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Failure of Boom in Self Propelled Surface Drilling Machine

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Abstract— *Operating requirements of machines used in surface mining are very severe. They are often subjected to percussive loads radically different from machines operating on the surface. The boom of these machine supports the rotation head and feed mechanism for drilling hole in earth surface. It has been observed that the boom fails during operation due to fracture and as a consequence several components of other assemblies undergo plastic deformation. The main objective behind boom design is that it should sustain the weight of drill assembly which is approximately 3 ton. The paper presents failure analysis and analytical results of boom using computer simulation. Finite element method analysis of the boom showed stress concentrations exceeding the allowable level done and the need to redesign it considering load fluctuations caused by operating environment. The results showed that maximum stress is induced at the observed fracture location. Redesigning the boom with increased cross section and improved material property overcomes the boom failure problem.*

Keywords— *Boom failure, Boom design, Drilling, Underground mining, FEM analysis*

I. INTRODUCTION

Self propelled mining machines are large crawlers that rip huge rocks, stones, coals, non-metals and directly load it on to conveyor. Operating conditions for surface mining equipment are radically different from machines operating on the surface. These machines are subject to adverse, severe and variable operating conditions and are often subject to percussive loads. Design of mining machines and components therefore demand a reliable construction that withstands the required loads whilst also being economical. This can be achieved through the use of modern integrated CAD/FEM systems.

Drill shot holes or anchor holes also made by drilling machines comprising a universal assembly, a front platform with an operator protecting structure. Such machines have a straight-line boom, to which distinct work tools can be attached, and feed mechanism to support rotation head for generating required torque for drilling hole in earth surface in mines [1]. A general defect found in such machines is damage and a cross fracture of the jib boom. This causes whole distinction of the front part of the jib from the rest of the machine. This paper deals with numerical model of the complete boom assembly and identifying the spatial stress field generated by the feed and rotation operational torque as regards its effect on the life of the boom. The efforts lead to evaluation of the loads acting upon the boom considering the most stringent load conditions.

II. SCOPE OF STUDY

The existing boom assembly has a box structure with additional lower support of various hydraulic cylinders whose function is to provide motion to the boom to attain various positions to capture drilling area. In the preliminary model, the weight of the drill mast is around 2.1 ton which the boom apparently does not sustain under the severe conditions the machine is required to operate. A common fault, which was found in machines of this type was damage of a cross fracture of the jib boom, causing whole distinction of the front part of the jib from the rest of the machine [2]. As was observed, the boom bends upwards in the vertical plane due to eccentric action of forces. It was found that the changing loads adversely affected the load-carrying capacity of the structure and resulted in formation of cracks at stress concentration sites in welded joint areas. Sometimes the machines operate in the range of resonant vibration which accelerated the degradation. This meant an unpredictable increase in load of feed and rotation torque at extreme load conditions [3, 4]. Boom design necessitates incorporating the requirement of sustaining the increased load of feed and rotation torque at extreme load positions besides its static analysis.

The purpose of redesigning the boom was to overcome the failure problem discussed above and to develop a universal boom that would sustain the weight of drill guide assembly with higher capacity of approximate 3 ton on machines with mast length ranging between 4 and 7m depending on the need. To

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minimize the time related to changing location of the machines, the boom mounted on self propelled mining machines need to possess sufficient number of degrees of freedom. Since the boom with the mast is a significant load for the structure, it also becomes necessary that this mass should be as small as possible. Boom redesigning, therefore, necessitated a consideration of engineering problems stemming from the operation, manufacturing technology, material limitations etc. The other objective was to standardize the product so as to be compatible for other similar kinds of surface drilling machines marketed by the manufacturer in the context of ease of mounting. This implies that the boom should be suitable enough to be used without demanding any helpful technologies or modifications on the wide range of machines. Such an approach not only achieves a reduction in the overall cost of manufacturing of the complete machine but also improves flexibility and ease of machine maintenance.

III. ANALYSIS

Existing boom assembly is shown in Figure 1. The present boom section is 200 mm x 200 mm built out of 12 mm thickness plates by welding is shown in Figure 2.

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. The force directions and loads acting on the structure are presented below in Figure 3 and Figure 4 for case (1) and case (3). It is seen that the maximum induced bending stress for FE410WA steel for case (1) is 180.21 N/mm^2 while the same for case (2) it is 79.87 N/mm^2 . To this gets added the stress induced by the force exerted by the actuating hydraulic cylinder which has been evaluated as 168.38 N/mm^2. Combined stress induced which is the addition of stresses induced in case (1) as it is more than case (2) and case (3) in boom is 348.59 N/mm^2. The results of the analysis as above are presented in the Table 1 below. Considering that yield stress for rolled steel section FE410W is given as 245 Mpa and the allowable stress as per IS:800 - 1984 Steel Handbook is 245 N/mm^2, it is concluded 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. This justifies the need for redesigning the boom.

