Finite element analysis of the chassis in telescopic crawler crane

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Abstract. In modern industry, crawler crane is an essential equipment. Chassis is an important component for crawler crane stability. In this paper, the finite element software ANSYS is used to establish the research model of SCC250TB crawler crane chassis. According to the stress characteristics of crawler crane, six typical working conditions are selected, and the corresponding stress nephogram is obtained through calculation. The model provides the basis for the strength and lightweight design of the crane’s chassis. It plays an important role in practical engineering projects.

1. Introduction

In recent years, the development trend of large-scale engineering equipment has made crawler crane products continue to be researched and developed in the direction of over-sized crawler crane or even super over-sized crawler crane. However, the design method and processing technology of domestic crawler cranes are still not well developed. Many potential quality problems of the product design still exist, and has caused huge losses[1]. The larger size and weight of the crawler crane has caused a substantial increase in production costs, transportation costs and operation costs[2]. Therefore, under the premise of safety and reliability, the lightweight design of the structure is the development trend in the industry. This not only can reduce the manufacturing cost and promote energy conservation and environmental protection, but also has important significance for improving market competitiveness.

The finite element simulation strategy for industrial design is currently one of the most important production methods for enterprise designers. With the continuous advancement of computer technology and computing software, the technology of using computers for collaborative manufacturing (Computer Aided Engineering) has become the most important part of product development and design[3]. In the field of mechanical engineering, finite element method has been maturely applied and achieved some results in crawler crane development and design[4-5].

2. Finite element analysis

2.1. Introduction to the chassis part of crawler cranes

The schematic diagram of the complete vehicle of the SCC250TB crawler crane is shown in Figure 1.
Its structure is divided into three parts: rotating platform, boom system and the chassis system\cite{6}. The base and crawler frame, as the main components of the chassis model, are welded by high-strength alloy steel plates. The function of the lower part is to handle the lifting load from the bogie and transfer its own weight to the ground through the base, track frame, track shoe and rollers. Therefore, the structural safety and reliability of the base and the crawler frame directly affect the working performance and safety performance of the whole machine.

![Figure 1 Schematic diagram of the complete crawler crane](image1)

![Figure 2 The lower part of crawler crane](image2)

2.2. The selection of working conditions
The chassis part of crawler crane mainly includes two parts, as shown in Figure 2.

The base is connected with the rotating system through a slewing support, and the track frame is connected with a pin shaft and an extrusion block. Its function is to transfer the load borne by the upper vehicle to the crawler frame to ensure the stability of the crane.

The crawler frame uses pins and extrusion blocks to connect the bogie, and is connected to the reducer and the four-wheel belt through bolts and pins to transmit the force to the ground.

Taking the operation requirements into account, we select six typical working conditions according to the characteristics of the off-car force, as shown in Table 1.

| Working condition | Working condition description |
|-------------------|------------------------------|
| 1                 | In maximum torque of 110 t·m, load 29.375t (overload) condition, operating radius 5m, and the car turns 0 degree |
| 2                 | In maximum torque of 110 t·m, load 29.375t (overload) condition, operating radius 5m, and the car turns 45 degrees |
| 3                 | In maximum torque of 110 t·m, load 23.5t + 3%side-loading getting off condition, operating radius 5m, the car turns 0 degree |
| 4                 | Under the maximum load of 31.25t(overload), the working radius is 3m, the car turns 0 degree |
| 5                 | Under the maximum load of 31.25t(overload), the working radius is 3m, and the car turns 45 degrees |
| 6                 | Under the maximum load of 25t+3%side load, the working radius is 3m, and the car turns 0 degree |

2.3. Force analysis of the chassis structure
The load on the crawler crane can be simplified as the vertical force F and the tipping moment M acting on the slewing center of the base\cite{7}, as shown in Figure 3. The load is transmitted between the base and the turntable mainly through slewing bearings and high-strength bolts.
The schematic diagram of the load on the turn table of the crawler crane is shown in Figure 4. P is the boom load, Q is the rear counterweight, and Th is the wire rope pulling force. The corresponding loads at the lifting point, rear counterweight and main hoisting place under the six working conditions (as shown Table 1 are calculated). Since the boom stress is not considered, no distinction is made between the boom head lifting point and the guide pulley point.

The calculation results are shown in Table 2, where x, y and z is the horizontal, vertical and normal direction. When analyzing the condition of turning 45° (working conditions 2 and 5), the load size and direction of the super structure are unchanged, and the turning and getting off is directly considered.

Table 2 Loads at various points under different working conditions when the car is turned at 0°

| Load/t | Lifting Point | Rear counterweight | Main hoist |
|--------|---------------|--------------------|------------|
|        | $F_x$ | $F_y$ | $F_z$ | $F_x$ | $F_y$ | $F_z$ |
| Working condition | 1 | -2.12 | -32.37 | — | -10 | 2.12 | 2.99 |
| | 2 | -2.12 | -32.37 | — | -10 | 2.12 | 2.99 |
| | 3 | -1.69 | 25.89 | 0.71 | -10 | 1.69 | 2.39 |
| | 4 | -1.57 | -34.83 | — | -10 | 1.57 | 3.58 |
| | 5 | -1.57 | -34.83 | — | -10 | 1.57 | 3.58 |
| | 6 | -1.26 | -27.86 | 0.75 | -10 | 1.26 | 2.86 |

2.4. Establishment of finite element model
In order to facilitate mesh division and stress calculation, the unnecessary welding parts in the 3D model, the chambers, and small holes and other parts are eliminated, the filling of the weld groove is completed, and the finite element solid model of the welded body is formed. Solid185 unit cell between 8-15mm is used in crawler frame welded body. Different loads of each working condition are applied to the mass 21 respectively, connected to the computational model by coupling constrains. The working condition 1 of the vehicle finite element models is shown Figure 5.

