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

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Abstract- A surface Drilling Rig having a Boom and feed mechanism to support rotation head which used for generate required torque for drilling hole in mines. The main purpose of Boom design is to sustain the weight of Drill Guide assembly of approximate 3 ton. The Boom has a box structure with additional lower support by various cylinders whose function is to apply motion to the Boom at various positions to capture drilling area. The drawback of this design is that the boom is designed with single plate thickness which results in the eccentric action of forces whereby the boom bents upwards in the vertical plane. The results of the strength calculations confirmed that stress concentrations occur in the fracture location and the standard allowable stress (based on the welded joint notch class, the level of mean stress and the stress change range) for changing position at extreme downside.

Keywords – Boom Design, 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 support for sufficient progress in the mining works. These are the vehicles used for stabbing, 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).

In the case of drill rigs, this part allows one to position the drill rig mast so as to ensure stable perpendicularity of the drill rig axis to the surface of walls. The size of the machine and its working range are, in this case, determined by the size of the drilling mast mounted on the boom. The mast length ranges between 4 and 7m depending on the needs, determining the size of the boom mounted on the self- propelled mining machines. In order to reduce the costs of manufacturing the entire machine, it is important to standardize particular parts and units and to design a universal boom which could be mounted on various types of machines. Because the boom with the mast is a significant load for the structure, its mass should be as smallest as possible. This brings about serious engineering problems stemming from the operation, manufacturing technology, material limitations, etc.

II. PROBLEM IDENTIFICATION

The redesign of Boom [figure1.] is required for sustaining the load of Feed and Rotation torque at various load condition. The three different load conditions are given with their calculations as follows.

Case 1: Boom is in Horizontal condition& drill guide at 22deg to vertical.
Case 2: Boom is in Horizontal condition & drill guide in horizontal.
Case 3: Stress coming on the boom when lift cylinder is actuated

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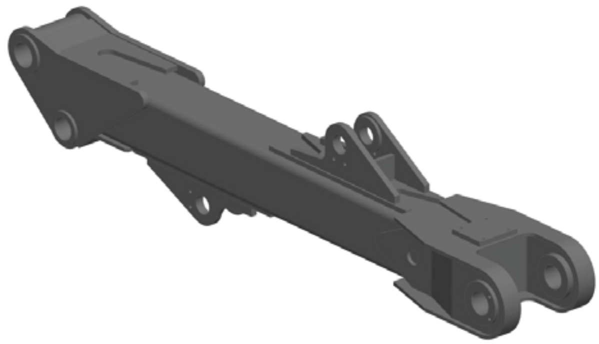

Figure 1. Boom

Input Material sizes & weight:-

Boom tube size - 200x200x12 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 [figure2]
Sq. tube outer side, a = 200 mm, Thickness = 12 mm
Sq. tube inner side, b = 176 mm
Moment of inertia = $(a^4-b^4/12)$ = 53373952 mm4
Z for section of boom= $(a^4-b^4/6a)$ = 533739.52 mm3

