IOSR Journal of Mechanical & Civil Engineering (IOSRJMCE)
e-ISSN: 2278-1684,p-ISSN: 2320-334X
PP 37-44
www.iosrjournals.org

Quasistatic and Dynamic Analysis of Rectangular Tube

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Abstract : *The axial folding of metal tubes has been known for several decades as an excellent energy absorbing mechanism. Today, components based upon this principle are utilized in high-volume industrial products such as cars, trains, and other sectors where energy, during a crush situation, needs to be absorbed in a controlled way. A rectangular tube of mild steel is analyzed for its energy absorption capability using the finite element based software LS-DYNA. For preprocessing & postprocessing, software HYPERWORKS is used. Meshing is done using Belytschko Tsay shell element with 5 mm element size. Quasistatic & Dynamic Analysis are carried out in this work for axial loading. Mean crushing load is taken as a parameter for validation of the results. The analytical equations of square tubes are used for comparison of mean force. Energy absorption, Peak crushing force and deformation of the tube are observed during the analysis.*

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Keywords - Quasistatic analysis, Dynamic Analysis, Rectangular Tube, energy absorption

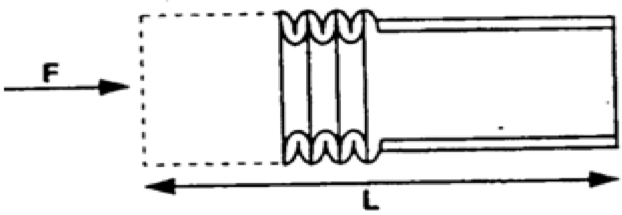
I. INTRODUCTION TO ENERGY ABSORBERS

The box tubular structures are among the energy absorbing devices which can be used in many applications such as cars, ships, and aircrafts and so on. The advantage of the tubular is that it has simple geometry, low cost and high energy absorbing capability. Crashworthiness studies have attracted much attention in recent years particularly to analyzing the deformation characteristics and determining the energy absorbing efficiency of various thin wall crashworthy components of different materials. For example, metals, plastic, composite etc. and for various geometrical shapes, cylindrical tubes, cones, conical frusta, square tubes, square frusta and pyramids all grooved or un-grooved when subjected to axial crushing/compression loading [1]

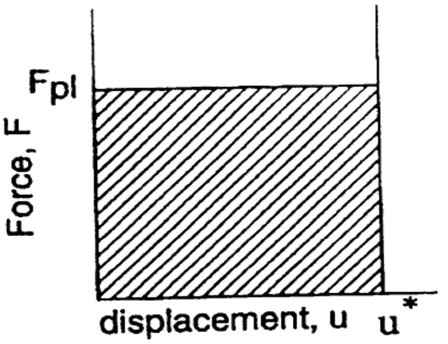
Energy absorption: Fig.1 is indicating a tube type of energy absorber. The dotted lines are indicating undeformed picture.

Ideal energy absorber has a long, flat loaded-deflection curve: the absorber collapse plastically at a constant force called the 'plateau force' as shown in the Fig. 2 Energy absorber for crash protection is chosen so that the plateau force is just below that which will cause damage to the protected object: the best choice is the one which has the longest plateau (i.e. the largest value of lock-up displacement u*), and therefore absorbs the most energy. Solid sections do not perform well in this role. Hollow tubes, shells, and metal honeycomb have the appropriate shape of load-deflection curve.

An ideal energy absorber is defined as one which maintains the maximum allowable retarding force throughout the stroke, apart from elastic loading and unloading effects. However a designer must often trade off considerations of cost, volume, stroke, weight, deceleration, etc., against efficiency. Above all, an energy absorber must be reliable and in many cases versatile enough to absorb dynamic load which may strike at a random location.



