Iran University of Science and Technology

Automotive Science and Engineering

Journal Homepage: ase.iust.ac.ir

ISSN: 2008-9899

Crashworthiness investigation of tubes filled with gradient foams

Hamidreza Zarei [1*], Mohammad Nazari [1]

[1]Postgraduate Center, Shahid Sattari University of Aeronautical Engineering, Tehran 13846-73411, IRAN

ARTICLE INFO

Article history:

Received: 7 Jan 2022

Accepted: 16 Nov 2022

Published: 16 Nov 2022

Keywords:

Gradient

Foam filled

Crash impact

Energy absorption

A B S T R A C T

This manuscript investigates the energy absorption behavior of the empty aluminum and ALPORAS foam-filled square tubes. An axial impact test apparatus conducts the experimental tests. To discover more details about crushing behavior, LS DYNA software is used for numerical simulation of the tests. The results of both methods are in satisfactory compliance. As a novelty, the crash performance of tubes filled with different foam densities is investigated. To examine the effect of foam density on energy absorption of the tubes, multi-layer foams with three different densities of 230, 330, and 430 kg/m^3 have been applied and the arrangement of the three foams is also a point of attention. It has been proven that filling the tubes with gradient foam improves the crash characteristics of the tubes. Numerical results revealed that tubes filled with gradient foam filler can absorb more energy than empty tubes (106%). Also, it can be seen that tubes filled with different individual foams have higher (63.7-87%) energy absorption capacities than empty tube. In numerical simulations, the required foam parameters are estimated from existing formulas and test results.

[DOI: 10.22068/ase.2022.604]

*Corresponding Author
Email Address: zarei@ssau.ac.ir
https://doi.org/10.22068/ase.2022.604

1. Introduction

Square aluminum tubes are used as crash-absorbing components to prevent severe damage to the body of the structures by dissipating initial kinetic energy during an impact loading and keeping the mean crushing load within an acceptable level. Aluminum tubes are collapsed and deformed plastically by impact loading and subsequent successive folding advances energy absorption (EA) level [1, 2].

Hanssen et al. performed static and dynamic crash tests on the square [3] and circular [4] aluminum foam-filled columns. Their work proposed some equations to predict the mean force, peak force, and effective crushing distance of filled columns. Axial crash tests were conducted to inspect the crash response of a thin-walled tube with foam filler by Ghamarian et al. [5] and Zarei et al. [6]. Their studies indicated that filled structures behave better than thin-walled empty tubes. Additionally, they compared the behavior of cylindrical and conical tubes and reported that energy absorption in cylindrical tubes is 18.4% less than conical samples. Xiuzhe et al. [7] studied the crash response of sheet-filled thin-walled structures. Their research revealed that the crash behavior of functionally lateral graded thickness tubes was relatively higher than traditional counterparts with uniform thickness. Multi-objective robust design optimization (MORDO) method was applied by Fang et al. [8] to find the optimized crash behavior of foam-filled bi-tubal structures.

Chang [9] et al studied tapered square tubes experiencing the inclined impact crash load. The research presented that specific energy absorption (SEA) and peak crushing force (PCF) of multi-cell tapered tubes were better than multi-cell straight tubes. They concluded that critical load angle in tapered samples is in the range of 30°-40° indicating that tapering the tubes is a stabilization solution. Crash investigation on empty [11, 12] and filled [8] square aluminum samples was done by Reyes [10] et al. They concluded that the EA capacity of tubes is extremely dropped when a global bending mode is triggered (in place of continual buckling). Nagel and Thambiratnam [13– 15] examined the EA reaction of straight and tapered thin-walled rectangular specimens under quasi-static and dynamic axial and oblique impact loading. They found that the tapered tubes behaved better under inclined impact. Besides, taking the geometry parameters into account revealed that wall thickness is the dominating feature compared to taper angle by more than 80%. To achieve the optimal SEA and PCF of thin-walled tubes, structural optimization routes were used in the pieces of literature [16–19]. Zarei and Kroger [17] proposed a new class of optimization. They analyzed the crush reaction of the thin-walled samples under axial and impact load via LS-DYNA finite element (FE) code. They executed a parametric examination based on the sampling design to develop the EA of the tubes under inclined impact loading. Najibi [20] et al. explored the EA features of the functionally graded foam-filled thin-walled layered tubes through quasi-static finite element analysis. The results proved that a 13-layered specimen with a high-low-high density arrangement (SEA=17.57kJ/kg) is an acceptable approximation for the continuous axial variations of density (SEA=17.06 kJ/kg).