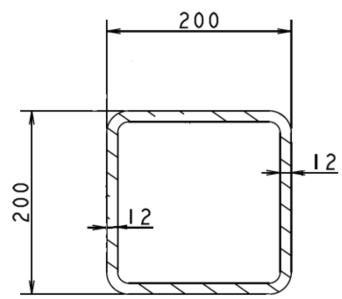

Figure 2. Cross Section of Boom

A. Case 1: Boom is in Horizontal condition& drill guide at 22deg to vertical (Figure3)-

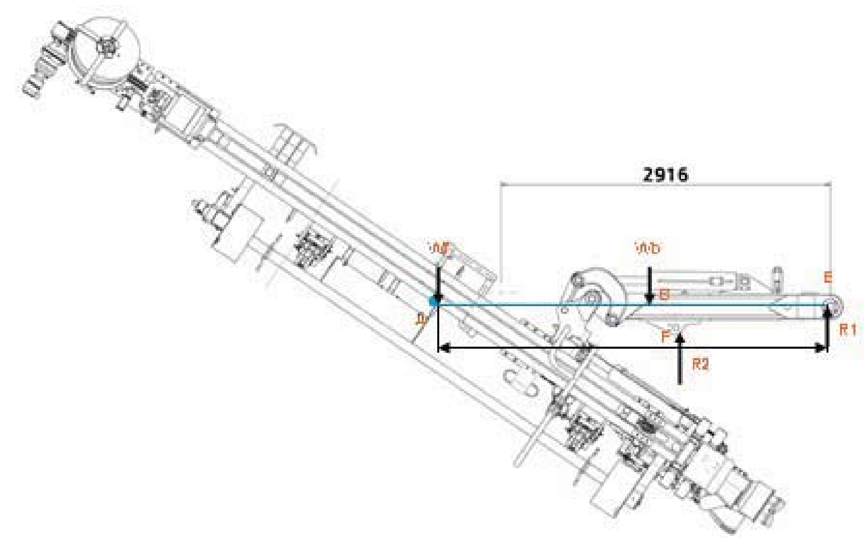

Figure 3. Boom in Horizontal Condition & Drill Guide at 22 deg. to Vertical

Hence by Calculations, Moment of Inertia, M_x =- 93123364.437 Nmm
Z for combine section of boom at X = 533739.52 mm^3
Hence bending stress induced in the boom = M_x / Z = 174.47 N/mm^2

Hence maximum induced bending stress at case 1 is 174.47 N/mm^2

B. Case 2: Boom is in Horizontal condition & drill guide in horizontal [Figure4]-

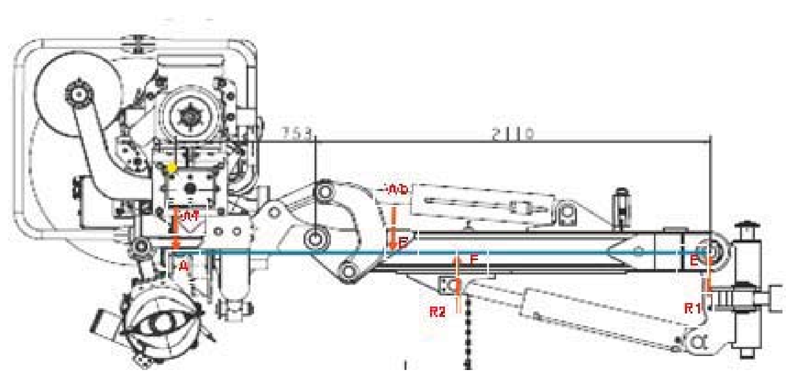

Figure4. Boom in Horizontal Condition & Drill Guide in Horizontal

Hence by Calculations, Moment of Inertia, M_x = -39892447.404 Nmm
Total Moment of inertia, I = 53483952 mm^4
Z for section of boom, Z= 608908.5455 mm^3
Hence bending stress induced in the boom = M_x / Z= 65.51 N/mm^2

Hence maximum induced bending stress at case 2 is 65.51 N/mm^2

C. Case 3: Calculating Stress coming on the boom when lift cylinder is actuated

Piston dia of the cylinder = 13 cm= 130 mm
Pressure on the hydraulic line =200Bar=$20x10^6$ N/m^2
Force acting when the cylinder is actuated = P x A = 260277.4485 N
From Graphical Analysis for boom angle 55˚, Φ = 30˚ [Figure5]

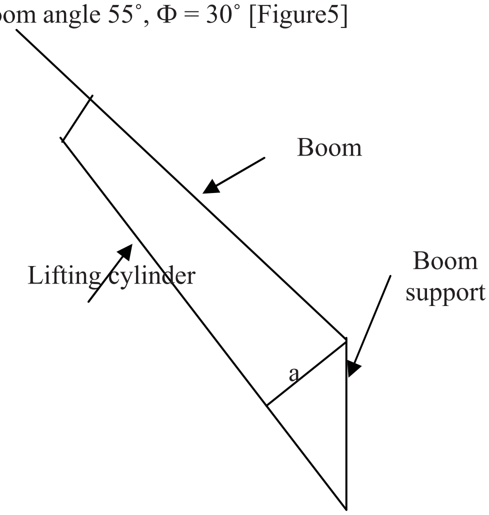

Figure.5 Ray Diagram of the Boom and Lift Cylinder

Perpendicular distance from hinge point to line of action of force (a) = 379 mm
Moment of cylinder force acting on boom = F x a = 98645152.9815 Nmm
Stress on the boom tube because of cylinder force (S= M_x*y/I) = 162.64 Nmm^2

Hence Combined Stress induced in Boom (including all cases) = 337.1198 N/mm^2

Now, Yield stress for rolled steel section FE410W = 245 Mpa
Allowable stress IS: 800 - 1984, Steel handbook = 24.5 Kg/mm2= 245 N/mm^2

Hence, The Total stress induced in the Boom is more than the permissible yield stress of the material without considering any factor of safety only for static load conditions. Hence it is recommended to use higher strength material or modify boom cross section for safe functioning.

