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Effect of Corner Bluntness on Energy Absorbing Capability of Non-circular Metallic Tubes Subjected to Axial Impact

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ABSTRACT

The energy absorbing capability is one of the most important aspects of crushing behavior of a structure subjected to axial impact. In this paper, a simple practical method is introduced to enhance the crushing behavior of this kind of structure. The dynamic explicit simulation of axial impact of metallic energy absorbing thin-walled tubes with special shaped cross-section is provided using LS-DYNA software. The effect of change in the corners bluntness of non-circular tubes on their energy absorbing capability has been studied. Moreover, the mean crushing force, the maximum deformation, and the mass specific energy absorption (*MSEA*) of the tubes were compared. Results show that the energy absorbing capability can be significantly improved by choosing proper bluntness of the corners for quasi-triangle- and quasi-square shaped tubes. Furthermore, results show an improvement in the energy absorbing capability of quasi-square shaped tubes even in comparison with circular one.

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NOMENCLATURE

X	x- coordinate of any circumferential point on the cross-section shape	p	Constant power parameter in Cowper-Symonds strain rate equation
Y	y- coordinate of any circumferential point on the cross-section shape	m_0	Rigid wall mass
θ	The angle of any circumferential point on the cross-section shape with respect to the x- axis	V_0	Rigid wall initial velocity
λ	Cutout size controlling number	E	Absorbed impact energy
n	Side number indicator of cross-section shape	A	Tube cross-sectional area
c	Cross-section aspect ratio	ΔL	Tube deformed length
w	Bluntness factor	ΔV	Tube deformed volume
$\dot{\varepsilon}$	Strain rate	ΔM	Tube deformed mass
σ_0	Basic static strength	ρ	Tube material density
σ'_0	Basic dynamic strength	P_{mean}	Mean crush force
D	Constant coefficient in Cowper-Symonds strain rate equation	$MSEA$	Mass specific energy absorption

1. INTRODUCTION

To improve the passengers safety, a vehicle needs a preservative body accompanied with some additional deformable components (e.g. crush boxes) embedded in the vehicle bumpers which can absorb the crush energy during the impact. These energy dissipating members such as the circular and square tubes, multi-corner columns, and top-hat sections can effectively absorb the impact energy by their folding and plastic deformation during the frontal as well as oblique impacts. Use of thin-walled tubes as energy absorbers in the automotive applications is widely spread due to their high strength to weight ratio and energy absorption capacity.

To decrease the crush damages, metallic energy absorbing thin-walled tubes are widely used in many structures subjected to impact. In the crashworthiness of a structure, strength to weight ratio, ease of fabrication

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and manufacturing, and costs are some of the important design parameters, which must be considered. However, the weight of the structure is one of the most important aspects of design in modern high-performance structures such as spacecrafts, airplanes, and automobiles.

Recently, many researchers have studied the crushing behavior of structures particularly thin-walled tubes. Lu and Yu [1] indicated the Andrews's theoretical model in which the absorbed crush energy can be divided into two elastic and plastic forms. Abramowicz and Jones [2-4] presented the experimental evaluation of circular and square-shaped cross-section thin-walled tubes of various dimensions which were subjected to the axial impact. Otubushin [5] compared the experimental results of axial crush of a square tube with the numerical simulation using DYNA3D software. A good agreement has been reported between these numerical and experimental results.

Al Galib et al. [6] conducted experimental and numerical investigations on axial crush of thin-walled aluminum tubes. Tai et al. [7] simulated the axial impact of mild and high strength steel tubes of various sizes, which are widely used in automotive industries. Some researchers [8-10] studied the effects of different types of buckling initiators on the efficiency of tubes subjected to axial impact. Buckling initiators are usually created by imposing some geometrical imperfections in the structure. In practice, these initial imperfections are generally used to lead the folding patterns, or to make simulation results more realistic [8-10].

Furthermore, the axial crush of aluminum foam-filled thin-walled tube was investigated in order to study effects of foam filling on the crushing behavior [9-11]. In a resembling study, Ahmad et al. [12] presented experimental as well as numerical investigations on oblique impact of empty and foam-filled conical tubes in order to study the crushing angle. They reported that filled tubes show better energy absorption characteristics by increasing the load orientation. To find out the effect of geometry on the structural efficiency, crushing behavior of several tubes of various cross-section shapes such as polygonal [13, 14], square and ellipse [15] have been investigated.

Salehghaffari et al. [16] within an experimental study presented two methods to improve the crushing behavior of metallic circular tubes. In their first method, they fitted a steel ring on the top edge of an aluminum tube; while in their second method, they cut some grooves from the steel thick-walled tubes, and made them partially thin-walled along the length. These methods showed improved characteristics and folding modes during the axial impact.

Within analytical analysis as well as experimental investigations, Jandaghi Shahi and Marzbanrad [17] studied the axial crush of (segmented) tailor-made tubes (TMTs) with varying mechanical properties and wall thickness along the tube's length. This research showed that TMTs have greater capacity to absorb the impact energy per weight compared with simple tubes with constant wall thickness. Tang et al. [18] through a geometrical investigation introduced some non-convex multi-corner thin-walled tubes which can absorb more energy during the axial impact. They also presented an analytical predicting formulation for mean crush force and supported their outcome via numerical simulation.

As mentioned above, in recent years, several methods are presented to improve the structural efficiency of an energy absorbing thin-walled tube. Improving the material properties, reinforcing the tube, and shape optimization are the most common ways to achieve this end.

