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Experimental and Numerical Simulation Investigation on Crushing Response of Foam-Filled Conical Tubes Stiffened with Annular Rings

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ABSTRACT

In this paper, crashworthiness characteristics of conical steel tubes stiffened by annular rings and rigid polyurethane foam are investigated. For this purpose, wide circumferential rings are created from the outer surface of the conical tube at some determined areas along tube length. In fact, this method divides a long conical tube into several tubes of shorter length. When this structure is subjected to axial compression, folds are shaped within the space of these annular rings. In this study, several numerical simulations using ABAQUS 5.6 finite element explicit code are carried out to study of crashworthiness characteristics of the empty and the foam-filled thin-walled conical tubes. In order to verify these numerical results, a series of quasi-static axial compression tests are performed. Moreover, load-displacement curves, deformation mechanism of the structure, energy absorption, crush force efficiency (CFE), initial peak load with different number of rings are described under axial compression. The results show that a conical tube with stiff rings as a shock absorber could be improved or adjusted the crushing mode of deformation and energy absorption ability.

Keywords : Conical; Annular rings; Polyurethane foam; Energy absorption; Maximum crushing load; CFE.

1 INTRODUCTION

CRASHWORTHINESS studies provide the mechanisms by which a proportion of impact energy is absorbed by the collapsing structure, whilst a small amount is transferred to the passenger. Impact energy absorption systems are used to protect automotive and its occupants from the effects of sudden impacts which occur during collisions. Many experimental and theoretical studies have been carried out for this design aim, dissipate the kinetic energy rather than convert and store elastically. Researches for high energy absorption per-unit weight or volume, which is very important in vehicles, may well be also justified. Energy absorbers are categorized into two major groups, the first is reversible energy absorber, like the hydraulic dashpot or elastic damper, and the second is irreversible energy absorber, like energy dissipation in plastic deformation of thin walled with different cross sections such as cylindrical, pyramidal, square, rectangular, triangular, hexagonal and frusta. During last two decades some essential studies have been carried out in the wide range of irreversible energy absorption devices such as thin-walled tube structures [1-6]. The results have been shown that circular tubes have the most energy absorption capacity and average force among all sections.

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Salehghaffari et al.[7] studied a new and efficient design method to encourage axisymmetric collapsing mode in long tubes. They found that cutting wide external grooves along circular metal tube length can significantly improve their energy absorption characteristics. Later Salehghaffari et al.[8] carried out experimentally two new design methods to improve energy absorption characteristics of cylindrical metal tubes. They have shown that the presented developed design methods were efficient in improving crashworthiness characteristics of cylindrical metal tubes such as sensitivity to external parameters like loading uniformity and direction, crushing stability, crush force efficiency and collapse mode while subjecting to axial compression.

Recently, increased focus has been given to tapered and conical tubes in which one or more sides of the tube are oblique to the longitudinal axis. Such structures are preferable to straight tubes since they are more likely to provide a desirable constant mean load–deflection response under dynamic impact loading. Furthermore, they are capable of withstanding oblique impact loads as effectively as axial loads [9]. Also, Mamalis et al. [10-12] compared a straight-circular with conical tube and found that a conical tube has more stable crush load–deflection curves and is more suitable as a structure since it has a higher Euler buckling load. Thin-walled conical tubes, based on their stable crush response and high energy absorption performance, have been widely used in various types of impact and crashworthiness applications. For instance, they are used in marine, civil and aeronautical structures [13-15]. As stated in Gupta et al. [16], one of the major advantages of using a conical tube compared to a cylindrical tube as an energy absorbing device is that it minimizes the chance of collapse via global bending. El-Sobky et al. [17] and Prasad [18] found that a conical tube, in general, collapses in the diamond mode expect for extremely low and high semi-apical angles. The experimental and numerical axial quasi-static crash tests on end-capped cylindrical and conical tubes have been performed by Ghamarian et al.[19]. They carried out the comprehensive parametric study to find the effect of the geometrical parameters of the tubes on their crashworthiness behavior. Rezvani et al. found that experimental deformation and load–displacement curves of conical tubes were affected by creating grooves on internal and external thetube wall [20-22]. Their results showed that grooves made crushing process easier and decreased maximum crushing load. The load fluctuations during deformation of the grooved specimens were also much less than those of the groove-less specimen. The grooves reduce the overall wave amplitude and the mean collapse load than the groove-less specimen, a property that is favorable in an efficient energy absorption device.

