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FINITE ELEMENT SIMULATION MODELING OF AXIAL DYNAMIC ENERGY ABSORPTION ON CONICAL TITANIUM ALLOY

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Abstract
This paper presents the finite element simulation of conical unfilled and foam filled titanium alloy subjected to dynamic axial loading. Validation of finite element analysis packages PAM-CRASH 2G version 2005 was compared with existing results from experimental analysis has been done by previous investigation to ensure that the numerical analysis is sufficiently accurate. On the other hand, proper thin-walled titanium alloy as well as polymeric foam material offer vast potential for optimally tailoring a design to meet crashworthiness performance requirements. The energy absorption characteristics of thin-walled alloy and polymeric foam with variable cross-sections in terms of vertex angle and wall-thickness are numerically studied. Results showed that the tube's energy absorption capability was affected significantly by variations of velocity-impact and wall-thickness as well as vertex angle of conical cross-sections. It was also found that as the filling polymeric foam into thin-walled tube increases the amount of absorbed energy than the empty tubes.

Keywords: Energy Absorption, Thin-Walled Structure, Finite Element Analysis, Foam-Filled

1. Introduction

Structural hollow sections have recently been introduced to the vehicle of aerospace, automotive and marine construction sector. These structural elements can offer greater structural efficiency than circular hollow sections when subjected to bending or combined loading, or when used as columns with intermediate restraint about the weaker axis, since they possess different major and minor axis flexural properties. Despite recent investigations involving the testing, numerical modeling and development of design rules for cross-section structure, a number of aspects of their structural response remain unexplored. In particular, the behavior of profiles in the post ultimate region has not yet been examined. Besides defining the unloading branch of the load–deformation curve, an accurate description of this region also allows the possibility of quantifying the ductility of the system. The aim of the present work is to develop a finite element model to predict the full load–deformation response of titanium alloy with and without filling polymeric foam as filler under pure compression. Towards this end, rigid plastic theory has been applied to profiles based on the method presented by Murray for plates. There is comparatively little information available concerning the crushing behavior of core and core-less thin-walled titanium alloy shells. But it would appear that no previous investigation has been made in the literature to study the crushing behavior of thin-walled cones with and without polymeric foam filling. Studies on energy absorption capability as well as the load carrying capacity of thin-walled titanium alloy tubes and cones are however still scarce. This was one of the motivating factors behind this paper. The purpose of this study is to investigate and determine the capability and influence of thin-walled titanium alloy conical with and without polymeric foam filled as collapsible energy absorber devices.

1.1 Finite element modeling

It is well understood that numerical analysis using finite element analysis allows wide range of optimization. Therefore, PAM CRASH 2G version 2005 was used throughout the analysis. Throughout the simulation process, three differences geometry of tubes have been modeled, which are straight tube, conical tubes with different vertex angles, and a solid element as a foam structure inside shell element surfaces. Shell elements are used to model structures in which one dimension the thickness is significantly smaller than the other dimensions, and the stresses in the thickness direction are negligible. Generally three-dimensional shell and solid elements are available with three different formulations; general purpose, thin only and thick only. General-purpose shell elements are valid for use with both thick and thin shell problems. Furthermore, all general purpose shell and solid elements consider finite membrane strains. All special purpose shell and solid elements assume finite rotations; however, they assume small strains. The thin only shell elements enforce the constraints; that is, plane sections normal to the midsection of the shell remain normal to the mid surface. The constraint is enforced either analytically in the element formulation or numerically through the use of penalty constraint. The thick only shell and solid elements are second order quadrilaterals that may produce more accurate results than the general purpose shell and solid elements in small strain application where the loading is such that the solution is smoothly varying over the span of the shell.

