Failure Analysis & Redesign of Boom under Static Analysis of Self-Propelled Surface Drilling Machine

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ABSTRACT
Machines used in surface mining, such as derricks, roof bolting machines, loaders, transportation vehicles as well as others are generally used for core exploitation, loading and transportation, i.e. basic mining tasks. Construction design practice, exploitation and tests prove that such machines as well as their sub-components are subject to requirements radically different from machines operating on the surface. In general operating requirements of machines used in surface mining are very severe or heavier. They are often subjected to percussive loads radically different from machines operating on the surface. The boom of these machines supports the rotation head and feed mechanism for drilling hole in earth surface. It has been observed that the boom fails due to fracture at extreme level load condition during actual operation of machines. As a consequence several components of other assemblies are subjected to overloading and, therefore, undergo plastic deformation.

The principal objective behind boom redesigning is that it should sustain the weight of drill assembly which is approximately 3 ton. In this thesis is presented the failure analysis of the boom assembly and computer simulation of failure considering collapse conditions. Measurements of hardness and micro hardness in the vicinity of the weld have been performed. After FEM analyses and material tests, it is seen that the root cause of the collapse lies in the existing design as also minor welding faults during fabrication.

Finite element method analysis of the boom showed stress concentrations exceeding the allowable level and the need to redesign it considering load fluctuations caused by operating environment. The analysis shows that the changing loads adversely affect the load-carrying structure causing its degradation, i.e. the appearance of cracks at stress concentration sites in welded joint areas. The results show that maximum stress is induced at the fracture locations observed during failure examination. Redesigning the boom with increased cross section and improved material property overcomes the boom failure problem observed in the present machines.

Keywords – Boom Analysis, Boom Design, Boom Structure, Open Pit Mining Rig, Surface mining boom.

I. INTRODUCTION
The Difficult geological conditions and more intense mining processes taking place today in many building sites lead to a high mechanization level of building works. Because of this, specialized self-propelled drilling machines are constructed, which enable a sufficient progress in the mining works. Among the machines most frequently used in the mining and building sites, there are those directly used in the preliminary works. These are the vehicles used for slabbing, drill rigs and bolt setters. The common feature of those machines is the fact that the working tools are placed on a boom. The boom mounted on self-propelled mining machines should have a sufficient number of degrees of freedom to minimize the time related to changing the location of the machines. Most frequently, this is a straight-line structure ended (in the case of drill rigs and bolt setters) with a rotating head (turnover fixture). It is assembled with feed assembly which is actuated through two hydraulic cylinders as shown below.

Figure 1.1 Location of boom in surface drilling machine
Manufacturer of mining equipment and machineries, has observed one particular problem with the boom. It was noted that the boom fails during operation due to fracture and as a consequence several components of other adjoining assemblies undergo plastic deformation. In view of the increased load of drill guide from 2.1 to 3 ton, and also to reduce the overall costs of manufacturing, the company has desired that entire machine be made out of standardized components and machine parts and units for interchangeability reasons. The company desires to design a universal boom which could be mounted on various types of machines. The redesigning of boom, therefore, required identifying the worst loading conditions using simple force analysis. The two cases are:

(1) Maximum tilt angle of drill guide is 22deg when boom is horizontal and
(2) When drill guide is horizontal.

The third typical situation is when force is exerted on the boom when lift cylinder is actuated. The company desires to design a universal boom which could be mounted on various types of machines. The redesigning of boom, therefore, required identifying the worst loading conditions using simple force analysis.

II. PROBLEM IDENTIFICATION

2.1 Analysis of Existing Model

Existing boom assembly is shown in Figure 2.1. The present boom section is 200 mm x 200 mm built out of 12 mm thickness plates by welding is shown in Figure 3.3. The design of boom required identifying the worst loading conditions using simple force analysis. The two cases are (1) maximum tilt angle of drill guide is 22deg when boom is horizontal and (2) when drill guide is horizontal. The third typical situation is when force is exerted on the boom when lift cylinder is actuated.