Figure 1: Existing boom assembly

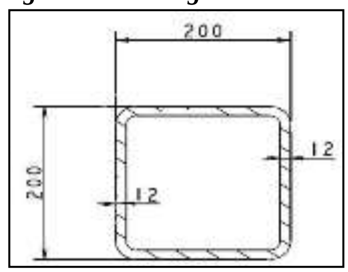

Figure 2: Cross section of boom

Case I: Boom is horizontal and drill guide at 22° to vertical with boundary conditions as shown below in Figure 3.

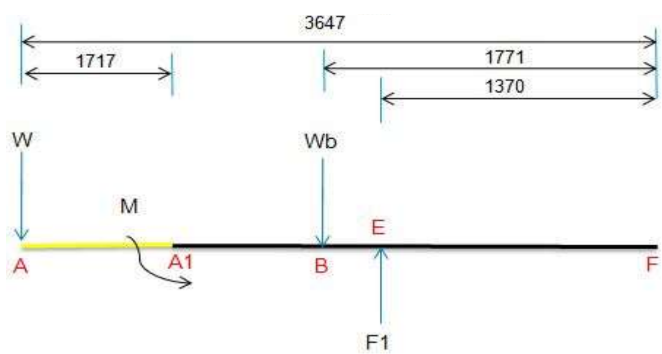

Figure 3: Loading and boundary conditions for Case I

Case III: Stress developed in the boom when lift cylinder is actuated with boundary conditions as shown below in Figure 4.

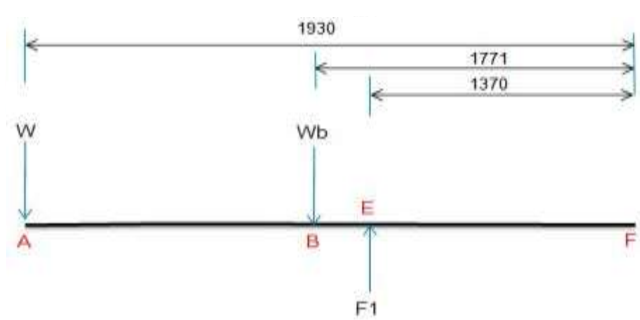

Figure 4: Loading and boundary conditions for Case III

IV. PROPOSED CHANGES IN BOOM DESIGN

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 FE510WC next grade of existing material FE410WA which is having yield strength 37.7 Kg/mm^2 or 377 Mpa. Similarly, 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 size – 200 mm x 200 mm x12 mm thickness has been retained. New 12 mm plates have been welded upon as shown in Figure 5. The weight of feed beam and boom assembly now, as estimated, is 3105 Kg and 750 Kg respectively. Section modulus for boom is also evaluated. The calculated stresses for the proposed cross section for the above stated two identified worst load conditions cases on similar lines of existing design are given in the Table 1 below. The total stress induced in the boom is less than the permissible yield stress of the material for static load conditions. It can therefore be stated that the design is safe with factor of safety 1.35.

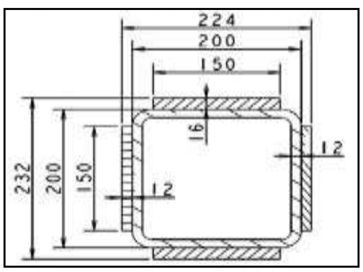

Figure 5: Cross section of proposed boom

Table 1: Analytical estimate of stress induced in boom in existing and proposed design

Loading condition	Existing boom, (N/mm^2)	Proposed new design, (N/mm^2)
Case 1: Boom is in horizontal condition and drill guide at 22° to vertical	**180.21**	**109.971**
Case 2: Boom is in horizontal condition and drill guide in horizontal	**79.87**	**71.25**
Case 3: Maximum stress coming on the boom when lift cylinder is actuated	**168.38**	**168.38**
Combined stress induced in boom	**348.59**	**278.351**

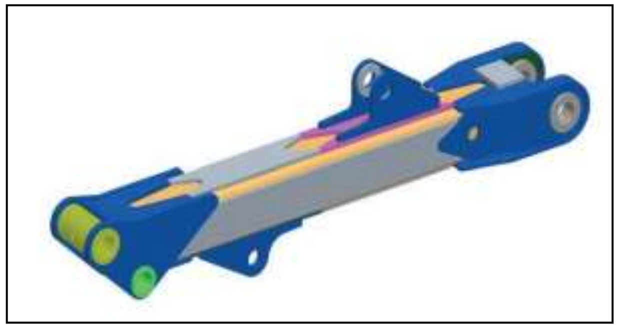

Figure 6.Assembly of proposed model by UNIGRAPHICS

V. FEM ANALYSIS

The revised design of the boom was analyzed further using FEM analysis. Figure 6 shows the assembly of proposed model using UNIGRAPHICS software for cases I & III together. Standard preprocessing material properties for FE 510W (S355JR) have been used in the analysis with type of element considered being hexahedral, tetrahedral for meshing. Please see Figure 7. The number of nodes equal 385447 while the number of elements are 140102 for loading & boundary conditions as discussed above for numerical calculations.