In a recent study [19], the authors investigated the axial crushing behavior of energy absorbing hybrid thin-walled tubes (i.e., metallic tubes overlaid by composite laminate). Within the numerical simulations verified by experimental data, they studied the effects of cross-sectional geometry, and investigated special selections of material properties as well in order to find their combined effects on the low-velocity crushing behavior of tubes. As what concluded in [19], hybriding thin-walled tubes can improve their structural behavior due to higher strength to weight ratio, and consequently, hybrid tubes can absorb more energy per unit of their deformed mass than common metallic tubes.

In the present study, a new simple concept of shape optimization is employed to improve the crushing efficiency of the metallic energy absorbing thin-walled tubes. The main objective of this study is to investigate the influence of cross-section bluntness on the energy absorbing capability of the tubes subjected to axial impact.

The finite element software LS-DYNA is used to simulate the axial impact of the tubes. Tubes with different special cross-section shapes are obtained by changing the corners' bluntness of some common N-gonal (Triangle, Square, Pentagon, and Hexagon) shapes. These shapes are produced using a special mapping function. Moreover, a buckling initiator is modeled to lead the folding pattern. Mild steel with piecewise-linear-plasticity is selected which its strain hardening behavior is modeled using the Cowper-Symonds strain rate relation. Results show that making these special changes in the corners' bluntness can significantly improve the crushing behavior of the non-circular tubes.

2. PROBLEM DEFINITION AND SOLUTION METHOD

2. 1. Cross-section Modeling As previously mentioned, several researchers have studied the axial impact of tubes, but these efforts are limited to some common cross-section shapes such as circle, square, etc. In order to study the effects of geometry more carefully, a simple mapping function [20] is used to model a wide variety of cross-sectional shapes. Using this function changing the bluntness of the corners of a simple polygonal-shaped tube is possible. This function is defined as below:

$$\begin{cases} X = \lambda(\cos\theta + w\cos(n\theta)) \\ Y = -\lambda(c\sin\theta - w\sin(n\theta)) \end{cases} \tag{1}$$

In these relations, λ is a positive real number which controls the cutout size, and n demonstrates the number of sides of the shape mines one (e.g., $n = 2$ for triangle and $n = 3$ for square). Moreover, c determines the cross-section aspect ratio which is set to be one ($c = 1$) in this study.

Finally, w is the bluntness factor which controls the curvatures at the corners. By reducing the bluntness factor w, all shapes gradually change to a circle when $w = 0$. Table 1 represents the geometrical parameters of quasi-squared shapes, which are obtained using relation (1).

As Figure 1 illustrates, a wide variety of quasi-squared and quasi-triangular shapes are obtained using this function.

In the present work, the effects of these geometrical parameters are investigated for the triangle, square, pentagon, and hexagon-shaped tubes. An equal circumference of 30π mm, which is equal to the circumference of a 30 mm diameter circle, is considered for all tubes. Moreover, the length and the thickness of all tubes are equal to 90 mm and 0.8 mm, respectively. With these conditions, the results are compared for tubes with identical length, weight and thickness. It should be noted that the modeling is limited to $n = 5$ (hexagon-shaped tubes), since for higher values of n, the shapes are close to a circle. Therefore, there were no significant differences in results of $n > 5$.

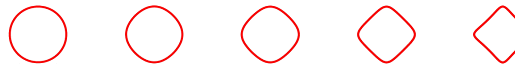
2. 2. Material Model and Mechanical Properties

Since the tube is subjected to a time-varying impact force, and also due to the strain hardening behavior of the metal, the strain rate effects must be considered. To achieve these conditions, the material model No. 24 of the LS-DYNA material library [21] is used to determine the mechanical properties of the metal. This model assumes a piecewise-linear-plasticity behavior which represents all plastic characteristics of the material. Moreover, the model uses the Cowper-Symonds strain rate equation as follows [7]:

$$\dot{\varepsilon} = D\left(\frac{\sigma_0'}{\sigma_0} - 1\right)^p \quad , \quad \sigma_0' \geq \sigma_0 \tag{2}$$

In this relation, $\dot{\varepsilon}$ represents the strain rate. σ_0 and σ_0' indicate the basic static and dynamic strengths, respectively. For mild steel, constant values of the Cowper-Symonds equation (D and p) are experimentally determined. These constant values which are required as inputs for the Cowper-Symonds equation, are reported as $D = 40$ and $p = 5$ [7]. The mechanical properties and the experimental stress-strain diagram for mild steel considered in this study are presented in Table 2 and Figure 2, respectively. Figure 2, in fact, gives the necessary information about the plastic stress and corresponding plastic strain values for mild steel as the inputs of the material model No. 24.

TABLE 1. Geometrical parameters of some quasi-squared shapes



w	0.0000	0.0250	0.0500	0.1000	0.1500
λ	15.000	14.979	14.916	14.668	14.268

TABLE 2. Mechanical properties of mild steel [7]

Density (kg/mm^3)	Elastic Modulus (GPa)	Yield Stress (MPa)	Poisson's Ratio
7.82×10^{-6}	207.2	325	0.33

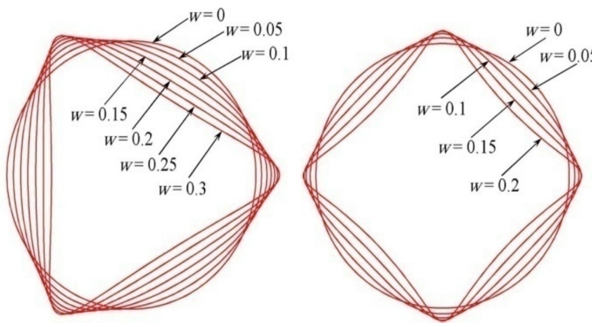


Figure 1. Quasi-triangular and quasi-squared shapes with different values of the bluntness factor