![Figure 2.1 existing boom assembly](image1)

![Figure 2.2 Line diagram of boom](image2)

![Figure 2.3 Cross section A-A of existing Boom](image3)

A1-Connected to Drill guide
E-Connected to hydraulic cylinder
G-Connected to hydraulic cylinder
F-Connected to collar
Input material sizes & weight:- Boom tube size – 200 x 200 x 12 th
Weight of feed beam assembly - 3105 Kg = 30460.05 N
Weight of boom assembly - 702 Kg = 6886.2 N

Calculating section modulus (Z) for boom [figure 2.3]

**Moment of Inertia**

\[
I = \frac{a^4 - b^4}{12}
\]

\[
= 533739.52 \text{ mm}^4
\]

**Z for section of boom**

\[
Z = \frac{a^4 - b^4}{6a}
\]

\[
= 533739.52 \text{ mm}^3
\]
Case 1 - Boom is in Horizontal condition & drill guide at 22deg to vertical [Fig. 1.2]
EB (B is CG point of boom assembly) = 1771 mm,
EA (A is the CG point of feed assembly) = 3647 mm,
EF= 1370 mm
BF = 401 mm
Calculating reaction at R1 & R2
R2 = 90597.0177 N,
R1 = -52779.4677 N
Vertical shear forces at different point.
Fa = -30460.05 N, Fb = -37817.55 N, Ff = 52777.8 N; Fe = 0
Between B & F points shear force changes its sign, let at J point where maximum bending moment will occurs.
By calculating,
We get distance BJ = 167.38 mm. Distance AJ = 2043.38 mm.
Calculating maximum moment at J
\[ M_x = \frac{-W_f \times AJ - W_b \times BJ}{Z} \]
= -9312364.437Nmm
Z for combine section of boom at X = 533739.52 mm^3
Hence bending stress induced in the boom = \( \frac{M_x}{Z} \) = 174.47 N/mm^2
Calculating the direct tensile stress due to forces in Z direction, considering area where these forces are applied
\[ F_v (vertical force component) = 130138.5 \]
\[ W_b (weight of boom assembly) = 6886.2N \]
\[ Area = L \times W = 1787 \times 12 = 21444 \text{ mm}^2 \]
\[ \sigma_t = \frac{F_v - W_b}{Area} \]
\[ \sigma_t = \frac{130138.5 - 6886.2}{1787 \times 12} = 5.74 \text{ Mpa} \]
The resultant equivalent stress is given by,
\[ \sigma_{max} = \sigma_b + \sigma_t = 174.47 + 5.74 = 180.21 \text{ Mpa} \]
Hence maximum induced bending stress at case 1 is 180.21N/mm^2.

Case 2 - Boom is in Horizontal condition & drill guide in horizontal
EB (B is CG pt of boom assembly) = 1592 mm
EAz (A is the CG point of feed assembly) = 3180 mm
EF = 1370 mm, BF = 222 mm, EAx (Along horizontal) = 1254 mm
Calculating reaction at R1 & R2
R2 = 36424.53 N, R1 = 1386.8397 N
Vertical shear forces at different point.
Fa = -30460.05 N, Fb = -37817.55 N, Ff = -1386.8397N, Fe = 0
Between B & F points shear force changes its sign, let at J point where maximum bending moment will occurs.
By calculating,
We get distance BJ = 230.45 mm. Distance AJ = 1818.45 mm.
Calculating maximum moment at J
\[ M_x = \frac{-W_f \times EAx - W_b \times BJ}{Z} \]
= -39568717.404 Nmm
Inertia of boom at section J = 53373952 mm^4
Z for combine section of boom at X = 533739.52 mm^3
Hence bending stress induced in the boom = \( \frac{M_x}{Z} \) = 74.13 N/mm^2
- Calculating the direct tensile stress due to forces in Z direction, considering area where these forces are applied
\[ F_v (vertical force component) = 130138.5 \]
\[ W_b (weight of boom assembly) = 6886.2N \]
\[ Area = L \times W = 1787 \times 12 = 21444 \text{ mm}^2 \]
\[ \sigma_t = \frac{F_v - W_b}{Area} = \frac{130138.5 - 6886.2}{1787 \times 12} = 5.74 \text{ Mpa} \]
The resultant equivalent stress is given by,
\[ \sigma_{max} = \sigma_b + \sigma_t = 74.13 + 5.74 = 79.87 \text{ Mpa} \]
Hence maximum induced bending stress at case 2 is 79.87 N/mm^2

Calculating stress coming on the boom when lift cylinder is actuated
Piston diameter of the cylinder = 13 cm = 0.13 m
Pressure on the hydraulic line = 200Bar = 20x10^6 N/m^2
Force acting when the cylinder is actuated = \( P \times A \) = 260277.4485 N
From graphical analysis for boom angle 55°, \( \Phi = 30° \)

