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Improving the crashworthiness characteristics of cylindrical tubes subjected to axial compression by cutting wide grooves from their outer surface

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This paper investigates a new and efficient design method to encourage axisymmetric collapsing mode in tubes with higher length than the critical length to collapse in concertina mode when subjected to axial compression. In this innovative design method, wide circumferential grooves are cut from the outer surface of the tube at some determined areas along tube length. When this structure is subjected to axial compression, folds are shaped within the space of these grooves. In fact, this method divides a long tube without the potential of collapsing in axisymmetric mode into several tubes of shorter length, being assembled together coaxially, with the potential to collapse in axisymmetric mode. In the present study, several numerical simulations using LS-DYNA finite element explicit code are performed to study the energy absorption characteristics of this structure under quasi-static compression and to determine the effect of various geometric parameters on its collapsing modes of deformation. In order to verify these numerical results, some experimental tests are performed. An analytical model to predict mean crush force of the new shock absorber is also developed in the present study. Through experimental, numerical and analytical studies, major parameters in design are identified, and typical modes of deformation that may occur during axial compression of tubes with wide external grooves are characterised. Also, load–displacement history and deformation mechanism of the structure under axial compression are described. It has been shown that with consideration of design parameters, characterised in this paper, cutting wide external grooves along circular metal tube length can significantly improve their energy absorption characteristics.

Keywords: circular tubes; external grooves; energy absorbers; collapse modes

Nomenclature

D_i	Inside diameter of tube
D_o	Outside diameter of tube
d	Depth of grooves
E_T	Total absorbed energy of the shock absorber
G	Total weight of the shock absorber
H	Total crushed length
L	Tube length
M_o	Plastic bending moment per unit length
N	Number of grooves
P_m	Mean crushing load
S_e	Specific energy absorption factor
t	Tube wall thickness
W	Length of thick portions
W_D	Required energy for creation of one fold
W_T	Work done by external work
λ	Length of wide grooves
ρ	Material density
σ_o	Flow stress
σ_u	Ultimate tensile stress
σ_y	Yield stress
ϕ	Bending angle

1. Introduction

These days, with the ever increasing transport vehicle speed, incidents of fatal traffic accidents are growing significantly. Therefore, over the past decades, to reduce human injuries and decrease financial burdens, designs of transport structures with the most favourable crashworthiness characteristics have received significant attention by researchers and designers. Recently, as a part of these comprehensive researches to improve crashworthiness characteristics of transport vehicles, many experimental and theoretical studies with the aim of designing devices of high potential to dissipate the impact energy into another form of energy have been performed. These devices have become famous as energy absorbers. In fact, there are two different types of energy absorbers: reversible ones, which may convert kinetic energy into pressure energy in comprehensible fluids or into elastic strain energy in solids, and irreversible ones, mostly dissipating kinetic energy into plastic deformation energy.

Various structures like circular and square tubes, octagonal cross-section tubes, spherical shells, frusta, taper tubes, s-shaped tubes, composite tubes, honeycomb cells

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and foam-filled and wood-filled tubes may be used as irreversible energy absorbers. Amongst them, metallic cylindrical tubes have attracted much more attention because of their high stiffness and strength combined with the low weight and ease of manufacturing process, which lead to low cost of the energy-dissipating device. These structures can dissipate kinetic energy into various forms of plastic deformation, friction or tearing energy, using different energy-dissipating methods [5,15].

One of these energy-dissipating methods is to compress a tube against rigid die, which may result in several different modes of deformations. Amongst them, the most favourable deformation mode from the point of energy absorption is inversion mode, occurring within the specific range of geometrical dimensions for a tube and die as well as specific friction condition between them. Therefore, several researchers have performed both theoretical and experimental studies to improve the possibility of shaping the successful inversion mode while compressing a tube against rigid die [6,9,18,20,22]. Despite the fact that inversion mode of deformation provides a favourable constant crush force, only a half of the tube length can contribute in plastic deformation and energy absorption. This low stroke efficiency of inversion mode of deformation along with its strong sensitivity to external parameters like the loading condition limits the application of this energy-dissipating method in various structures subjected to axial impact load. Axial splitting and curling of tubes against canonical dies is the other recognised method of energy dissipation, having been studied by a number of researchers [14,21,25]. In this energy-dissipating method, a great percentage of tube length contributes to the dissipation of external energy, and the collapsing force is relatively steady. However, this method provides low crush load, and again its performance is strongly affected by external parameters like loading direction. Recently, expansion of circular tubes by rigid tubes has been introduced as a new and efficient method of kinetic energy dissipation without any serious sensitivity to loading direction and other external parameters [24]. However, this method has low stroke efficiency and provides low specific energy absorption, while lower values of mean crush load are required to protect the structure subjected to axial impact load from damages. Combination of different methods of energy dissipation, inversion and axial crushing of metal tubes in a single collapsible shock absorber was also investigated for the first time by Chirwa [8].