Recently, thin-walled structures have been filled with foams. Foams are a new class of materials with extremely low densities and unique combination of excellent mechanical which help the stability of the tubes and improve the energy absorption. Many studies were performed using foams under both quasi-static and dynamic axial crushing load [21, 23-27]. The behaviour of foam-filled thin walled tubes with different cross-sectional shapes under axial compression was investigated by Thornton [28]. The results indicated that mean crushing load of the foam-filled tube was more excess than the sum of those individual tube and the foam due to the interaction effects between both components. The effect of filling thin-walled circular metal tubes with low density polyurethane foam under both quasi-static and dynamic axial loading was studied by Reid et al. [29]. They have been shown that the stability of crushing was improved by using polyurethane foam. Therefore the crushing load will increase, either due to the crush resistance of the foam or because of its influence on changing the buckling mode of the tube wall from diamond or mixed mode to ring mode. Darvizeh et al.[30] performed analytical and experimental investigations on the energy absorption characteristics of grooved thich-walled circular tubes filled with low density. The results showed that grooved thick-walled tubes filled with low strength foams could offer favorable energy absorption capacity and stability. Euler buckling was also prevented due to the grooves and specific energy absorption was increased approximately twice that of the empty tubes. Ahmad et al. [31] have been investigated crushing response of empty and foam-filled conical tubes by using experimental and numerical simulation. The crushing mode of deformation occurred in non-axisymmetric mode at ring-less conical tubes (Fig. 1). Their results also showed that the energy absorption of a foam-filled conical tubes were significantly higher than that of the tubes without foam.

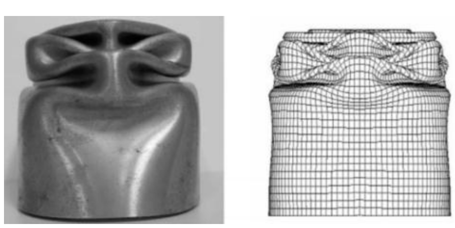


Fig.1
Typical deformation modes of foam-filled conical tube [31].

Many thin-walled circular tubes stiffened with annular rings in order to improve energy absorption, specific energy absorption and to avoid Euler buckling [7, 8, 30, 32]. The quasi-static axial collapde response of cylindrical tubes which were externally stiffened with rings investigated by Salehghaffari et al.[33]. They performed multi-objective optimization problem to find an optimal geometric design for rings. The effect of initiator on the foam-filled thin-walled circular tubes with stiffened annular rings has been performed to increase the energy absorption and prevent the sudden force applied to the main part of the automotive and its occupants by Rezvani et al. [34]. The results have shown that by installing the initiator at the end of foam-filled circular tubes, an increase in the energy absorption and a decrease in the sudden intense force would be occurred. Therefore, this present study develops the effectiveness of creating annular rings outer surface of the empty and foam-filled conical tubes under axial loading. In this innovative idea experimental studies and numerical simulations are carried out to investigate the effects of (i) filling the tubes with polyurethane foam, (ii) varying the number of rings along the tube length, on crashworthiness characteristics and crushing mode of the thin-walled conical tubes under axial compression.

2 EXPERIMETAL SETUP

2.1 Material and fabrication process

Present work investigates the axial quasi-static crushing response of empty and foam-filled conical tubes stiffened by annular rings on the outer surface of tube. The test specimens were made of the mild steel. Furthermore, in order to obtain accurate material information, typical quasi-static engineering tensile stress-strain curve of tube material tested in accordance with ASTM B557M standard, which is shown in Fig. 2. The elastic modulus of this material is $E = 210GPa$, the density is $\rho = 7800 Kg/m^3$ and the Poisson ratio is $\nu = 0.29$. Also, the yield stress and ultimate stress for this alloy are equal to $\sigma_y = 265MPa$ and $\sigma_u = 421MPa$, respectively.

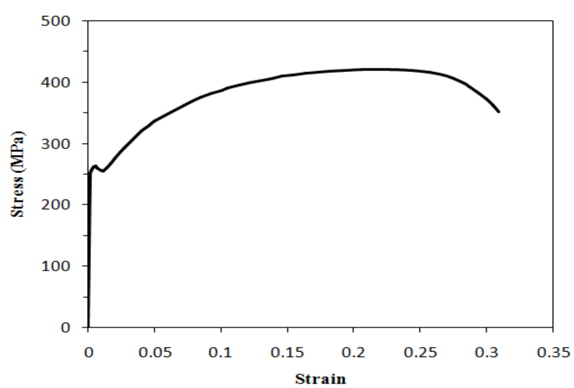


Fig.2
Typical engineering tensile stress–strain curve for mild steel.

In order to produce rigid polyurethane foam use in energy absorption application, rigid PU foam with density of $\rho = 192 Kg/m^3$ is manufactured. In this study, rigid PU foam was manufactured by simultaneously mixing of two liquids, polyisosynate and polyol. These liquid are mixed together for about 60 seconds and poured into tubes. In order to maintain the internal pressure created during the chemical reaction of two liquid, both ends of the tubes are closed tightly by fixture. Fig. 3 shows stress–strain curve of the rigid PU foam, obtained by compressing a cubic specimen quasi-statically along one direction, according to the ASTM D1621-94 standard. As is evident from the figure, the compression curves of polymeric foams contained three distinct regions: linear elastic region, plateau region and densification region. At small strains, usually <5%, the behavior is linear elastic, with a slope equal to the Young modulus of the foam. Therefore, the linear elastic region, deformation is controlled by cell walls bending or stretching. This region is followed by a plastic collapse region which proceeded by spreading the local deformation and collapse to non-deformed region of the sample. This region is characterized by a load plateau with either a constant value or slight increase with displacement. In densification region, cell walls start to touch each other and the sample is densified.