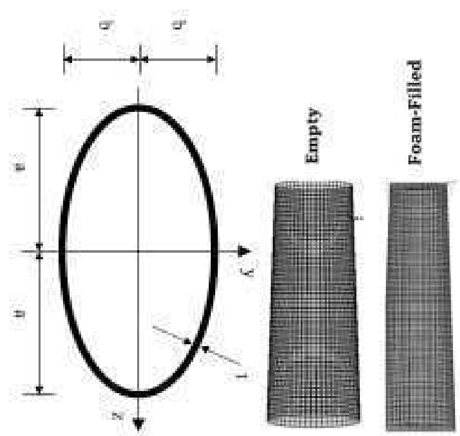

Figure 1: Dimension of conical modeling

Table 1: Dimension of conical finite element modeling

Type (Conical)	Length (mm)	a (mm) Bottom	a (mm) Top	b (mm) Bottom	b (mm) Top	t Thickness (mm)	Polyurethane Foam Density	Vertex Angle (α)
Type A	200	40	40	30	30	1, 1.5, 2	0, 60	0
Type B	200	40	33	30	23	1, 1.5, 2	0, 60	2
Type C	200	40	26	30	16	1, 1.5, 2	0, 60	4

All models were developed using the nonlinear finite element software PAM CRASH 2G version 2005. The numerical solutions were also carried out using the explicit finite element code PAM-CRASH 2G. PAM-VIEW was used as the visualization tool postprocessor. The titanium alloy tubes were modeled using Belytschko-Tsay-4 node-thin shell elements. These elements are three-nodded reduced integration shells with six degrees of freedom per node and are suitable for thick and thin applications. A uniform mesh size of all modeling 3 mm was employed, with maximum element sizes of 5 × 5 mm. The foam filler and the compression test plates were modeled with 8-node solid elements. Finite element meshes of the dynamic axial compression testing of empty and foam-filled tubes are shown in Figure 2. The cross-sectional dimensions of the modeled conical sections were kept constant at 40 × 30 mm, while three variations of wall-thickness of 1, 1.5 and 2 mm were considered, providing a range of cross-section slenderness values. The member length was fixed at 200 mm, which was sufficiently short to ensure no global buckling, and all sections were subjected to concentric compression. All models were assigned rigid plastic material properties without strain hardening or residual stresses, to allow direct comparison.

1.2 Parametric finite element analysis

To carry out real crushing finite element formulation, the value of the velocity and time need to be suitable. The initial velocity '*v*' of the tube are set to be 20, 30 and 40 m/s, hence calculated value for crushing time, *'t'* was found to be 0.02 s. The other important parameter of representing crushing event is boundary condition. In structural analyses, boundary conditions are applied to those regions of the model where there displacements and/or rotations are known. Such regions may be constrained to remain fixed or may have specified nonzero displacements and/or rotations. In this model the fixed-free condition has been used where the top section of the tube is constrained completely and, thus, cannot move in any direction also. The bottom section, however, is fixed in the horizontal direction but is free to move in vertical direction. The direction in which motion is possible is called degree of freedom *'dof'*, hence this model only has a single degree of freedom.

2. Material properties

Material type 103 corresponding to the elastic–plastic isotropic thin shell material model was used for the tube material. Material type 103 uses an enhanced plasticity algorithm that includes transverse shear effects. It exactly satisfies Hill's criterion and precisely updates the element thickness during the plastic deformation. The base and upper plates were modeled using an elastic–plastic solid element model called material type 1. This material model corresponds to elastic–plastic behavior with isotropic hardening and elastic–plastic behavior can be introduced by specifying the yield stress and tangent modulus or specifying the effective plastic strains and stresses.

The material behavior of polyurethane foam was modeled using the crushable foam solid model, namely material type 2. Material type 2 corresponds to the solid materials exhibiting coupled volumetric bulk and deviatoric shear plasticity. The coupling between both parts of the material response is established by a pressure dependent von-Mises yield surface. Three contact models were used in the modeling: (i) a tied contact between the base plate and the empty tube, (ii) a node-to-segment contact between the upper plate and the tube ends and (iii) a self-contact (self-impacting contact with edge treatment) in order to prevent the interpenetration between folds in the tube wall during plastic deformation. The self-contact impact algorithm of type 36 allows all slave segments defined in a given sliding interface. No segment orientation is needed to be specified, since the algorithm automatically detects penetrations and keeps in memory the segment side from which a slave node comes into contact. The self-contact impact algorithm also uses a search algorithm so-called 3D Bucket sort algorithm in which the 3D slave surface is subdivided into a number of buckets and the slave nodes are recalculated in terms of bucket coordinates.