Among various methods of energy dissipation, axial crushing of tubes under impact load has been the most common means of dissipating impact energy, as this method provides a reasonable constant load and fairly high crush force efficiency. In comparison to other energy-dissipating methods, a greater percentage of tube material can contribute in plastic deformation while it crushes axially, providing high specific energy absorption factor, which has great significance in protection of structures with weight limitation. However, all of these favourable crashworthiness characteristics can be achieved when the tube crushes in axisymmetric or concertina mode of deformation. A large number of experimental and theoretical studies [2,4,7,11,13,16,17,19] have shown that depending on tube geometry, material properties of tube, boundary conditions and loading conditions, it may buckle in other modes of deformation such as diamond, Euler and mixed modes. In fact, the Euler mode of deformation, occurring when the tube length is greater than the critical length for a given diameter and thickness of tube, should be avoided in crashworthiness applications, as this mode provides inefficient and unreliable characteristics of the energy absorber. On the other hand, while diamond and mixed modes are considerably more likely to occur than other collapsing modes for a common dimension of tube, little changes in loading and boundary conditions can easily cause a tube with the potential of crushing in diamond or mixed modes to buckle in the Euler mode of deformation. In addition, even under fixed boundary and loading conditions, miscalculation is inevitable when the shock absorber is designed to collapse in diamond or mixed mode of deformation, as the exact shape of the tube in these cases is quite unpredictable. Concertina or axisymmetric collapsing mode has none of these limitations and has been considered as the most reliable and efficient deformation mode in dissipating impact energy. Unfortunately, experimental studies show that among the wide range of tube dimensions, it can crush in concertina mode within the considerable limited region of L/D and D/t diagram, over which the tube has less contribution in energy dissipation compared to larger ones.

However, contrary to comprehensive researches performed so far to study the effect of various parameters on the collapsing mode of circular thin-walled structures subjected to axial impact or quasi-static loads, there have been a very limited number of researches with the aim of extending the concertina collapsing region. In a recent research [23], cutting an initial circumferential edge groove outside the tube and using one- and two-circumferential stiffeners have been suggested as two design methods to activate the axisymmetric plastic buckling mode. In this research, several validated finite element (FE) simulations have also shown the extension of the axisymmetric collapsing region after using these methods in the L/D and D/t diagrams. In another research [10], cutting circumferential grooves alternatively inside and outside the tube at predetermined intervals has been introduced as a solution to force the plastic deformation to occur at these predetermined intervals along the tube. This proposed method could be a good candidate as a controllable energy absorption element, but it reduces the amount of material participating in plastic deformation and energy absorption. Cutting a given tubular structure in several portions and coaxially assembling them by separating a non-deformable disc is the other solution to encourage the axisymmetric mode in axial crushing of tubes, as investigated by Abdul-Latif et al. [1].

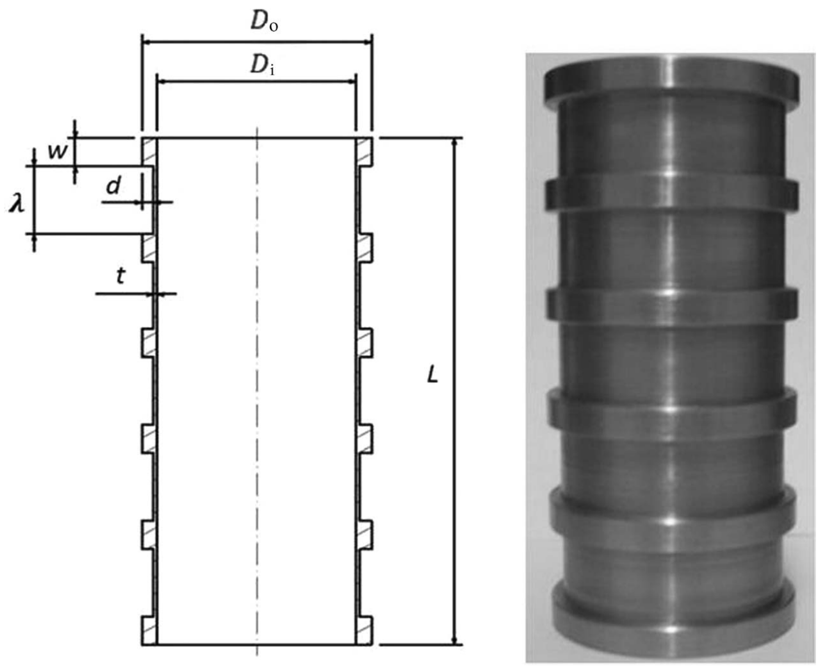

Figure 1. Shape of a tube with wide external grooves with its detailed design.

