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METHODOLOGY FOR THE DESIGN OF A LIGHTER FOAM FILLED TUBE STRUCTURE FOR IMPROVED CRASHWORTHINESS PARAMETERS SUBJECTED TO QUASI STATIC AXIAL COMPRESSION

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Abstract

The objective of this research paper is to propose a general methodology for the design of a foam filled tube which is lighter and more efficient in terms of Specific Energy Absorption (SEA) and Crush Force Efficiency (CFE) than an existing square aluminium empty tube for energy absorption application. The analytical expressions are used to determine the mean load during Quasi-Static axial compression loading of the foam filled tubes. The fact the thinner tubes develop lower peak load and the higher plateau stress of the foam results in higher mean load are used in the analytical calculations. Hence, the tube thickness and the foam density are chosen as the variables. The analytical results are compared with the experimental results of the initial design to choose the best design. An FE model is used as a tool to verify and predict the characteristics of the proposed design. Finally, the experimental model of the proposed design is built and its crushing behaviour is compared against the initial design. The foam used to fill the tube is high density Polyurethane (PU) foam. The experimental validation of intermediate designs is eliminated. Thus, the time of overall design cycle is reduced.

Keywords: Quasi-static compression, PU Foam, Mean load, Specific energy absorption, Crush force efficiency.

1. Introduction

The impact of transport vehicles is undesirable event but occurs frequently. One of the most important automotive parts for crash energy absorption is the crash

Nomenclatures

A	Area, m^2
b	Width, mm
e_0	Crush efficiency
E	Energy absorbed, J
l	Length, mm
P_{max}	Peak load, N
P_{mean}	Mean load, N
t	Thickness, mm
Wt	Weight, kg
x	Displacement, mm

Greek Symbols

δ	Deformation, mm
η	Structural effectiveness
σ_{pl}	Plateau stress, Pa
σ_y	Stress, Pa

Abbreviations

CFE	Crush Force Efficiency
SEA	Specific Energy Absorption, J/kg

box. In the automotive cars, the crash box is located at the front end of its front side frame as shown in Fig. 1. In the case of head-on collision, crash box progressively collapses and absorb impact energy prior to the other body parts minimizing the damage of the main cabin frame thus protecting the passengers. Hence, it is important to design the transport structures to withstand impacts and crashes.

There is a continuous demand for the design of light weight structures which puts greater demands on the designer since more aspects of design become critical as the working stresses become closer to the ultimate strengths of the material. These light weight structures eventually improve fuel economy and emissions along with reduced cost.

Aluminium alloy has high specific strength. Its property of corrosion resistance coupled with ability to be recycled easily makes it a good alternative material in automotive application. The prospect shows that in Europe, consumption of aluminium will increase to 30% by 2020 [1] as shown in Fig. 2.

There have been ongoing attempts to increase the energy absorption capability of tubular structures. The introduction of grooves at the predefined location along the tube not only reduce peak load but also it can stabilize the deformation behaviour [2]. The effect of slot geometry on the peak load was studied by Ghani et al. [3] and the results suggested that the peak load was significantly reduced and thus increasing the CFE.

In last few decades, foam-filled tubes are given much attention in crashworthiness applications. Aluminium foam and Polyurethane foam have gained popularity due to their high specific strength, cost and ease of manufacturing. Foam-like lattices are often used for cushioning, packaging or to protect against impact, utilizing the long, flat plateau of their stress-strain curves [4]. The typical stress strain curves for foams are shown in Fig. 3.

