CRASHWORTHINESS INVESTIGATION OF METALLIC THIN-WALLED TUBES UNDER AXIAL IMPACT LOADING

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Abstract

In this study, the crush behavior of different aluminum and steel based thin-walled structures with square and circular cross sections, different semi-apical angles and thicknesses are investigated and compared under axial impact loading. In order to investigate crush behavior of these structures, impact with a fixed rigid wall is simulated by using commercial dynamic/explicit finite element (FE) code, Abaqus. The elastic-plastic material models are used and the effect of strain rate is included in the FE model by using the Cowper-Symonds overstress power law. The FE model is validated with the previously established solutions in literature. The maximum and mean crush forces, crush force efficiency (CFE), total absorbed energy \( E_{\text{total}} \), specific energy absorption (SEA) and deformation modes are computed to compare the crashworthiness performance of the thin-walled structures.

Keywords: Crashworthiness, thin-walled structures, axial crushing, energy absorption, finite element method

1. Introduction

Design engineers have been developing new active and passive vehicle safety systems to minimize injuries and loss of life when an accident occurs. Among them, thin-walled structures are mostly used in passive vehicle safety systems as crash energy absorber elements. These elements transform kinetic energy into internal energy via plastic deformations of structure [1]. Fig. 1 demonstrates examples of the metallic shells as energy absorber elements in front of trains and in longitudinal frames of automobiles [2].

The crashworthiness investigations of various metallic thin-walled structures have been widely studied in literature [1-20]. Among them, square and circular cross sectional, straight and tapered thin-walled structures are mainly considered as energy absorbing elements. All tube sidewalls are parallel to the longitudinal axis of the column in the straight thin-walled structures while one or more sides of the tube are oblique to the longitudinal axis in the tapered thin-walled structures [1, 3-6]. The tapered energy absorbers are preferred to the straight thin-walled structures since they decrease the possibility of global buckling which decreases the energy absorption capacity of the structure and are capable of withstanding both axial and oblique loads [7-9]. The studies on tapered energy absorbers are limited compared to the studies related to straight tubes [1-12]. Relevant works about crashworthiness investigation of thin-walled structures in literature are summarized as follows. Guler et al. [1] examined the crush behavior of steel based thin-walled straight and conical shell structures under axial impact loading by using finite element (FE) method. Ghamariani and Zarei [10] studied the crush performance of the end-capped cylindrical and conical tubes. Karbhari and Chaoling [4] investigated energy absorbing characteristics and crush mechanism with various parameters such as type of material, frustra geometry and angle of loading. The energy absorption characteristics of straight and tapered rectangular tubes are studied under impact loading by Nagel and Thambiratnam [7]. Energy absorbing elements have various wall thicknesses and tapered angles which are modeled as tubes with one, two, three and four tapered sides in [7]. Singace et al. [11] studied cylinders and frustra structures under higher Euler buckling load with the same weight and volume. Mamalis and Johnson [12] investigated truncated circular cones and thin-walled circular aluminum alloy under axial static loading.

Even though the quantity of the absorbed energy is the most important parameter for an energy absorber, the amount of the crush forces is also essential [1]. The purpose of an energy absorber is to continue low crush forces conversely reaching high energy levels. When the crush force is high, the force transmit to the passengers and vehicle components will be high [7]. There is a limit for the safety of both crashworthiness of the vehicle components and the passengers. In order to obtain that, the initial peak crush force should be decreased as much as possible. For this reason, researchers use a tapered shape in axial direction since it absorbs large amount of energy and reduces the first peak load. Moreover various design strategies were demonstrated in the literature to decrease the initial peak crush force [15-19]. The specific energy absorption (SEA) is another crucial parameter in the designing of the crush elements which is defined as the energy absorption per unit mass. If the weight of the structure is an important parameter, the energy absorber...
Baykasoglu, C., Tunay Çetin, M. and Yalcin, O. should be researched with respect to its weight efficiency and SEA value ought to be checked [20].

The aim of this study is to investigate the effects of different geometrical parameters (such as wall thickness, semi-apical angle, cross section geometry and material), on energy absorption characteristics of thin-walled structures. To this aim, an explicit-dynamic FE method is used. The FE model is validated with the previously established solutions in the literature. CFE and SEA, the maximum and mean crush forces, deformation modes and total absorbed energy ($E_{\text{total}}$) values are calculated for each energy absorber design.