In the present study, a new design method to control energy absorption characteristics of thin-walled structures under compression is introduced. In this method, cutting wide circumferential grooves from the outer surface of a tube converts it into several thinner and shorter tubes, which are assembled together coaxially. When this structure is subjected to axial compression, it collapses and folds are shaped within the space of external grooves. Figure 1 shows the shape of a tube with wide external grooves along with its detailed design. As a matter of fact, the present method divides a long tube without the potential to collapse in axisymmetric mode into several shorter tubes with the potential to collapse in axisymmetric mode. In this study, the axially quasi-static compression of the discussed shock absorber is simulated numerically using LS-DYNA explicit FE code. To verify numerical simulations and study the crashworthiness characteristics of the shock absorber, some compression tests of specimens with different geometric parameters are established in the present study. Through the observation, obtained from numerical and experimental studies of the shock absorber, major parameters in design are identified, and typical modes of deformation that may occur during axial compression of tubes with wide external grooves are characterised. Also, load–displacement history and deformation mechanism of the structure under axial compression are described. Based on these results, an analytical model to predict the mean crush load of the energy absorber under axial compression is presented. Acceptably, theoretical results are in good agreement with experimental results. It has been shown that, unlike inversion, axial splitting and crushing of thin-walled structures, external parameters do not affect the energy absorption characteristics of the studied shock absorber significantly, resulting in a high safety factor. Furthermore, cutting wide grooves along the tube length can considerably decrease the elastic resistance of the structure against axial impact load, which leads to a considerable increase in crush force efficiency of the shock absorber. As another advantage of the proposed design method, with the consideration of major geometric parameters in design, various mean crush forces, required to protect different structures subjected to axial impact load, can be obtained.

2. Experiments: description and results

Seamless steel tubes of commercial quality with an *outside* diameter of 62 mm and an *inside* diameter of 48 mm were machined to the required size and length of 144 mm. Then, wide circumferential grooves of different lengths and depths were cut from the outer surface of the tubes to prepare specimens for compression tests. Afterwards, cut surface areas of all specimens were also grinded to improve their surface quality. Figure 1 shows the typical shape of the prepared specimen for compression test, and Table 1 shows geometric parameters of each prepared specimens. As it is realised from this table, three geometric parameters, namely depth, length, and number of grooves, vary from one specimen to another. Also, to show the extension of axisymmetric collapsing mode in axial crushing of tubes by using the proposed design method, two groove-less tubes

Table 1. Specimens' dimensions.

Specimen no.	L (mm)	D_o(mm)	D_i(mm)	t (mm)	d (mm)	λ(mm)	W (mm)	N
A1[a]	144	54	52	1	–	–	–	–
S1	144	60	52	1	3	16.8	10	5
S2	144	60	52	1	3	13.5	9	6
S3	144	60	52	1	3	19.2	8	5
A2[a]	144	55	52	1.5	–	–	–	–
S4	144	60	52	1.5	2.5	16.8	10	5
S5	144	60	52	1.5	2.5	13.5	9	6
S6	144	60	52	1.5	2.5	19.2	8	5

[a]Groove-less specimen.

of similar L/D and D/t with that of tubes with cut wide grooves were also made for compression tests.

In order to obtain the material data of the specimens, a quasi-static material test was performed on a strip cut from them using a standard tensile test machine, and the resulting stress–strain relationship is shown in Figure 2. The elastic modulus of this material was $E = 210$ GPa and its density was $\rho = 7800$ kg/m^3. It is assumed that the mechanical properties of this steel alloy are not sensitive to strain rate at room temperature.

In this study, all quasi-static axial compression tests of specimens were performed on a 20-ton ZWICK hydraulic testing machine at a nominal crushed speed of 10 mm/min, and the specimens were crushed between parallel steel plates of the test machine without any additional fixing. A repeated compression test of each specimen was also performed to ensure the validity of experimental results.

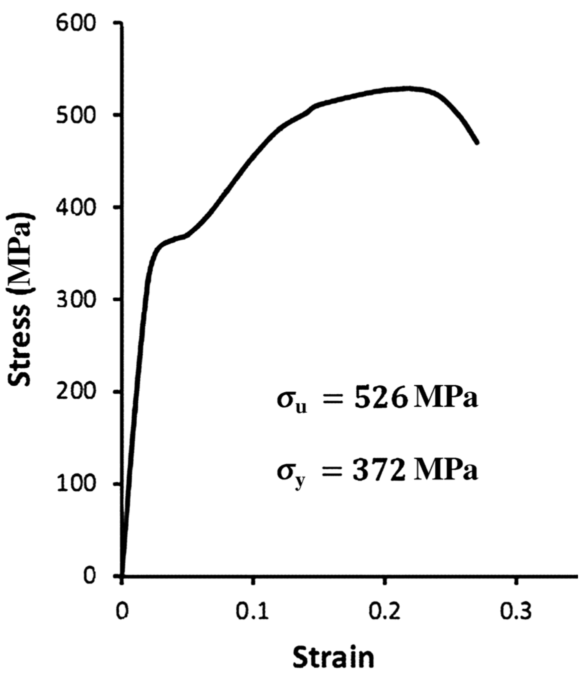

Figure 2. Stress–strain relationship of specimen's material.

The final shapes of all the specimens after the compression test are shown in Figures 3 and 4, and their corresponding load–displacement curves, resulting from experimental tests, are plotted in Figure 5. Values of energy absorption and mean crushing loads were calculated by measuring the area under the obtained load–displacement curves, and the summary of test results is shown in Table 2. Comparison between crushed shapes of groove-less tubes with that of grooved tubes show the reliability of the proposed design method developed in the present study, as it significantly extends the limits of axisymmetric collapsing modes of deformation under axial compression. In the present study, these experimental results are also used to verify theoretical studies of the proposed shock absorber.