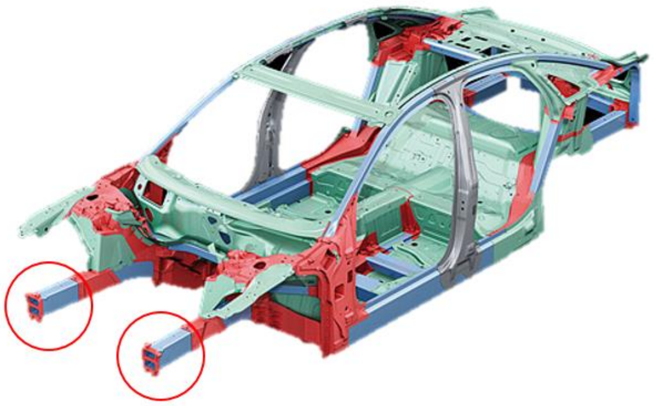


Fig. 1. Position of an automotive crash box in the body-in-white. (Source: Audi)

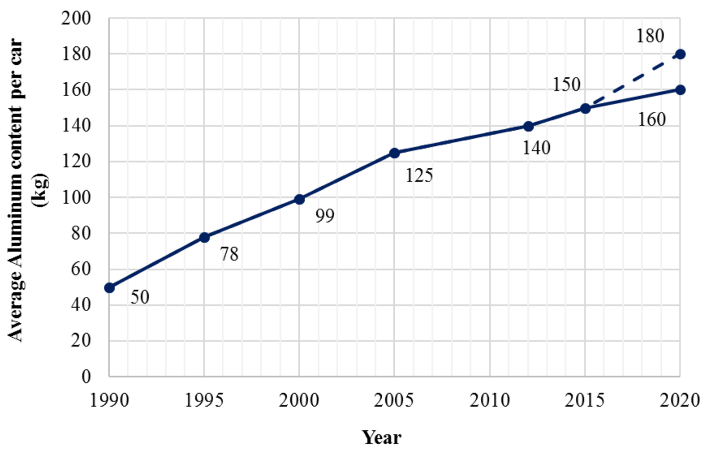


Fig. 2. Evaluation of average Aluminium content per car produced in Europe (Source: European Aluminium Association).

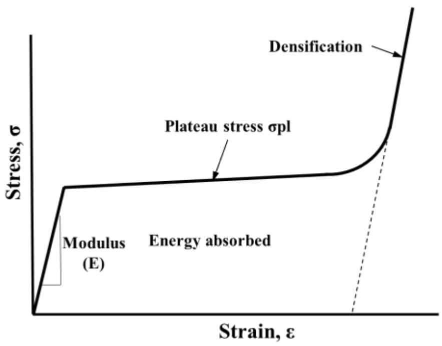


Fig. 3. Typical stress-strain curve for foams.

The foam-filled absorbers are especially favourable when high values of the energy absorption are required [5].

Many investigations have been carried out on tube structures filled with Polyurethane foams. The load versus displacement curve dependents on the foam density and the thickness of the tube (up to 2.5 mm) [6]. The tube with higher density foam can absorb more energy and the higher density of foam increases the number of folds in collapsed tubes [7].

Bargav et al. [8] studied the behaviour of Polystyrene and polyethylene foam filled tubes are studied and the results showed that the lower plateau stress of the foam does not contribute significantly in the increase of mean load.

In this paper, a general methodology for the design of foam filled tubes for energy absorption application is given. An alternate design for the 3.24 mm thick 50.8 mm wide square empty tube is proposed. The proposed design is a foam filled tube of thickness 2.25 mm.

2. Terminology of axial crushing

A typical Load-displacement response during an impact of tubular structures shows an initial peak load (P_{max}) which is usually much higher than the calculated mean load. Ideally the load must rise to a threshold value that will cause no harm to passengers and remain constant for the subsequent deformation [9].

There are several parameters that are significant in any crash analysis. They are: Peak Load (P_{max}), Energy absorption (E_a), Crush Force Efficiency (CFE), Specific Energy Absorption (SEA), Mean Load (P_{mean}), Crush efficiency (e_0) and structural effectiveness (η). These are important parameters for comparing the effectiveness of the energy absorber.

