Design and Analysis of Telescopic Boom for Mobile Cranes

Akhilesh Chandgude, Akshit Agarwal, Clint Mathew, Sai Chavan

Department of Mechanical Engineering, SIES GST, Mumbai University, Navi Mumbai, India

Abstract— *This paper includes the design of five sections of the telescopic boom, automatic boom extension rope mechanism and self-compensating rope mechanism used in the crane. The models of the booms were designed using CAD software and verified using finite element analysis software. Stiffener plates were used to improve the strength of critical sections found by analysis. Also this paper incorporates selecting two rope mechanisms which eliminate the human involvement in the extensions and hence reduces chance of error in working of the booms.*

Keywords— Stiffener Plate, Rope Mechanism, Finite Element Analysis, Telescopic Boom

I. INTRODUCTION

A crane is a load carrying machine that has either wire ropes or chains, and sheaves, that lifts and lowers material and moves them from one place to another. The telescopic boom assembly is the most important component of pick and carry cranes. This assembly may consist three to as much as over ten sections. The present work deals with a crane having five boom sections with two rope mechanisms which are responsible for the rope and the boom movements. The design and analysis have been done considering the maximum loading conditions. The objective of the work may include:

1. Selecting extension mechanism
2. Selecting rope compensation mechanism
3. Calculation of dimensions of boom cross-sections
4. Selecting material to optimize weight and strength
5. Modelling of boom sections using SolidWorks software
6. Analysis of boom sections using ANSYS software
7. Adding stiffener plates to critical sections identified in analysis

The calculations for finding the dimensions were carried out considering shear and tensile failure of the boom sections referring as stated [1].

The various sections of the assembly can be categorized as:

1. Mother Boom Section
2. Intermediate Boom Sections (3 in number)
3. Tip Section

II. ROPE MECHANISM FOR EXTENSION OF BOOMS

In conventional system, the extensions are operated individually by separate levers. This causes non-uniform distribution of the load over the length of the booms. In 1968 J.L.Grove [2] showed that the mechanism for automatic and simultaneous extension of two booms using a single hydraulic cylinder which is further developed for 5 booms in this paper. This mechanism uses a system of pulleys, winding drums and a single piston to cause motion of each boom relative to the preceding boom section. It helps in even distribution of the load over all the boom sections. It also eliminates the need for a separate hydraulic system for each boom. The hydraulic piston, which is pivoted inside the mother boom, will push the first intermediate section and the tension this motion develops in the rope will cause the motion of the other booms. Each boom moves relative to the previous boom, distance equal to the extension of the hydraulic piston. The schematic of the mechanism is shown in Fig.1.

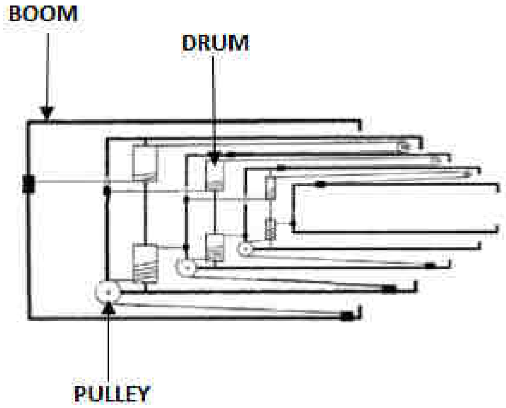

Fig. 1: Sectional top view of rope mechanism for extension of booms

III. THE SELF-COMPENSATING ROPE MECHANISM

In conventional mechanisms the rope has to be extended manually by the operator when the boom sections are to be extended. This might lead to excessive tension in the rope due to any error by the operator. To avoid this problem a mechanism involving a system of pulleys is used. This mechanism keeps a particular length of rope in reserve and the change in the distance between the winch and the hook in compensated for by this additional rope. A schematic sketch of the mechanism is shown in Fig.2. The pulleys are arranged so as to eliminate the need for unwinding the rope from the winch, when the booms are extended. The winch is used only when the hook has to be lowered or raised.

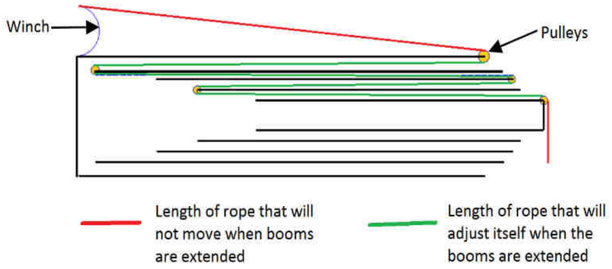

Fig. 2: Sectional side view of self-compensating rope mechanism

IV. DESIGN CALCULATIONS

The calculations were carried out according to SAE J1078 that includes checking for bending, shear and tensile failure of the boom sections. The design calculations[3]were carried out and a number of iterations were made by making changes in the material, thickness of the metal sheets and the cross section.