(F = Impact force, L = Undeformed length, δ= deformation)
Fig. 1 Representation of a crushed structure [2]



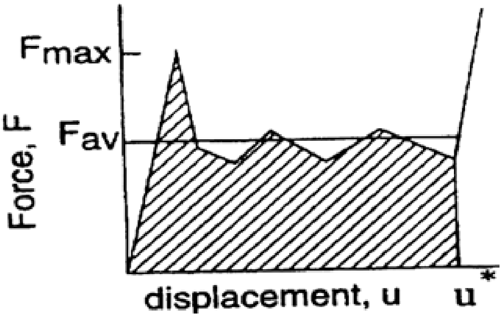


Fig. 2 Force-Displacement Characteristic of an Ideal Energy Absorber [2]

Fig.3 Typical Force-Deflection Characteristics of a Practical Energy Absorber [2]

Practical energy absorbers have a characteristic load-deflection response as sketched in Fig. 3. An initial (or subsequent) peak value of load F_{max} exceeds the average value F_{avg}, and leads to an increased acceleration and potential damage of object.

Integration of the force versus deformation curve gives the energy absorption for the corresponding deformation.

The mean or average crush load is defined as the absorbed energy divided by the current deformation. [2]

II. REVIEW OF ACCIDENTS IN INDIA

The With the increase in number of vehicles every year, there is increase in number of accidental deaths also. Table 1 table shows the increase in the number of accidental deaths in recent years.

Table 1 No. of road accidental deaths in india [3]

Sr. No.	Year	No. of Road Accidental Deaths
1	2009	1,26,896
2	2010	1,33,938
3	2011	1,36,834
4	2012	1,39,091
5	2013	1,37,423

Table 2 shows the accidental deaths as per the type of vehicle.

Table 2 Road Accident Deaths by Type of Vehicle during 2013 [3]

Sl. No.	Type of Vehicle	No. of Road Accidental Deaths	% share of total Vehicles
1	Truck/Lorry	24,081	17.5
2	Bus	12,055	8.8
3	Tempo/Vans	8,138	5.9
4	Jeep	8,596	6.3
5	Car	14,803	10.8
6	Three Wheeler	6,492	4.7
7	Two Wheeler	34,187	24.9
8	Bicycle	2,587	1.9
9	Pedestrians	12,385	9.0
10	Others	14,099	10.3
Total		1,37,42	100.0

From this data, it seen that Jeep & Cars together are contributing nearly 17.1 % of total deaths.

Need of the research: From the extensive literature review it is observed that most of the researchers have done study of square tube of aluminum or aluminum alloy whereas rectangular tubes are more suitable for mounting of different parts in light commercial vehicles like jeep and others [4]. Also compared to aluminum material, steel is having more energy absorption for a given volume of material which is important in case of vehicle structure.

Thus the need is to concentrate more research on energy absorption capability of rectangular steel tubes.

III. MATERIAL PROPERTIES

The cross section of the tube taken for study is 110 mm x 60 mm with thickness 2.5 mm and length 250 mm. The material of the tube is mild steel. The material properties of earlier researcher [5] are used directly for this study.

True stress –True strain curve for this material is as below.

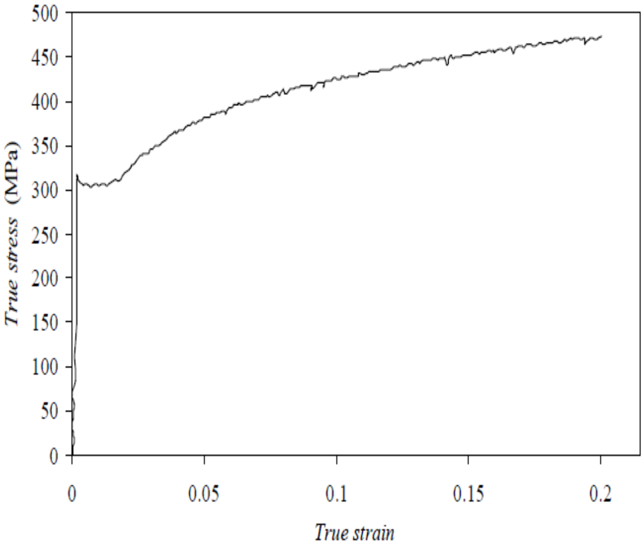


Fig. 4 True static stress - strain curve for mild steel [5]

Table 4 Material Properties [5]

Yield strength (σ_y)	304 MPa
Young's Modulus (E)	205GPa
Poisson's ratio (υ)	0.3
Density (ρ)	7700 kg/m^3

Assumption: Isotropic, No strain hardening & strain rate effect on yield strength of steel as low overall average strain rate is used in tensile test.