Energy absorption is calculated as the area under the load-displacement curve during impact process Eq. (1)

$$E_a = \int_0^x P\, dx \tag{1}$$

Specific Energy Absorption (SEA) is defined as the energy absorbed per unit weight of the absorber Eq. (2)

$$SEA = \frac{E_a}{Wt} \tag{2}$$

P_{max} and P_{mean} are used to describe the characteristics of an absorber. The P_{max} values provides an indication of the force required to initiate collapse and hence beging the energy absoption process. This maximum load should be within the acceptable limit. P_{mean} is defined as that constant load acting on the absorbed which results in same energy absorption for a given deformation as seen during actual loading case Eq. (3).

$$P_{mean} = \frac{E_a}{x} \tag{3}$$

An empirical formula for estimating P_{mean} for the case of square tubes subject to axial compression is given in Eq. (4) and shows that it is a function of the yield strength (σ_y) of the material of the tube, thickness (*t*) and width (*b*) of the tube [10].

$$P_{mean} = 13.06\, \sigma_y t^{5/3} b^{1/3} \quad (4)$$

The larger value of P_{mean} corresponds to larger value of the energy absorbed and the goal of the design is to maximize it. On the other hand, there is a continuous attempt to decrease the value of P_{max}. Accordingly, CFE is defined as the ratio of the mean force to the initial peak force Eq. (5)

$$CFE = \frac{P_{mean}}{P_{max}} \times 100 \quad (5)$$

Due to the characteristics of the collapse process, not all of the energy absorbing structure is utilized in the collapse process and a good indicator for the amount of material used during collapse is the crush efficiency and is defined as Eq. (6)

$$e_0 = \frac{\delta_f}{l} \quad (6)$$

For a structure that bottoms-out, i.e., all of the material in the structure has been deformed in a stable manner, the crush efficiency will be large. Conversely, for a poor energy absorbing design, the crush efficiency is small. The displacement used to calculate the crush efficiency is the value at which unstable collapse begins or when the specimen bottoms-out [11].

The structural effectiveness of a structure (η), allows comparisons to be made between structures having different material properties and is defined as [11]

$$\eta = \frac{P_{mean}}{A\sigma_y} \quad (7)$$

The increased number of lobes created by introducing foam filler causes the force level of the foam-filled tubes to be significantly higher than that of the combined effect of empty tube and foam separately [12], i.e., ($E_{filled} > E_{empty} + E_{foam}$). The increase in the number of lobes for the foam filled tubes is due to the interaction effect between the tube and the foam [13]. An equation for the mean crushing load ($P_{mean,filled}$) of foam filled columns by including contributions of the average crushing load of empty tube (P_{empty}), foam plateau stress (σ_{pl}) and interaction effect is given by [14]

$$P_{mean,filled} = P_{empty} + \sigma_{pl} b^2 + 5bt\sqrt{\sigma_{pl}\sigma_y} \quad (8)$$

where σ_y, b and t are the yield strength of the tube material, tube width and thickness. The second term of the right hand side of Eq. (8) accounts for the axial compression of the foam and the last term for the interaction effect.

3. Methodology

The initial 3.24 mm thick empty tube is subjected to quasi-static compression test. The parameters such as P_{max} and P_{mean}, energy absorbed, SEA and CFE are calculated from the load-displacement curve. These parameters are set as the design goal. The FE model of this initial empty tube is validated using the mechanical properties of the initial Aluminium tube. Using Eq. (8), the mean load for the filled tube is calculated analytically by varying the tube thickness, yield strength of the tube material and plateau stress of the foam. The optimum design is then chosen which weighs less than the target weight and the mean load should be higher than the target mean load. The proposed design is verified numerically using a validated FE model of foam filled tube. The stress-strain curve of the proposed density of the foam is obtained by the extrapolating the existing data. Finally, experimental load-displacement curve of the proposed design is compared against the initial design.

4. Design Process

4.1. Initial design

The initial design is a 50.8 mm wide square tube of thickness 3.24mm. The material of the tube is Aluminium 6063. The material of the tube has a yield strength of 116.8 MPa and *UTS* of 148.2 MPa. The height of the tube is 150 mm and weighs 243.1g. The tube is subjected to quasi-static axial compression loading and load versus deflection curve is recorded. The parameters such as peak load, energy absorbed and mean load are calculated. SEA and CFE are estimated using Eqs. (2) and (5). The targets for the weight and mean loads are established. The criteria for the selection is the design should weigh less than the target weight and the mean load should be higher than the target mean load.