Abdullahim and Gao [21] introduced a two-stage discrete and recurrent optimization process to optimize the crash behavior of the multi-cell tubal structure. Their calculation indicated that increasing the mass ratio from 0.5 to 0.75, increases the SEA from 34.25 kJ/kg to 42.96 kJ/kg. Khalkhali et al, numerically studied the crash performance of thin-walled square tubes under 3-D inclined loading [22]. Their results showed that there is not an identical tendency for all circumstances of 3D inclined loading in comparison to axial loading. Zarei examined the crash behavior of simple and hybrid composite samples with one, two, and three layers under axial and oblique impact loading [23]. Here, the EA and SEA are considered to find modified crash absorbers in terms of weight. Djamaluddin et al. reviewed the presented research about optimizing the crash reaction of foam-filled thin-walled structures [24].

In this paper, axial impact tests are carried out on the empty tubes at first. Then, the tubes are filled with aluminum foams and the tests are conducted again to investigate the effect of

[DOI: 10.22068/ase.2022.604]

aluminum foams. Also, successive crushing behaviors of the empty and filled tubes were simulated using the finite element method. The effect of foam density on the EA of the filled tubes was investigated. Afterward, each tube is filled with three foams with different densities. The novelty of the paper is comparing the crush response of multi-foam tubes. Two configurations are high-medium-low and low-medium-high densities both of which were compared with filled tubes with individual foam.

2. Material properties

2.1. Aluminum tube

Extruded heat-treated 6060 aluminum tubes were used. Standard tensile tests were done to find the mechanical properties of the aluminum tubes. The yield strength S_y= 231 MPa and ultimate strength S_{ult} = 254 MPa have been measured.

2.2. Foam

The Alporas aluminum sheet (foam) produced by the Shinko Wire has been applied in this work. The nominal thickness and density (ρf) of sheets were 50 mm and 230 kg/m3, respectively.

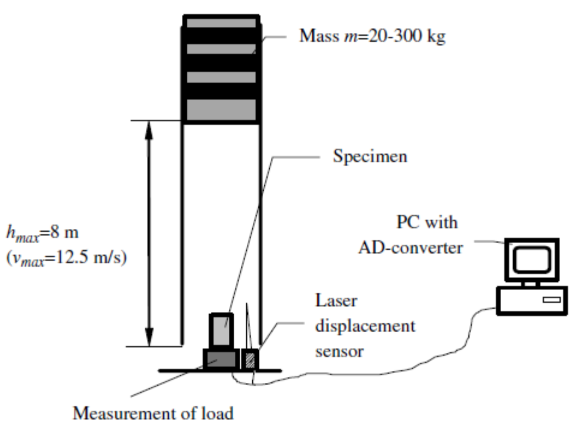

Figure 1. Test setup.

3. Experimental and Numerical studies

3.1. Experimental procedure

Axial Crash performance of foam-filled and unfilled tubes was analyzed experimentally. The tubes are 270 mm long, and 2mm thick, with two different cross-section dimensions of 55×55 mm and 60×60 mm. The tubes were filled with 50 mm thick foams layer-by-layer. The experimental test apparatus can be seen in Fig. 1 available in the Institute of Dynamics and Vibration at the University of Hannover. The bottom end of the specimens was clamped within a gap of 30 mm. An object of 154 Kg with a maximum drop height of 8m is selected, while mass initial velocities are different. Each test has been repeated three times to improve the validity of the results. The research concern was the average impact force Pm and the absorbed energy. Pm is expressed as:

$$P_m = \frac{1}{\delta_{\max}} \int_0^{\delta_{\max}} P(\delta) d\delta \tag{1}$$

Here P(δ) is the immediate impact force resulting from the immediate crush displacement. The EA is the area under the crush load-displacement diagram. The amount of captivated energy per structure mass unit is called SEA.

