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# Impact Behavior of Windowed Polygonal Tubes

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***Abstract*—The present paper aims to present a parametrical analysis from polygonal tubes under dynamical compressive loads, representing an impact collision between a barrier and an impact absorber. The finite element simulations are performed for quadrilateral and hexagonal cross sections, considering solid standard tubes as well as tubes with openings in their side walls, the so called windowed tubes. The parametric analysis asses the total absorbed energy, the specific energy absorbed (SEA) in addition to the initial peak load.**

***Keywords*—impact loading, dynamical analysis, FE simulation, polygonal tubes, energy absorption**

## I. Introduction

The vehicle safety is a research area in automobile design responsible for assure the passengers integrity after a collision event and to avoid the impact or reduce its probability. In a vehicle design different systems are developed in order to guarantee safety, among them we can name sensors systems and structural development as the main methods considered by researchers and engineers.

Although the vehicle structure must respect various requirements to increase the energy dissipation is the major objective related to the vehicle safety. The reduction in the transmitted kinetic impact energy to the occupants is achieved applying impact absorbers and by designing optimized energy minded structure.

Different types of energy absorber systems have been extensively studied applying brand new techniques. However, thin-walled tubes still show an elevated absorption capacity when compared to their easily manufacturing condition. The energy dissipation characteristics of thin-walled tubes are commonly related to the progressive folding mechanism and wherefore it is fundamental to guarantee local buckling, so leading to a maximal energy absorbed.

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Furthermore, the design of thin-walled tube absorbers may consider many cross sections, for instance the square tubes, which have been studied for [1], the polygons tubes presented by [2], the top-hat or double hat geometries evaluated by [3] and so on. Each one of them has its own behavior when subjected to an axial load thus presenting distinct buckling modes and energy absorption levels.

## II. Dynamic progressive buckling

Thin walled tubes under compressive loads may present a deformation pattern know as progressive buckling, where a series of folds are formed while the load is applied.

Fig. 1, presented by [4], shows a typical curve for progressive buckling , where the loads grows in the elastic stage until the initial peak load ($P_k$) is reached in point A, then tube structure lost its stability and the first fold formation begins. In point B the fold is completed and the structure recovers its stiffness starting a new cycle. Each peaks and valleys correspond to one fold developed in the tube and the area below the curve represents the total energy absorbed.

Theoretical models present equations in order to estimate the mean load ($P_m$) and the absorbed energy ($E_a$) according to the geometry of the tube and to the deformation mechanism expected to occur.

### A. *Buckling Mechanism for Square Tubes*

The theory for square tubes has been substantially studied by [5] and [6] where its buckling modes are presented and its dissipation mechanisms are explained. There are three distinct dissipation regions present in the square tubes, as can be seen in the hatched regions of Fig. 2. It shows an idealized collapse mechanism for part of a square tube.

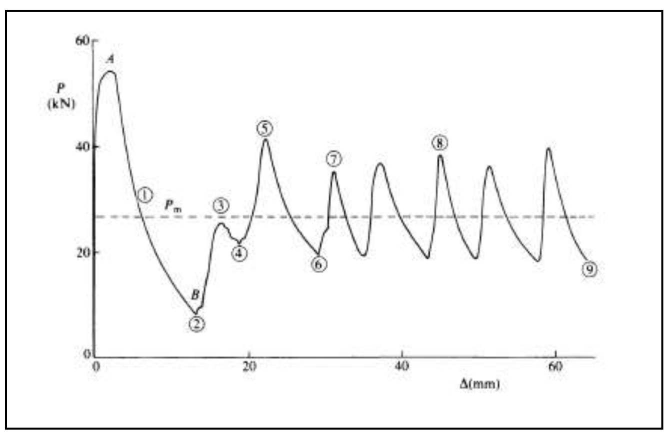


Figure 1. Typical load versus displacement curve for progressive buckling in thin walled tubes. [4]

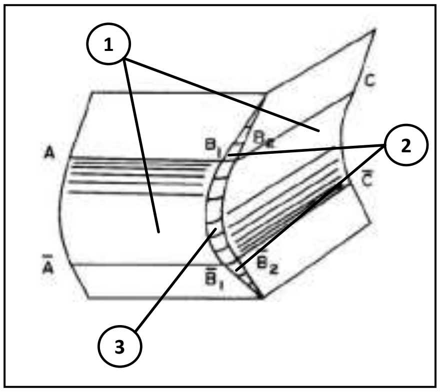


Figure 2. Collapse mechanism for square tubes. [6]

The two circular sections defined by the number 1 in Fig. 2 are formed by the folding of a wall from the square side. The other two regions, 2 and 3, in Fig. 2 originate from the movement of the square hinges since the continuity of the geometry needs to be maintained. The regions 2 and 3 require an excessive energy to allow the hinges motion and therefore are responsible for dissipating greater part of the input energy. In other words, the region 1 holds less importance in the energy absorption when compared to the travelling hinge mechanism composed by regions 2 and 3.

