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Analysis of Structure of Telescopic Arm of QY25 Truck Crane

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Abstract: The telescopic arm is the main force component of the truck crane, and its weight accounts for about 20% to 30% of the whole crane. The performance of the telescopic boom has a great impact on the large-tonnage crane and the high-lift and high-rise working conditions. Therefore, it is more and more important to increase the weight and reduce the weight of the telescopic arm by optimizing the design. This article takes the telescopic boom of QY25 crane crane as the research object, discusses its mechanical performance through finite element analysis, and uses ANSYS Workbench software to optimize it to find a more reasonable shape of telescopic arm cross section.

Keywords: Truck crane telescopic arm, three working conditions, static analysis, optimum design.

1. FINITE ELEMENT MODEL ESTABLISHMENT

The maximum rated lifting capacity of a QY25 truck crane is 25t, the minimum rated amplitude is 3m, the maximum lifting moment is 75t • m, the longest main boom is 23m, the maximum rotation speed is 2r/min, and the maximum speed of the main lifting mechanism is 110m/ Min. The telescopic arm has a U-shaped cross-section and its structural diagram is shown in FIG. 1 -1

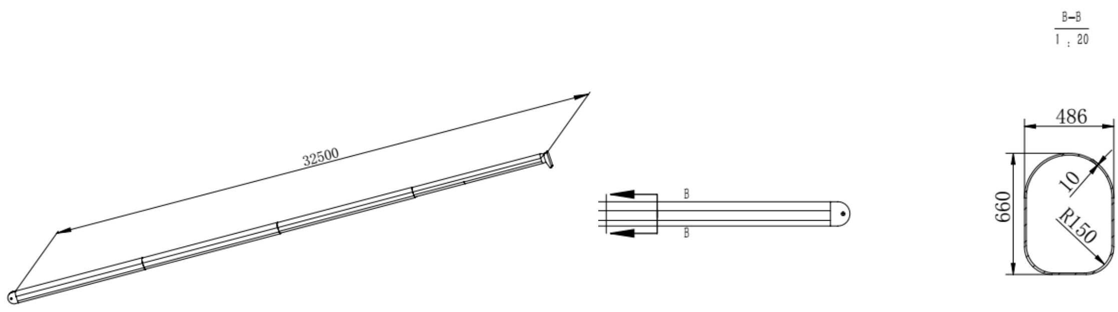


Fig 1. Sketch of crane boom

1.1 Model simplification and unit type selection

The edge reinforcement part of the telescopic arm is to ensure the welding process and has little influence on the calculation result, so the simplification of the finite element model is established. The actual reinforcement of the arm head and the ribs of the arm root and the turntable hinge are more and the force is more complex. Therefore, the reinforced parts of the arm head and the arm root are simplified to be thickened surfaces. The jib weld is located on the middle face of the boom height and the force is relatively small and is ignored. Both the slider and the cover plate are relatively regular faces, so the grid meshing is used in the meshing and the number of iterations can be reduced in the calculation.

During the modeling process, the element type of the slider is so- lid45, the element type of the lid is shell63, the thickness of the upper and lower cover plates of the model is determined by the real constant, which is more in line with the actual situation of the telescopic arm.

1.2 Telescopic arm constraint processing

Since the telescopic arm only slides relative to the slider in the telescopic direction, the other five degrees of freedom need to be constrained by the coupling. The model of this paper selects the contact surface node and uses the cap command to realize the coupling of the contact surface. This coupling adopts the linear calculation method, which is simpler than the method of establishing the contact pair, and the calculation result is accurate. A stiffness node is established at the hinge point of the hydraulic cylinder and at the hinge point of the arm, and only the degree of freedom of rotation of the telescopic arm about the hinge point in the plane of the variation is released.

2. LOAD CALCULATION

The load on the crane includes lifting weight, deadweight, lateral load, tension on the arm of the wire rope, wind load, etc. The wind load is added on the side of the boom, while other loads must be added to the corresponding position or key point, key face on. Firstly, the inertial load is added. The inertial load of the jib is automatically detected by the software. Before loading, the parameters of the material must be set. The Young's modulus is 2.06×MPa, and the Poisson's ratio is 0.31 and the density is 33/107.85 mg. The lifting load is added to the pulley shaft hole of the boom. The tension of the steel rope is added at the shaft hole of the guide wheel. The direction is defined by the local coordinates. The force analysis can decompose these loads into axial force and lateral force.

In the luffing plane

Axial force

$$N_1 = \phi_2 P_Q \sin\theta + G_b \sin\theta + \frac{\phi_2 P_Q}{\eta_i} \quad (1)$$

Lateral force

$$H_1 = \phi_2 P_Q \cos\theta + G_b \cos\theta \quad (2)$$

Bending moment

$$M_1 = e_2 \phi_2 P_2 \sin\theta - \frac{e_1 \phi_2 P_Q}{i\eta} \quad (3)$$

In the plane of boom rotation, the load on the jib is

Lateral force

$$H_2 = P_Q \tan\alpha + P_{W11} + \phi_5 m_b \alpha_T \quad (4)$$

Torque

$$T = P_Q e_2 \tan\alpha \quad (5)$$

In the formula

i——hook pulley block magnification;

η——pulley efficiency;

ϕ_2 ——lifting dynamic load coefficient;

ϕ_5 ——dynamic effect load factor;

θ——boom elevation angle;

P_0——rated lifting load ;

$P_Q = (m_Q + m_0)g$, g——Gravity acceleration, m_Q——rifting quality for luffing planes,

m_0 For spreader quality; G_b——Boom weight $G_b = m_b g$, m_b For the boom quality;

α ——The maximum yaw angle of crane wire rope, take 3° ~ 6° ;

α_T——The tangential acceleration of the boom end;

e_1, e_2 ——Lifting rope guide pulley axis and hook fixed pulley axis to the neutral axis of the arm

P_{w11}——Working state maximum wind load acting on the crane.