Important parameters considered in design:

1. Yield strength of the material
2. Thickness of the metal sheets
3. Cross sectional dimensions (height and width)

TABLE 1 shows the dimensions of boom sections obtained from these calculations. Variation in cross section can be observed when there is change in material or thickness.

Table 1: Dimensions of boom sections obtained from calculations

Part	Material yield Strength (MPa)	Thick-ness (mm)	Height (mm)	Width (mm)
Tip Boom	410	8	150	110
Boom 4	410	8	242	208
Boom 3	410	8	334	300
Boom 2	410	8	426	393
Mother Boom	410	8	520	486
Tip Boom	250	8	185	150
Boom 4	250	8	277	242
Boom 3	250	8	370	333
Boom 2	250	8	462	426
Mother Boom	250	8	555	518
Tip Boom	250	10	175	140
Boom 4	250	10	271	236
Boom 3	250	10	368	333
Boom 2	250	10	464	429
Mother Boom	250	10	560	526

4.1 Material of metal sheets

The selection of material is based on multiple parameters such as availability, machinability, weldability, yield strength and cost. The material selected is mild steel IS2062 E-250 and its properties[4] are as given below:

Ultimate tensile strength (min.)	410 MPa
Yield Strength (min.)	230-250 MPa
Percentage Elongation	23
Bend Test	25 mm
Mass Density	7.85 kg/m3
Poisson's Ratio	0.29

Fig. 3 shows the variation in weight of material used with change in material used. The graph is plotted for two different thicknesses of 8mm and 10mm respectively. The graph has yield strength (N/mm^2) on X-axis and weight(kgs) on Y-axis.

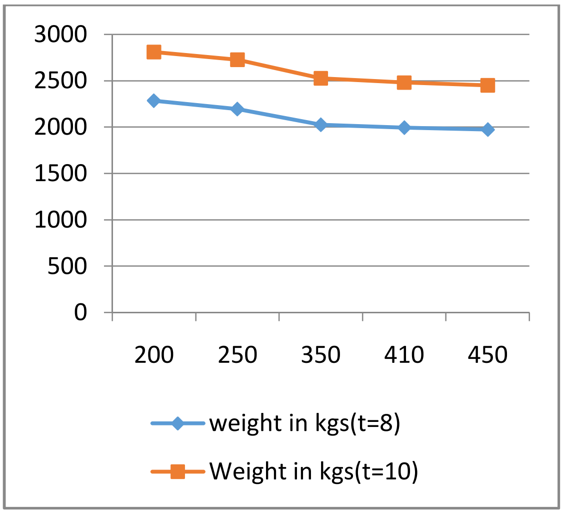

Fig. 3: Graph showing variation of total weight of material used for all boom sections with change in material

4.2 Thickness of metal sheets

The weldability considerations call for the thickness to be greater than 4mm, since lower thickness will require the components to be spot welded, which causes stress concentration at the joint as well as low fatigue strength in the boom. Whereas, to keep the weight of the material in check the thickness should be kept under 12 mm. The variation in cross section for same material with change in thickness can be seen in TABLE 1. The final thickness of the plates that was selected is 10mm.

4.3 Cross-section of booms

The dimensions of cross-section of each boom is calculated individually according to SAE J1078. The dimensions of tip section were calculated by optimizing the parameters such as thickness, material yield strength and cross-section (height and width). The cross-sectional dimensions for the remaining booms were calculated by incorporating clearances between the booms. A clearance of about 1.5" to 2" is provided between the booms, to incorporate the pulleys and the rope for the mechanisms and bearing pads. These dimensions were further verified according to SAE J1078.

V. STIFFENERS

A bar, angle, channel, other additional plate or section attached to a metal plate or sheet to increase its resistance or prevent loss of strength due to web buckling. The comparison of stiffened and un-stiffened elements [5] can be seen in the Analysis section of the paper. Bearing stiffeners are used to strengthen the critical sections of the booms. Fig. 4 shows the stiffeners attached to the boom. The locations for adding the stiffeners were identified after analysis of model in ANSYS software. Fig.11 and Fig.12show the deformation and Equivalent stresses (Von Mises stresses)respectively in the intermediate boom without adding stiffener plates. Fig.13 and Fig14. show the deformation and Equivalent stresses (Von Mises stresses)respectively in the intermediate boom after adding stiffener plates. It is observed that the deformation and stresses reduce considerably after stiffener plates are added.