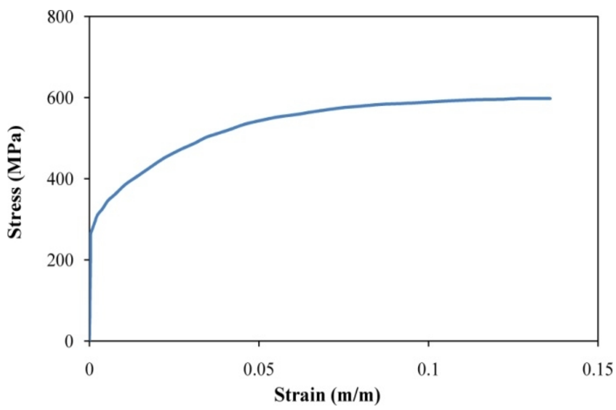


Figure 2. Experimental stress-strain diagram for mild steel [7]

2. 3. Finite Element Model and Boundary Conditions To model the axial impact conditions, it is assumed that in the absence of gravitational force, a rigid wall of specified mass (m_0) and initial velocity (V_0) impacts the tube. Its kinetic energy is absorbed during the folding and deforming the tube until it stops. The value of m_0 and V_0 is 25 kg and 10 m/s, respectively. The bottom edge of the tube is completely constrained, and the rigid wall moves and axially crushes the tube. This means the lower edge of the tube is fully clamped, while the upper edge is free. However, by starting the facial impact, a contact algorithm must be introduced between the rigid wall and the upper edge of the tube in order to characterize the contact conditions. How to define and impose this algorithm will be discussed in the following context.

The Belytschko-Tsay 4-node shell element with five points of integration through the thickness is used to model the thin-walled tube. The element size should be chosen in such a way that the outcomes of numerical study fit in the experimental investigations. Based on what reference [7] has reported, the element size must be smaller than half of the length of the plastic hinge. As what stated in [7], the appropriate mesh size for tubes in category of present study is 2 mm. Furthermore, the contact conditions are defined between the tube and the wall, and also between the folded layers of the tubes to prevent the penetration of contacting elements.

To achieve this, two contact algorithms, CCNTS and CASS, are used [21]. The CCNTS (CONTACT_CONSTRAINT_NODES_TO_SURFACE) with a friction coefficient of 0.9 simulates the interfacial contact conditions between the rigid wall and the tube to prevent the penetration. Moreover, the contact algorithm CASS (CONTACT_AUTOMATIC_ SINGLE_SURFACE) with the friction coefficient of 0.3 simulates the contact conditions between the continuous folds of the tube (which overlap during the folding) to prevent the interpenetration.

In order to lead the folding pattern during the axial crush, a buckling initiator is created in the model. To model the buckling initiator, a 0.3 mm depression is imposed to the circumferential elements located at 10 mm from the upper end of the tube. It is to be noted that, although the position and shape of the buckling initiator may have effect(s) on the results, these parameters are not under consideration in the present study. According to the literature review, there are some useful references (e.g., [5, 8-10]) which discuss the effects of different types of buckling initiators. Figure 3 shows the position and dimensions of the buckling initiator.

Usually, in a crashworthiness analysis, important parameters such as maximum deformations, mean forces, and mass specific energy absorption (*MSEA*) are used to evaluate the effectiveness of an energy absorbing member. Energy absorbing capability of tubes can be determined by the *MSEA*, the amount of impact energy per unit of deformed mass, which is absorbed during the folding of the tube. The mean force represents the average constant force which the tube can endure during the impact. The absorbed crushing energy increases when the tube can endure a higher mean force. It means that, higher maximum deformation yields less mass specific energy absorption. As it can be seen in Figure 4, these parameters may be obtained using the force-deformation (*F*-*D*) curve.

The *F*-*D* curve can be drawn by cross-plotting the force-time (*F*-*T*) and deformation-time (*D*-*T*) curves. Note that the absorbed energy can be calculated as the area under the *F*-*D* curve through the length of deformation.

The maximum deformation (ΔL) can be obtained directly from the *F*-*D* curve. The mass specific energy absorption (*MSEA*) also can be calculated as follows [7]:

$$MSEA=\frac{E}{\Delta M}=\frac{E}{\rho \Delta V}=\frac{E}{\rho A \Delta L} \tag{3}$$

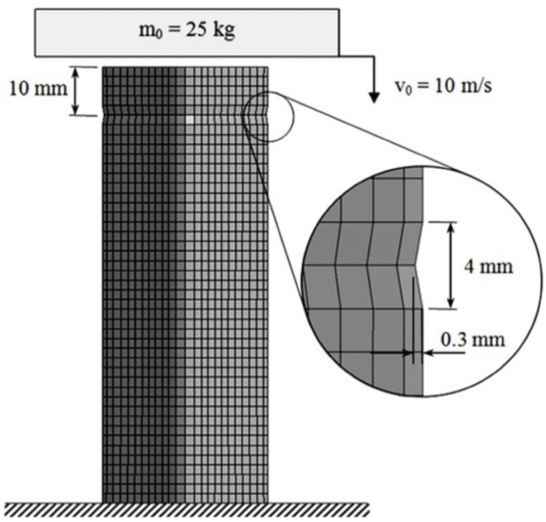


Figure 3. Finite element model and schematic of buckling initiator, its position and dimensions