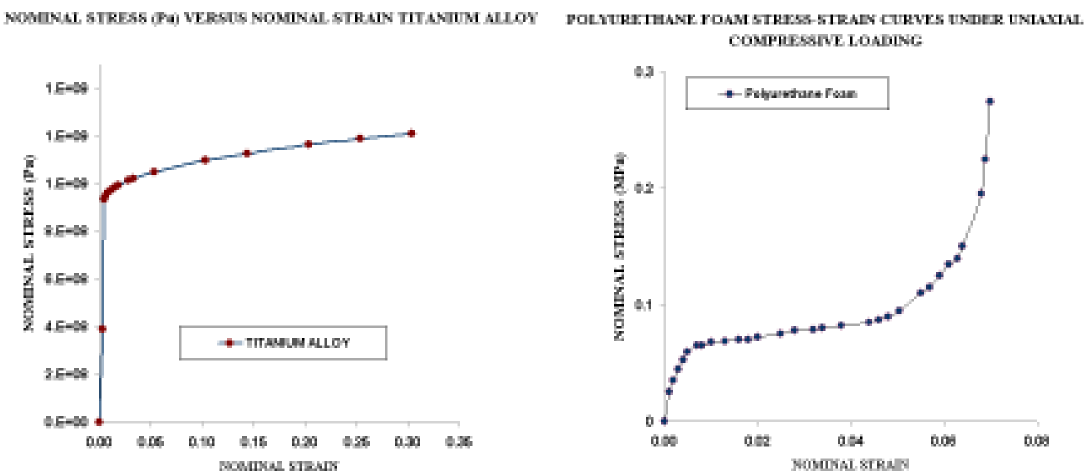

Figure 2: Nominal stress and strain curve of titanium alloy and polyurethane foam

2.1 Crushing process

The simulation, which normally is run as a background process, is a stage in which PAM CRASH 2G/Explicit solves the numerical problem defined in the model. PAM CRASH 2G/Explicit is a special-purpose analysis product that uses an explicit dynamic finite element formulation. It is suitable for short, transient dynamic events, such as impact and blast problems, and is also very efficient for highly nonlinear problems involving changing contact conditions, such as forming simulation. It is well known that a nonlinear structural problem is one in which the structure's stiffness changes as it deforms. All physical structures are nonlinear. In this simulation the stiffness is fully dependent on the displacement; the initial flexibility can no longer be multiplied by the applied load to calculate the spring's displacement for any load. Results evaluation can be done once the simulation has been completed and the reaction forces, displacements, energy or other fundamental variables have been calculated. The evaluation is generally done interactively using the visualization module of PAM CRASH 2G/CAE. The visualization module, which reads the neutral binary output database file, has a variety of options for displaying the results, including color contour plots, animations, deformed shaped plots, and X–Y plots. The total work done (W) during the axial crushing of the cones are equal to the area under the load/displacement curve and is evaluated as:

$$W = \int P ds \quad (1)$$

where, P is the force acting on the tube. Therefore the specific energy absorption per unit mass, E is recognized as:

$$E = \frac{W}{m} \quad 2)$$

where, m is the crushed mass of the thin-walled tube.

It was assumed that thin-walled titanium alloy has only isotropic strain hardening, and for quasi-static loading, the strain rate effects on the yield strength were neglected due to the relatively low overall average load rate used in the tensile tests. For dynamic loading, the effect of strain rate was included in the finite element model using the Cowper–Symonds constitutive equation given by the following relation:

$$\dot{\varepsilon}_p = D^t \left(\frac{\sigma_d}{\sigma_s} - 1 \right)^{q^t} for \mapsto \sigma_d \geq \sigma_s \quad (3)$$

where σ_d is the dynamic flow stress at a uniaxial plastic strain rate ε_p, σ_s, the associated static flow stress, and the constants D^t and q^t are tube material parameters. Equation (3) is an overstress power law that was incorporated into the finite element model.