The absorbed energy in a square tube can be found as the sum from the three regions shown in Fig. 2. Thus, the energy from the circular walls in regions 1 is summed to the conical surface dissipated energy in regions 2 and the toroidal surface generated in order to keep the geometrical continuity at the hinges. As shown by [6] the energy can be obtained as the integration of the curvature deformation as well as the surface extension presented in the three different regions.

The deformation mechanism depends not only on the cross section but also on others geometrical characteristic as the wall thickness and on boundary conditions like the impact velocity. In the present paper windowed and not-windowed tubes are compared and different modes can be seen when the window size is changed.

Accordingly to [7], two different characteristics modes are performed by square tubes with windowed geometries. Symmetrical mode occurs when two opposite side walls bend inwards while the other two bend outwards, the extensional mode is characterized by the outward bends in all the four corners.

### B. *Specific Energy Absorption*

The total energy absorbed is the main measure to be evaluated when studying absorbers behavior and characteristics. However, when different geometry of tubes are taken into account this direct data cannot determine completely the most suitable absorber since it is not considering the geometries differences. Hence it is common that efficiency factors or parametric measures are applied when assessing different absorbers.

[4] introduces another measure of efficiency known as the specific energy absorption factor (SEA). SEA is the total energy absorbed per unit of mass of the crash box. By this way thin walled tubes with diverse cross sections or even with openings in their side walls can be compared.

In the present paper the SEA is used to assess the absorbing characteristics from the square and hexagonal windowed and not-windowed tubes.

### C. *Initial Peak Load*

The other parameter is the initial peak load that can be seen in Fig. 1 as the load at the first peak A. This measure must be reduced to avoid the development of high level loads in the absorbers. In a standard vehicle structure the parts are always connected to each other and in this system the first structural part, known as crash box, has to deform progressively, in order to absorb the biggest amount of energy reducing the damage in the connected structures. However, a high initial peak load can induce a deformation in the subsequent part before the beginning of the progressive buckling in the crash box.

The initial peak load has been studied by [8] for thin-walled grooved cylindrical tubes under axial compression. Their experiments show the influence of the insertion of triggers on the tube to reduce the initial peak load permitting the buckling to begins before the yielding load.

A trigger is usually a hole inserted in the tube to reduce the initial peak load and favor the development of folds. A hole removes material in the side wall of the tube reducing the cross sectional area in that region leading to a localized increase in the stress levels and consequently reducing the load necessary to former a fold.

## III. Analysis Definition

The analysis considered squares and hexagonal thin-walled tubes subject to an axial load caused by a striking mass of 500kg at 5m/s. Inertia effects and strain rate material sensitivity were also taken into account. The analysis was performed for a common thin-walled tube and for other three tubes with six windows at their side walls for each cross section. Thereby, the assessment of the windows influence at the crushing behavior could be defined. The buckling mode alteration due to the windows dimensions variation and the peak load decrease were thoroughly investigated.

Initially, has been considered standard tubes without holes, models SQ0 and HX0, with dimensions *L* and *t*, as shown in Fig. 3, respectively height 180mm and thickness 1.4mm. The remaining analysis comprise six square holes in the side walls with dimensions *a* and *b* (see Fig. 3 and Table 1), set apart by a distanced *H* = 30mm.

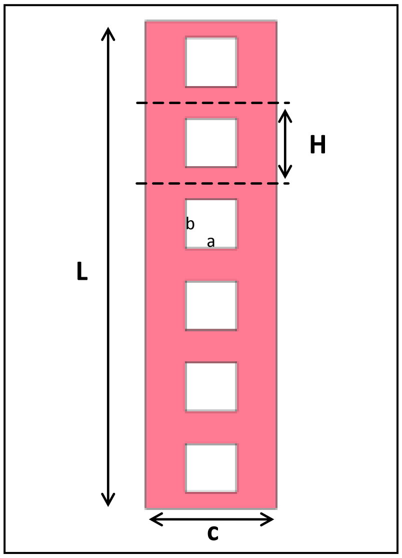


Figure 3. Geometry dimensions.

TABLE I. TUBE DIMENSIONS

| Model | Dimension | | |
|---|---|---|---|
| | *c (mm)* | *a (mm)* | *b (mm)* |
| SQ0 | 48.50 | - | - |
| SQ1 | 48.50 | 14.55 | 21.00 |
| SQ2 | 48.50 | 24.25 | 15.00 |
| SQ3 | 48.50 | 33.95 | 9.00 |
| HX0 | 30.38 | - | - |
| HX1 | 30.38 | 9.11 | 21.00 |
| HX2 | 30.38 | 15.19 | 15.00 |
| HX3 | 30.38 | 21.27 | 9.00 |

The material considered is mild steel with density ρ=7129kg/m3, Young's modulus E=211GPa, Poisson coefficient ν=0.3, yield stress σy=26.5MPa and ultimate stress σu=431.6MPa. The Cowper-Simonds constitutive equation is applied in order to consider the dynamic flow stress, with D=16.640, p=3.53.