2. FE models

The crush simulations have been performed using nonlinear explicit FE code, Abaqus [21]. The FE crush models consist of a thin-walled specimen, a moving and a fixed rigid plate. Fig. 2 shows the components of the FE model. Specific mass ($m$) and axial velocity ($v$) are assigned to the moving rigid plate to simulate the mass and velocity of the crashing body. 120 kg mass and 15 m/s velocity are used for the steel based thin-walled energy absorber, while 120 kg mass and 12 m/s velocity are used for the aluminum based thin-walled energy absorber. It is assumed that, there is no initial imperfection in the thin-walled structures.

The models of the straight and tapered thin-walled energy absorbers are developed using four nodes quadrilateral reduced integration shell elements (S4R), which are suitable for large strain analyses. To obtain the best balanced accuracy and efficiency for numerical simulations, a standard convergence test is applied. After convergence, an element size of 1.5 mm is found to produce suitable results.

The mesh of FE models is uniform along the absorber axis as shown in Fig. 2. The length of the straight and tapered tubes is selected to be 200 mm. Surface- to- surface and self-contact algorithms are used to define interactions between the moving rigid plate and the energy absorber, and for the self-contacts of the energy absorber. A friction coefficient ($\mu$) of 0.3 is considered for both surface- to-surface and self-contacts. The elastic-plastic material models are used for steel (St32, $E=210000$ MPa, $\sigma_y=235$ MPa) and aluminum (AlMgSi0.5, $E=75230$ MPa, $\sigma_y=200$ MPa). Table 1 shows the stress-strain data points used for steel and aluminum in the FE models, which are obtained from quasi-static and dynamic tension tests [2].

The effect of strain rate on the steel based thin-walled energy absorber is included in the FE model by using the Cowper-Symonds overstress power law given [21].

$$\dot{\varepsilon}^p = D (R - 1)^n \quad \text{for} \quad \dot{\varepsilon} \geq \dot{\varepsilon}^0$$

where $\dot{\varepsilon}^p$ is the inelastic strain rate, $R$ is the ratio of the dynamic flow stress ($\dot{\sigma}$) to the static flow stress ($\sigma^0$), $D$ and $n$ are material strain rate parameters. The strain rate parameters, $D$ and $n$, in Eq. 1, are taken to be 3000 s$^{-1}$ and 4, respectively. These values are taken from the previous study for the dynamic axial crushing of St35 cylindrical shell [2]. On the other hand, the strain rate dependency for the given aluminum material is not taken into account since it is negligible [2].

In analysis, two different types of cross-section geometries, namely square and circular, are used to compare the crashworthiness performance of both steel and aluminum-based thin-walled energy absorbers. In addition, four different wall thicknesses, namely 0.5 mm, 1 mm, 1.5 mm and 2 mm; and six different semi-apical angles, namely 0°, 2.5°, 5°, 7.5°, 10° and 12.5° are also considered for each of the thin-walled energy absorbers. The diameter of the circular cross section (d) and the width of the square cross section (w) are taken to be 180 mm. On the other hand, the identical axial length of 200 mm is used for all energy absorbers. Fig. 3 shows the geometries of the square and circular energy absorber structures.

3. Results and Discussion

The parameters necessary for characterization of crashworthiness performance of energy absorbers can be described as follows:

- Maximum crush force which is generally defined as peak force, $F_{\text{max}}$
- The post-crush displacement, $\delta$ is displacement of crushed structure in load-displacement curve
- Total absorbed energy which is referred to the area under the load-displacement curve,
\[ E_{total} = \int_{0}^{\delta_{max}} F \, d\delta \]  
(2)

- Mean crush force which is obtained by following equation,

\[ F_{mean} = \frac{E_{total}}{\delta_{max}} \]  
(3)

- Crush force efficiency (CFE), which is the ratio of the mean crush force, \( F_{mean} \) to the maximum crush force, \( F_{max} \).

\[ CFE = \frac{F_{mean}}{F_{max}} \]  
(4)

- Specific energy absorption (SEA) which is defined as the total absorbed crash energy per unit of the crushed structure mass (m)

\[ SEA = \frac{E_{total}}{m} \]  
(5)

| Table 1. The stress-strain data points used for steel and aluminum in the FE models [2] |
|-----------------|-------|-------|-------|-------|-------|-------|-------|
| Steel           | \( \sigma \) [MPa] | 200   | 206.154 | 215.385 | 218.462 | 228.718 | 230.769 | 235.897 | 235.897 |
|                 | \( \varepsilon_{pl} \) | 0.0049 | 0.0112 | 0.0175 | 0.0305 | 0.0426 | 0.0556 | 0.0697 | 0.0998 | 0.1304 | 0.1641 | 0.1862 |
| Aluminum        | \( \sigma \) [MPa] | 235   | 251.020 | 283.673 | 304.082 | 325.510 | 333.673 | 339.796 | 340.816 |
|                 | \( \varepsilon_{pl} \) | 0.0346 | 0.0494 | 0.0697 | 0.0998 | 0.1304 | 0.1641 | 0.1862 |