2.2 Validation of the finite element model

To ensure the FE model was sufficiently accurate, it was validated using existing theoretical models which predict the mean crush loads for the axial crushing of a straight round tube under dynamic axial loading has been mentioned by L. Aktay et al. A model of aluminum straight rectangular tube was developed using finite element code PAM CRASH 2G/Explicit with the free length of the tube being 300 mm, while the diameter of the tube was 30 mm (see Figure 3). The tube thickness is 3 mm and loaded with impact mass of 90 kg at a velocity of 15 m/s. Figure 3 shows a complete crushing of a rectangular tube and the failure mechanism of the tube, which is almost identical by the descriptions of Aktay et al.

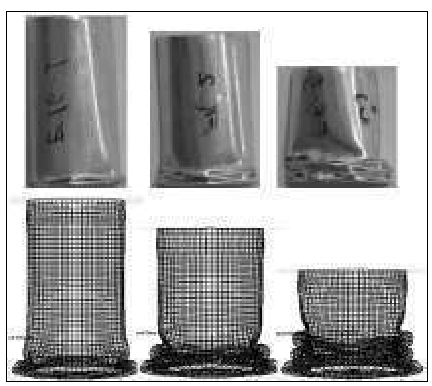

Figure 3: Verification and calibration of experimental analysis and numerical modeling

The validation is of particular importance, since it proves that the simulation model can be effectively used to examine the failure mechanism of a crushed tube. The tube response obtained by the finite element model was compared with that of the experimental analysis for a range of crash parameters under dynamic loading as shown in Figure 3. The crash parameters that have been analyzed were total compression, peak load, mean load, and total energy absorbed by the tube. Numerical results show a higher value as compared to experimental one. It can also be seen that the difference between value of experiment and numerical analysis from the table below can be considered as reasonable as listed in Table 2. This discrepancy might be due to the assumptions that have been made throughout the simulation process.

Table 2: Comparison of experimental and numerical analysis

Parameters	Experimental	Numerical	Error (%)
Total compression (mm)	185	195	5
Peak load (kN)	295	275	6
Mean load (kN)	211	192.3	8.8
Absorbed energy (kJ)	39.1	33.2	15

3. Result and Discussion

3.1 Empty titanium alloy conical tubes

The dynamic load displacement curves of empty aluminum conical tube models constructed with the numbers of elements 3 X 3 mm, are shown in Figure 4 for wall-thickness of 1 mm tube, respectively. In these figures, for the purpose of comparison both of deformation crushing element model and dynamic load versus displacement curve empty of empty titanium alloy conical tube is also shown. As seen in Figure 4, the dynamic load versus displacement curves of models with deformation crushing of element model on 0, 30, 60, 90, 120 and 150 mm give reasonably good agreements with the different of shape of conical with 0, 2 and 4 degree of vertex angle.

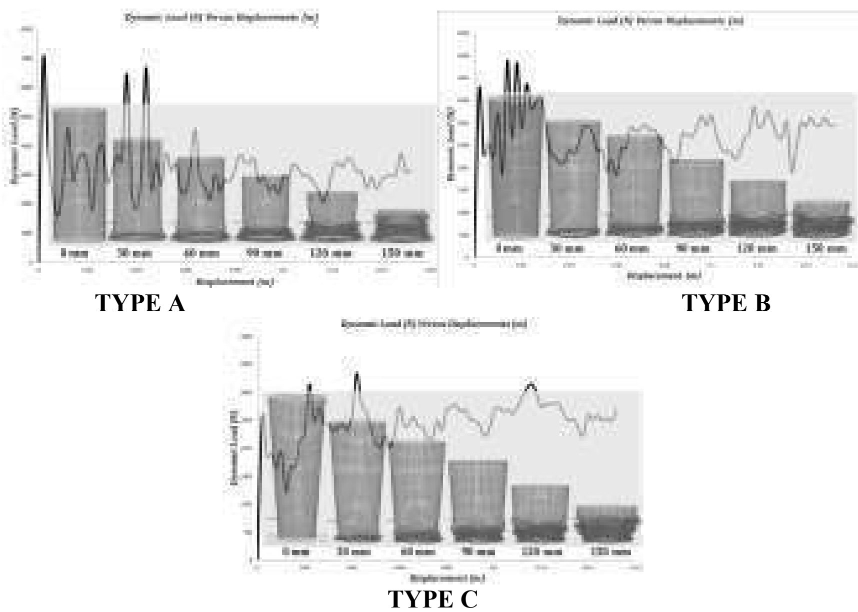

Figure 4: Dynamic load (N) versus displacement (m) type of A, B and C unfilled-foam conical profile (Wall-Thickness = 1 mm: Velocity = 10 m/s)