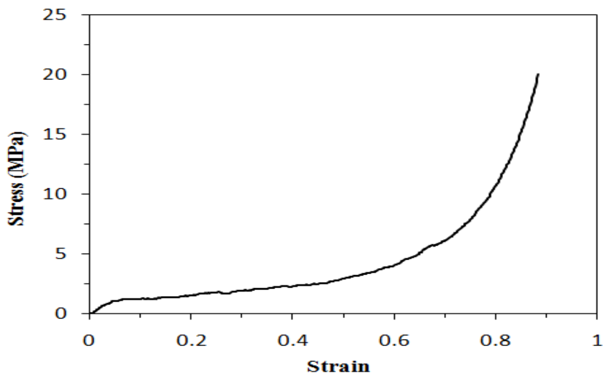


Fig.3
Stress-strain curves of polyurethane foam with density of $192 Kg/m^3$.

We manufactured several conical tubes to ascertain the effect of annular rings and rigid PU foam on energy absorption. The specimens are made by machining with a lathe, as illustrated in Fig. 4. The details and dimensions of the manufactured specimens are given in Table 1.

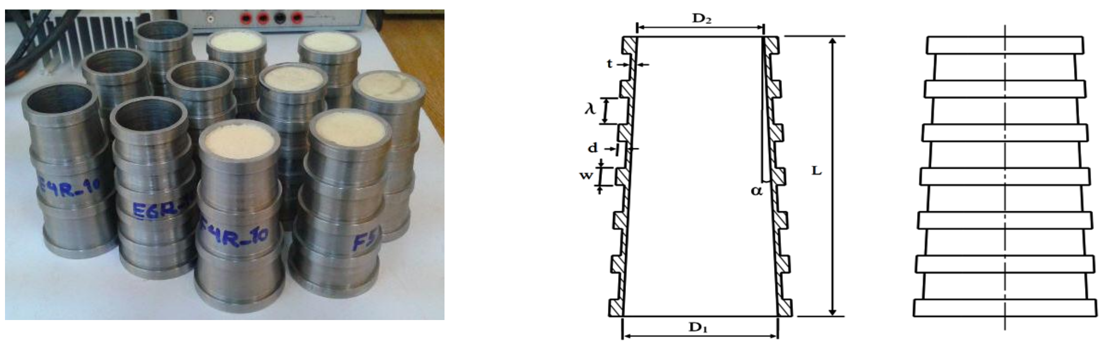


Fig.4
Thin-walled conical tubes stiffened by rings with its detailed design.

Table 1
Specimen's dimension.

Specimen		$D_2(mm)$	$D_1(mm)$	$L(mm)$	$t(mm)$	$W(mm)$	$d(mm)$	$\lambda(mm)$	$\theta(°)$
C-E-4R	Empty	36	55.98	100	1	7	2	24.25	4
C-E-5R	Empty	36	55.98	100	1	7	2	16	4
C-E-6R	Empty	36	55.98	100	1	7	2	11.6	4
C-F-4R	Foam-filled	36	55.98	100	1	7	2	24.25	4
C-F-5R	Foam-filled	36	55.98	100	1	7	2	16	4
C-F-6R	Foam-filled	36	55.98	100	1	7	2	11.6	4

2.2 Test procedure

In order to study the effects of rings number, density of rigid PU foam on the behavior of the conical tubes, quasi-static axial compression tests are performed in 30 tons DARTEC testing machine. To ensure the repeatability of the experimental results, the tests were repeated three times for each of the empty and the foam-filled tubes. The crosshead went down to compress the top of the conical tube with the constant strain rate of 1.42×10^{-3} (equal to 10 *mm/min*). As observed in Fig. 5, the specimen is placed between the plates of test machine without any additional fixing. Pictures of the specimen after the experimental crushing test are shown in Fig. 6. Furthermore, the load-displacement curves, resulting from experimental tests, are plotted in Fig. 7.

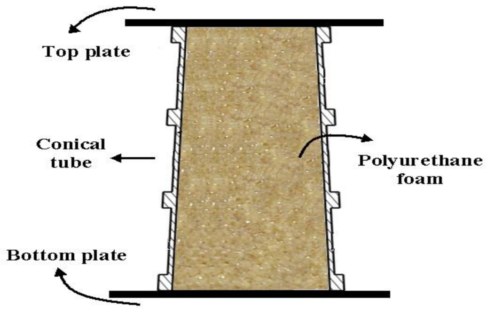


Fig.5
A typical arrangement of conical tube between the steel plate.