Table 5 True stress- plastic strain data points for steel in FE model [5]

σ_t (MPa)	304.6	344.19	385.51	424.88	450.39	470.28
ε_p	0	0.0244	0.0485	0.0951	0.1384	0.1910

IV. ANALYTICAL SOLUTION

To The quasistatic mean load for rectangular tube is obtained using the expression proposed by W. Abramowicz and N. Jones [6].

These equations used for the validation of analysis of rectangular tube are strictly speaking only applicable to square tubes, however it has been found to produce reasonable results for rectangular tubes [7].

Analytical solution of quasistatic analysis:

The mean crushing load (P_m) is given by

$P_m = M_o \times 52.22\ (c / h)^{1/3}$

Here,

M_o = Fully plastic bending moment per unit length for sheet metal

$M_o = \sigma_0 h^2/4$ …….Here σ_0 is yield stress of the material.

h = thickness of tube & c = side length of tube

Here as tube is rectangular, 'c' is taken as mean of the lengths of two sides

So, $c = (110 + 60)/2 = 85$mm.

$h = 2.5$ mm & $\sigma_0 = 304$ MPa

Hence , $M_o = 304 \times (2.5)^2/ 4 = 475$

Thus,

$P_m = 475 \times 52.22\ (85/2.5)^{1/3}$

$\mathbf{P_m = 80.356}$ **KN**

Analytical solution of dynamic analysis:

Here also equation corresponding to the square tube is used for the validation of the results of dynamic analysis.

$P_m^d = M_0 \times 52.22\ \{1 + (0.33V/c\ D)^{1/P}\}\ (c/h)^{1/3}$

In dynamic analysis, strain rate hardening is to be taken into account. The effects of strain hardening can be accounted for by using the average of the yield & ultimate stress as the value of $\sigma_{0.}$ [8][9].

Thus σ_0 is flow stress of the material in dynamic analysis.

Here σ_0 = flow stress = $(\sigma_{0yield} + \sigma_{0ultimate})/2$

$= (304 + 470) / 2 = 387$

$M_o = 387 \times h^2/4 = 387 \times (2.5)^2/ 4 = 604.69$

$P_m^d = 604.69 \times 52.22\{1+(0.33\times \frac{15}{85 \times 6844})^{1/3.91}\}\ (85/2.5)^{1/3}$

$\mathbf{P_m^d = 107.46}$ **KN**

V. QUASISTATIC ANALYSIS

Finite element analysis is carried out to determine the performance of the rectangular tubes. Software HYPERWORKS is used for pre-processing & post-processing while LS DYNA is used for processing.

A. Finite Element Model & Meshing

Model of the rectangular tube is as shown in the Fig.5. Meshing is done using Belytschko Tsay shell element. Total No. of Elements was 3760 & No. of Nodes was 3873.

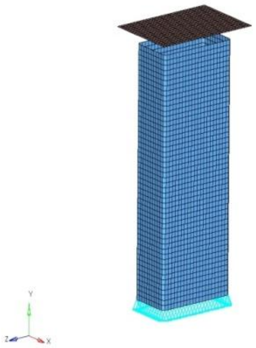


Fig. 5 Meshed model of rectangular tube

A fillet of 3 mm is used at the corners of the model. Element size used was 5 mm x 5 mm as is used by Nagel [5] for a similar geometry. On the top side a rigid plate is modelled.

B. Boundary conditions & Load conditions

Boundary condition: The tube is constrained at the bottom in all translational and rotational directions as in Fig. 5.

Load condition: The Rigid plate is given prescribed velocity of 10 mm per minute in vertically downward direction.

VI. DYNAMIC ANALYSIS

In the dynamic analysis the same meshed model is used.

Boundary Conditions: The rectangular tube is constrained at the bottom in all directions as in the earlier case.

