Abstract—The mine shaft loading stations are generally equipped with rails that lead into the shaft area to facilitate the loading of the hoisting mine cages with rail-stock. To prevent any inadvertent entry leading to accidents the rails are bared with a steel barrier. This paper presents the theoretical and finite element analysis approach for impact analysis for three different barrier sections.

Keywords—Elasto-plastic material, finite element analysis; impact energy, plastic bending.

I. INTRODUCTION

THE tragedy in May 1995 at Vaal Reefs shaft Number 2 saw a locomotive with personnel carriage crashing into the shaft gates. It fell down the shaft tearing the mine elevator cage with it, both plummeting to the bottom of the shaft [1]. The locomotive went past the barriers that were actually intended to stop much smaller and lighter machinery. Safety was called into question by the Commission of Enquiry, which found that no device was in place to adequately prevent the locomotive and carriage from entering the shaft. Numerous recommendations followed, which cover safety devices in the shaft station area, and require manufacturers and suppliers to ensure that equipments do not pose any safety or health risk [2]. Guidelines for shaft station stopping devices followed, calling for stopping devices to be:

(a) analytically verified, as well as
(b) field tested for integrity [3].

Various types of stopping devices are in use in the different mining industries. The mine shaft loading stations are generally equipped with rails that lead into the shaft area to facilitate the loading of the hoisting mine cages with rail-stock. To prevent any inadvertent entry of a locomotive or any other rail stock, the rails are bared with a steel barrier (Fig. 1) [4, 5].

This paper presents the FEM and analytical approach of impact energy calculations for different barrier sections. It is expected that a run-away train of mass 10 T with a speed of 20 km/h can be arrested using the 300W structural steel section [3]. This analysis concerns itself with the energy absorption ability of the “CTO Stop Block” type, consisting of a 203 mm x 152 mm x 52 mm taper flange I-section beam rated for 154 kJ (Fig. 2a). Fig. 2b presents the I section stiffened by two 150 mm x 30 mm rectangular beams to yield the composite CTO Stop Block.

Fig. 1 Typical shaft entrance with Stop Block barriers chained in place between rails

Fig. 2 (a) I section and (b) stiffened composite CTO Stop Block

The schematic arrangement of the steel stop block is shown in Fig. 3. It is assumed to be 500 mm deep inside the concrete foundation and 500 mm high above the ground. The height of impact is assumed to be 450 mm above the ground. For absorbing a part of the energy there is an additional rubber block at the front end of the stop block around the point of impact (Figs. 1, 6).
II. MATERIAL MODELING

As shown in Fig. 4 the stress-strain curve obtained for a typical 300W steel specimen after testing on a UTM is idealized (in the form of distinct elastic and plastic regions and the curved portion linearized to an ideal plastic material for simplified calculations). The area under the stress-strain curve represents the total energy stored.

![Fig. 4 Modified stress-strain curve of 300W Steel](image)

III. THEORETICAL ANALYSIS

Establishing the equivalent static load that will cause yielding of the I beam for the 203x152x52 kg/m 300W Steel section gives:

\[
\sigma_y = 300 \text{ MPa} \\
E = 200 \text{ GPa} \\
G = 82 \text{ GPa} \\
I_c = 47.8 \times 10^{-6} \text{ m}^4 \\
Z_{pl} = 539 \times 10^{-6} \text{ m}^3 \\
A = 6.64 \times 10^{-3} \text{ m}^2
\]

From theoretical calculations it was established that the plastic bending energy is significantly larger than the elastic bending energy. However, since a simple I section was unable to arrest the kinetic energy of the locomotive (154 kJ), the final design required the I beam to be stiffened with additional 2 off rectangular steel plates (150 mm x 30 mm) welded for additional rigidity, to yield a composite section (Fig. 2b) along with a rubber buffer. Other sections (254 mm x 152 mm x 59 mm and 305 mm x 152 mm x 66 mm) were also similarly stiffened with rectangular plates of appropriate section. The free body diagram is generated as shown in Fig. 5.

![Fig. 5 Free body diagram of stop block](image)

A comparative summary of the stresses, plastic bending and energies calculated for the given bend angles for the three composite steel I sections analysed in this paper is presented in Table 1.

<table>
<thead>
<tr>
<th>I-section</th>
<th>203x152x52 mm</th>
<th>254x152x59 mm</th>
<th>305x152x66 mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bending Stress</td>
<td>2.54E+08</td>
<td>2.40E+08</td>
<td>2.24E+08</td>
</tr>
<tr>
<td>Shear Stress</td>
<td>92E+06</td>
<td>104E+06</td>
<td>115E+06</td>
</tr>
<tr>
<td>Principal Stress 1</td>
<td>283E+06</td>
<td>278E+06</td>
<td>272E+06</td>
</tr>
<tr>
<td>Principal Stress 2</td>
<td>-30E+06</td>
<td>-38E+06</td>
<td>-48E+06</td>
</tr>
<tr>
<td>Shape factor (I, rect)</td>
<td>1.2, 5.37</td>
<td>2.2, 4.45</td>
<td>3.2, 3.45</td>
</tr>
<tr>
<td>Total Energy (yield)</td>
<td>9.71E+02</td>
<td>1.27E+03</td>
<td>1.82E+03</td>
</tr>
<tr>
<td>Plastic bending Permanent set (mm)</td>
<td>20</td>
<td>21</td>
<td>22</td>
</tr>
<tr>
<td>Bend angle</td>
<td>0.05747</td>
<td>0.06034</td>
<td>0.06321</td>
</tr>
<tr>
<td>Total Plastic Moment</td>
<td>2.63E+05</td>
<td>3.91E+05</td>
<td>5.56E+05</td>
</tr>
<tr>
<td>Total Plastic Energy</td>
<td>1.51E+04</td>
<td>2.36E+04</td>
<td>3.52E+04</td>
</tr>
<tr>
<td>Total Energy</td>
<td>1.61E+04</td>
<td>2.49E+04</td>
<td>3.70E+04</td>
</tr>
</tbody>
</table>