Figure 3.4 Ray diagram of the boom and lift hydraulic cylinder
Perpendicular distance from hinge point to line of action of force (a) = 379 mm
Moment of cylinder force acting on boom = \( F \times a = 98645152.9815 \text{ Nmm} \)
Stress on the boom tube because of cylinder force \((S= Mx*y/I)=162.6498 \text{N/mm}^2\)

- Calculating the direct tensile stress due to forces in Z direction, considering area where these forces are applied

\[
F_v (\text{vertical force component})=130138.5 \text{W}\\
W_b (\text{weight of boom assembly})=6886.2 \text{N}\\
\text{Area} = L*W=1787*12 = 21444 \text{ mm}^2\\
\sigma_t = \frac{F_v - W_b}{\text{length*width}} = \frac{130138.5 - 6886.2}{1787*12} = 5.74 \text{ Mpa}\\
\]

The resultant equivalent stress is given by,

\[
\sigma_{max} = \sigma + \sigma_t = 162.6498 + 5.74 = 168.38 \text{ Mpa}\\
\]

Hence maximum induced bending stress at case 2 is 168.38 \text{ N/mm}^2.

Hence from all above calculations we get,

**Case 1: Boom is in horizontal condition & drill guide at 22deg to vertical**

Stress induced in boom = 180.21 \text{ N/mm}^2

**Case 2: Boom is in horizontal condition & drill guide in horizontal**

Stress induced in boom = 79.87 \text{ N/mm}^2

**Stress coming on the boom when lift cylinder is actuated**

Stress induced in boom = 168.38 \text{ N/mm}^2

Hence combined stress induced in boom = 348.59 \text{ N/mm}^2

Table 3.1 Stress induced in boom analytically of existing design

<table>
<thead>
<tr>
<th>Loading condition</th>
<th>Stresses induced in boom analytically ((\text{N/mm}^2))</th>
</tr>
</thead>
<tbody>
<tr>
<td>Case 1: Boom is in horizontal condition &amp; drill guide at 22deg to vertical</td>
<td>180.21</td>
</tr>
<tr>
<td>Case 2: Boom is in horizontal condition &amp; drill guide in horizontal</td>
<td>79.87</td>
</tr>
<tr>
<td>Maximum stress coming on the boom when lift cylinder is actuated</td>
<td>168.38</td>
</tr>
<tr>
<td>Combined stress induced in boom</td>
<td>348.59</td>
</tr>
</tbody>
</table>

Considering that yield stress for rolled steel section \(\text{FE410W} = 245 \text{ Mpa}\) and the allowable stress as per IS:800 - 1984, \(\text{Steel Handbook} = 24.5 \text{ Kg/mm}^2=245 \text{ N/mm}^2\), it is found that the total stress induced in the boom is more than the permissible yield stress of the material under static load conditions and without considering any factor of safety. Hence, it is recommended to use higher strength material or modify boom cross section for safe functioning.

### III. THEOROTICAL APPROACH

The possible approaches that have been considered to overcome the overloading issue include redesigning of the boom by changing the material or the section modulus or both. It is recommended to use higher strength material \(\text{FE510WC next grade of existing material FE410WA}\). Section modulus can be improved by changing cross section of the boom by welding additional plates of thickness 12 mm on the outer periphery of the existing design. This would increase moment of inertia and enable sustaining increased load for safe functioning.

The original boom tube size – 200 mm x 200 mm x 12 mm thickness has been retained. New plates have been welded upon as shown in Figure 4.1. The weight of feed beam and boom assembly now, as estimated, is 3105 Kg (30460.05 N) and 750 Kg (6916.05 N) respectively. Section modulus (Z) for boom is evaluated next.

The original boom tube size – 200 mm x 200 mm x 12 mm thickness has been retained. New plates have been welded upon as shown in Figure 4.1. The weight of feed beam and boom assembly now, as estimated, is 3105 Kg (30460.05 N) and 750 Kg (6916.05 N) respectively. Section modulus (Z) for boom is evaluated next.