Crush force-displacement and absorbed energy-displacement curves are obtained from performed the crash simulations in Abaqus/Explicit. Then, the mean crush force, CFE and SEA values are calculated by using Equations 2-5 for a 90 mm deformation length. CFE and SEA values are obtained for square and circular energy absorbers for each wall thickness and semi-apical angle. In order to verify the proposed FE model, a straight model executed with the same material, geometry, boundary conditions and impact velocity as given in the Nagel and Thambiratman [7] and Guler et al. [1]. Table 2 shows the maximum crush force, mean crush force and energy absorption (at a deformation length of 200 mm) values of the straight model. As seen in Table 2, the presented results are in good agreement with [7] and [1].

| Table 2. Maximum crush force, mean crush force and energy absorption values of the straight model |
|-----------------|-------|-------|-------|
| Nagel and Thambiratman [7] | 200   | 45.5  | 9.1   |
| Guler et al. [1]        | 204   | 48    | 9.6   |
| Present                | 197   | 48.5  | 9.7   |

Fig. 4 shows the comparison of the CFE with semi-apical angle values for different wall thicknesses and shapes for aluminum. As seen from Fig. 4, square model is more efficient at 12.5° semi-apical angle while the circular model is more efficient at 10° semi-apical angle for 0.5 mm wall thickness. On the other hand, the circular model is more efficient at 7.5° semi-apical angle whereas the square model is more efficient at 12.5° semi-apical angle for 1.25 mm wall thickness. Both circular and square models have the highest CFE values in all cases for 1.5 mm wall thickness. In addition, the most efficient energy absorber is the circular cross-sectional absorber with 1.5 mm wall thickness and 10° semi-apical angle as seen from Fig. 4.

According to Fig. 5, the circular energy absorber is more efficient than the square energy absorber for all wall thickness values. For both 1 mm and 1.5 mm wall thickness values, the square model is more efficient than the circular model at 12.5° semi-apical angle. Moreover the circular model has higher CFE than the square model for all semi-apical angles for 1.25 mm wall thickness. Fig. 5 also shows that the circular absorber is the most efficient one with 1.5 mm wall thickness and 10° semi-apical angle. It can be concluded from these graphs that conical absorbers are generally more efficient than the straight ones for both square and circular cross-sectional geometries.
In addition to CFE, there is another important parameter which is called SEA. The SEA values of aluminum are obtained for both square and circular cross-sectional geometries and various wall thicknesses as seen from Fig. 6. According to Fig. 6 circular energy absorber is more efficient than the square model except for the one with 1.5 mm wall thickness.

Fig. 7 shows the SEA- semi-apical angle values of steel with various wall thickness values and two different cross-section geometries. The highest SEA value for circular energy absorber occurs at 7.5° semi-apical angle and 1.5 mm wall thickness value. On the other hand, for all semi-apical angle values the circular model is more efficient than the square model except for the one with 1.5 mm wall thickness.

5. Conclusions

In this study, the crush behavior of different aluminum and steel based thin-walled structures with square and circular cross-sections, different semi-apical angles and thicknesses were examined numerically and compared with each other. For this purpose, 60 specimens with different wall thicknesses and semi-apical angles are used. The following conclusions can be derived from the analyses:

- When the wall thickness of absorber increase, the absorbed energy also increases for all models.
- As the absorbed energy increases, the crush force also increases for all models.
- Both steel and aluminum based energy absorber models are crushed progressively in all cases.
- For aluminum model, the highest values are observed with 1.5 mm thickness and 10° semi-apical angle in both CFE and SEA graphs.
- According to CFE graph, the highest value is observed with 1.5 mm thickness and 10° semi-apical angle while in SEA graph, the highest value is
observed with 1.5 mm thickness and 7.5° semi-apical angle for steel model.

In future, we plan to study observed with crush behavior of composite based thin-walled structures with different cross sections, semi-apical angles and thicknesses to reduce the weight of the energy absorber. The results presented in this paper could be useful for a better understanding of the crush behavior of metallic thin-walled structures and potentially beneficial for designers of these structures.

References