4.2. Analytical calculation

The mean loads are calculated for each of the foam densities by varying the tube thicknesses. The stress-strain curve for foam of densities 60 kg/m^3, 100 kg/m^3 and 145 kg/m^3 [15] and for density 160 kg/m^3 [16] is shown in Fig. 4. The strain curve for foam of density 180 kg/m^3 is obtained by extrapolation. It is observed that the plateau stress increases with the increase in the density of the foam. The plateau stress is estimated as the stress value corresponding to strain of 0.4. The Plateau stresses for each of the foam densities is shown in Table (1).

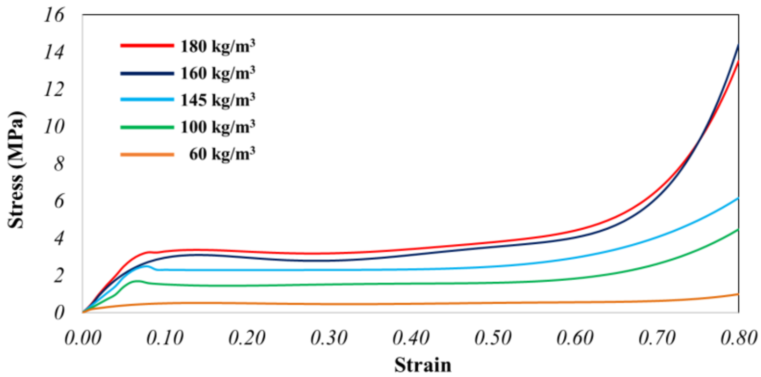


Fig. 4. Stress vs. strain curves for foams of various densities.

The variation of mean load with the tube thickness is plotted for different levels of yield strength of the tube material. The design meeting the criteria of weight and mean load are chosen. Among the selected designs the one with the highest mean load is selected for further study.

4.3. Experimentation

The load test under compression was carried out on a UTM (Model No: TUE-C-1000) of 100 tonne capacity. The test was carried out under Quasi-Static condition with the loading rate of 0.6 kN/s. The experiments were terminated manually when the specimens have crushed to beyond 100 mm from its initial length. The deformation pattern is observed. The load versus displacement curves is plotted and the energy absorbed during the compression process is calculated. The comparison is made between the empty and foam filled tube.

4.4. Numerical simulation

A finite element model of the initial tube is modelled using the material properties obtained from the tensile test. The tube is subjected to Axial compression under quasi-static condition. The finite element model is validated by comparing the load-displacement curve from simulation with the experimental curves.

A finite element model of the proposed design is developed based on the dimensions of the design predicted analytically. Analysis is performed using LS dyna. The tube is subjected to quasi-static compression loading. Load versus displacement curve is obtained and is compared with that of the initial design.

The finite element commercial software LS DYNA R971 with implicit solver (CONTROL_IMPLICIT_GENERAL) is used to simulate the quasi static crushing with time step of 0.001 s. Noting the fact that Aluminium is strain rate independent material, a suitable strain rate of 0.1/s is used in the analysis. The tubes were modelled in Hypermesh using 4-noded shell elements with 2 integration points along the thickness of the tube and are of element size 2 mm. The foams were modelled using Hex element of size 4 mm. The material model MAT_PIECEWISE_LINEAR_PLASTICITY (MATL24) was used to model Aluminium tubes and MAT_LOW_DENSITY_FOAM (MATL57) was used to model foam behaviour. The contact behaviour was modelled using the cards AUTOMATIC_GENERAL and CONTACT_INTERIOR respectively between Tube-Foam interface and self-interacting surfaces (internal behaviour of the foam). A rigid wall at the bottom of the tube representing the base of the UTM was created using RIGIDWALL_GEOMETRIC_CYLINDER (with Motion type: NONE) at bottom of the tube. The Tube specimen is placed on this base, constraining the motion of the tube at the bottom along the direction of loading. For Simulating the crusher, another rigid wall at the top of the tube representing the crosshead of the UTM is modelled using RIGIDWALL_GEOMETRIC_CYLINDER_MOTION (with Motion type: Displacement). The crosshead travel is controlled by assigning a LOADCURVE to the top rigid wall. The motion of the crosshead is along the axis of the tube crushing the tube by 100 mm during a time period of 5 s. The simulation was terminated after the rigid wall crushed the tube for about 100 mm. The FE model with Boundary Conditions is shown in Fig. 5.