3. FE simulation

In the present study, numerical simulation of axial crushing of all specimens has been carried out using explicit FE code LS-DYNA, version 971. To perform these numerical simulations, the basic 3D geometry of the model (Figure 6) was firstly created according to the user code [12]. As it can be observed from Figure 6, in this 3D model, a rigid plate is placed on both the top and bottom of the tube with wide external grooves. The bottom rigid plate is constrained in all degrees of freedom, while the top plate is described as

Figure 3. Groove-less specimens after axial crushing.

Figure 4. Grooved specimens after axial crushing.

a rigid body with five constraints, assuming that only displacement along the vertical axis is allowed with a constant velocity of 1 m/s. Thus, under this loading condition, no inertial effects on forming mechanism are likely to occur and no dynamics effects in deformation mechanics need to be taken into account.

To define contact between the movable rigid plate and the tube as well as the contact between the tube and the fixed plate, a 'surface-to-surface' contact type is used. In this 'surface-to-surface' contact description, the tube nodes are defined as 'slave' nodes, whereas rigid plate nodes are defined as 'master' nodes. A 'single-surface' interface is also selected to simulate the collapse of specimens when elements of the tube wall contact each other, creating a new interface. This contact type uses nodal normal projections; hence, it prevents the elements from penetrating the specimen surface during collapse.

Solid elements (hex8) are used to model all the analysed tubes with wide external grooves. After convergence, an element size of 1 mm is found to produce suitable results. The material properties of the deformable tube are defined as elastic-plastic with kinematic hardening (material model type 3). Both fixed and movable plates are simulated using the 'material model type 20'. To remove Hourglass mode, 'Control-Hourglass' is used, and for Hourglass viscosity type (IHQ), Equation (6) is considered as a suitable option.

All numerical simulations of specimens were performed on a Pentium PC 3.2 GHz. Figure 7 shows numerical simulation of collapsed shapes of all specimens after axial crushing.

4. Analytical formulations

The presented analysis of the shock absorber is based on Chirwa's theoretical solution [8] to estimate the work done by axisymmetric buckling in thin-walled inversion-buckling metal tubes with slight modification.

Assuming that one fold is shaped within the space of each wide groove, cut from the outer surface of the tube, during axial crushing, a simple collapse model of the structure to predict the mean crush force is shown in Figure 8. As it is realised from this figure, similar to the model proposed by Alexander [3] for axial crushing of thin-walled structures, during the formation of one fold, three circumferential plastic hinges occur within the space of one external

Table 2. Experimental, analytical and numerical results after axial crushing.

			P_m(KN)			E_T(J)		
Specimen no.	H (mm)	Buckling mode[a]	Test	FE	Analytic	Test	FE	Analytic
A1	72	D	26.52	–	–	1910.45	–	–
S1	62	5C	28.38	27.21	23.41	1759.56	1687.02	1732.71
S2	63	6C	29.23	29.15	23.75	1841.49	1836.45	1639.04
S3	75	3C → D	30.40	28.86	23.81	2280.00	2164.50	2047.86
A2	64	D	44.06	–	–	2820.17	–	–
S4	62	5C	53.03	52.79	42.57	3287.86	3272.98	3150.59
S5	60	6C	57.36	57.29	45.04	3441.60	3437.40	3108.23
S6	73	5C	50.32	50.19	42.21	3673.36	3663.87	3630.67

[a] C, concertina; D, diamond.