The numerical simulations have been performed at LS-Dyna 9.71 using 106200 solid elements for the square tube without windows. In the other models the quantity of elements is slightly lower due to the windows. Imperfections were considered applying the command *perturbation node* from LS-Dyna. An elastic-plastic material has been considered for the tube, applying the material command *piecewise linear plasticity*.

## IV. Results and Discussions

The energy absorption and the SEA as well as the initial peak load and the characteristic buckling mode have been evaluated for all the models simulated.

Fig. 4 shows the models simulated at 2ms, on the right can be seen respectively models SQ0, SQ1, SQ2, SQ3 and on the left HX0, HX1, HX2, HX3.

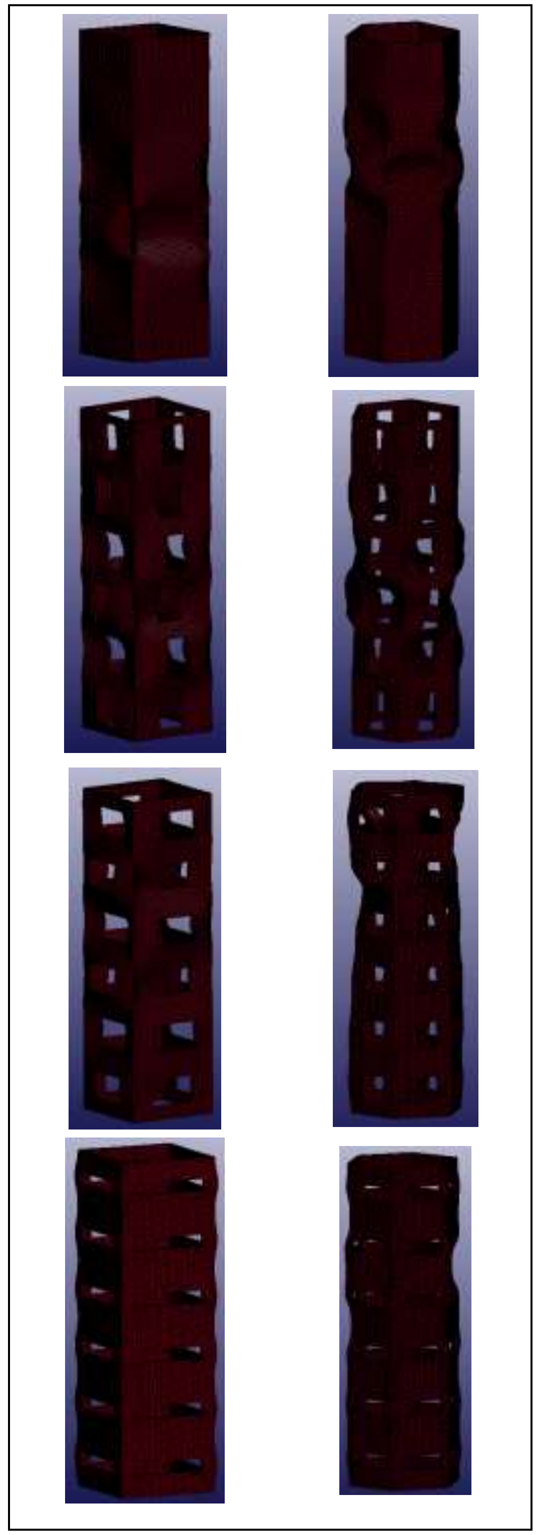

Figure 4. Simulated models at 2ms. Right: respectively SQ0, SQ1, SQ2, SQ3. Left: respectivelly: HX0, HX1, HX2, HX3.

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It is possible to identify in Fig. 4 that the models from SQ0, SQ1, SQ2 and HX0, HX1, HX2 are developing a symmetrical mode during the load application. On the other hand, models SQ3 and HX4 have buckled trough the extensional mode. The results agree with [7] since the transition from the symmetrical to the extensional mode happens when the width of the windows rise.

When the value of *a* increases, the amount of material to support the axial load reduces around the window. Thus, above certain values, the corners of the square could not sustain the load and the hinges are forced to bend outwards, resulting in the extensional mode.

The Fig. 5 shows the load versus displacement curves until the first fold is completely formed. The square tubes present a higher initial peak load when compared to the hexagonal tubes, it can be explained by the lower stiffness in the hexagonal corners leading to a lower resistance to buckling.