Table 1. Plateau stress.

Foam Density (kg/m^3)	σ_{pl} (MPa)
60	0.483
100	0.561
145 [15]	2.325
160 [16]	3.096
180	3.414

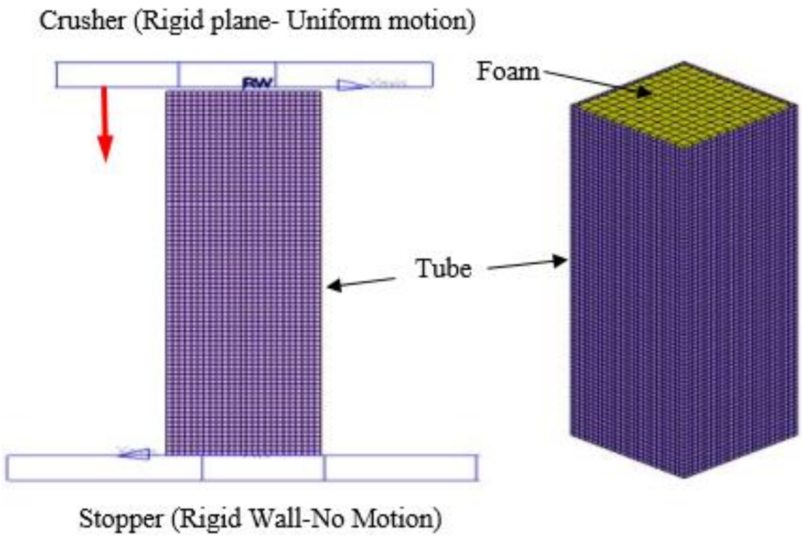


Fig. 5. FE Modelling showing boundary conditions.

4.5. Proposed design

The proposed design consists of 50.8 mm wide square aluminium tube of thickness 2.25 mm filled with PU foam of density 180 kg/m^3. An actual model is built which is subjected to axial quasi-static compression loading and load versus deflection curve is recorded. The parameters such as peak load, Energy absorbed and Mean load are calculated. Finally, the values of SEA and CFE are compared with that of the initial design.

5. Results and discussion

The experimental load - displacement curves of the initial 3.24 mm thick empty tube shows the peak load of 84.4 kN and Mean load of 42 kN. From the experimental curves, the *SAE* and CFE are obtained respectively as 17.2 kJ/kg and 50%. Weight analysis is performed analytically and the results are plotted as shown in Fig. 6. The density of 2700 kg/m^3 is used to calculate the weight of the tube. Here, the target is the weight of the initial design of the tube of 3.24 mm thickness. Any design falling above the target line indicates heavier than the initial design.

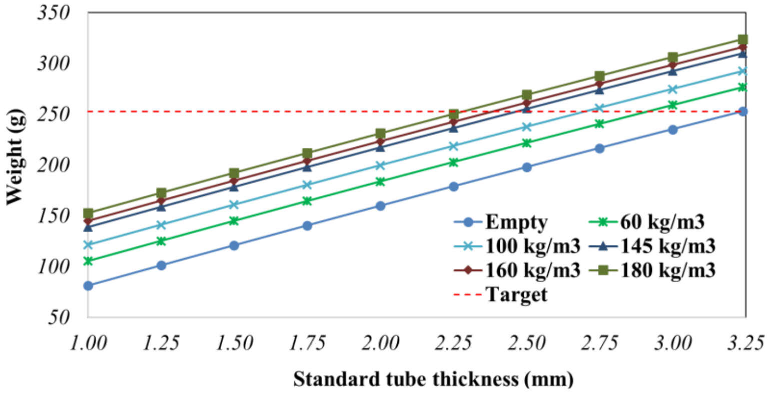


Fig. 6. Weight vs. thickness of empty and filled tube.