Load condition: The rigid plate of mass 90 kg is given initial velocity of 15 m/s in vertically downward direction.

In this analysis, the strain hardening effect was included using Cowper-Symonds constitutive equation

$\varepsilon_p = D\,[\,\sigma_d/\sigma_s - 1]^q$ and $D = 6844 s^{-1}$ & $q = 3.91$

where, constants D & q are material parameters, σ_d is dynamic flow stress at a uniaxial plastic strain rate. σ_s is associated static flow stress as is taken by Nagel [5].

VII. RESULTS & DISCUSSIONS

The crush force – deflection curve is obtained after the analysis. The effective crushing distance was approximated by assuming axial deformation to be only 73 % of the axial length, 2H of a folding unit in accordance with Abramowicz & Jones [6]. This distance comes out to be 175 mm approximately.

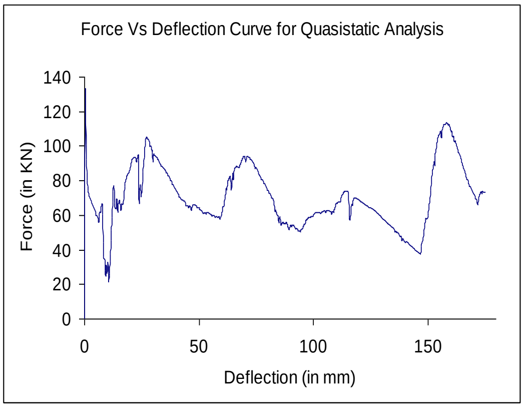


Fig. 6 Force –Deflection curve of quasistatic analysis

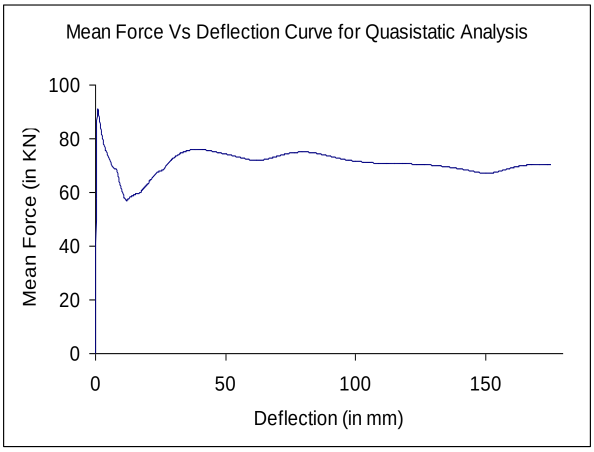


Fig. 7 Mean load-deflection curve for quasistatic analysis

The area under the curve gives the energy absorption value. The higher values of the crush forces in the above curve indicate that the forces required to form the fold are more.

From the data points of the above force –deflection curve, mean load deflection curve is obtained using Microsoft Excel software. The mean load-deflection curve is shown in Fig.7.

Fig. 8 shows the deformed shape of the tube. To ensure the quasistatic analysis the ratio of kinetic energy to internal energy was observed. This was very less indicating quasi-static analysis.

Fig. 9 shows profiles of internal energy & kinetic energy with reference to time. The ratio of artificial strain energy to internal energy was less than 5 % indicating no hourglass problem.

