and 15 m/s impact velocity were presented in Figure 9, including the positions of the model at three different instants; a) initial contact, b) contact of cone surface and striker, c) contact of top and bottom surfaces of cones.

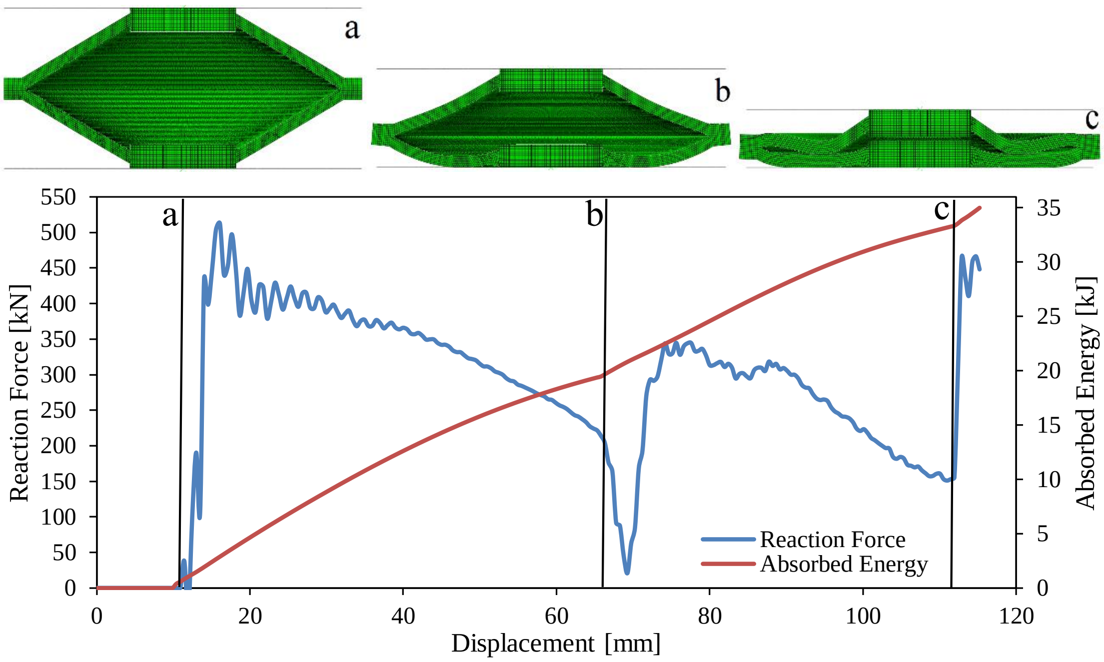

Figure 9. Deformation mode for the structure of 30° at 15 m/s at selected points on reaction force and absorbed energy to displacement plot.

3.4 Specific Energy Absorption (SEA)

Specific Energy Absorption SEA is another important parameter for an energy absorber. It is defined as the ratio of total energy absorbed in the system to the mass of the structure. The higher SEA value refers to a more lightweight absorber, which means more energy can be absorbed with a lighter structure. SEA values of this study were calculated by using the data collected from the output results of the simulation. Figure 10 shows the dependence of the SEA to both geometry and the impact velocities. As a result of the study, the effects of both β angle and the impact velocity on the dynamic response under impact loading cannot be underestimated.

$$SEA = \frac{E_A}{\rho} \quad (2)$$

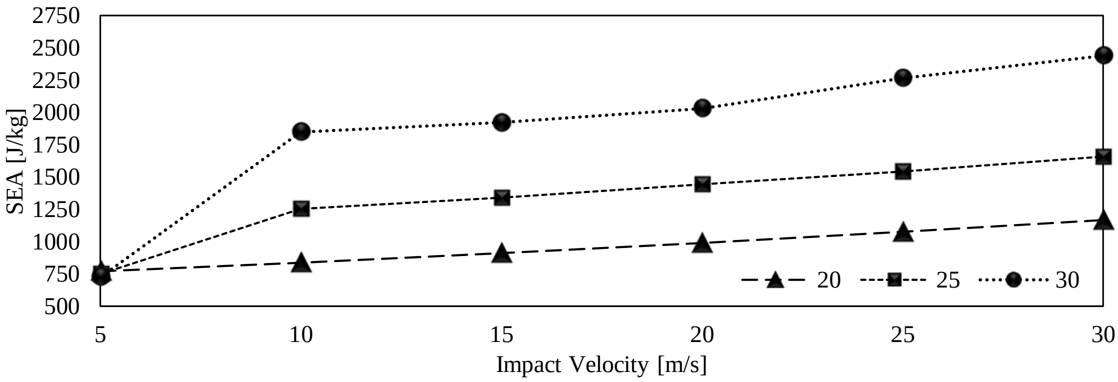

Figure 10. Comparison of SEA with different β angles and impact velocities.

4. Concluding Remarks

A numerical investigation of the conical energy absorbing structure was examined in this study. In order to obtain detailed information about the crash behavior of the absorber structure, axial impact was simulated by using explicit package of ABAQUS software. Some of the significant conclusions of the study are summarized below.

- The peak reaction force increases significantly with increasing β angle, but has almost no change with increasing impact velocity. The mean reaction force has similar characteristic with peak reaction force.
- The amount of the absorbed energy increases in all models when the β angle and impact velocity becomes higher.
- For CFE of the model, the highest value of CFE was observed in the model with β=30° and impact velocity of 5 m/s.
- The specific energy absorption (SEA) values have a slightly increasing behavior with increasing impact velocity in all models but the impact velocity of 5m/s.

Results obtained from this study are very useful to understand the response of the structure under impact loading. In future studies, the model of the structure will be improved to obtain longer deformable zone to increase the energy absorbing capacity and to enhance mean force to peak force ratio. Accordingly, a suitable experimental setup will be designed for the comparison and validation of the results with real conditions.

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