Chapter 3
DEV Development AND VALIDATION OF FINITE ELEMENT MODEL OF ENERGY ABSORBER

3.1 Introduction

In this chapter, initially discussion about the material properties is done. Also, this chapter describes validation of finite element model used for the analysis. Hot rolled mild steel is the material of the tube used for this research work. Software Hyperworks is used for preprocessing and post-processing. For processing LS DYNA is used.

3.2 Material Properties

For the validation of the results of the analysis of bumper, experimental work was to be carried out. Hence material properties are tested using a universal testing machine.

3.2.1 Testing of Material

Figure 3.1 shows the details of the specimen used for tensile testing.

Figure 3.1 Tensile Test Specimen used for determining material properties
Here the specimen is cut transverse to the direction of rolling as per IS 1079: 2009. The dimensions of the test piece were decided as per IS1608: 2005. The set up used is as shown in the Figure 3.2. Engineering stress-strain curve is shown in Figure 3.3

**Figure 3.2 Set up used to determine material properties**

**Figure 3.3 Engineering Stress-Strain Curve**
This material is having yield strength of 293.809 MPa which is near to the yield strength of material used by Nagel (2005).

### 3.2.2 Conversion of Engineering Stress/Strain into True Stress/Strain

In linear analysis, Stress = D x Strain. This relation is sufficient to do the analysis, where D is constant matrix and depends on material properties like Young’s modulus, Poisson’s ratio. But in nonlinear analysis, D is no longer constant which needs to be updated and there we need the true strain vs true stress curves to update D matrix. And once material reaches in residual stress, again D matrix becomes constant. But, non-linear (elastoplastic or viscoelastoplastic) stage lies in between linear and residual where we update D matrix. So in order to update D matrix we need flow rule which again depends on true strain versus true stress curve.

From the engineering stress strain curve, values of different points are noted. Using these values and software facilities, values of true stress-strain are obtained.

These values are as shown in Table 3.1. The relation between True stress and True strain is shown in Figure 3.4.

<table>
<thead>
<tr>
<th>True strain</th>
<th>True stress (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>293</td>
</tr>
<tr>
<td>0.0198</td>
<td>386.586</td>
</tr>
<tr>
<td>0.058</td>
<td>461.36</td>
</tr>
<tr>
<td>0.0953</td>
<td>507.5</td>
</tr>
<tr>
<td>0.131</td>
<td>540.0</td>
</tr>
<tr>
<td>0.161</td>
<td>567.15</td>
</tr>
</tbody>
</table>

Table 3.1 Data points for True Stress-True Strain
3.3 Estimation of Mean Crushing Load for Axial Quasi-Static Loading

Estimation of mean crushing force for quasi-static axial loading of the tube is done before the finite element analysis. The quasi-static mean crushing force ($P_m$) for the rectangular tube is determined by utilizing the expression recommend by N. Jones and W. Abramowicz (1984).

The above proposed expression used for validating model of rectangular tube is strictly speaking applicable to only square tubes. But it is seen that this expression produces reasonably good results for rectangular tube with same perimeter as of square tube. [(S. R. Reid and T. Y. Reddy- (1986)]. The same expression is used by other researcher for rectangular tubes. [(Nagel 2005) (Mohammad Abedi 2012) (T. H. Kim 2001)]

$$P_m / M_o = 52.22 \times (c/h)^{1/3}$$  \hspace{1cm} (3.1)

Here

c = side length of tube = (60+110) / 2 = 85 mm

h = thickness of tube = 2.5 mm

and $M_o$ = fully plastic bending moment per unit length of sheet metal

$$M_o = \sigma_0 h^2 / 4$$  \hspace{1cm} Here $\sigma_0$ is flow stress (yield stress) of the material.

Thus, $\sigma_0 = (\sigma_{0\text{yield}}) = 293.8 \text{ MPa}$
Hence, \[ M_o = 293.8 \times (2.5)^2 / 4 = 459 \text{ MPa} \]
Thus, \[ P_m = M_o \times 52.22 \times (c / h)^{1/3} \]
\[ P_m = 459 \times 52.22 \times (85 / 2.5)^{1/3} \]
\[ P_m = 77.65 \text{ KN} \]

### 3.4 Finite Element Analysis of energy absorber

Finite element analysis is carried out using nonlinear explicit software LS DYNA.

#### 3.4.1 Finite Element Model and Meshing

Rectangular tube is modeled using Hyperworks software. Meshing is done using Belytschko Tsay shell element. This element is used by all researchers.

![Figure 3.5 Meshed model with load and boundary conditions](image)

At the corners of the rectangular tube 3 mm fillet is used in modeling. Element size used is 5 mm x5 mm as is used by G. M. Nagel (2005) for a similar geometry. Total number of elements was 3760 and number of nodes was 3873. On the top of the tube, a rigid plate is modeled.