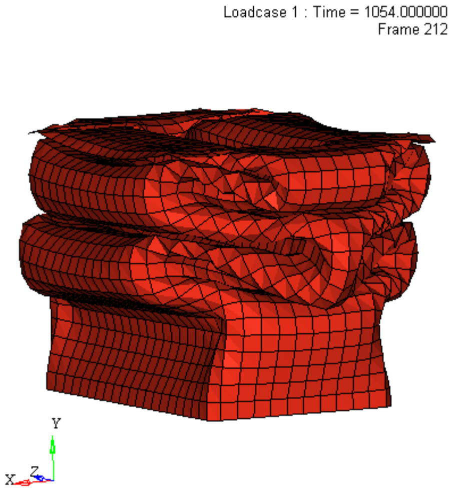


Fig. 8 Deformed shape during quasistatic analysis

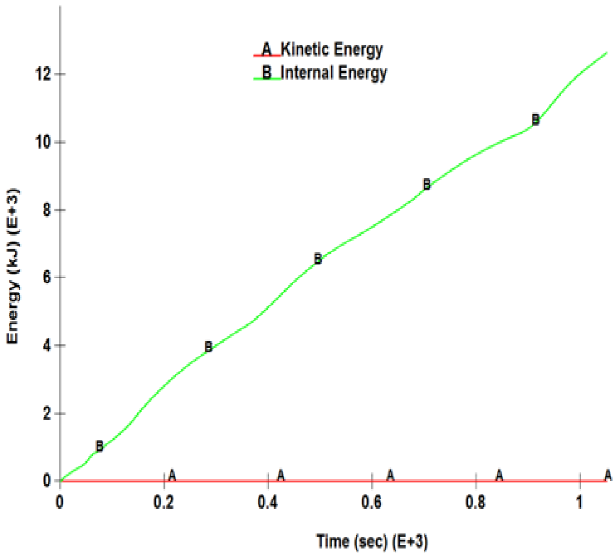


Fig. 9 Kinetic energy & Internal energy –time profiles for quasistatic analysis

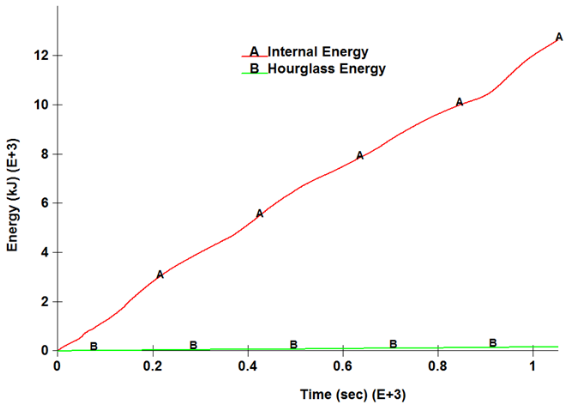


Fig. 10 Internal energy & hourglass energy – time profiles for quasistatic analysis

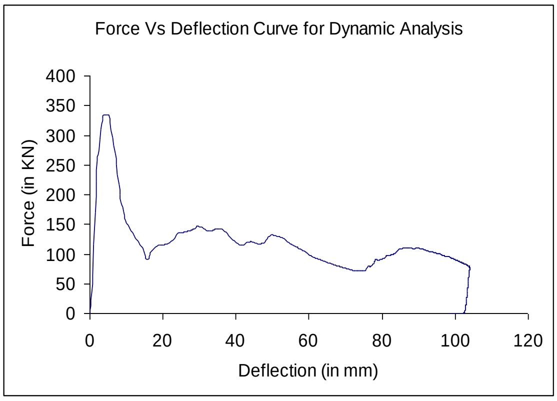


Fig. 11 Force-Deflection curve of dynamic analysis

Fig. 12 shows load-deflection curve of dynamic analysis. Mean force-deflection curve is also obtained for dynamic analysis. Figure 13 shows mean force-deflection curve for dynamic analysis.