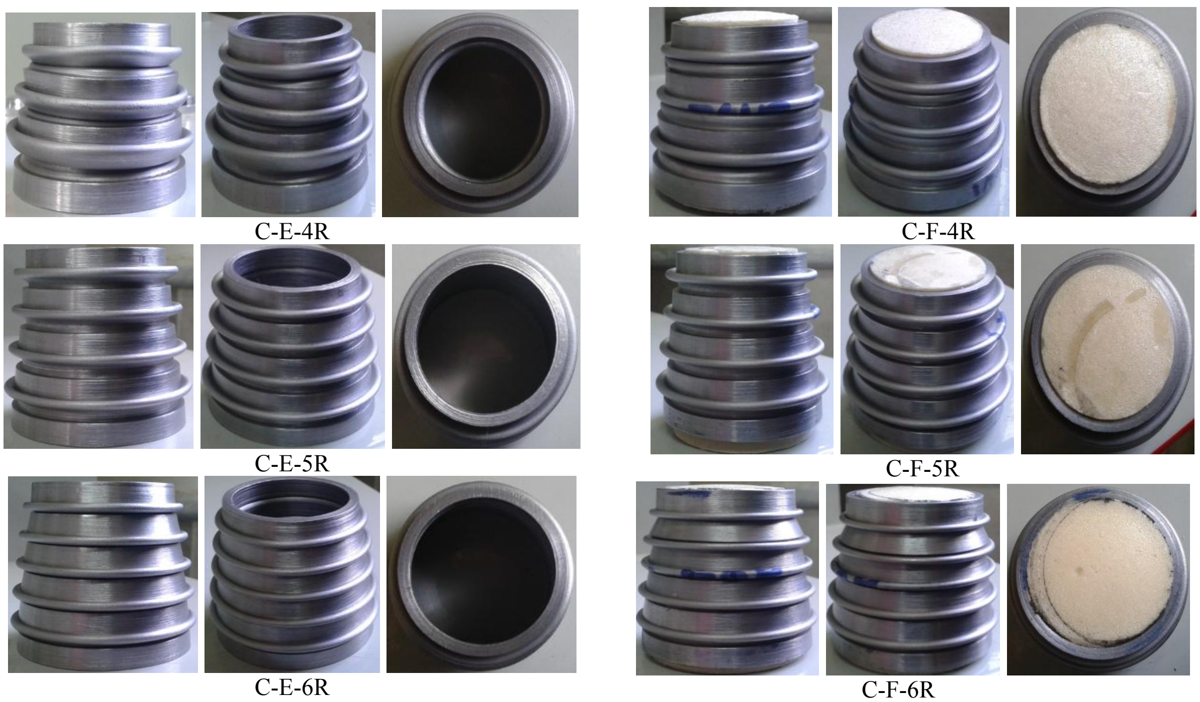


Fig.6
Crushed shapes of specimens after axial compression test.

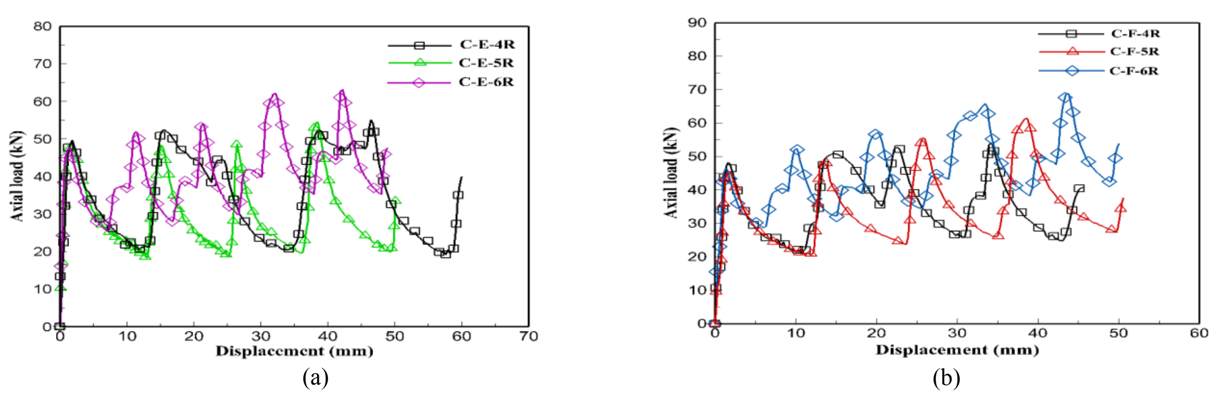


Fig.7
Load–displacement curves obtained from experimental test for specimens: a) Empty conical tube, b) Foam-filled conical tube.

3 DESCRIPTION OF FINITE ELEMENT MODEL

The nonlinear explicit FE code ABAQUS 5.6 is used to simulate the crashing behavior and energy-absorbing capability of the foam-filled conical tubes stiffened by annular rings subjected to quasi-static axial compression. When a quasi static load is applied, the loading rate is almost constant and therefore, the strain rate and inertia effects are neglected. As it can be observed from Fig. 8, in this model, two rigid plates consisting of movable rigid and constrained rigid plate are placed on the top and on the bottom of the conical tube, respectively. The element type of rigid bodies is R3D4 (4-node 3D bilinear rigid quadrilateral). The bottom rigid plate is constrained in all degrees of freedom, while the top rigid plate is described as a rigid body free to translate only along the y axis with a constant velocity of 10 *mm/min*.