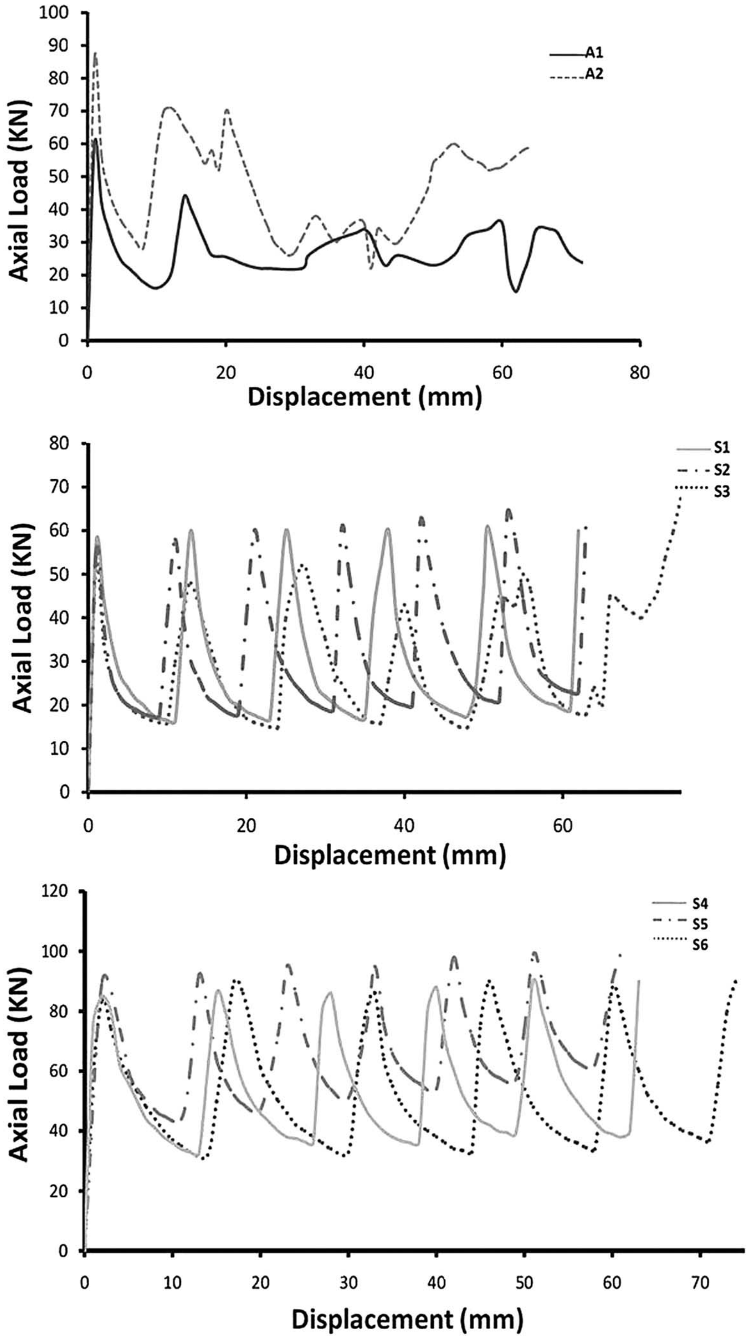

Figure 5. Load–displacement curves of all specimens derived from experimental tests.

wide groove and the metal between its surface experiences stretching. Therefore, the involved energies in the formation of one axisymmetric convolution between one-grooved spaces of the shock absorber are

(1) Work done due to the plastic deformation (W_D)
(2) Work done by external work ($W_T = P_m \times (\lambda - 2t)$)

The interrelation of the process variables is determined by setting the energy balance to zero. Thus,

$$\sum W_i = 0, \text{ or } W_D - W_T = 0 \tag{1}$$

The work done due to the plastic deformation is achieved only through axisymmetric buckling process in one-grooved space, W_{BL}. Thus,

$$W_D = W_{BL}, \tag{2}$$

where the work by axisymmetric buckling is done in two separate plastic deformations, namely, (1) work done in

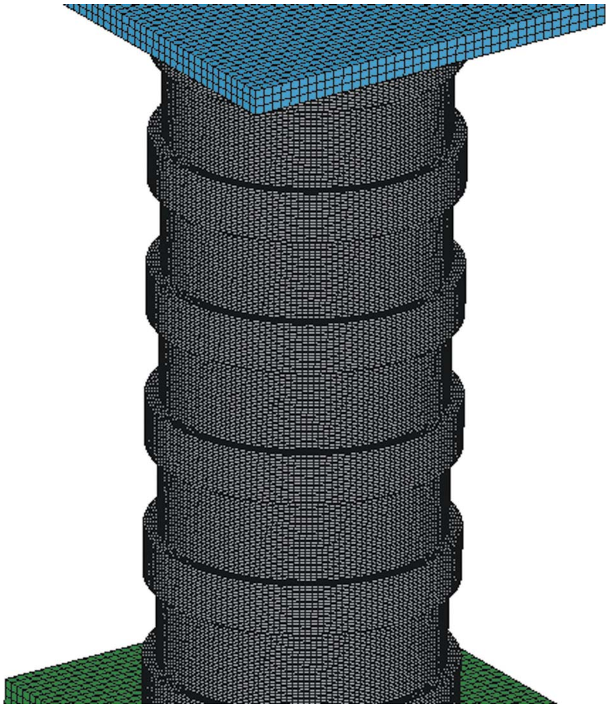

Figure 6. The basic 3D geometry of FE model.

formation of three assumed stationary circumferential plastic hinges (Figure 8), W_{BBL}, and (2) work done in expansion due to the high circumferential stresses between the hinges, W_{EBL}. Hence,

$$W_{\mathrm{BL}} = W_{\mathrm{BBL}} + W_{\mathrm{EBL}}. \tag{3}$$

Substituting Equations (2) and (3) into Equation (1), the energy balance equation becomes

$$W_{\mathrm{T}} = W_{\mathrm{BBL}} + W_{\mathrm{EBL}}. \tag{4}$$

Before analysing Equation (4), it is appropriate to consider the following assumption to the analysis:

(1) The material of grooved tube behaves as rigid, perfectly plastic with an average flow stress σ_{o}.
(2) Interaction between bending and stretching in the yielding criterion is negligible.
(3) Change in wall thickness at any point of the grooved tube is small.
(4) Circumferential hinges in axisymmetric buckling are considered to be stationary.
(5) Angle ϕ increases from zero to a value of approximately $\pi/2$. Therefore, the buckling process in the formation of one convolution within a groove space of the shock absorber is complete when the straight-sided convolutions come in contact.