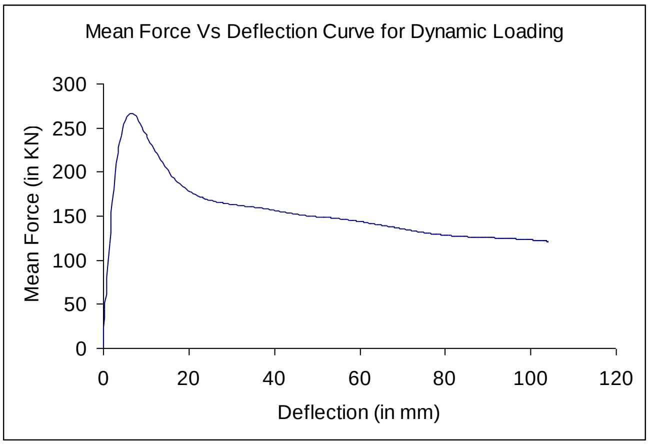


Fig. 12 Mean force- deflection curve for dynamic analysis

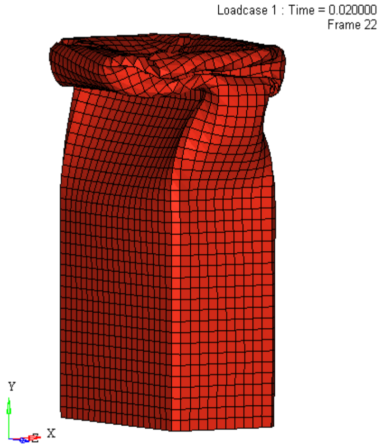


Fig. 13 Deformed shape of the tube during dynamic analysis

Figure 13 shows deformed shape of the tube for dynamic analysis. Figure 14 shows profiles of internal energy & kinetic energy with reference to time. Figure 15 shows internal energy & artificial strain energy profiles for dynamic analysis.

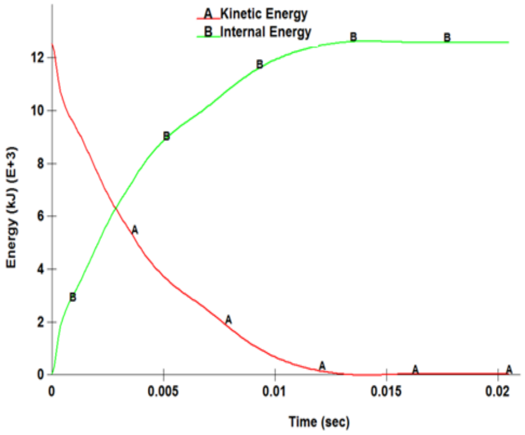


Fig. 14 Kinetic energy and Internal energy – time profiles for dynamic analysis

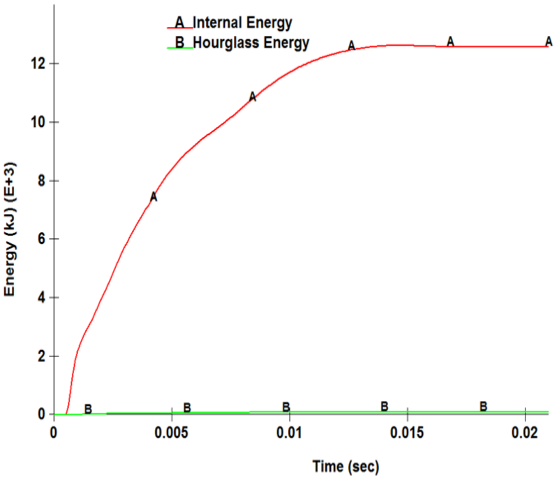


Fig. 15 Internal energy & hourglass energy – time profiles for quasistatic analysis

Finally the area under the load-deflection curve was measured with Microsoft excel software & mean load was calculated. The results are summarized in the Table 3 in the form of comparison.

Table 3 Comparison of Mean load values

Type of analysis	**Mean crushing load**		
	Analytical (for square tube) (KN)	**F. E. A. (for rectangular tube) (KN)**	**Diff. %**
Quasistatic	80.356	70.435	12.34
Dynamic	107.46	121.64	13.19

VIII. CONCLUSIONS

Quasistatic and Dynamic analysis of a rectangular tube of mild steel under axial loading was carried out. Following are the conclusions.

- The values of mean crushing load for rectangular tube obtained by finite element analysis are in reasonably good agreement with the values obtained by using analytical equations of square tube.
- The dynamic mean load is higher than quasistatic mean load. This is due to strain rate effect. But this may also be due to inertia effects & associated higher input kinetic energy.
- During dynamic loading, deformation is lesser as compared to quasistatic loading.
- Rectangular tubes are capable of giving nearly constant mean load –deflection response which is a desirable characteristic of an energy absorber.

Acknowledgements

We are thankful to the Principal and Management of Walchand Institute of Technology, Solapur for supporting us during this study. We are also thankful to the Principal and Management of M.E.S. College of Engineering, Pune for providing a platform to present our work.

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