3.2. Numerical Simulations

FE software of LS-DYNA was used to do the numerical studies. Due to the symmetry of the tubes, just half of the samples are modeled. Belytschko–Tsay shell elements were applied to meshing the tubes. Solid elements were used for foam filler. The impactor was modeled as a rigid body. Since initial geometrical defects affect the uttermost loads, these defects have been axially taken into account in the calculations by Eq. 2 [25]:

$$w = w_0 sin\ (n\pi x/L) \tag{2}$$

Where w0 and n respectively are amplitude and number of half-sine waves along the length. The variables were determined as w_0 =0.1 mm and n = 5 in this work.

The number of LS-DYNA elastoplastic material 103 was implemented for aluminum tubes. Berstad et al. studied the effects of varying hardening parameters on the absorbed energy of aluminum tubes [26]. He showed that although different hardening parameters result from some diversities in the way tubes respond, the EA for each magnitude of the hardening parameter is almost equal [27]. Hence, in this

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work, the non-linear isotropic hardening model is applied. Deshpande and Fleck foam material model (material number 154 in LS-DYNA) was used for aluminum foam [28]

Deshpande and Fleck expressed yield stress function based on tests conducted on aluminum foams as:

$$\sigma = \frac{\sigma_{vm}^2 + \alpha^2\sigma_m^2}{1+\left(\frac{\alpha}{3}\right)^2} - \sigma_y \tag{3}$$

$$\sigma_{vm} = \sqrt{\frac{2}{3}}\left(\sigma - \sigma_m I\right) \tag{4}$$

$$\sigma_y = \sigma_{pl} + \gamma\left(\frac{\varepsilon}{\varepsilon_D}\right) + \alpha_2 Ln\left(\frac{1}{1-\left(\frac{\varepsilon}{\varepsilon_D}\right)^{\beta}}\right) \tag{5}$$

Where σ_{vm} and σ_m are Von Mises and mean stress and σ_{pl}, γ, α_2, and β are material dependent constants. Densification Strain has also been defined as:

$$\varepsilon_D = -Ln\left(\frac{\rho_f}{\rho_{f_0}}\right) \tag{6}$$

Here foam density denoted by ρ_f and ρ_{f0} is the base material density.

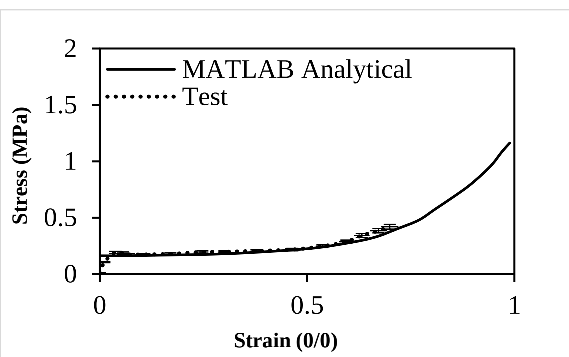

Figure 2. Stress-strain diagram of ALPORAS aluminum foam resulted from test (dotted) and MATLAB analytical (solid).

4. Results and discussion

Experimental and calculated crush load–displacement and deformation patterns of hollow and foam-filled tubes are presented in Fig. 2 and 3. Here, it is evident that the deformation mode is accurately simulated. Experimental and numerical crush load-displacement curves are close to each other. Here it can be found that the numerical mean crush load is slightly greater than the experimental outcomes.