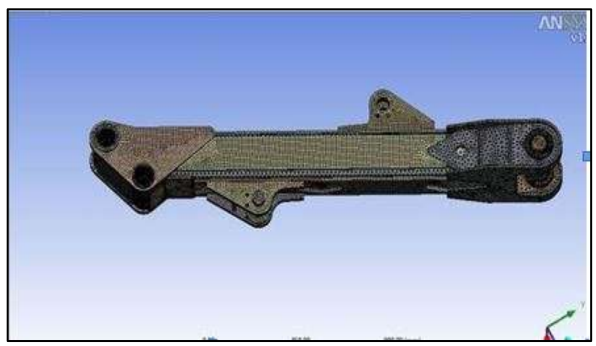

Figure 7: Meshing of boom assembly for FEM analysis

Weight of feed assembly (W), equaling 30460.05 N, acts through the centre of gravity of feed assembly. Considering its effect on boom assembly in terms of moment we get Moment of inertia M = 52.3 KN. Weight of boom assembly (W_b) is 6886.2 N and the force acting when the cylinder (F) is evaluated as 260277.4485 N at an angle of 30˚ on point E. Resolving this cylinder and considering only its vertical component, one obtains vertical force component (F_1) = F*sin 30˚=260277.4485*0.5 = 130.138 KN. Among the two cases of boom, stress developed in case I is more than that in case II. Combined stress or equivalent stress have therefore been evaluated considering only case I combined with case III. The results for the two critical cases are presented below pictorially in Figure 8 and 9. They have been compared with the conventional analysis in Table 2 below.

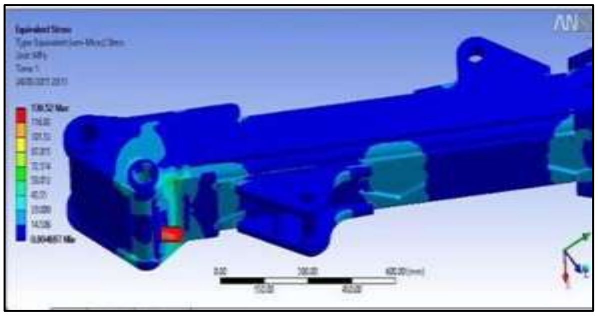

Figure 8. ANSYS results for Case I -Boom is horizontal and drill guide at 22deg to vertical, the maximum stress induced in a boom assembly is 130.52 Mpa

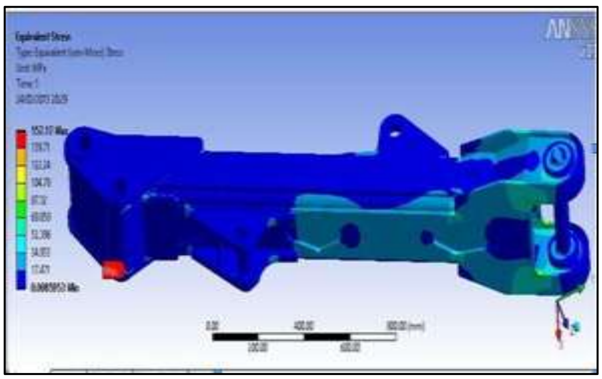

Figure 9. ANSYS results for Case III- Stress developed in the boom when lift cylinder is actuated is 157.17 Mpa

Table 2. Comparison of results for analytical estimate of stress induced and after FEM analysis

Loading condition	**Analytical estimate (N/mm^2)**	**FEM analysis (N/mm^2)**
Case 1: Boom is in horizontal condition & drill guide at 22° to vertical	**109.971**	**130.52**
Case 2: Boom is in horizontal condition & drill guide in horizontal	**71.25**	**-**
Case 3: Stress developed in the boom when lift cylinder is actuated	**162.64**	**157.17**
Combined stress induced in the boom	**278.351**	**287.69**

V. CONCLUSIONS

Analytical assessment of stresses developed in the boom under critical loading conditions by conventional methods and using FEM analysis show that only for static load condition the total stress induced in the boom is less than the permissible yield stress of the material. The equivalent (Von-Mises) stress is 278.351 Mpa and is found to be maximum in the vicinity of the pin. It is, however, less than the permissible yield stress of boom material of 377 Mpa, It is thus established that that new design is safe by considering distortion energy (Von-Mises) theory. It is also observed that the equivalent stresses developed in the boom estimated by conventional analytical method (278.351Mpa) closely matches with the results obtained using FEM analysis (287.69Mpa).

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