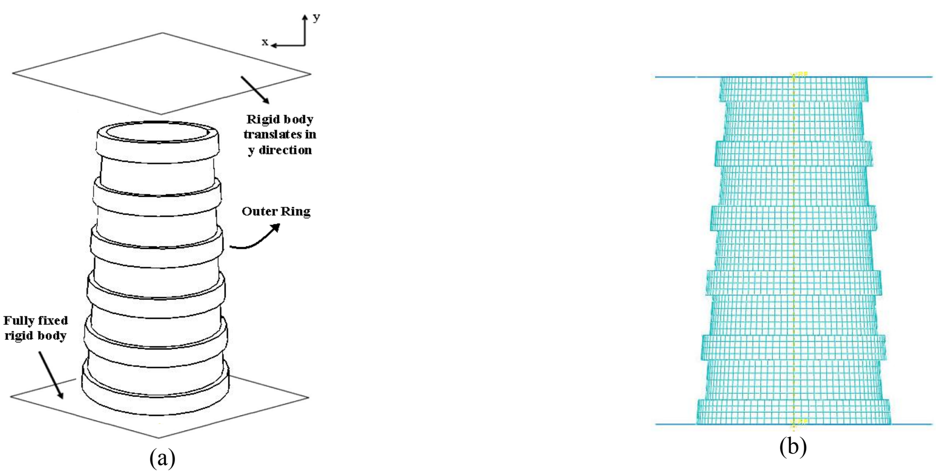
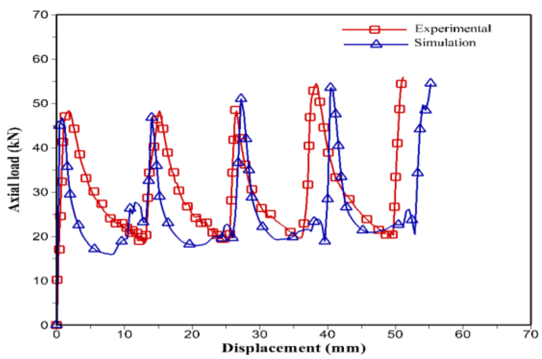
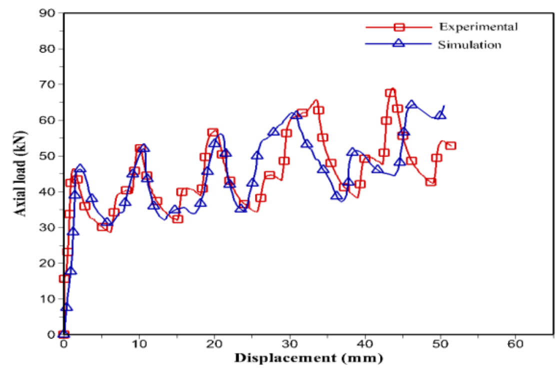


Fig.8
a) Loading and boundary condition arrangement, b) Finite element mesh.

The steel tubes are modeled by C3D8R (8-node linear brick, reduced integration, hourglass control) and the rigid PUfoam is modeled using 8 noded solid elements with the reduced integration techniques in combination with hourglass control. After convergence, element sizes of 1.8 *mm* for the conical tubes and 3 mm for the foam filler are found to produce acceptable results. The material properties of steel tubes and foam filler are defined as linear elastic followed by nonlinear work hardening in the plastic region. The isotropic plasticity and crushable foam material models are used to model the conical tube and foam filler respectively. The true stress-strain curves for both filler and steel materials are used to introduce the approximated true stress-plastic strain data points in numerical simulations. Table 2. lists the material parameters for polyurethane foam based on the crushable foam model [35, 36]. A 'self-contact' interface is also selected to simulate the collapse of specimens when elements of the tube wall contact each other. To define contact between the movable rigid plate, constrained rigid plate, foam and the tube, "surface to surface" contacts is defined with coefficient of friction of 0.3. Simulation of the problem with test conditions in software is extremely time-consuming and so we need an economical solution for solving the problem. Being aware of natural frequency, we can find a down limit of time so that we consider the situation quasi-static. For selecting the load rating in quasi-static condition, we start with the biggest load rating (time is equal to the period of natural frequency). Then, by checking the internal energy and kinetic energy, we find the best load rating scale. Therefore, the numerical model predicts the load–displacement curves observed in experimental results with a reasonable accuracy, as shown in Fig. 9. The numerical simulations of collapsed shapes after axial crushing are illustrated in Fig. 10. The results for energy absorption, mean crushing load, initial peak load and CFE are also summarized in Table 3. and compared with experimental data. As can be seen, there is an acceptable agreement between FE analysis and experimental tests.



(a)



(b)

Fig.9
Comparison between load–displacement curves obtained from experimental test and numerical simulation for specimens: a) C-E-5R, b) C-F-6R.