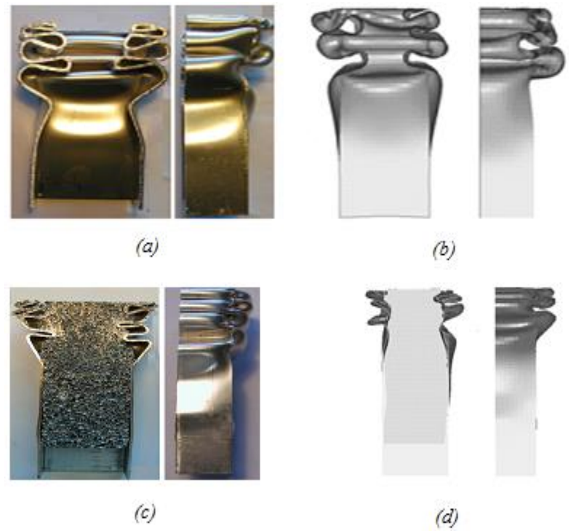

Figure 3. Impact deformation of empty aluminum tube deformation in (a) experimental test (b) simulation. Foam-filled aluminum tube deformation in in (c) experiment (d) simulation.

4.1. Gradient foam filler

Previously shown that foam filler generates a force resisting the squeeze of walls [17]. This force is dependent on the density of the foam. Also, it has been found that the denser the honeycomb, the more energy absorbed. In addition, it has also been expressed that using dense honeycombs always increases the EA, but there is a critical density hindering the use of excessively dense honeycombs [18]. The same results were presented for foam-filled tubes [17]. This study concluded that an optimum foam density presents higher EA and SEA simultaneously.

To improve the crash performance of foam-filled tubes, in this paper tubes were filled with gradient foams. The tubes were filled with three foams with different densities of 230, 330, and

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430 kg/m^3 each of which filled one-third of the effective length of the tube before and after applying the impact load. Two different arrangement is selected for foam filler. In the first solution, the foams are arranged as 230, 330 and 430 kg/m3 from the top of the tube closed to impact mass and in the second solution, the arrangement was reversed.

Numerical simulations were applied to determine the crash behavior of the gradient-filled tubes. In this paper, the properties of the foam for several densities are assessed by use of the theoretical formulation that was derived by Gibson and Ashby [29]. For this estimation the foam stress-strain diagram is subdivided into three zones: elastic zone, plateau zone, and densification zone. In the first zone, stress and strain are related by a linear elastic equation. The foam modulus of elasticity is calculated by the equation below:

$$E = E^{*}\left[\left[\phi^{2}\left(\frac{\rho_f}{\rho_{f_0}}\right)^{2}\right]+\left[(1-\phi)\left(\frac{\rho_f}{\rho_{f_0}}\right)\right]\right]\sqrt{a^{2}+b^{2}} \tag{7}$$

In the second zone, foam deforms mainly under steady stress $\sigma_{pl.}$ Here σ_{pl} can be calculated by equation proposed by Ashby & Gibson [29]:

$$\sigma_{pl} = \sigma_{y0}\left[0.33\left\{\phi\frac{\rho_f}{\rho_{f_0}}\right\}+0.44(1-\phi)\frac{\rho_f}{\rho_{f_0}}\right] \tag{8}$$

Where σ_{ys} is the filler foam yield stress, φ is fracture coefficient at the foam cell wall, ρ_f and ρ_{f0} are respectively the foam density and nominal density. Eq. 6-8 are used to estimate stress-strain diagram of the tubes filled by foams with varying densities.

Since eq. 6-8 have some material constants that should be determined from the compression tests on foam with different densities, compression crash test was done on foam with a density of 230 kg/m^3. Experimental Compression results presented in Ashby & Gibson [29] were used to calibrate equations 6-8. In other words, the results of [29] were used to best estimate the behavior of foams with other densities.