The plastic energy increases with increasing deformation (or bend angle) from 0-35° as depicted in Fig. 6. It was estimated that the stop can arrest the carrier only up to a bend angle of nearly 35°, beyond which it fails.

IV. FINITE ELEMENT ANALYSIS

The same parametric analysis was carried out using ABAQUS Version 6.11, a commercial finite element analysis
(FEA) package to validate the analytical approach. The following assumptions were made in the FEA:

- Fillets on the I sections are neglected and flange thicknesses of I sections is assumed constant.
- Perfect I section to rectangular plates welds (Assumed - Tied or Glued contact).
- The stop’s hinge point is infinitely strong.
- The weldments are sufficiently strong.
- Rubber buffer was not considered.
- Load Case 1 – Gradually applied Static load of 161 kN.
- Load Case 2 - Suddenly applied load = 2 X Gradually applied load.
- Load Case 3 – Impact load (run-away train of mass 10 T travelling at 20 km/h = 154 kJ of kinetic energy).

The following material properties for 300W steel were input in Abaqus -

- \( E=200 \text{ GPa}; \) Poisson’s ratio=0.3;
- maximum tensile strength=300 MPa, density =7850 kg/m³

![Fig. 6](image) Variation of theoretical plastic energy absorbed with different bend angles for a composite 203 x 152 x 52 I section

Fig. 6 shows the full and Fig. 7b shows the half composite I section modeled with the applied load and boundary conditions (bc) in Abaqus. As an initial trial the material was assumed to be elastic and static analysis was carried out. The contact condition at the I section-rectangular plate interface was assumed to be Tied. The full meshed model is shown in Fig. 8a. However, half model was also analyzed to take the advantage of model symmetry and refined meshing. After mesh convergence study a total of 14672 eight noded linear hexahedral elements of size 9 mm and a total number of 17841 nodes were used for model meshing (Fig. 8b).

![Fig. 7](image) Full and (b) half model of composite I section with modified loads and bc

![Fig. 8](image) (a) Full and (b) half meshed model

V. RESULTS AND DISCUSSION

The Von-Mises (\( \sigma_{VM} \)) stresses found out are plotted as shown in Fig. 9. For the Load case 1 of gradually applied load \( \sigma_{VM} = 453.8 \text{ MPa} \) and for Load case 2 of suddenly applied load \( \sigma_{VM} = 909 \text{ MPa} \) was evaluated (for a coarser mesh - Fig. 9). Plastic yielding occurs but not over the full section and hence the stopper is able to stop the rail trolley.

A plot of the resultant displacements is shown in Fig. 10. For Case - 1 the displacement = 54.7 mm and the bend angle is calculated as = 6.24 degree and for Case - 2, the bend angle is = 12.46 degree. However, the deformation is only limited to the top portion of the stop where a concentrated load is applied.

![Fig. 9](image) Variation of theoretical plastic energy absorbed with different bend angles for a composite 203 x 152 x 52 I section

![Fig. 10](image) V. RESULTS AND DISCUSSION
Further, impact (dynamic explicit) FE analysis was also carried out on half model with 15764 nodes and 11995 eight noded hexagonal elements of size 7.5 mm. The elasto-plastic material model with isotropic hardening was used to simulate elastic-plastic impact analysis. The plastic strain was modeled as $\varepsilon = 0.0$ at the onset of yield (300 MPa) and as $\varepsilon = 1.0$ at twice the yield value. Adiabatic heating option was also invoked. The different energy plots for whole model i.e. the kinetic energy, strain energy and the total energy (Fig. 11) are checked. The trends are as expected with the kinetic and strain energy stabilizing to near zero values at the end of a time period of nearly 0.1 s.

The total energy of the stop model is found to be nearly 345 kJ which is quite larger than the 171 kJ predicted by the theoretical approach. This can be likely accounted to the various assumptions made in the theoretical analysis. However, this limiting condition will be reached early for multiple impacts on already bend stop blocks (caution).

VI. CONCLUSIONS

In this paper the impact analysis of a mine rail stop device was attempted both by the theoretical and FEA approach. The limiting bend angle obtained by the theoretical approach = 31 degrees while by the FEA approach it is = 45.2 degrees. A comparative parametric study of different I section stops was also conducted.

FEA is able to successfully simulate a real life impact problem. Such simulation studies are likely to help in averting tragic mining accidents as that of Vaal Reefs in 1995. Mining industry could have wider applications of FEA towards deploying safer equipments. Some of the ongoing work includes -

- On site testing and feedback based modification
- Simulation of different trolley speeds (20, 25 and 30 km/h) and weights (10, 12 and 15 T).
- Simulation of onset of limiting bend condition for different stop blocks of already bend configurations due to being subjected to multiple impacts.
- Evaluate alternate stop block designs for improved safety.

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REFERENCES