The variation of mean load with the tube thickness for tubes filled with PU foam of different densities for two levels of yield strengths is shown in Fig. 7. The target for the mean load is based on the experimental results of the initial design. Any design below the target line implies lower mean load and hence are neglected. It can be seen that the tube of thickness 2.25 mm filled with PU foam of density 180 kg/m^3, just meets the weight and mean load criteria. The details of designs meeting both the criteria of weight and mean load are shown in Table 2.

Table 2. Details of design meeting the weight and mean load criteria.

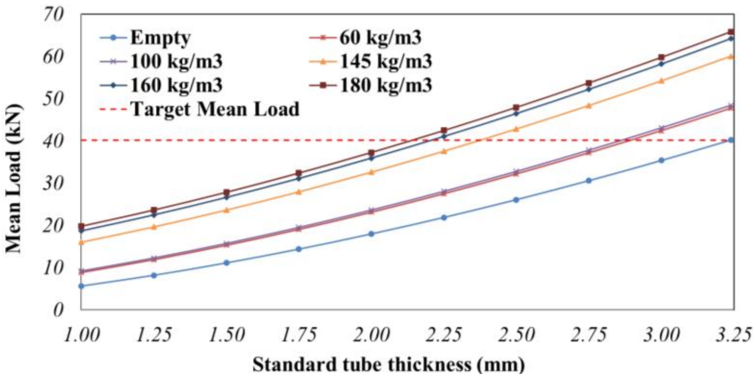
Design	Weight	P_{mean}	t (mm)	foam density (kg/m^3)	Ratio P_m/W_t (N/kg)
Target	253	40236	3.24	Empty	0.16
1	241	47483	2.75	60	0.20
2	222	41036	2.50	60	0.19
3	238	41672	2.50	100	0.18
4	237	45774	2.25	145	0.19
5	243	49491	2.25	160	0.20
6	223	42972	2.00	160	0.19
7	250	50961	2.25	180	0.20
8	231	44373	2.00	180	0.19

The plot of Mean load against the weight of the designs meeting the criteria is shown in Fig. 8. The plot shows that design 7 has the highest mean load and is lighter than the initial design. Hence it is chosen as the proposed design.

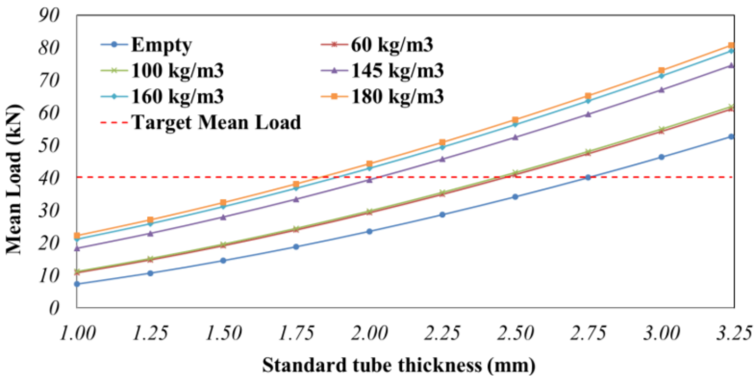
Tensile tests were performed as per IS 1608. Stress-Strain Curves for the materials of the tubes is shown in Fig. 9. The yield strength of the material of the proposed design is higher than that of the initial design.

The deformation patterns for both initial and proposed designs are shown in Fig. 10. Both specimens underwent systematic progressive folding forming 2-3 lobes. This type of crushing mode is known in as Diamond mode or non-Axisymmetric sequential folding and is the preferred mode of crushing for higher energy absorption.

The load versus displacement curves is shown in Fig. 11. It can be seen that the peak load is higher for the initial design which is due to the fact that the initial design is thicker than the proposed design. The energy absorbed which is the area under the load-displacement curves is more or less same for initial and proposed designs. The lower peak load for the proposed design is one of the desired objectives in the design of energy absorber in order to increase the CFE.