**Boom Section**

SQ. tube outside, \(a_1 = 200 \text{ mm}\)

Thickness = 12 mm

SQ. tube inner side, \(b_1 = 176 \text{ mm}\)

Moment of inertia \(I_1 = (a_1+ b_1)/12 = 53373952 \text{ mm}^4\)

Plate \(150 \times 12, b_2 = 12 \text{ mm}, h_2 = 150 \text{ mm}\)

Hence, moment of inertia \(I_2 = 45000 \text{ mm}^4\) plate \(150 \times 16\)

\(b_3 = 16 \text{ mm}, h_3 = 150 \text{ mm}\)

Hence, moment of inertia \(I_3 = 60000 \text{ mm}^4\)

Total moment of inertia \(I = 53483952 \text{ mm}^4\)

\(Z\) for section of boom = \(608908.5455 \text{ mm}^3\)

**Case 1-Boom is in horizontal condition & drill guide at 22deg to vertical**

EB (B is CG pt of boom assembly) = 1771 mm,

EA (A is the CG point of feed assembly) = 3647 mm,

EF = 1370 mm, BF = 401 mm

Calculating reaction at \(R_1\) & \(R_2\)

\(R_2 = 90597.0177 \text{ N} \quad R_1 = -52779.4677 \text{ N} \quad \)

Vertical shear forces at different point.

\(F_a = -3090.15 \text{ N} \quad F_b = -37817.55 \text{ N} \quad F_i = 52777.8 \text{ N} \quad F_e = 0\)
Between B & F points shear force changes its sign, let at J point where maximum bending moment will occurs.

By calculating,

We get, distance BJ = 167.38 mm. distance AJ = 2043.38 mm.

Calculating maximum moment at J

\[ M_x = -W_f \times AJ - W_b \times BJ = -6470260 \text{ Kgmm} \]

Total Moment of inertia, \( I = 53483952 \text{ mm}^3 \)

Hence, bending stress induced in the boom = \( M / Z \) for section of boom, \( Z = 608908.5455 \text{ mm} \)

Total Moment of inertia, \( I = 53483952 \text{ mm}^3 \)

Calculating maximum moment at J

\[ M_x = -W_f \times AJ - W_b \times BJ = -6470260 \text{ Kgmm} \]

Hence bending stress induced in the boom = \( M / Z \) = 104.231N/mm²

- Calculating the direct tensile stress due to forces in Z direction, considering area where these forces are applied

\[ F_v (vertical \ force \ component) = 130138.5 \ N \]

\[ W_b (weight \ of \ boom \ assembly) = 6886.2N \]

\[ Area = L \times W = 1787 \times 12 = 21444 \text{ mm}^2 \]

\[ \sigma_t = \frac{F_v - W_b}{Area} = \frac{130138.5 - 6886.2}{21444} = 5.74 \text{ Mpa} \]

\[ \sigma_t = \frac{F_v - W_b}{Area} = \frac{130138.5 - 6886.2}{21444} = 5.74 \text{ Mpa} \]

- The resultant equivalent stress is given by,

\[ \sigma_{max} = \sigma_b + \sigma_t = 104.231 + 5.74 = 109.971 \text{ Mpa} \]

Hence maximum induced bending stress at case 1 is 109.971N/mm²

**Case 2 - Boom is in Horizontal condition & drill guide in horizontal**

EB (B is CG pt of boom assembly) = 1592 mm

EAz (A is the CG point of feed assembly) = 3180 mm

EF = 1370 mm, BF = 222 mm,

EA (Along horizontal) = 1254 mm

Calculating reaction at R₃ & R₂

\[ R_2 = 36424.53N, R_1 = 1386.8397N \]

Vertical Shear forces at different point.

\[ F_v = -30460.05N, F_b = -37817.55N, F_f = 1386.8397N, F_a = 0 \]

Between B & F points shear force changes its sign, let at J point where maximum bending moment will occurs.

By calculating,

We get, distance BJ = 230.45 mm. distance AJ = 1818.45 mm.

Calculating maximum moment at J

\[ M_x = -W_f \times EA_x - W_b \times BJ = -39892447.404 N \text{ mm} \]

Total moment of inertia, \( I = 53483952 \text{ mm}^3 \)

Z for section of boom, \( Z = 608908.5455 \text{ mm} \)

Hence bending stress induced in the boom = \( M / Z = 65.51N/mm² \)

- Calculating the direct tensile stress due to forces in Z direction, considering area where these forces are applied

\[ F_v (vertical \ force \ component) = 130138.5 \]

\[ W_b (weight \ of \ boom \ assembly) = 6886.2N \]

\[ Area = L \times W = 1787 \times 12 = 21444 \text{ mm}^2 \]

\[ \sigma_t = \frac{F_v - W_b}{Area} = \frac{130138.5 - 6886.2}{21444} = 5.74 \text{ Mpa} \]