3. TELESCOPIC ARM STATIC ANALYSIS

3.1 Working condition analysis

Table 1. Three kinds of working conditions

Condition	arm length	amplitude	maximum lifting weight
Condition1	10.5m	3m	25T
Condition 2	17.7m	7m	11.5T
Condition 3	32.5m	8m	6T

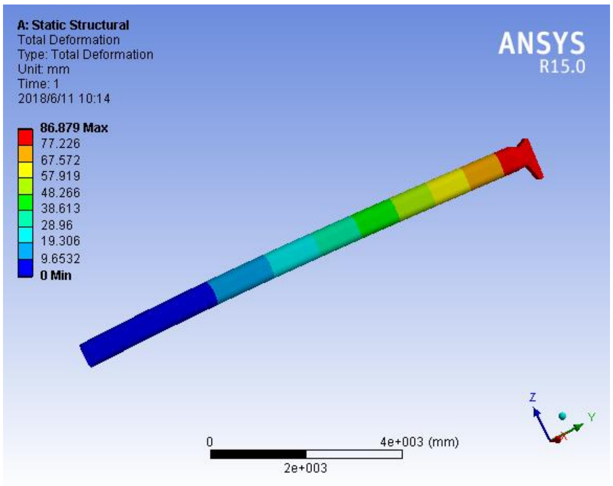
3.2 Telescopic static analysis results

When all the settings are completed, the next step is to solve the model. There are two kinds of solvers: direct solver and iterative solver. In general, the solver can be selected by itself or set in advance. In this paper, the automatic selection solver is used to calculate the total deformation of the telescopic arm under different working conditions. The total deformation of the telescopic arm under three conditions is shown in Figure 1-2.

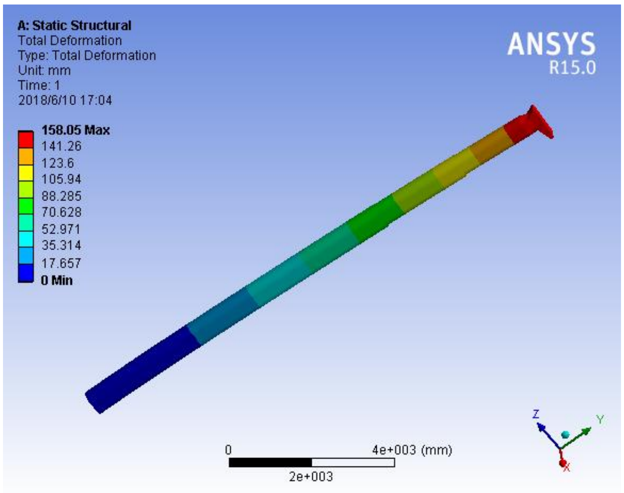
3.3 Result analysis

(1) According to the analysis results, the maximum deformation of condition 1 is 86.879mm, the maximum deformation of condition 2 is 158.05mm, and the maximum deformation of condition 3 is 1492.5mm.

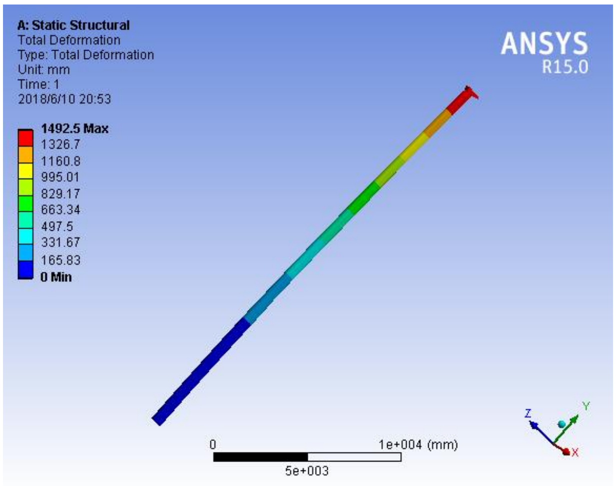
(2) Due to the different lengths of the telescopic arms in the three working conditions, their allowable displacement deformations are not the same. From the calculation, the allowable deformation of condition 1 is 110.25mm, and the allowable deformation of condition 2 The quantity is 207.36mm. The allowable deformation of condition 3 is 1656.45mm. The comparison with the finite element analysis results shows that the deformation of the model meets the design requirements.



(a) Deformation diagram of condition 1



(b) Deformation diagram of condition 2



(c) Deformation diagram of condition 3

Fig 2. Deformation diagram of telescopic boom under three conditions

4. SUMMARY

In this paper, finite element analysis software ANSYS Workbench was used to perform the finite element static analysis of the telescopic boom of the truck crane. By reasonably simplifying the boom structure, a geometric model of the crane boom structure is established. Then the finite mesh is divided according to the Poisson's ratio, the elastic modulus, and the load in the actual working conditions. The boundary conditions are imposed on the characteristics of the arm movement and force transmission, and the finite element modeling process of the boom structure from geometric modeling, mesh division to applied load and boundary conditions is completed, and finally the deformation of the telescopic arm under three working conditions is obtained. Illustration. Let us understand the position of the telescopic arm and the size of the deformation, and provide a certain reference for the design and optimization of the structure.

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