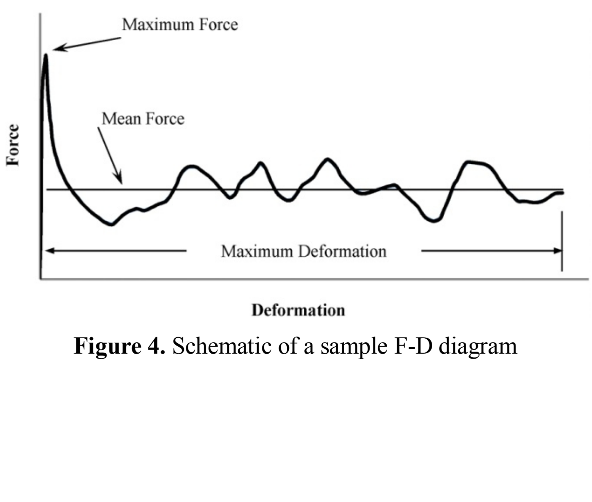


Figure 4. Schematic of a sample F-D diagram

in which,

$$E = \frac{1}{2} m_0 V_0^2 \tag{4}$$

In these relations, *MSEA* represents the mass specific energy absorption which indicates the energy absorbing capability per unit deformed length. *E* represents the total absorbed energy which is equal to the initial kinetic energy of the rigid wall. Moreover, ΔV and ΔM represents the deformed volume and mass, respectively. In fact, ΔM is the mass of the deformed length (ΔL) of the tube. Furthermore, the mean force (P_{mean}) which is the absorbed energy per unit deformed length can be obtained as follows [7]:

$$P_{mean} = \frac{E}{\Delta L} = \frac{m_0 V_0^2}{2 \Delta L} \tag{5}$$

3. RESULTS AND DISCUSSION

3. 1. Validation The finite element modeling and solution procedure used in this study are verified via the sample No. HS-07 reported by Tai et al. [7]. It is to be noted that, this circular tube has length of 90 mm, wall thickness of 0.8 mm, and diameter of 31 mm, and is subjected to an axial impact by a mass of 30 kg which has an impact velocity of 15 m/s. The good agreement between curves shown in Figure 5 signifies the validity of the simulation results.

It is also to be noted that, an additional verification sample [19] is presented in Appendix-A in order to verify the modeling procedure via experimental results.

3. 2. Numerical Results Wherever a tube of non-circular cross-section shape is preferred, the design parameters constrain the designer to select the tubes with triangular, square, pentagonal, or hexagonal cross-section shapes. Hence, the proficiency of these kinds of tubes must be evaluated and improved as much as it is possible.

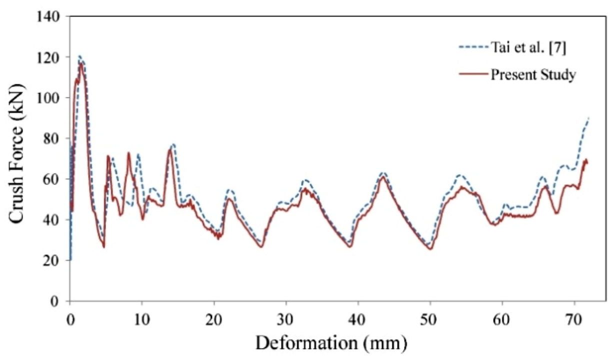


Figure 5. Validation of the results including the force-deformation curves

Results of numerical simulations of axial impact of some N-gonal shape tubes are presented in the following figures. In these figures, the effect of corner bluntness on the maximum deformation, mean force, mass specific energy absorption (*MSEA*), and final deformed shaped of the tubes are illustrated. Note that, to make a better comparison, all results are normalized with respect to corresponding values of equivalent circular shaped tubes.

It is to be noted here that, the results presented in this paper are normalized with respect to the corresponding values for circular tube in order to investigate the pure effects of geometry changes on the crushing behavior. In fact, for each tube $\Delta M_i = \rho_i A_i \Delta L_i$ (the subscript index *i* indicates each tube). Generally, the density (ρ_i) and the cross-sectional area (A_i) for a tube can be different from other ones. In such situation, the normalized *MSEA* and normalized mean force are not equal in value. Therefore, one cannot draw a correct conclusion on the shape optimization.

However, in this paper, all tubes are assumed to have identical density and cross-sectional area, the results are normalized, and the terms of density and cross-sectional area are vanished. Therefore, the normalized *MSEA* and mean force are independent of density and cross-sectional area of tubes, and as a result, these parameters have equal values.

Moreover, representating both of these equal normalized values are due to their explanations. The normalized *MSEA* is reported to indicate the capacity of tubes in absorbing the impact energy compared to circular tube. While, the normalized mean force represents the average axial load imposed to the tubes (and consequently to the bumper and chasis). With this in mind, the designer can gain an overall veiw about the external impact loads applied to the vehicle.