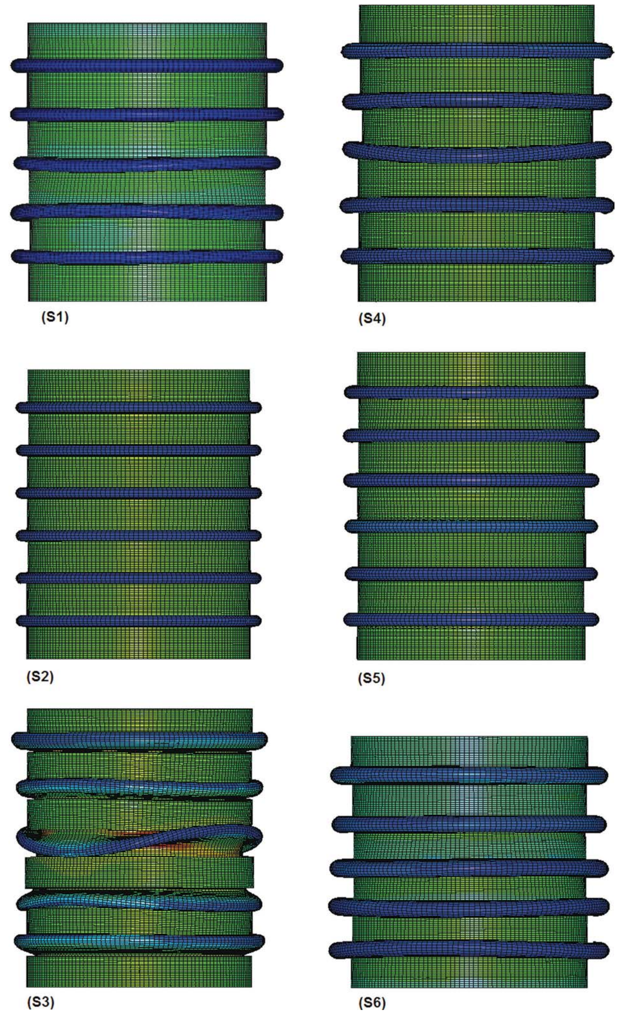

Figure 7. Crushed shapes of grooved specimens derived from numerical simulation.

The energy absorbed in three stationary circumferential plastic hinges (hinges 1, 2 and 3 in Figure 8) can be estimated as follows:

$$\begin{aligned} W_{\mathrm{BBL}} &= 2\int_0^{\pi/2} 2\pi R M_{\mathrm{P}} \mathrm{d}\phi \\ &\quad + 2\int_0^{\pi/2} 2\pi \left(R + \frac{\lambda}{2}\sin\phi\right) M_{\mathrm{P}} \mathrm{d}\phi \\ &= 2\pi M_{\mathrm{P}}(2R\pi + \lambda), \end{aligned} \tag{5}$$

where the plastic bending moment per unit circumferential length of the hinge using Von Mises criterion is $M_{\mathrm{P}} = (2/\sqrt{3})\sigma_{\mathrm{o}}(t^2/4)$.

The energy absorbed in the expansion between the plastic hinges can be estimated as follows:

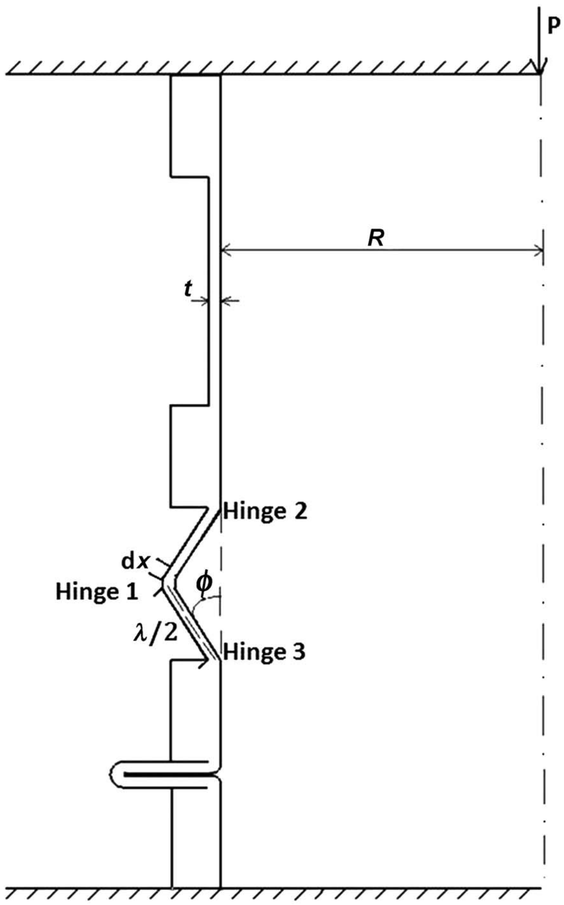

Figure 8. Collapse model of a tube divided into several portions using external wide grooves.

$$W_{\mathrm{EBL}} = 2\int_0^{\lambda/2} 2\pi R t \sigma_{\mathrm{o}} \ln\left(\frac{R + x\sin\phi}{R}\right) \mathrm{d}x = \pi t \sigma_{\mathrm{o}} \frac{\lambda^2}{2}, \tag{6}$$

where the following estimation, for $\phi = \pi/2$, is used:

$$\ln\left(\frac{R + x\sin\phi}{R}\right) \approx \frac{x}{R}.$$

Substituting Equations (5) and (6) into Equation (4), the energy balance equation is re-written as

$$W_{\mathrm{T}} = \frac{\pi t^2 \sigma_{\mathrm{o}}}{\sqrt{3}} \left[(2R\pi + \lambda) + \left(\frac{\sqrt{3}}{2}\right)\left(\frac{\lambda^2}{t}\right)\right]. \tag{7}$$