The effects of vertex angle on deformation arrangement can be seen in figure which is the reduction of dynamic load after first crash initiator before appeared a folding in surface element model. The lobe of folding were easily be made from level to level deformation of element found corresponding to the points of the first and the last fold formation seen in the Figure 4. Finite element models were, however, continued with calculated the crushing process from un-deformed until final progressive collapses. The initial peak load were higher in vertex angle of 0 degree was 7000 N cause by the frontal surface impact having a large diameter compared to vertex angle of conical 2 and 4 degree were 3700 N and 2500 N, respectively. Whereas having smaller diameter of frontal surface impact will crush progressively without collapse buckling. The agreement between three different of vertex angle of conical were similarity of average load values. It is quite satisfactory except in the initial and mean load of the load displacement curves, corresponding to the points of the first and the last fold formation. The range element model deformed shapes of empty 150 and 190 mm conical corresponding to the deformation ratios of 20%, 50% until 90% are sequentially shown in Figures 4.

3.2 Polyurethane Foam Filled titanium alloy tubes

The density polyurethane foam-filled conical 60 kg/m^3 was selected in this study and the model resulted in diamond mode of collapse same as the collapse mode of empty conical. Figure 5 showed the agreements between the models deformed shapes of tubes at 30 until 150 mm deformations. The effect of foam filling is to reduce the fold length and hence to increase the number of folds formed a result which was also previously found in foam filled titanium alloy and steel tubes.

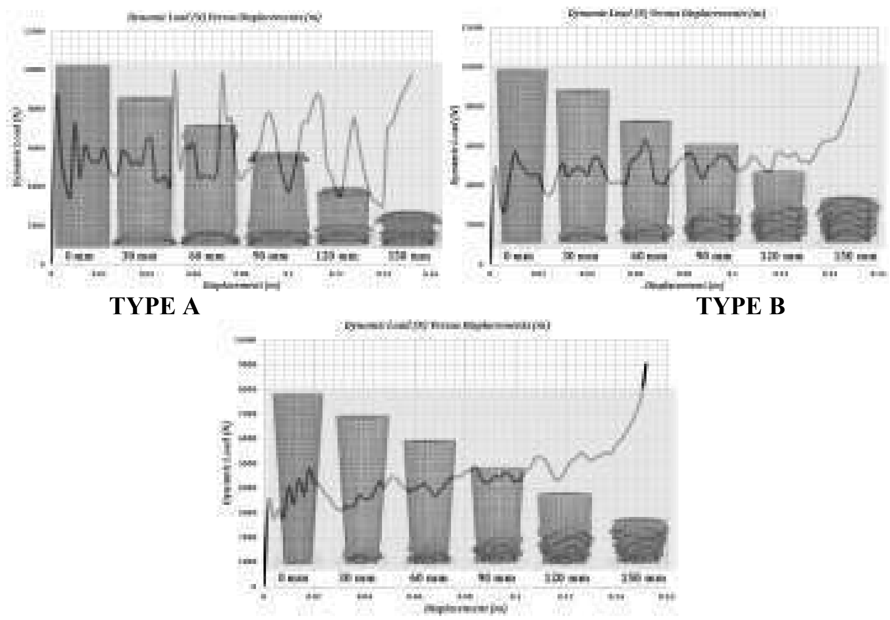

Figure 5: Dynamic load (N) versus displacement (m) type of A, B and C filled-foam conical profile (Wall-Thickness = 1 mm: Velocity = 10 m/s)