As it can be seen in Figure 6, for quasi-triangular shaped tubes, when $w = 0.3$ the *MSEA* is equal 0.55. By reducing the bluntness curvature to 0.2, the *MSEA* increases to 0.81, which shows about 46% increase in *MSEA* without a major change in the cross section shape. Moreover, Figure 7 illustrates the effect of change in bluntness on the deformed shape of the triangular tubes. Clearly, by reducing the bluntness (w), the deformation shape (folding pattern) is more similar to the folding pattern of a circular tube and the major plastic deformation moves toward the end of the tube. By increasing *w*, the plastic deformation extents along the tube length similar to a triangular tube deformation. Moreover, the end shortening of the tube decreases by reduction of the bluntness factor. This greater deformation is clearly a result of smaller localized bending stiffness at the corners due to the sharper angles. Moreover, as Figure 7 obviously shows, in quasi-triangle-shaped tubes, by increasing the bluntness factor, the folding pattern significantly changes from a regular symmetrical diamond mode (for $w = 0$) to a complex buckling mode (for w = 0.3). The deformed length of a quasi-triangular shaped tube with w = 0.3, is 80% more than similar circular tubes. By reducing the bluntness curvature to 0.2 the deformed length is only 23% higher than a similar circular tube. This is about 57% reductions in deformed shaped of the tube only by changing the bluntness factor. Furthermore, the mean force which is, in fact, the length specific energy absorption decreases by increasing the deformed length. Hence, the smaller the deformed length, the greater the absorbed crush energy to be expected. Similarly, as it is illustrated in Figure 8, for quasi-square shaped tubes, when w = 0.15 the *MSEA* is equal 0.72. By reducing the bluntness curvature to 0.0625, the *MSEA* increases to 1.02, which shows a 41% increases in *MSEA*. Moreover, a quasi-square tube with this bluntness factor has a *MSEA* higher than a circular tube. Figure 9 illustrates the effect of change in bluntness on the deformed shapes of the square tubes. Clearly, for w < 0.0625, the deformation shapes and end shortening are very similar to a circular tube. According to Figure 8, there are significant improvements in the amounts of mean force and *MSEA* for some of the quasi-square-shaped tubes in comparison with square-shaped ones. For some specific values of the bluntness factor w (0.0125 < w < 0.0625) the amounts of mean force and *MSEA* of the corresponding tubes are even more than those of a similar circular tube. Applying some specific values of corner bluntnesses make the folding pattern in these quasi-square shaped tubes different from that of the circular one. The thin-walled tubes subjected to axial compressive load absorb the impact energy during their plastic deformation. Therefore, the destruction mode influences its energy absorbing efficiency [7]. The optimized functionalities observed in Figure 8 for 0.0125 < w < 0.0625 may be described as a result of this significant influence. However, the advantage of such shape optimization is to improve the energy absorbing capacity of tubes without making any change in the mass, length, and thickness of tube except a minor change in cross-sectional geometry. Clearly, the simple method presented in this study, results in high improvement in the mass specific energy absorbing capability of the tubes. As shown in Figures 10 and 12, for the quasi-pentagon and quasi-hexagon-shaped tubes, there is no considerable improvement in the amounts of the mean force and *MSEA*. This is clearly due to the greater resemblance between their cross-sectional shapes and the circular tubes cross-section shape. The deformed shapes of the tubes are shown in Figures 11 and 13. There is no major change in folding patterns of quasi-pentagonal/hexagonal shaped tube as bluntness factor changes.

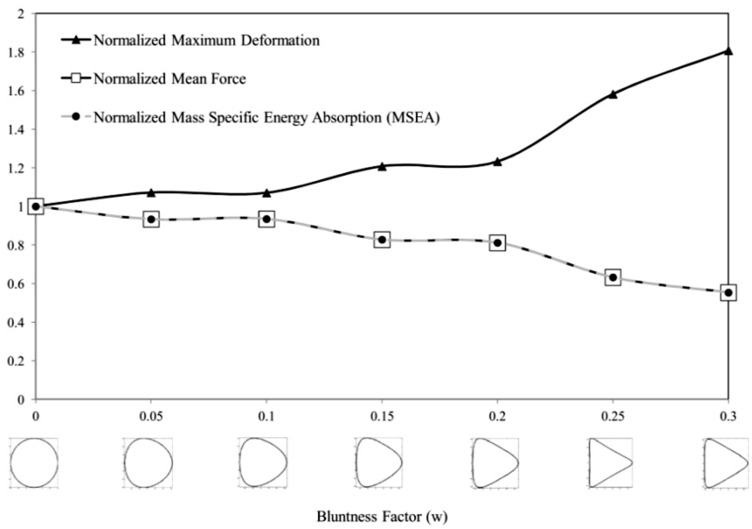


Figure 6. Normalized maximum deformation, mean force and MSEA vs. bluntness factor for quasi-triangular shaped tubes