4.1. *Mean crushing load*

Substituting $W_{\mathrm{T}} = P_{\mathrm{m}} \times (\lambda - 2t)$ into Equation (7), the mean crush load of the shock absorber can be calculated as

$$P_{\mathrm{m}} = \frac{\pi t^2 \sigma_{\mathrm{o}}}{\sqrt{3}(\lambda - 2t)} \left[(2R\pi + \lambda) + \left(\frac{\sqrt{3}}{2}\right)\left(\frac{\lambda^2}{t}\right)\right]. \tag{8}$$

4.2. *Total absorbed energy in plastic deformation*

Substituting Equation (7) into Equation (1), the total amount of plastic energy dissipated in formation of one fold within a grooved space of the shock absorber can be calculated from the following equation:

$$W_{\mathrm{D}} = \frac{\pi t^2 \sigma_{\mathrm{o}}}{\sqrt{3}} \left[(2R\pi + \lambda) + \left(\frac{\sqrt{3}}{2}\right)\left(\frac{\lambda^2}{t}\right)\right]. \tag{9}$$

Multiplying Equation (9) by the number of grooved spaces of the shock absorber, N, we will have the following equation for the total amount of energy absorbed by the shock absorber in plastic collapse after the compression test becomes complete:

$$E_{\mathrm{T}} = \frac{\pi N t^2 \sigma_{\mathrm{o}}}{\sqrt{3}} \left[(2R\pi + \lambda) + \left(\frac{\sqrt{3}}{2}\right)\left(\frac{\lambda^2}{t}\right)\right]. \tag{10}$$

4.3. *Specific energy absorption factor*

The total weight of the presented shock absorber will be (see Figure 1)

$$G = \rho\pi \left[N\lambda D_{\mathrm{i}} t + (N+1) W \frac{(D_{\mathrm{i}}^2 - D_{\mathrm{o}}^2)}{4}\right]. \tag{11}$$

Therefore, the total absorbed energy per weight for the presented shock absorber will be

$$S_{\mathrm{e}} = \left(\frac{N t^2 \sigma_{\mathrm{o}}}{\sqrt{3}\rho}\right) \left(\frac{(\pi D_{\mathrm{i}} + \lambda) + \left(\frac{\sqrt{3}}{2}\right)\left(\frac{\lambda^2}{t}\right)}{N\lambda D_{\mathrm{i}} t + (N+1) W \frac{(D_{\mathrm{i}}^2 - D_{\mathrm{o}}^2)}{4}}\right). \tag{12}$$

5. Results and discussion

Collapsed shapes of all specimens after axial compression, resulting from numerical simulations, are shown in Figure 7. Comparisons between these simulated collapsed shapes with those derived from experimental compression tests (Figure 4) indicate that the presented FE model can simulate the crushed shape of the shock absorber with sufficient accuracy. Also, as can be observed from Table 2, the predicted values of mean crush load by numerical simulations are in good agreement with experimental results. Therefore, the performed numerical simulations can predict the collapsing shape and the crashworthiness parameters of the shock absorber with sufficient accuracy. Table 2 also indicates that the proposed analytical model can predict values of mean crush load very well. It should be noted that to predict the mean crush load of specimens analytically, the average flow stress of $\sigma_{\mathrm{o}} = 450$ MPa is used in Equation (7).

Figure 9. Different stage of buckling for specimens A1, S2, S3 and S6.

Figure 9 shows different stages of deformation in specimens with wide external grooves during their axial buckling. It can be seen that at the first stage of buckling, one fold tends to shape within the length of one groove, being mostly near the top or end of the specimen, when the shock absorber is compressed axially. During this stage of deformation, load increases due to the elastic resistance of the divided structure along this groove length and declines dramatically while a plastic fold shapes completely within this groove space (see Figure 5). At the next stage of buckling, the second fold tends to shape within the length of another groove, mostly near the former groove. At this stage of deformation, again, load increases as a result of elastic resistance of the divided structure along this groove length and declines dramatically while a plastic fold shapes completely within this groove space (see Figure 5). As compression of the shock absorber continues, the same phenomenon tends to occur within the space of other wide grooves, leading to sharp increase and decrease of load, till the buckling process of the shock absorber is complete (see Figures 5 and 9).