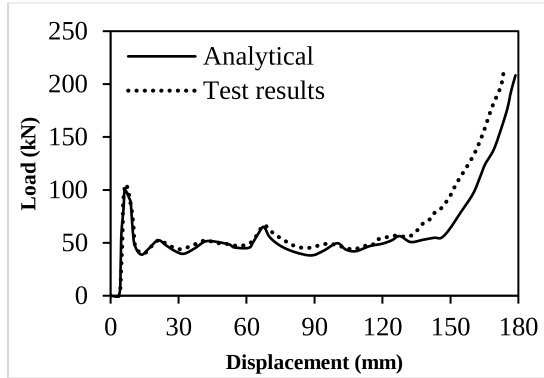

Figure 4. Load-displacement diagram of foam-filled tubes resulted by test and analytical.

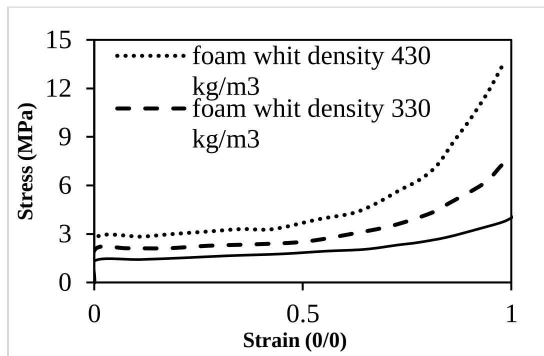

Figure 5. Stress-strain diagram of tubes filled by foams with densities of 230, 330, and 430 kg/m3 resulted estimation method.

Fig. 4 shows the estimated and experimental stress-strain plot of foam with a density of 230 kg/m^3. The same stress-strain diagram was estimated for foam with densities of 330 and 430 kg/m^3. Table 1 shows the material constant of Deshpande and Fleck formulas for foam with different densities. Fig. 5 presents a stress-strain diagram of foam with a density of 330 and 430 kg/m^3 which were determined from the estimation method. These curves were used in crash simulations of foam-filled tubes.

Table.1 Estimated data for the foams with density of 230, 330, and 430 kg/m^3.

ρ (kg/m^3)	σ_p (MPa)	α	α_2	γ	β
230	1.5	2.12	22E6	1.41E6	4.8
330	2	2.12	23.5E6	1.8E6	5
430	2.7	2.12	25E6	2E6	5.2

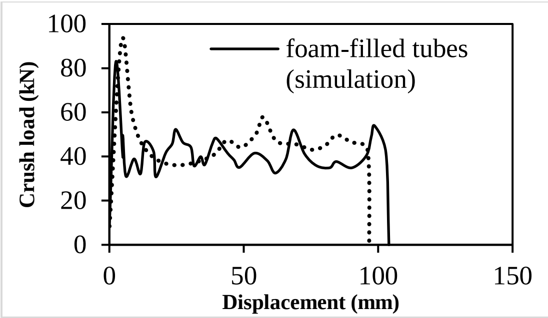

Figure 6. Numerical and experimental crash load-displacement diagram of foam-filled tubes.

Fig. 6 presents load-displacement curves of crash test of samples filled by the foam of 230 kg/m^3 density wherein experimental and estimated stress-strain curves were used for simulation of the foam. The curves demonstrated are based on the average values of three tests. The figure conveys that the simulation results being based upon experimental and estimated stress-strain curve well matches the experimental diagram. Therefore, this estimation method was applied to simulate the crash of foams with different densities.

Numerical simulations were used to find the crash reaction of tubes filled with gradient foams. To validate the results, the crash performance of tubes filled with individual foam with densities of 230, 330, and 430 kg/m^3 were simulated. Table 2 compares the EA and SEA of empty and filled tubes with different densities. Here it can be seen that the EA of filled tubes was increased when the foam density was increased. But the SEA first increased and then decrease when foam with a density higher than 230 kg/m^3 was used.

Fig. 7 showed absorbed energy-displacement curves of tubes filled with individual foams. The same results are presented in fig. 8 for two arrangements of tubes filled with gradient filler.

From this figure, it can be seen that the second arrangement which denser foam is closer to the impact mass can absorb more energy than the first arrangement.

Also, Table 3 compares the EA and SEA of the hollow tube filled with gradient foam with the second arrangement in which denser foam is closer to the impact mass. Here it can be seen that filling a tube with gradient foam can absorb more energy in lower weight than other individually foam-filled specimens.