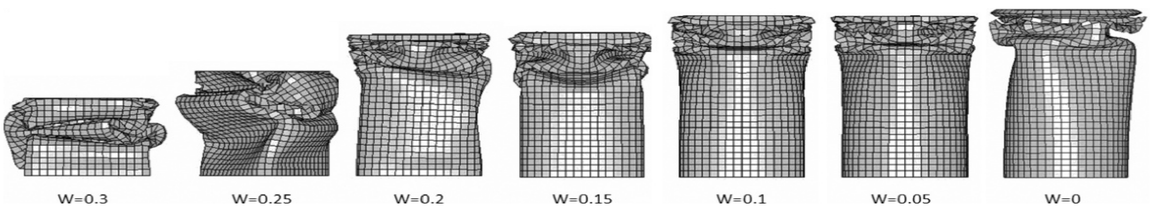


Figure 7. Deformed shapes of the quasi-triangle-shaped tubes

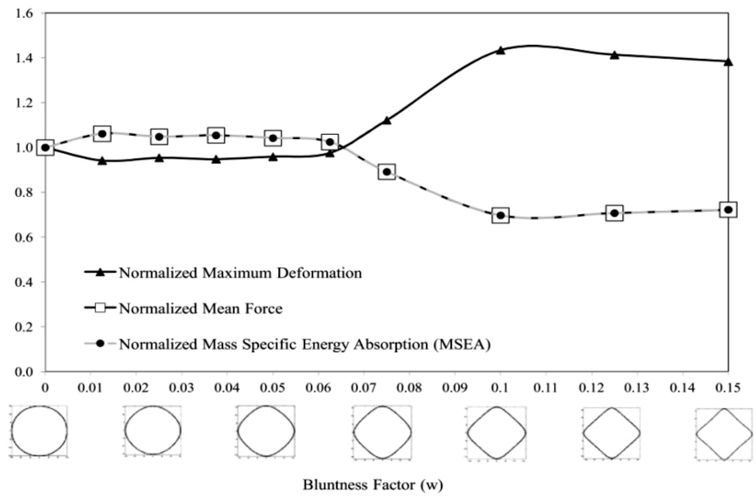


Figure 8. Normalized maximum deformation, mean force and MSEA vs. bluntness factor for quasi-square shaped tubes

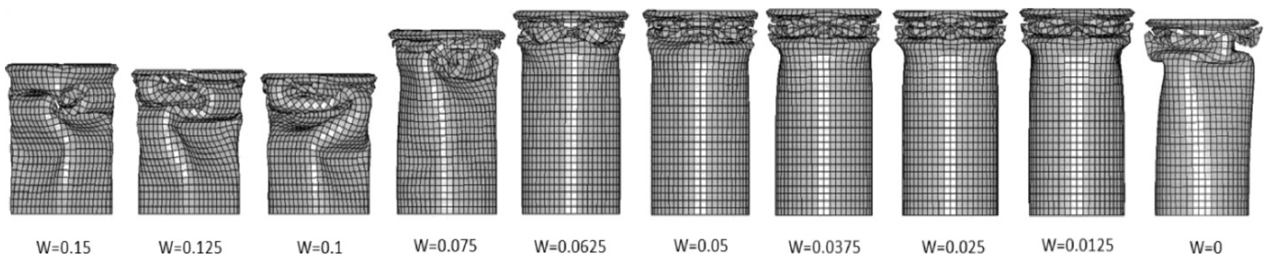


Figure 9. Deformed shapes of the quasi-square-shaped tubes

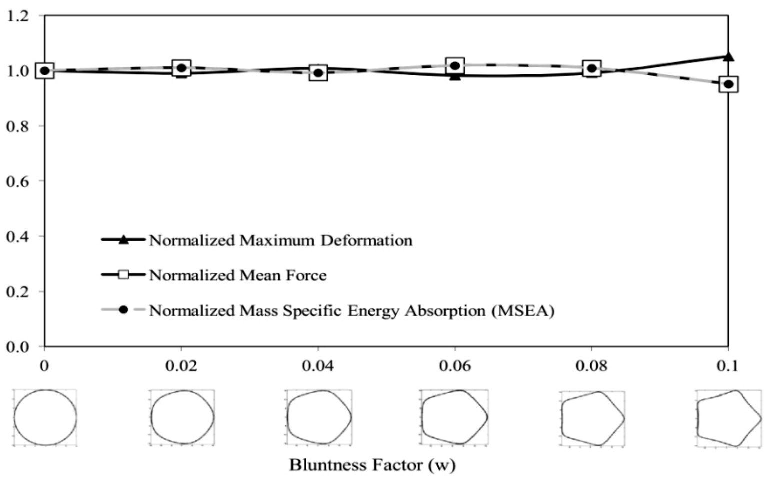


Figure 10. Normalized maximum deformation, mean force and MSEA vs. bluntness factor for quasi-pentagonal shaped tubes