According to Figure 4, except specimen number S3, all experimented specimens buckled completely in concertina mode of deformation. Experimental and numerical results show that with the given geometry of a tube, both length and depth of a groove, cut from its outer surface, can significantly affect buckling deformation mode of the shock absorber while subjected to axial compression. These results show that increasing the length of external grooves in specimens with fixed groove depth can decrease the possibility of shaping axisymmetric folds. In specimens with a groove depth of 3 mm (S1, S2, S3), increasing the length of groove from 13.5 mm in S2 to 19.2 mm in S3 leads to the change of buckling deformation mode from concertina in specimens S1 and S2 to diamond in S3. However, decreasing the groove depth from 3 mm in S3 to 2.5 mm in S6 changes the diamond buckling mode of deformation in S3 to concertina mode in S6 (these specimens have equal groove length). These observations show that in specimens with large groove length, decreasing the depth of groove, cut from the outer surface of tube, can control buckling mode of deformation. Table 2 also indicates that both values of length and depth of grooves, cut from the outer surface of tube, can affect the mean crush load of the shock absorber; however, the effect of groove depth is considerably more significant. Therefore, mean crush load and buckling mode of deformation in the proposed energy absorber of this study can be controlled by the geometry of tube and both values of groove depth and length, cut from its outer surface.

In the present study, to show the efficiency of the proposed method in extending the limits of axisymmetric mode of deformation, for each set of grooved specimens with fixed groove depth, a groove-less specimen of the same geometric parameters has been compressed axially. Corresponding results show that (see Figures 3 and 4) division of tubes by cutting wide grooves from their outer surface can control the axisymmetric collapsing mode of the shock absorber considerably. Even in the case of specimen number S3 with non-axisymmetric mode of deformation, the grooved shock absorber provides considerably more favourable crashworthiness characteristics and stability during axial crushing in comparison with the groove-less specimen of A1 (see Figures 3, 4 and 9). Moreover, in contrast to previous design of grooved shock absorbers [10], which reduced the energy absorption compared to groove-less specimen, the present design method does not affect this important crashworthiness parameter. In fact, in the present study, the objective to cut wide grooves from the outer surface of tubes is to divide their length to several portions and improve their crashworthiness characteristics. Moreover, division of tube length with wide external grooves can significantly decrease maximum collapsing load of the tube, resulting in high crush force efficiency, as introduction of wide external grooves affects the initial elastic resistance of the tube during axial compression.

As can be seen, although specimens are not chosen in best diameter thickness to collapse in concertina mode, collapsed shape of grooved specimens shows the significant efficiency of the present design method in extending the limits of concertina region, having been realised by several experimental tests for groove-less circular tubes. As a matter of fact, this design method provides solution to designers with geometric limitation in designing collapsible shock absorbers for an energy-dissipating device which collapses in concertina mode, but not necessarily having geometric parameters in limited concertina region. As another advantage, this proposed design method helps designers to obtain mean crushing loads, which may not be achieved by collapsing of a tube with geometric parameters in a limited concertina region.

Nevertheless, from Equation (12), it can be understood that specimens with wide external grooves provide less specific energy absorption than groove-less specimens. This is because thick portions in grooved specimens do not contribute in energy absorption during axial crushing. However, these parts provide significant advantages, which in some special engineering applications outweigh the disadvantage of making the presented energy-absorbing device heavy. They play a significant role in improving the stability of the shock absorber while crushing. In some engineering structures, the shock absorber is considered to be a part of the structure, which also should be capable of dissipating kinetic energy to protect other main parts from damages while the structure is subjected to shock loading. For these applications, a shock absorber with extreme deformation is not suitable. Thick portions in the proposed energy-dissipating device meet these demands and prevent extreme plastic deformation of the shock absorber, causing unfavourable instability while crushing. As another advantage, these portions make a significant contribution to encourage the shock absorber to collapse in concertina mode, leading to a significant increase in safety factor. It should be noted that further theoretical and experimental investigation of the presented shock absorber is required to improve its specific energy absorption by decreasing both the thickness and length of thick portions while maintaining the mentioned favourable crashworthiness characteristics.

6. Conclusion

In the present study, cutting wide circumferential grooves from the other surface of tubes with the aim of dividing their length into several portions is introduced as an efficient and reliable method to improve their crashworthiness characteristics. Numerical simulations of axial crushing in grooved specimens of various geometric parameters under axially quasi-static loading were performed. Also, an analytical model to predict the mean crush load of the shock absorber was proposed. Comparison of these results with experimental results, having been performed in this study, shows the high accuracy and reliability of theoretical studies. Through experimental, numerical and analytical studies, major parameters in design are identified, and typical modes of deformation that may occur during axial compression of the shock absorber are characterised. Also, load–displacement history and deformation mechanism of the shock absorber under axial compression are described.

It has been shown that this design method can significantly control shaping of axisymmetric folds while the shock absorber crushes axially, and extend the limits of this most favourable collapsing mode of deformation. The effect of various geometric parameters of the shock absorber such as groove length and depth is studied both theoretically and experimentally. Results have shown that by adjusting these geometric parameters, various required mean crushing load to protect different structures subjected to impact load from serious damages can be achieved. Also, with the consideration of major geometric parameters in design, shaping concertina mode of deformation can be guaranteed in axial collapsing of the shock absorber, resulting in high safety factor. Moreover, unlike conventional methods of energy absorption like axial splitting and axial crushing of thin-walled structures, the proposed method has less sensitivity to loading direction and condition. Favourably, it can be seen that cutting wide grooves along the tube length can considerably decrease elastic resistance of the structure against axial impact load, which leads to a considerable increase in crush force efficiency of the shock absorber.

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