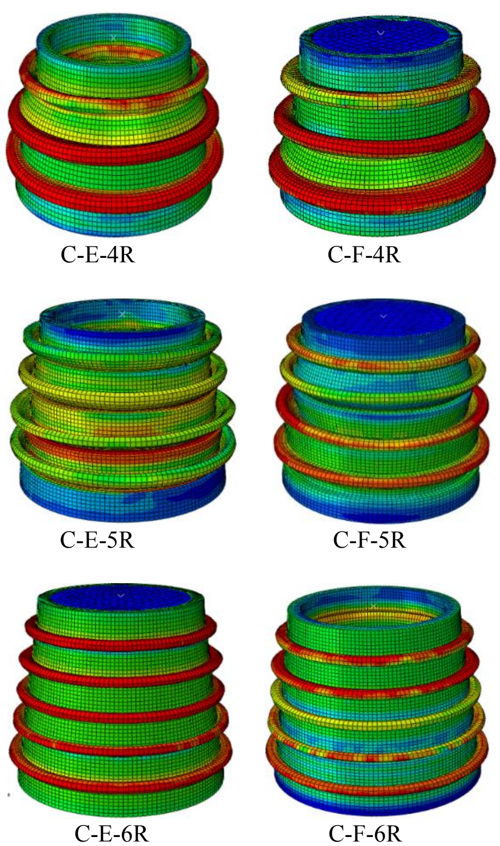


Fig.10
Final crushing mode of thin-walled ring-stiffened conical tube after axial loading.

Table 2
Material parameters of crushable foam model considered for polyurethane foam [35, 36].

Density	Elastic parameters		Plastic parameters	
$\rho^f\ (kg\ /\ m^3)$	$E^f\ (Mpa)$	ν^f	k^f	ν_p^f
192	16.5	0	1	0

Table 3
Experimental and numerical results after axial crushing.

Specimen	Experimental results				Simulation results			
	$p_m\,(kN)$	$E\,(\mathrm{J})$	$p_{\max}\,(kN)$	CFE (%)	$p_m\,(kN)$	$E\,(\mathrm{J})$	$p_{\max}\,(kN)$	CFE (%)
C-E-4R	36.77	1841.52	49.74	74	35.55	1791.28	47.11	75
C-F-4R	37.72	1921.13	48.41	78	40.95	1950.14	49.59	82
C-E-5R	29.70	1521.84	48.40	61	25.14	1389.31	46.78	54
C-F-5R	34.21	1764.32	45.71	75	35.96	1796.22	44.34	81
C-E-6R	40.81	2063.01	47.72	85	37.16	1962.36	46.14	80
C-F-6R	45.07	2320.27	46.54	97	45.39	2292.45	46.30	98

4 ENERGY ABSORBER CHARACTERISTIC CRITERIONS

4.1 Absorbed energy

Absorbed energy, is defined as an integration of a load-displacement curve as beneath:

$$E_{abs} = \int p d\delta \tag{1}$$

where p and δ are the crush force and crush distance, respectively.

4.2 Mean crush load

The mean crush load is defined by the division of absorbed energy E_{abs} and total deformation δ, respectively, as below:

$$p_m = \frac{E_{abs}}{\delta} \tag{2}$$

4.3 Crush force efficiency

The crush force efficiency is defined as below:

$$CFE = \frac{p_m}{p_{\max}} \tag{3}$$

where p_m and $p_{\max}$ are the mean crush load and initial peak load, respectively. For an ideal energy absorber this parameter should be as close to 100% as possible.

5 RESULTS AND DISCUSSION

5.1 Crush mode

In this study, the effects of the rigid PU foam, the rings number on the crushing behavior and the energy absorption of conical thin-walled tubes are investigated. The annular rings on the outer surface are divided the conical tube into several shorter conical portions. Hence, the crushing mode of the tube is stable by increasing the rings number.

In Figs. 6 and 10, the final deformations are shown in all of the specimens, resulting from the experimental test and the numerical simulation under axial compression. The comparison between them are shown that the presented FE model can simulate the crushed shape of the shock absorber with sufficient accuracy. According to Fig. 6, the specimens of 5 and 6 rings are crushed the concertina (axisymmetric) mode of deformation. By increasing the distance between two rings, for example specimen of C-E-4R, the stretching between the metal rings increases. As the distance between two rings (λ) increases, the concertina mode of deformation cannot be sustained and buckling mode is changed to the diamond mode. Furthermore, due to the non-axisymmetric collapse, the possibility of rupture increases by increment in the crushing length.

5.2 The load-displacement curve

The crashworthiness parameters are calculated by experimental test and numerical simulation to study the effects of polyurethane foam and the rings number. As shown in the Fig. 7, after an initial elastic deformation in the conical tube, load increases during the plastic deformation due to strain hardening process and declines dramatically while a plastic fold shapes completely within the two rings space. By further increasing the crush force, second fold is created within the other two rings space, and this procedure is repeated until the specimen is completely crumpled.

5.3 The effect of rings number and polyurethane foam

Fig. 11 shows the absorbed energy, maximum crushing load and crush force efficiency for the empty and foam-filled conical tubes stiffened by annular rings. As shown in the figure, the crashworthiness parameters of this shock absorber have a minimum value at the rings number. Therefore, there is a critical value for the distance between two rings and according to the application viewpoint, to produce a good energy absorber, the rings distance is selected far from the λ_{cr} . The critical rings distance is obtained as $\lambda_{cr} = 16mm$ with the rings number of $N = 5$.