(a) σ_y=116.8 MPa.



(b) σ_y=153.1 MPa.

Fig. 7. Variation of Mean Load with the thickness of empty and foam filled tubes.

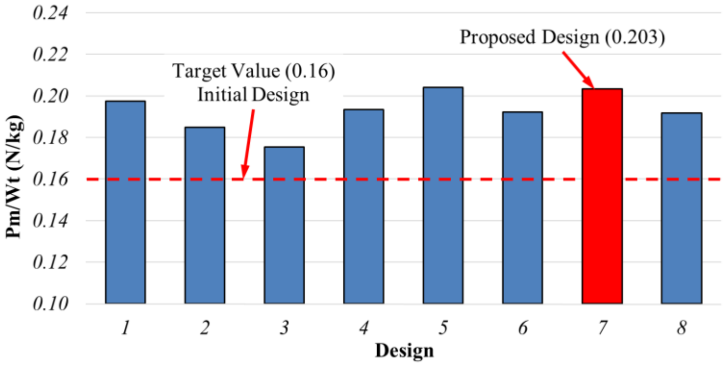


Fig. 8. Designs meeting the criteria of weight and mean load.

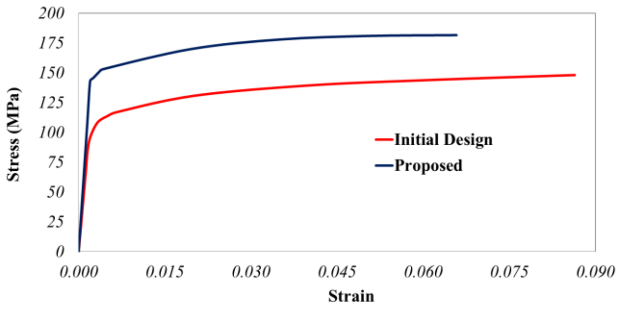


Fig. 9. Stress-Strain Curves for the materials of the tubes.

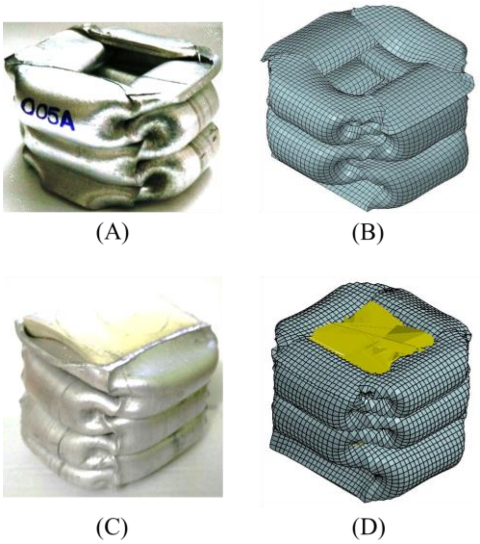


Fig. 10. Deformation pattern
Initial Design (Empty tube)- (A) Experimental; (B) FEM,
Proposed Design (Foam Filled tube)-(C) Experimental; (D) FEM.

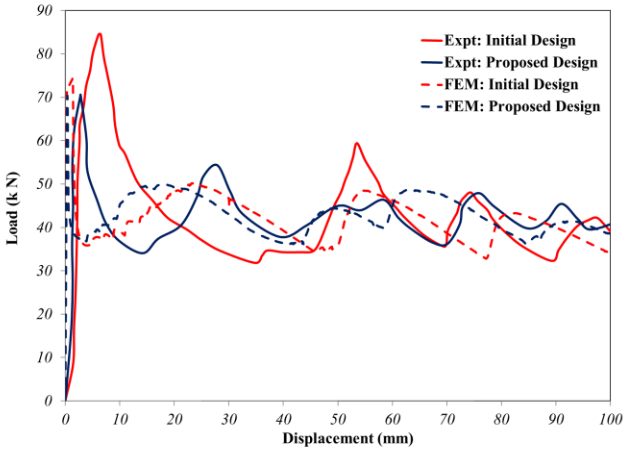


Fig. 11. Load vs. displacement curves.