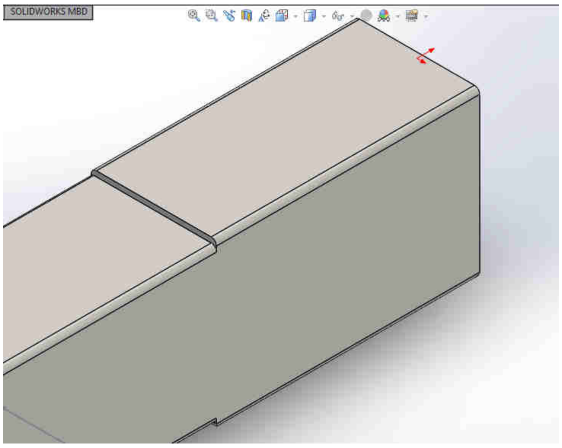

Fig.4: Stiffener fixed on top and bottom of intermediate boom section

VI. BEARING MATERIAL

Bearings pads or wear pads are provided between successive booms to avoid metal to metal contact of booms. The wear pad [6] units which are used may consist of plural pad elements individually supported on cushioning elements who’s spring constant or strength maybe varied to meet particular conditions of loading or stress at that particular pad element. The bearing pads used in this paper are of dimension 150X30Xt mm, where ‘t’ is the thickness of the bearing. The thickness[7] depends on the clearance given between the booms. A groove of 2 mm is provided on the boom surface to allow proper sliding of the boom assembly. The material of these pads is Nylon 6,6. Fig. 5 shows the assembly of the booms with bearing in between.

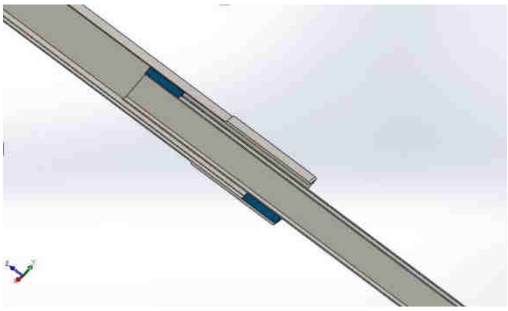

Fig.5: Sectional view of boom assembly with bearings

VII. ANALYSIS

The analysis[8] of the telescopic boom assembly was carried out on individual booms rather than on complete boom assembly. This was done in order to avoid unnecessary complications and easy identification of point of failure, if any. The booms were considered in fully extended positions for finding critical points.

7.1 Boom Models

The boom models were created using SolidWorks'13. The material selected was ISC2048 E-250. The corresponding dimensions of the booms were taken from design calculations. Fig. 6 shows the model of tip section boom.

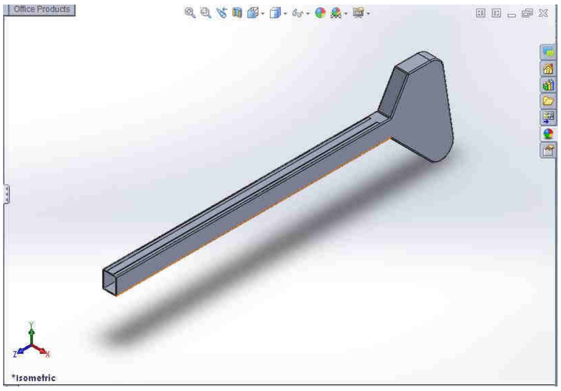

Fig.6: Model of tip section

7.2 Boundary and loading conditions

The analysis was carried out on ANSYS16.2. The overhang of the preceding boom is considered to be fixed as the boom hinges at these two points. The overhang of the succeeding boom is considered for applying the loads. The reactions obtained from the boom are applied as force on the preceding boom. The self-weight of the boom is considered at its center of gravity. This crane is assumed to be of a capacity of 14 tons and at maximum extended position of booms it is assumed to carry a load of 1.5tons.