From Fig. 11, it is evident that by increasing the rings number the crush force efficiency of all specimens for both the empty and foam-filled tubes increases. Furthermore, the tubes filled with foam have significantly larger crush force efficiency than the empty ones. The largest and lowest value of crush force efficiency in the foam-filled conical tubes, 78 and 97%, belong to specimens of C-F-4R and C-F-6R, respectively. As it can be observed from Fig. 11, by increasing the rings number for both the empty and foam-filled conical tubes, the absorbed energy increases and it has a minimum value at the rings number of $N = 5$. The maximum crushing load for both structures also decreased with increasing the number of rings. The maximum crushing load in the empty conical tubes is greater than the foam-filled ones. Therefore, the foam-filled conical tubes are preferred because they have a higher energy absorption capacity and crush force efficiency in comparison with the empty ones.

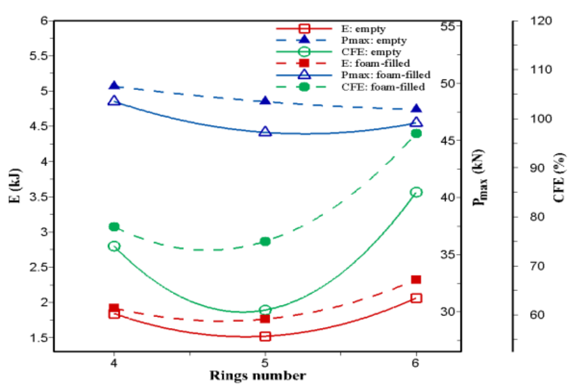


Fig.11
Comparison between empty conical tubes with foam-filled ones stiffened by rings on energy absorption, maximum crushing load and crush force efficiency.

6 CONCLUSIONS

In this paper, the effects of rings number and rigid polyurethane foam on the crashworthiness characteristics of thin-walled conical steel tubes stiffened with annular rings were studied. Quasi static experimental test and numerical simulation were performed for calculation of the energy absorption, initial peak load and crush force efficiency (CFE) of the empty and foam-filled tubes. Load–displacement curves and deformation mechanism of the tube under axial compression were described. Comparison between the experimental and the numerical simulation results showed the high accuracy and reliability of the presented numerical model. The main conclusions and design information are summarized below:

- Creating uniform plastic hinges by using of annular rings on the outer surface of tube.
- Controlling the mode of collapse in the tubes by creating annular rings on the outer surface of tube and rigid PU foam under axial compression.
- The crushing mode of the tube was stable by increasing the rings number.
- Choosing the appropriate number of rings in order to enhance the energy absorption performance.
- Increasing the energy absorption, CFE in the foam-filled conical tubes stiffened by annular rings in comparison with the empty ones.
- Decreasing the maximum crushing load in the foam-filled conical tubes stiffened by annular rings in comparison with the empty ones.
- Choosing the thin-walled conical tubes with stiffened annular rings would be a favorable choice for a controlled behavior of energy absorption device.

REFERENCES

[1] Abramowicz W., 2003, Thin-walled structures as impact energy absorbers, *Thin-Walled Structures* **41**: 91-107.

[2] Nagel G., Thambiratnam D., 2004, A numerical study on the impact response and energy absorption of tapered thin-walled tubes, *International Journal of Mechanical Sciences* **46**: 201-216.

[3] Nagel G., Thambiratnam D., 2005, Computer simulation and energy absorption of tapered thin-walled rectangular tubes, *Thin-Walled Structures* **43**: 1225-1242.

[4] Aljawi A., Alghamdi A., Abu-Mansour T., Akyurt M., 2005, Inward inversion of capped-end frusta as impact energy absorbers, *Thin-Walled Structures* **43**: 647-664.

[5] Guillow S., Lu G., Grzebieta R., 2001, Quasi-static axial compression of thin-walled circular aluminium tubes, *International Journal of Mechanical Sciences* **43**: 2103-2123.

[6] Alavi Nia A., Haddad Hamedani J., 2010, Comparative analysis of energy absorption and deformations of thin walled tubes with various section geometries, *Thin-Walled Structures* **48**: 946-954.

[7] Mokhtarnezhad F., Salehghaffari S., Tajdari M., 2009, Improving the crashworthiness characteristics of cylindrical tubes subjected to axial compression by cutting wide grooves from their outer surface, *International Journal of Crashworthiness* **14**: 601-611.

[8] Salehghaffari S., Tajdari M., Panahi M., Mokhtarnezhad F., 2010, Attempts to improve energy absorption characteristics of circular metal tubes subjected to axial loading, *Thin-Walled Structures* **48**: 379-390.

[9] Reid S., Reddy T., 1986, Static and dynamic crushing of tapered sheet metal tubes of rectangular cross-section, *International Journal of Mechanical Sciences* **28**: 623-637.

[10] Mamalis A., Johnson W., 1983, The quasi-static crumpling of thin-walled circular cylinders and frusta under axial compression, *International Journal of Mechanical Sciences* **25**:713-732.

[11] Mamalis A., Johnson W., Viegelahn G., 1984, The crumpling of steel thin-walled tubes and frusta under axial compression at elevated strain-rates: some experimental results, *International Journal of Mechanical Sciences* **26**: 537-547.