- The resultant equivalent stress is given by,

\[ \sigma_{max} = \sigma_b + \sigma_t = 104.231 + 5.74 = 109.971 \text{ Mpa} \]

Hence maximum induced bending stress at case 2 is 109.971N/mm²

**Calculating stress coming on the boom when lift cylinder is actuated**

Piston diameter of the cylinder = 13 cm = 130 mm

Pressure on the hydraulic line = 200 Bar = 20x10⁶ N/m²

Force acting when the cylinder is actuated = \( P \times A = 260277.4485 \text{ N} \)

From graphical analysis for boom angle 55°, \( \Phi = 30° \)

Perpendicular distance from hinge point to line of action of force (a) = 379 mm

Moment of cylinder force acting on boom = \( F \times a = 98645152.9815 \text{ Nmm} \)

Stress on the boom tube because of cylinder force (S = \( M_x^2/y/I = 162.64 \text{ Nmm}^2 \))

- Calculating the direct tensile stress due to forces in Z direction, considering area where these forces are applied

\[ F_v (vertical \ force \ component) = 130138.5 \]

\[ W_b (weight \ of \ boom \ assembly) = 6886.2N \]

\[ Area = L \times W = 1787 \times 12 = 21444 \text{ mm}^2 \]

\[ \sigma_t = \frac{F_v - W_b}{Area} = \frac{130138.5 - 6886.2}{21444} = 5.74 \text{ Mpa} \]

The resultant equivalent stress is given by,

\[ \sigma_{max} = \sigma + \sigma_t = 162.6498 + 5.74 = 168.38 \text{ Mpa} \]

Hence maximum induced bending stress at case 3 is 168.38N/mm²

**Case 1: Boom is in horizontal condition & drill guide at 22deg to vertical**

Stress induced in boom = 109.971N/mm²

**Case 2: Boom is in horizontal condition & drill guide in horizontal**

Stress induced in boom = 71.25 N/mm²

**Stress coming on the boom when lift cylinder is actuated**

Stress induced in boom = 168.38 N/mm²

Hence combined stress induced = 278.351N/mm²

Now, yield stress for rolled steel section FE510W = 377 Mpa

Allowable stress \( IS:800 = 37.7 \text{ Kg/mm}^2 = 377 \text{ N/mm}^2 \)

The total stress induced in the boom is less than the permissible yield stress of the material only for static
load conditions. Hence the design is safe with factor of safety of 1.35.

Table 34.1 Stress induced in boom analytically in proposed design

<table>
<thead>
<tr>
<th>Loading condition</th>
<th>Stresses induced in boom analytically(N/mm²)</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Case 1:</strong> Boom is in horizontal condition &amp; drill guide at 22deg to vertical</td>
<td>109.971</td>
</tr>
<tr>
<td><strong>Case 2:</strong> Boom is in horizontal condition &amp; drill guide in horizontal</td>
<td>71.25</td>
</tr>
<tr>
<td>Maximum stress coming on the boom when lift cylinder is actuated</td>
<td>168.38</td>
</tr>
<tr>
<td>Combined stress induced in boom</td>
<td>278.351</td>
</tr>
</tbody>
</table>

IV. SOFTWARE ANALYSIS

4.1 CAD Model Preparation

Boom assembly contains following parts
- Bush Pin
- Bush Plate
- Supporting Plate
- Square tube Plate

Following simple procedure is adopted for modelling assembly
- Preparation of individual part model and file is saved as .prt extension
- Open assembly section of Pro-Engineer 4.0
- Import part models into assembly and giving appropriate assembly constraints
- Save assembly with step extension: surfaces and volumes are saved by this type of extension.

Some of assembly parts are as shown below.

- (a) Square tube plate
- (b) Supporting plate 1
- (c) Bush plate
- (d) Pin support
- (e) Supporting plate 2
- (f) Section plate
- (g) Bush support
- (h) Bush eye

The whole boom assembly model is prepared as shown and is exported to further analysis.

4.2 Mesh Generation with Hexahedral

In this analysis mesh generation is auto mesh generation with element size is 10. This element size is used for all the body of boom. Hex-dominant method is used for all the parts of boom. Results obtained are shown in table 4.1

Meshed model is as follows.

Some of meshed model parts are as shown below.

Table 4.1 Mesh Result using ANSYS
4.3 Assign material properties

The engineering data should be specified. In this step the engineering materials and their parameters like density, Poisson’s ratio and other important properties are added. All the materials that are associated with geometry should be specified as this act as library for material selection in analysis. Material details for Boom assembly is FE510W (S355JR).