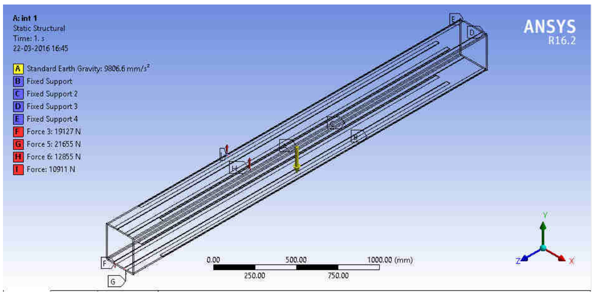

Fig.7: Boundary and loading conditions for intermediate boom without using stiffener

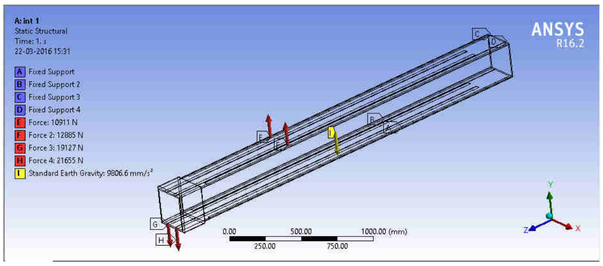

Fig.8: Boundary and loading conditions for intermediate boom using stiffener

Fig.7 and Fig.8 show the boundary and loading conditions applied to the boom without using stiffener and after adding the stiffener respectively.

7.3 Result

Fig.9 and Fig.10 show the deformation and equivalent stresses (Von Mises stresses) in the tip boom. The maximum deflection and stress observed were 20.56mm and 191.83N/mm^2 respectively. The tip boom was identified as the weakest section out of all the boom sections.

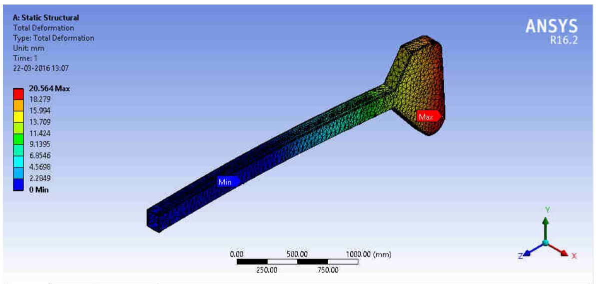

Fig.9: Total deformation of tip section

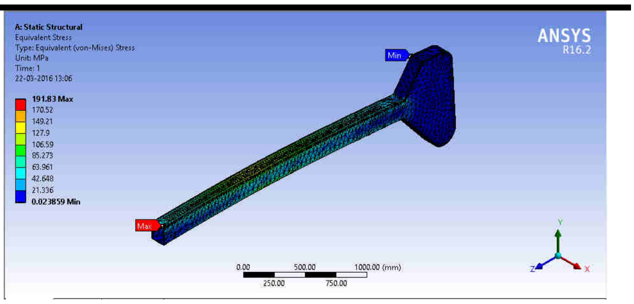

Fig.10: Equivalent stress (Von-Mises stress) on tip section

Fig.11 and Fig.13 show the comparison of deformation and Fig.12 and Fig.14 show the comparison of Von-Mises stresses when stiffener is not used opposed to when it is used respectively.

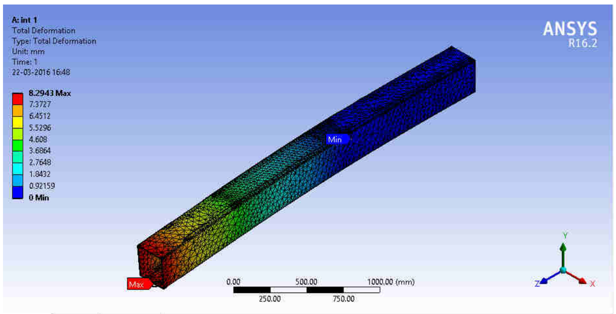

Fig.11: Total deformation of intermediate section without using stiffener

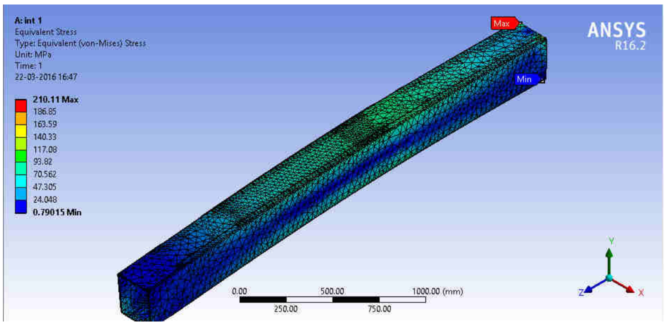

Fig12. Equivalent stress (Von-Mises stress) on intermediate section without using stiffener