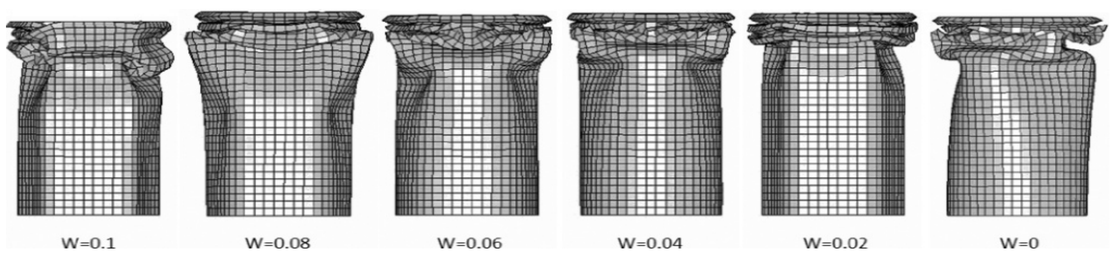


Figure 11. Deformed shapes of the quasi-pentagon-shaped tubes

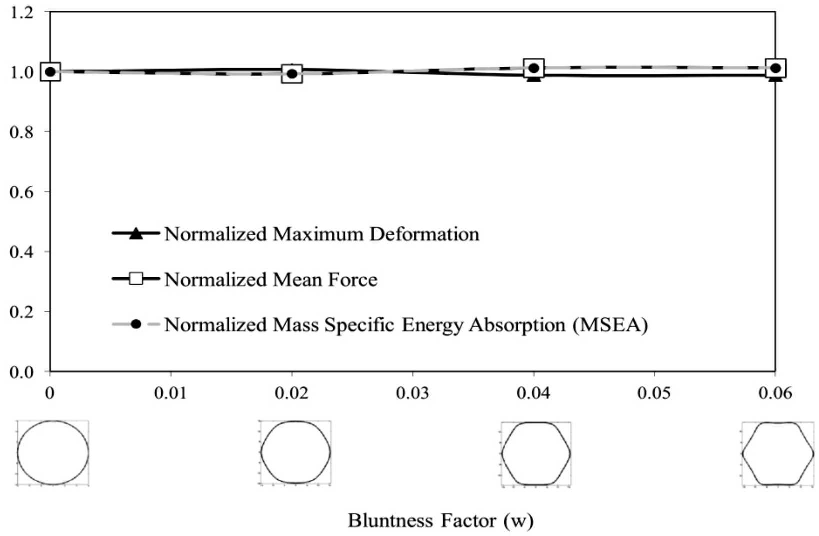


Figure 12. Normalized maximum deformation, mean force and MSEA vs. bluntness factor for quasi-hexagonal shaped tubes

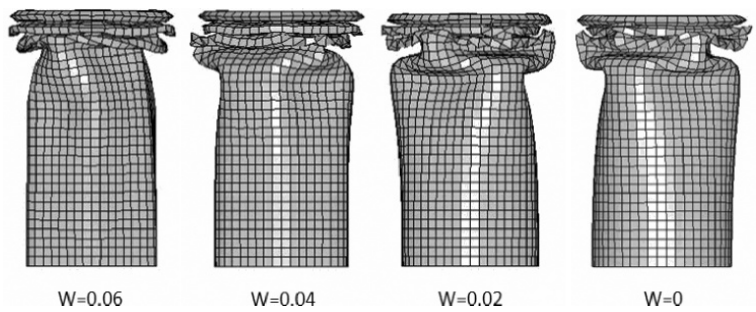


Figure 13. Deformed shapes of the quasi-hexagon-shaped tubes

4. CONCLUSIONS

In this study, the axial impact of energy absorbing thin-walled tubes with special shaped cross-sections has been simulated using the LS-DYNA software. A new simple concept of changing the corner bluntness of tubes is presented in order to change and improve their operational efficiencies during the axial impact. A considerable difference in the axial impact responses of the tubes with different values of the bluntness factor is observed. It should be noted that, all the tubes are made of the same materials and have identical initial, boundary, and loading conditions.

Results show a considerable improvement in the mass specific energy absorption of the tubes which is achieved only by making a simple change in the cross-sectional bluntness and geometry. Such simple changes in geometry result in some shapes which provide significantly better functionality, up to 60% for quasi-triangle-shaped tubes and 40% for quasi-square-shaped ones (in comparison with the corresponding minimum values). Furthermore, the results show a surprising improvement of about 6% in the structural behavior of some optimum quasi-square-shaped tubes ($0.0125 < w < 0.0625$) in comparison with circular ones, which are supposed to be the best. This unexpected result has been already observed in the investigation of the stress concentration in metallic plates with similar special shaped cutouts [20]. Clearly, the present study shows that more effective structures with higher capabilities to absorb the impact energy can be designed by slight modification of the corner bluntness without any change in the mass, length, and thickness of non-circular metallic tubes.

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5. REFERENCES

1. Lu, G. and Yu, T., "Energy absorption of structures and materials", Woodhead Publishing, (2003).

2. Abramowicz, W. and Jones, N., "Dynamic axial crushing of circular tubes", ***International Journal of Impact Engineering***, Vol. 2, No. 3, (1984), 263-281.

3. Abramowicz, W. and Jones, N., "Dynamic axial crushing of square tubes", ***International Journal of Impact Engineering***, Vol. 2, No. 2, (1984), 179-208.

4. Abramowicz, W. and Jones, N., "Dynamic progressive buckling of circular and square tubes", ***International Journal of Impact Engineering***, Vol. 4, No. 4, (1986), 243-270.

5. Otubushin, A., "Detailed validation of a non-linear finite element code using dynamic axial crushing of a square tube", ***International Journal of Impact Engineering***, Vol. 21, No. 5, (1998), 349-368.

6. Al Galib, D. and Limam, A., "Experimental and numerical investigation of static and dynamic axial crushing of circular aluminum tubes", ***Thin-Walled Structures***, Vol. 42, No. 8, (2004), 1103-1137.

7. Tai, Y., Huang, M. and Hu, H., "Axial compression and energy absorption characteristics of high-strength thin-walled cylinders under impact load", ***Theoretical and Applied Fracture Mechanics***, Vol. 53, No. 1, (2010), 1-8.

8. Langseth, M., Hopperstad, O. and Berstad, T., "Crashworthiness of aluminium extrusions: Validation of numerical simulation, effect of mass ratio and impact velocity", ***International Journal of Impact Engineering***, Vol. 22, No. 9, (1999), 829-854.

9. Hanssen, A. G., Langseth, M. and Hopperstad, O. S., "Static and dynamic crushing of square aluminum extrusions with aluminum foam filler", ***International Journal of Impact Engineering***, Vol. 24, (2000), 347-383.

10. Shariati, M., Allahbakhsh, H. and Saemi, J., "An experimental and numerical crashworthiness investigation of crash columns assembled by spot-weld", ***Mechanika***, Vol. 82, No. 2, (2010), 21-25.

11. Bi, J., Fang, H., Wang, Q. and Ren, X., "Modeling and optimization of foam-filled thin-walled columns for crashworthiness designs", ***Finite Elements in Analysis and Design***, Vol. 46, No. 9, (2010), 698-709.