The energy versus displacement curve is shown in Fig. 12. It is observed that during the start of the crushing process, the initial design absorbs higher energy than the proposed design. This is mainly due to the contribution of higher peak load observed for initial design. However, as the crushing process advances, the energy curves for both initial and proposed design tend to coincide and hence the overall energy absorbed is the same.

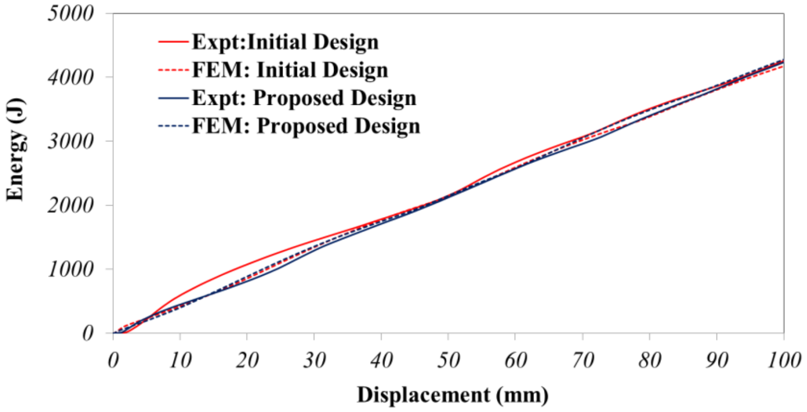


Fig. 12. Energy vs. displacement curves.

The experimental and numerical results of initial and proposed designs are shown in Table 3. FE results also predicted higher peak load for initial design than for proposed design. It also predicted higher *SAE* and CFE for proposed design.

Table 3. Experimental results.

	Weight (g)	Pmax (kN)	Pmean (kN)	SEA (kJ/kg)	CFE (%)
Initial Design	243.1	84.4 (74.4)	42.0 (42.0)	17.2 (16.8)	50.0 (56.0)
Proposed Design	237.2	70.6 (70.3)	42.0 (43.0)	17.9 (17.4)	60.0 (61.0)
Percentage improvement (%)	-2.4	-16.4	0.0	4.1	10

Note: Results shown within brackets are for FE simulations.

6. Conclusions

In this research paper, a foam filled tube is proposed as an alternative to the initial design of the tube for energy absorption application. The initial design was an empty tube 50.8 mm square tube of thickness 3.24 mm. The proposed design is a foam filled tube of thickness 2.25 mm and was filled with PU foam of density 180 kg/m^3. The results obtained from numerical analysis show similar trend as for experimental results. FE analysis shows higher peak load for initial design than for proposed design. It also predicted higher *SAE* and CFE for proposed design.

The proposed design is lighter than the initial design by 6 g and has better performance in terms of SEA and CFE. The reduction in weight of 6 g is significant in automotive applications which finally contributes to the better performance by improved specific fuel consumption. It is to be noted that there are many components in automobiles for energy absorption applications where such tube design can be replaced and the overall mass of the vehicle will come down. The improvement in CFE was achieved by lowering the peak load which is due to the reduction in thickness and maintaining the mean load level which is due to foam filling. The reduction in peak load is again a significant achievement for which the load transferred to the foundation is reduced and hence the foundations can further be optimized which results further reduction in overall weight and improvement in vehicle performance.

This research paper also outlines a general methodology for design of a lighter tube structures filled with foam which is an alternative to the thick empty tube. The methodology is summarized below:

- Obtain the Crashworthiness parameters of the initial design. Benchmark it. These will be the targets or the design goals.
- Obtain mechanical properties of the tube materials. These properties will be used in FE model validation.
- By using analytical expressions, perform the weight - mean load analysis and select the design that meets the targets. It includes proper tube thickness and foam density.
- Validate the design using FEA and fabricate the foam-filled tube structure

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