III. PROPOSED SOLUTION

Input Material sizes & weight:-

Boom tube size - 200x200x12 th
Weight of feed beam assembly - 3105 Kg=30460.05 N,
Weight of boom assembly - 750 Kg=6916.05 N (due to increase in weight of plates)
Calculating section modulus (Z) for boom [figure6]

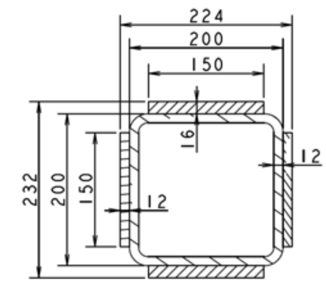

Figure6. Cross Section of Proposed Boom

Sq. tube outside, a_1 = 200 mm, Thickness = 12 mm, Sq. tube inner side, b_1 = 176 mm
Moment of inertia I1 = $(a_{14}-b_{14}/12)$ = 53373952 mm^4
Plate 150x12, b_2 = 12 mm, h_2= 150 mm
Hence, Moment of inertia I_2 = 45000 mm^4
Plate 150x16, b_3 = 16 mm, h_2= 150 mm
Hence, Moment of inertia I_3 = 60000 mm^4
Total Moment of inertia I = 53483952 mm^4
Z for section of boom= = 608908.5455 mm^3
Now, Calculating stress for above same cases with new Moment of Inertia & Section modulus, we get,

Case 1: Boom is in Horizontal condition& drill guide at 22deg to vertical.

Bending stress induced in the boom = M_x / Z = 104.231N/mm^2

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

Bending stress induced in the boom = M_x / Z = 65.51 N/mm^2

Case 3: Stress coming on the boom when lift cylinder is actuated

Stress induced in boom = 162.64 Nmm^2

Hence Combined Stress induced in Boom = 266.871 N/mm^2

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

Allowable stress IS: 800 - 1984, Steel handbook = 37.7 Kg/mm^2=377 N/mm^2

Material is also changed from Fe410WA to Fe510WC which is having yield strength 37.7 Kg/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 Design is safe with factor of safety 1.41.

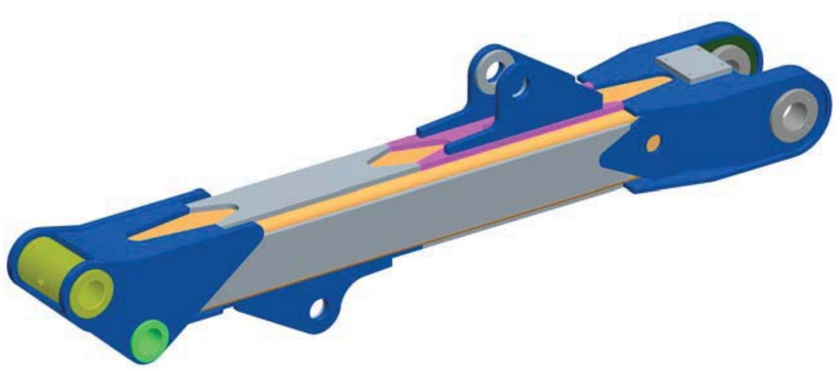

Figure 7. Designed Model of Proposed Boom

IV.CONCLUSION

Based on the results of Theoretical calculations, currently, it is concluded that the underlying causes of load substance due to increased weight of Drill mast is solved. Mainly, in the Boom critical zone it is the super positioning of influences that, more or less, has a detrimental effect on the local stress distribution: (a) influence of the support plates; and (b) influence of the proximity of the cross section affected by the load. Because of this the key idea of the Boom redesign is to dislocate the above mentioned stress concentrators in order to minimize as much as possible the detrimental effects of stress concentration super positioning. The Theoretical calculations results pointed out that the maximum stress value in the critical zone of the redesigned column head is 1.41 times lower than the permissible stress value. It is important to highlight that the reconstruction solution is designed in such a way as to be realized in field conditions.

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