2.5. Finite element calculation and analysis
In order to facilitate calculation and stress analysis, finite element analysis needs to separate the base, crawler frames, and perform position constraint and force loading separately. After analysis, the stress cloud diagram of the vehicle bases and crawler frame under 6 working conditions can be achieved.

In condition 1, the maximum stress of the base is 284MPa, and the position of it is at the edge of the adjustable cover plate (Figure 6 and 8) and the central bore of the cover (Figure 9). The maximum stress of the track frame is 314MPa, and the stress is at the corner of the bottom plate hole (Figure 7).

In condition 2, the maximum stress of the base is 325MPa, and the position of it is at the edge of the adjustable cover plate and the upper cover plate and the standing tube weld. Track frame of maximum stress is 305MPa, and the maximum stress is in the base plate aperture corner position.
In condition 3, the base of the maximum stress is 230MPa, and the position of it is at the edge of the cover plate at reducers, the upper covers plate and the standing tube weld. Track frame of maximum stress is 305MPa, and the maximum stress position is at the bottom corners of the hole.

In condition 4, the base of the maximum stress is 200MPa, and the position of it is at the edge of the cover plate at reducers (Figure 10). Track frame of maximum stress is 305MPa, and the maximum stress position is at the bottom corners of the hole (Figure 11).
In condition 5, the base of the maximum stress is 225MPa, and the position of the maximum stress is at the edge of the cover plate at reducers. The maximum stress of the track frame is 224MPa, and the maximum stress position is at the bottom corners of the hole.

In condition 6, the base of the maximum stress is 163MPa, and the position of the maximum stress is at the edge of the adjustable cover plate. The maximum stress of the track frame is 168MPa, and the maximum stress position is at the bottom corners of the hole.

From the stress analysis of the above 6 working conditions, the large stress area of the base is mainly concentrated on the edge of the upper cover plate and the central hole of the lower cover plate. The main reason is that this is the overturning point of the base. The large stress area of the crawler frame is mainly concentrated on the bottom of the crawler frame, especially the corners of the holes where the load-bearing wheels are assembled. The main reason is that the load above the base is transmitted to the bottom of the crawler frame of the vertical plate. The vertical plate and the load-bearing wheels on both sides form a small local simply supported beam structure. The stress cloud diagram results also show that most of the structural stress in the original design of the base and the crawler frame is far lower than the allowable stress of the material, with a large strength margin, and the design is conservative. On the premise of ensuring the rigidity of the base and the crawler frame, appropriately reducing part of the structure can give play to the performance of the steel, thereby reducing the weight of the vehicle and the entire crane, and saving manufacturing costs.

The main materials of the lower structure are Q345 and Q460. According to the finite element calculation results, the stress of each component meets the material strength requirements.

2.6. The strength check of the rotary bolt of lower structure

The number of bolts connecting the turntable and the base is 80. When the vertical load and the tipping moment act on the turntable, parts from the bolts on tension are subject to axial tension, and some of the bolts on compression are subject to the pressing force of the contact surface between the turntable and the base[8]. In the finite element analysis, the force of 80 uniformly distributed bolts is converted into vertical tension and pressure, and the force of the pressure part is converted to the force of the bolt connection point. The above simplified force model is used to analyse 6 conditions. Turning the bolt on a maximum torque rotation condition of 45° (condition 2), the maximum axial force of the bolt is 88.15kN. The stress cloud is shown in Figure 12. 10.9S grade M24 bolts are used in this design, and the calculation is as follows: The allowable bearing capacity in the axial direction is:

\[
P \leq \frac{0.2\sigma_s d}{1000\mu}
\]

(1)
Where yield point: \( \sigma_s = 900 \text{MPa} \)

Effective cross-sectional area of bolts: \( A_t = 353 \text{mm}^2 \)

Strength safety factor: \( n = 1.48 \)

Load distribution factor: \( \beta = 0.2583 \)

Obtaining the axial direction can be substituted for the permissible carrying capacity. Bolt’s maximum pulling force is 88.15kN, which is less than the allowable value.

3. Summary

The finite element model for the chassis structure of the crawler crane is established in this paper. The stress distribution of the base and the crawler frames under six specific conditions is analysed, and the strength checking to provide a basis for the strength design of the chassis structure is conducted. The stress analysis results also show that the stress of most of the chassis structure is much lower than the allowable stress of the material. While ensuring their rigidity, some structures and materials can be appropriately reduced, and lightweight design can be carried out to save energy, protect environment, and reduce costs. The strategy presents important values in improving market competitiveness.

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