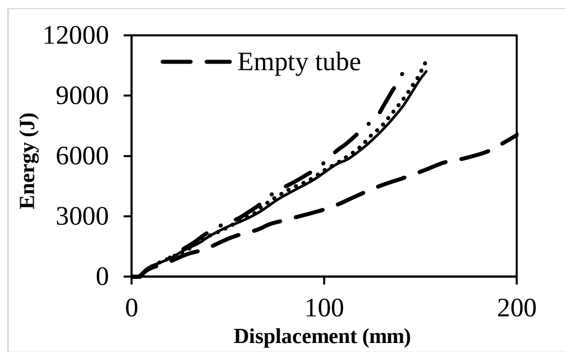

Figure 7. Absorbed energy diagram of the empty and the three tubes filled by foams with densities of 230, 330, and 430 kg/m3

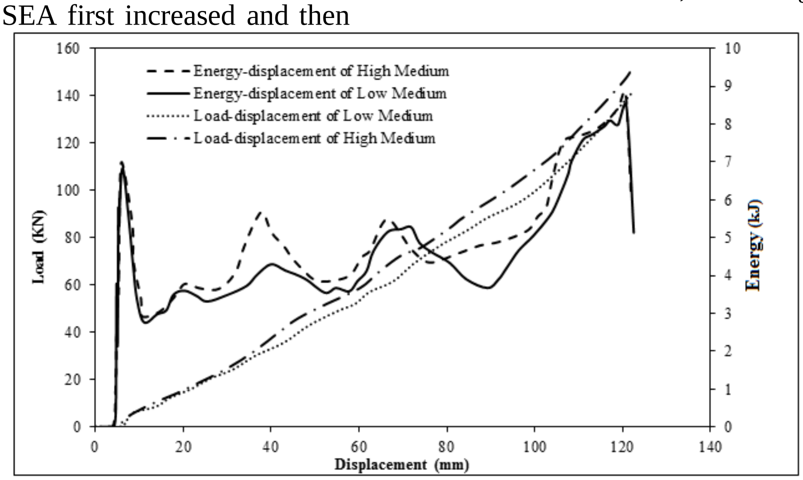

Figure 8. Load-displacement and energy-displacement diagram of the tubes filled by two samples filled by foams with different densities and arrangements.

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Table2. Influence of filling the aluminum tubes by different foams.

Model type	Energy (J)	Increase percent (%)	SEA (J/kg)	Increase percent (%)
Empty tube	4485	-	29113	-
Tube filled by foam with density of 230 kg/m^3	7344	63.7	31737	9.0
Tube filled by foam with density of 330 kg/m^3	7468	66.5	28118	-3.4
Tube filled by foam with density of 430 kg/m^3	8386	87.0	28034	-3.7

Table.3 Influence of filling the aluminum tube by gradient foams.

Model type	Energy (J)	Increase percent (%)	SEA (J/kg)	Increase percent (%)
Tube filled by three foams with different densities	9276	106.8	34926	20.0

5. Conclusions

In this paper, the influence of filling aluminum square tubes on energy dissipation characteristics has been investigated. Tubes were filled with metallic foams of different densities (230, 330 and 430 kg/m^3) either individually or together. The outcomes illuminated that filled tubes will absorb more energy (63.7-87% increase in magnitude) and this capacity is directly related to foam density. Furthermore, in the case of using three different foams simultaneously, additional improve in energy absorption is observed (106%).

By defining the specific energy absorption (SEA) as the magnitude of absorbed energy per unit mass, the tube filled with the foam with a density of 230kg/m3 demonstrated the greatest SEA among other foam densities (330 and 430 kg/m3) and greater than the hollow tube for 9%. This fact must be attributed to lower weight of the structure.

Filling the tube with foams with gradient densities causes a quasi-smart response by the foam filling of the tubes. The results showed that filling tubes with gradient foams presents higher EA and SEA compared to tubes filled with foam of one density.

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