12. Ahmad, Z., Thambiratnam, D. and Tan, A., "Dynamic energy absorption characteristics of foam-filled conical tubes under oblique impact loading", ***International Journal of Impact Engineering***, Vol. 37, No. 5, (2010), 475-488.

13. Yamashita, M., Gotoh, M. and Sawairi, Y., "Axial crush of hollow cylindrical structures with various polygonal cross-sections: Numerical simulation and experiment", ***Journal of Materials Processing Technology***, Vol. 140, No. 1, (2003), 59-64.

14. Rossi, A., Fawaz, Z. and Behdinan, K., "Numerical simulation of the axial collapse of thin-walled polygonal section tubes", ***Thin-Walled Structures***, Vol. 43, No. 10, (2005), 1646-1661.

15. Marzbanrad, J., Mehdikhanlo, M. and Pour, A. S., "An energy absorption comparison of square, circular, and elliptic steel and aluminum tubes under impact loading", ***Turkish Journal of Engineering and Environmental Sciences***, Vol. 33, (2009), 159-166.

16. Salehghaffari, S., Tajdari, M., Panahi, M. and Mokhtarnezhad, F., "Attempts to improve energy absorption characteristics of circular metal tubes subjected to axial loading", ***Thin-Walled Structures***, Vol. 48, No. 6, (2010), 379-390.

17. Jandaghi Shahi, V. and Marzbanrad, J., "Analytical and experimental studies on quasi-static axial crush behavior of thin-walled tailor-made aluminum tubes", ***Thin-Walled Structures***, Vol. 60, (2012), 24-37.

18. Tang, Z., Liu, S. and Zhang, Z., "Energy absorption properties of non-convex multi-corner thin-walled columns", ***Thin-Walled Structures***, Vol. 51, (2012), 112-120.

19. Yousefsani, S. A., Rezaeepazhand, J. and Maghami, S. A., "Axial crush of metallic and hybrid energy absorbing thin-walled tubes with polygonal cross-sections: Numerical analysis", ***International Journal of Automotive Engineering***, Vol. 3, No. 1, (2013), 293-304.

20. Rezaeepazhand, J. and Jafari, M., "Stress concentration in metallic plates with special shaped cutout", ***International Journal of Mechanical Sciences***, Vol. 52, No. 1, (2010), 96-102.

21. "Ls-dyna user's manual", ***Livermore Software Technology Corporation (LSTC)***, (2010).

22. Ei-Hage, H., Mallick, P. and Zamani, N., "Numerical modeling of quasi-static axial crush of square aluminum-composite hybrid tubes", ***International Journal of Crashworthiness***, Vol. 9, No. 6, (2004), 653-664.

APPENDIX-A

In a previous work, a more complex numerical simulation of axial impact of hybrid tubes (i.e., metallic tubes overlaid by composite laminate) was provided, and results were verified with experimental results for a sample (No. 2L45SH). Present study provides a similar model which is, in fact, a simple form of what presented in [19]. As Figure A-1 [19] demonstrates, this simulation shows good agreement with experimental and numerical results reported in [22].

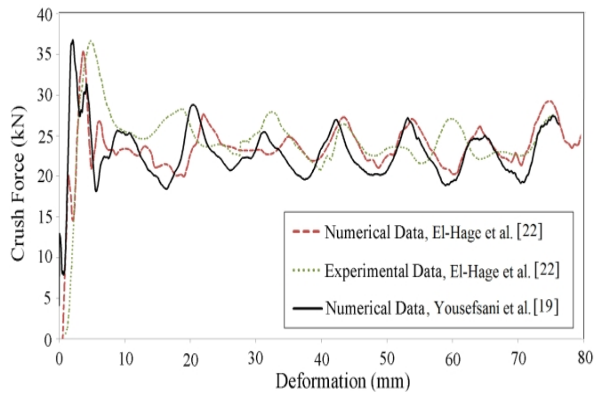

Figure A-1. Validation of present simulation via experimental investigations

Effect of Corner Bluntness on Energy Absorbing Capability of Non-circular Metallic Tubes Subjected to Axial Impact

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چکیده

قابلیت جذب انرژی برخورد در سازه های تحت ضربه محوری، یکی از مهم ترین جنبه های تحلیل رفتار مقاومت در برابر ضربه است. در این مقاله، یک روش عملی ساده برای بهبود عملکرد چنین سازه های در برابر ضربه معرفی شده است. بدین منظور، ضربه محوری لوله های جدارنازک جاذب انرژی فلزی با شکل مقطع های خاص به کمک نرم افزار تحلیل دینامیکی صریح LS-DYNA مدلسازی شده و اثرات تغییر در میزان گردی گوشه های مقطع لوله های با شکل مقطع غیردایروی بر قابلیت جذب انرژی برخورد مطالعه گردیده است. افزون بر این، نیرو برخورد متوسط، تغییرشکل بیشینه و مقدار جذب انرژی ویژه جرمی در این لوله ها مقایسه شده است. نتایج به دست آمده نشان می دهند که قابلیت جذب انرژی برخورد را می توان با انتخاب بهینه میزان گردی گوشه ها برای لوله های با شکل مقطع شبه–مثلثی و شبه–مربعی به طور معنی داری بهبود بخشید. همچنین، مشاهده می شود که برخی لوله های با مقطع شبه–مربعی حتی در مقایسه با لوله های دایروی دارای قابلیت بیشتر و عملکرد بهتری در جذب انرژی برخورد هستند.

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