#### 3.4.2 Load conditions and Boundary conditions

Bottom of the tube is constrained in all rotational and translational directions as shown in Figure 3.5. The rigid plate was moved with an initial velocity of 10 mm/minute in vertically downward direction. This velocity is also used by Nagel (2005). Boundary condition and load condition are as is used by F. Tarlochan (2013).

#### 3.4.3 Hourglass effect

To ensure the absence of hourglass effect, internal energy, hourglass energy time profiles are observed.
Figure 3.6 Hourglass energy and internal energy time profiles

Ratio of Hourglass energy (artificial strain energy) to internal energy was found to be less than 5%. This indicates absence of Hourglass effect.

3.4.4 Energy absorption and mean load

Figure 3.7 shows original shape and deformed shapes of tube. During the progressive folding of the tube, the energy absorption takes place.

Figure 3.7 Original and deformed shapes of tube
The force-displacement curve obtained by quasi-static analysis is shown in Figure 3.8. The area under the curve represents the energy absorbed. This area was measured using Microsoft-Excel software. The mean crushing force is determined using the energy absorption and corresponding displacement. Deflection of tube up to 2/3 rd of the total length of tube is considered here as is considered by Nagel (2005) and other researchers.

The mean crushing force for quasi-static analysis is 80.58 KN.

3.4.5 Comparison of mean load

Results of finite element analysis are compared with estimated results and variation error was calculated. This variation was 3.77 %. Thus, equations of mean crushing force of square tube gave good result for rectangular tube and model is validated.

3.5 Parameters for dynamic loading

Using validated model of quasi-static analysis, dynamic analysis was done as is done by A. Reyes (2002) using software for velocity of impact 10 m/s and 15 m/s. During the dynamic analysis, Cowper-Symonds constitutive equation was used in the Finite Element model to represent the effect of strain rate hardening. This equation is given by
\[ \dot{\epsilon}_p = D \left( \frac{\sigma_0'}{\sigma_0} - 1 \right)^q \text{ for } \sigma_0' \geq \sigma_0, \]

Where \( \sigma_0' \) is the dynamic flow stress at a uni-axial plastic strain rate \( \dot{\epsilon}_p \), \( \sigma_0 \) is the associated static flow stress. The constants \( D \) and \( q \) are material parameters. Above equation is an over-stress power law and was incorporated into the Finite Element model. The parameter values were \( D = 6844 \text{ s}^{-1} \) and \( q = 3.91 \), as used in earlier studies for the dynamic crushing of mild steel tubes [Reid and Reddy (1986), Nagel (2005), W. Abramowicz and N. Jones (1984) (1986)].

### 3.6 Analysis of bumper using experimental method

A bumper system was manufactured using the above tested material and is as shown in Figure 3.9.

![Figure 3.9 Bumper](image-url)

But during the analysis, cracks are developed in the tubes due to the hardening of the tube during welding. Due to this, sensors do not respond leading to stoppage of the load application in the universal testing machine.
Also as the thickness of the tubes is 2.6 mm, more force is required for folding of the tubes. The load application was at the centre and one large bar was kept on the bumper beam to transfer the load applied by machine to the tubes as is done by Nagel (2005). But bumper was deflected at the centre. So experiment was terminated here. Also there was lot of resistance from testing laboratories as force required for this testing was much more than the force commonly applied by them during daily work. Hence it was decided to manufacture one more bumper system with tubes of lesser thickness to validate the methodology.

So one thinner sheet of mild steel was purchased and material testing is done. Figure 3.11 shows engineering stress-strain relationship for this new material.
Figure 3.11 Engineering Stress-Strain curve of new material used for analysis of complete bumper

The new set up is as shown in Figure 3.12

Figure 3.12 Experimental set up for analysis of bumper with new material

Here also for the loading of the bumper a solid block of mild steel was used to avoid central deflection. But in the bumper system manufactured was having some defects
like some portion was concave instead of flat in one of the energy absorber. These led to some weak areas and folding was not proper. Also very small difference in the height of crash box led to loading of one tube first and second tube afterwards. Also the material was hardened due to welded joints on both sides. Cracks were generated in this tube also. As all things were not going in the correct manner, experiment was terminated after 20 mm. The load–displacement curve is shown in Figure 3.13. The deformed shape is shown in Figure 3.14

![Figure 3.13 Quasistatic analysis of bumper system](image1)

![Figure 3.14 Deformed shape of bumper during experiment](image2)

### 3.7 Summary

In this chapter, details of finite element modeling, experimental testing is discussed. The validation of the results is also done during this work. Problems of experimental testing are also discussed.