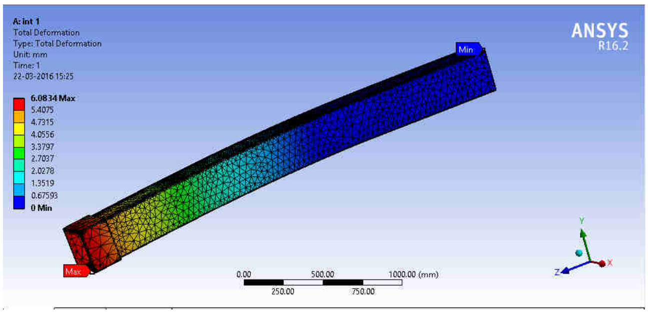

Fig.13: Total deformation of intermediate section using stiffener

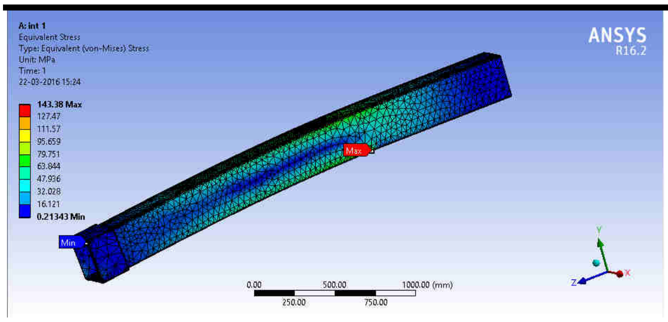

Fig.14: Equivalent stress (Von-Mises stress) on intermediate section using stiffener

The maximum deformation in the intermediate boom section without using stiffener was found to be 8mm. This deformation was reduced to 6.08mm after using stiffener.

The maximum equivalent stress in the intermediate boom section without using stiffener was found to be 210MPa. This stress was reduced to 143.38MPa after using stiffener.

VIII. CONCLUSION

The paperexplains the various mechanisms that can be used for smooth working of crane booms and further shows how the booms can be designed using CAD and verified using FEA. This approach can be used for future design and research. The models used in analysis does not include small attachments such as pulleys, tension springs, fixed hooks and other accessories. However, these components do not contribute much to the loading factor as the overall load is much higher. The model for the boom sections was created using SolidWorks'13 and then they were imported into ANSYS16.2 where the finite element analysis was carried out. The stiffener plates that were added after analysis, reduced the stress and deflection considerably and are required to be added to only the critical sections to reduce wastage of material. The maximum stresses were found to be lower than the ultimate tensile stress and yield stress of the material selected for the booms. Hence, the design can be safely used for lifting loads up to 14 tons. For future work different shapes of cross-sections can be compared to select the most optimum design for mobile crane booms.

REFERENCES

[1] 'A Recommended Method of Analytically Determining the Competence of a Hydraulic Telescopic Cantilevered Crane Booms'- SAE J1078 April 1994.

[2] J.L.Grove, Method of and Apparatus for Extending a Telescopic Crane Boom, 1968, Patent: US3386594.

[3] Navneet Kumar, Mohd. Parvez," Force Distribution on Telescopic Boom of Crane" International Jornal of Mechanical Engineering and Robotics Research, Vol.1, No.2, July 2012, pp.96-104.

[4] IS 2062-2011 Hot Rolled Medium And High Tensile Structural Steel — Specification (Seventh Revision)https://law.resource.org/pub/in/bis/S10/is.2062.2011.pdf

[5] Sunil. M. Hardwani, A. V. Patil, "Study, Test and Designing of Cold Formed Section as per AISI Code" International Journal of Applied Sciences and Engineering Research, Vol. 1, Issues 3, 2012, pp.522-531.

[6] Fu Tien Sung, Crane Boom Having Wear Pads, 1973, Patent: US3719403 A.

[7] Shinzong Chen, Yu Tang, Yuhou Wu, Ke Zhang, Jia Sun," Design and Finite Analysis of Wind Turbine Generator Internal Maintenance Lift Platforms Lifting Boom Mechanism" International Journal of Mechanic Systems Engineering, Volume 3, Issue 4, November 2013, pp.135-140.

[8] Bhupender Singh, Bhaskar Nagar, B.S. Kadam, Anuj Kumar," Modeling and Finite Element Analysis of Crane Boom" International Journal of Advanced Engineering Research and Studies, Vol. I, Issue I, October-December, 2011, pp.51-52.