[12] Mamalis A., Manolakos D., Saigal S., Viegelahn G., Johnson W., 1986, Extensible plastic collapse of thin-wall frusta as energy absorbers, *International Journal of Mechanical Sciences* **28**: 219-229.

[13] Gupta N., Sheriff N.M., Velmurugan R., 2006, A study on buckling of thin conical frusta under axial loads, *Thin-Walled Structures* **44**: 986-996.

[14] Sheriff N.M., Gupta N., Velmurugan R., Shanmugapriyan N., 2008, Optimization of thin conical frusta for impact energy absorption, *Thin-Walled Structures* **46**: 653-666.

[15] Spagnoli A., Chryssanthopoulos M., 1999, Elastic buckling and postbuckling behaviour of widely-stiffened conical shells under axial compression, *Engineering structures* **21**: 845-855.

[16] Gupta N., Prasad G.E., Gupta S., 1997, Plastic collapse of metallic conical frusta of large semi-apical angles, *International Journal of Crashworthiness* **2**: 349-366.

[17] El-Sobky H., Singace A., Petsios M., 2001, Mode of collapse and energy absorption characteristics of constrained frusta under axial impact loading, *International Journal of Mechanical Sciences* **43**: 743-757.

[18] Prasad G.E., Gupta N., 2005, An experimental study of deformation modes of domes and large-angled frusta at different rates of compression, *International Journal of Impact Engineering* **32**: 400-415.

[19] Ghamarian A., Zarei H., 2012, Crashworthiness investigation of conical and cylindrical end-capped tubes under quasi-static crash loading, *International Journal of Crashworthiness* **17**: 19-28.

[20] Rezvani M.J., Nouri M.D., 2013, Axial crumpling of aluminum frusta tubes with induced axisymmetric folding patterns, *Arabian Journal for Science and Engineering* **39**: 2179-2190.

[21] Damghani Nouri M., Rezvani M.J., 2012, Experimental investigation of polymeric foam and grooves effects on crashworthiness characteristics of Thin-walled conical tubes, *Experimental Techniques* **38**: 54-63.

[22] Rezvani M., Damghani Nouri M., 2015, Analytical Model for Energy Absorption and Plastic Collapse of Thin-Walled Grooved Frusta Tubes, *Mechanics of Advanced Materials and Structures* **22**: 338-348.

[23] Seitzberger M., Rammerstorfer F.G., Gradinger R., Degischer H., Blaimschein M., Walch C., 2000, Experimental studies on the quasi-static axial crushing of steel columns filled with aluminium foam, *International Journal of Solids and Structures* **37**: 4125-4147.

[24] Ahmad Z., Thambiratnam D.P., 2009, Dynamic computer simulation and energy absorption of foam-filled conical tubes under axial impact loading, *Computers & Structures* **87**: 186-197.

[25] Abramowicz W., Wierzbicki T., 1988, Axial crushing of foam-filled columns, *International Journal of Mechanical Sciences* **30**: 263-271.

[26] Yamada Y., Banno T., Xie Z., Wen C., 2005, Energy absorption and crushing behaviour of foam-filled aluminium tubes, *Materials transactions* **46**: 2633-2636.

[27] Reddy T., Wall R., 1988, Axial compression of foam-filled thin-walled circular tubes, *International Journal of Impact Engineering* **7**:151-166.

[28] Thornton P.,1980, Energy absorption by foam filled structures, *SAE International* 800081.

[29] Reid S., Reddy T., Gray M., 1986, Static and dynamic axial crushing of foam-filled sheet metal tubes, *International Journal of Mechanical Sciences* **28**: 295-322.

[30] Darvizeh A., Darvizeh M., Ansari R., Meshkinzar A., 2013, Effect of low density, low strength polyurethane foam on the energy absorption characteristics of circumferentially grooved thick-walled circular tubes, *Thin-Walled Structures* **71**: 81-90.

[31] Ahmad Z., Thambiratnam D., 2009, Crushing response of foam-filled conical tubes under quasi-static axial loading, *Materials & Design* **30**: 2393-2403.

[32] Adachi T., Tomiyama A., Araki W., Yamaji A., 2008, Energy absorption of a thin-walled cylinder with ribs subjected to axial impact, *International Journal of Impact Engineering* **35**: 65-79.

[33] Salehghaffari S., Rais-Rohani M., Najafi A., 2011, Analysis and optimization of externally stiffened crush tubes, *Thin-Walled Structures* **49**: 397-408.

[34] Rezvani M.J., Jahan A., 2015, Effect of initiator, design, and material on crashworthiness performance of thin-walled cylindrical tubes: A primary multi-criteria analysis in lightweight design, *Thin-Walled Structures* **96**:169-182.

[35] Ghamarian A., Abadi M.T., 2011, Axial crushing analysis of end-capped circular tubes, *Thin-Walled Structures* **49**: 743-752.

[36] Mirfendereski L., Salimi M., Ziaei-Rad S., 2008, Parametric study and numerical analysis of empty and foam-filled thin-walled tubes under static and dynamic loadings, *International Journal of Mechanical Sciences* **50**: 1042-1057.