The load values of model further show good coincidence with those of numerical analysis as depicted in Figure 5. In contrast all conical, the foam filling in final deformed tube switched the collapse mode from diamond to concertina mode and again model show very similar deformed shapes load–displacement curves in Figure 5. The reduction in fold length in filled tube also increased the number fold showed in figure. The restraining effect of foam filler to the fold formation was explained to be the major cause of the reduced fold length in filled tubes. The entrance of the conical column wall into foam filler results in shorter fold lengths and increases the number of fold formed. Santosa et al. stated that the encroachment of the column wall into the aluminum foam filler allows an additional compression in the foam and retards the sectional collapse of the column. A similar filler conical encroachment effect is also seen on the surface images of partially crushed foam-filled tubes fillers and tubes in Figure 5. The change of collapse mode from diamond into concertina with foam filling was also observed previously in Al foam-filled Al and steel tubes, polyurethane foam filled Al tubes and wood sawdust-filled plastic tubes and it was proposed to be due to the thickening effect of foam filling, which drives the deformation shift from diamond to concertina. It was also shown by Hanssen et al. that after a critical aluminum foam density the deformation mode in filled aluminum tubes shifted from diamond to concertina mode. A similar mode-shift was also found in polyurethane foam-filled thin-walled titanium alloy tubes with the increasing of foam density.

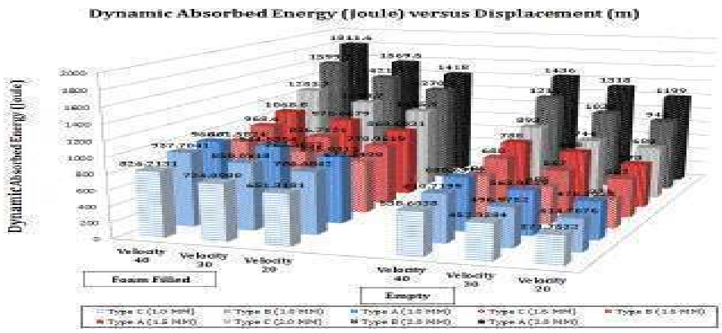

Figure 6: Dynamic absorbed energy (Joule) of unfilled and foam-filled conical profile

For the present work, the numerical energy absorption corresponding to the 75% deformations of empty and foam-filled tubes are shown sequentially for all model analysis conical tubes in Figure 6. Again, numerical dynamic energy absorption is well agreed with each other; the maximum differences between them are almost 5%, 8 % and 7% for 1, 1.5 and 2mm wall-thickness of conical tubes, respectively. It is also noted in Figure 6 that 0 kg/m3 density foam filling is almost ineffective in increasing the capability absorbed energy over those of filled foam ones. The percentage increase in absorbed energy density foam filling of 60 kg/m3 are found numerically 15% for 2 mm conical tube and numerically almost 10% for 1 and 1.5 mm conical tube, respectively. These results also show agreement with the study of Guillow et al. in that the average crushing load of foam-filled tube is higher than that of simply addition of crushing loads of empty tube alone and foam filler alone. It should, however, be stated here that the comparison of the energy absorptions between empty and filled tubes should be made on the equal mass basis by taking into consideration the thickening of the empty tube wall. Previous studies on aluminum foam-filled tubes and boxes have shown that there existed a critical total filled tube mass and the corresponding critical foam density above which the use of foam filling became more efficient than empty tube. The effect of polystyrene foam filler density and the critical total mass of filled tubes should be further investigated numerically and experimentally for efficient crash element designs.

4. Conclusions

The dynamic crushing behavior of polyurethane foam-filled titanium alloy extrusion conical tubes was investigated numerically. The numerical solutions were carried out using the explicit finite element code PAM-CRASH 2G version 2005. The kinetic and internal energy histories were investigated at different mass density scaling and deformation velocities and based on efficiency and time inexpensive modeling a mass density scaling with a factor of 1000 and a loading velocity of 20, 30 and 40 m/s were chosen for the modeling. In general, satisfactory agreements were found between the results of finite element modeling and experimental dynamic crushing tests of foam-filled thin-walled titanium alloy tubes. The model has also highlighted several effects of foam filling in thin-walled conical titanium alloy tubes. The number of folds formed in foam-filled tubes, both in diamond and concertina mode of deformation, increased with foam filling and also with increasing foam filler density. The energy absorptions in foam-filled tubes were shown to increase with increasing filler density and higher than the sum of the energy absorptions of empty tube (alone) and filler (alone).

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