<table>
<thead>
<tr>
<th>Material</th>
<th>Young’s modulus (GPa)</th>
<th>Poisson’s ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>FE510W</td>
<td>190-210</td>
<td>0.27-0.3</td>
</tr>
</tbody>
</table>

4.4 FEA Results

Pre-processing of model was carried out and processing was done in ANSYS. After converging solution results were analyzed.

From the figure, it can be observed that maximum stress zone occurs at the pin. Therefore, stress value is important in predicting failure. Maximum principle stress obtained from FE analysis is 130.52 Mpa for Case 1 and 157.17 Mpa for Case 3 as shown above. This stress value obtained which is less than allowable stress.

<table>
<thead>
<tr>
<th>Varying parameter</th>
<th>Stresses in boom (N/mm²)</th>
<th>Stresses in boom (N/mm²) by ANSYS</th>
</tr>
</thead>
<tbody>
<tr>
<td>Case 1:</td>
<td>109.971</td>
<td>130.52</td>
</tr>
<tr>
<td>Case 2:</td>
<td>71.25</td>
<td>-</td>
</tr>
<tr>
<td>Stress when lift cylinder is actuated</td>
<td>162.64</td>
<td>157.17</td>
</tr>
<tr>
<td>Combine stress induced in the boom</td>
<td>278.351</td>
<td>287.69</td>
</tr>
</tbody>
</table>

V. EXPERIMENTAL ANALYSIS

FE analysis on base model was carried out as explained earlier. To check resemblance of simulation result strain gauge test was performed on base model. Test for Redesign of Boom for vertical and Horizontal Condition as follows.

The boom was design under static loading condition and it was tested under actual loading condition. This report will describe the performed measurements and the measurement technique used during the stress and strain measurements of the boom on the Rock drills models. The aim of the measurements was to provide the stress levels and dynamic behavior of the boom system for Drill rig in order to provide data for a FEM model check.
The measurements were performed at the test area at plant. The measurements were made with four different types of measurement probes:

- Strain gauges for direct stress and strain measurements in the boom
- Dynamic pressure transducers in all major hydraulic cylinders in order to calculate the forces action on the boom system
- Wire transducers for measurement of the position of the different moving parts during the operation.
- Accelerometers for measurement of vibrations action of the different parts and calculation of inertia forces.

In the tests, performed in the test area of plant, the rig was operated in well-defined ways, called time sequences, in order to be representative to all expected situations for a rig on a customer site. This means the rig was operated in drilling, positioning and several tramming maneuvers, i.e. on varying ground conditions and with the boom and feed in different positions.

VI. RESULT

From the above testing, they found that results were satisfactory and ready for actual working. The design is useful for cost reduction point of view for any parts of Drilling rig and respective technical specification of respective parts.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Analytical Results</th>
<th>FEA Test Result</th>
<th>Strain gauge test Result</th>
<th>Remark</th>
</tr>
</thead>
<tbody>
<tr>
<td>Location of max Stress for Case 1</td>
<td>109.971 Mpa</td>
<td>OK</td>
<td>OK</td>
<td></td>
</tr>
<tr>
<td>Location of max stress for Case 2</td>
<td>120.52 Mpa</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Location of max Stress for Case 3</td>
<td>162.64 Mpa</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

VII. CONCLUSION

At the end of extensive literature survey and FEM analysis and testing the following conclusions have been reached.

- Study of the existing boom and its failures was the most important step to begin the analysis.
- Finite element analysis of boom gives the behavior of material or part on application of load.
- Analytical solution with aid of FEA result matches well within a range of testing results.
- It is observed that the total stress induced in the boom is less than the permissible yield stress of the material only for static load conditions, that is equivalent (Von-Mises) Stress Maximum in vicinity of pin is 278.351 Mpa which is less than the permissible yield stress of boom material 377 Mpa, which shows that our design is safe by considering distortion energy (Von-Mises) theory.
- Higher value of stress was a prime reason for failure of base model.
- The boom designed in this work shall be useful as a standard assembly for all surface drilling machines manufactured by various OEMs and would ease maintenance. Cost of manufacture of the basic machines is expected to be lowered.

With reference to this thesis; in future; it is useful for cost reduction point of view for any parts of drilling rig in consideration of procedure and respective technical specification of respective parts. It is also applicable for other rig model for internal reference purpose.

REFERENCES

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