The reduction of the cross sectional area produces a higher stress concentration around the edges decreasing the critical buckling load. In other words, the value of the initial peak load is related mainly to the elastic behavior of the component during compression. When the windows are inserted in the model, the reduction of the cross sectional area makes the elastic region smaller; consequently, the buckling initiates in a lower level of load. To illustrate this, in Fig. 5, it is shown that for higher values of *a* there is a drop in the initial peak load, this can be seen for square and hexagonal tubes.

The energy absorption analysis has been restricted to the initial instants of the impact to enable the assessment of the total energy dissipated in the formation of one fold and considering the influence of the size of the window and the characteristic mode developed in the tube. Although for the models where the extensional mode occurs have lower energy absorption in the beginning of the analysis, where the elastic deformation is predominant, after 4ms the plastic deformation starts and their energy dissipation overtake the other windows models.

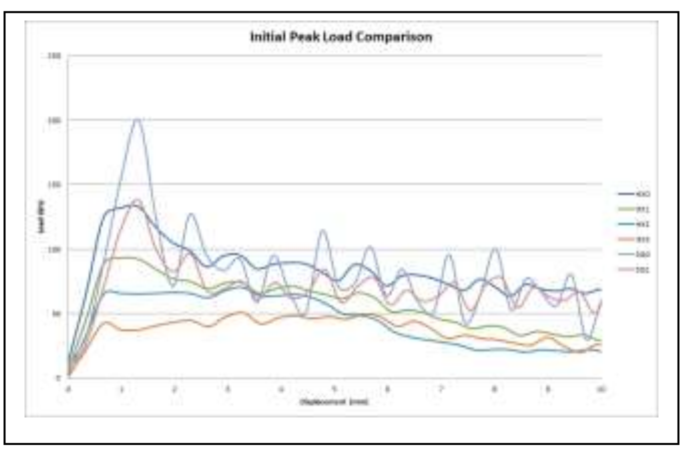


Figure 5. Load versus displacement at first formated fold for simuated modells.

Table 2 lists the energy absorption levels for all the models at 15ms, when the first fold is completely formed. Additionally, the Specific Energy Absorption (SEA) indicates that the windowed tubes have very similar values to the original tubes.

TABLE II. ENERGY RESULTS COMPARISON

| Model | Energy Measure | |
|---|---|---|
| | *E (kJ)* | *SEA (kJ/kg)* |
| SQ0 | 4.10 | 12.12 |
| SQ1 | 3.52 | 10.99 |
| SQ2 | 3.28 | 10.37 |
| SQ3 | 3.90 | 12.19 |
| HX0 | 3.8 | 7.02 |
| HX1 | 2.63 | 5.57 |
| HX2 | 2.24 | 4.88 |
| HX3 | 2.16 | 4.57 |

Summarizing, windowed square tubes can maintain the energy absorption levels while providing lighter structures and reducing the initial peak load. The efficiency of this geometrical configuration hinges on the windows being placed in the side walls of the square. A great part of the energy is dissipated by the travelling hinges mechanism, not much affected by the presence of windows.

On the other hand, hexagonal tubes showed a lower level capacity when compared to square tubes, mainly when take into account the SEA, that may be caused by the higher mass for hexagonal tubes. Furthermore, for hexagonal tubes the addition of openings in the side walls did not represent an improvement in absorption characteristics. In fact, in hexagonal tubes the hinges can sustain the stability at a lower level when compared to a square cross section and also the amount of material removed in the opening is higher since there are six side walls.

## V. Conclusion

The present paper studied the presence of windows on the side walls of thin-walled square and hexagonal tubes subject to an axial dynamic impact. It was shown that the size of the window influences the characteristic mode developed during the crash, both symmetric and extensional mode occurring, depending on the window dimensions.

Additionally, the windowed models have been compared to a conventional tubes and the energy absorption efficiency for the five models simulated was very similar to the common square cross section tube. When considering the specific energy absorption, the values for the model SQ3 were still higher than that for the reference model SQ0. This result is probably because the square holes are applied in the side walls, not affecting the travelling hinges mechanism, which is responsible for great energy dissipation.

Differently, for hexagonal tubes the openings reduced the energy absorption capacity during an impact. This may be caused by the great volume extracted from the side walls moreover to the lower stiffness in the hexagonal corners in comparison to a square tube.

The windowed tubes present significantly reduction in the initial peak load, allowing the control of the buckling initiation. The decrease in the cross sectional area around the windows enabled the initiation of buckling to occurs faster, reducing the elastic region in the load-displacement curve.

Window cut square tubes have proved an interesting solution in new crash box development since they have guarantee an elevated energy absorption level while allowing an adequate control of the buckling behavior and a low initial peak load. However the addition of windows in the hexagonal models would request further studies to evaluate a proper application where a trigger can be add to a tube without the reduction in energy absorption behavior.

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