Dynamic load-bearing capacity analysis for the main Transmission Mechanism of Traction Machine in a Cargo Lift

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Abstract. This paper discusses the dynamic load-bearing capacity of the main transmission mechanism of worm pair traction machine in a cargo lift. By fully considering the mass, rotational inertia and friction for transmission of the worm pair, multi-body dynamics was adopted to calculate the worm gear teeth contact force and motion characteristics. Based on the nonlinear finite element algorithm for friction-contact and combined with the dynamics computation results, the finite element calculation for the teeth contact strength and bending strength were performed. The finite element calculation results and theoretical calculation results were compared to analyze their difference. The mechanism quality was evaluated from the running stability and mechanical strength, and the evaluation results show that the main transmission mechanism of traction machine in the cargo lift has good dynamic load-bearing capacity and high safety, and meets the design requirements.

Key words. Cargo lift; Worm pair traction machine; Main transmission mechanism; Multi-body dynamics; Nonlinear finite element.

1. Introduction

Cargo lift is a kind of heavy-duty mechanical device that cannot be replaced by manpower for modern industries to work efficiently. It has a steel wire rope type and a hydraulic type depending on its power driving methods. The steel wire rope type is widely used in the market, with a traction machine as its driving host. The traction machine is composed of a motor, a reducer, a brake and a traction wheel, and is available for a toothed type and a non-toothed type depending on the type of the reducer. The toothed type traction machine produced in China is mostly a worm pair type[1].

As the driving host of the cargo lift, the traction machine is different from other transmission actuators. Its performance and quality, especially the load-bearing capacity and running stability, are directly related to the safety of the cargo lift[2]. Therefore, the requirement regarding load-bearing capacity and running stability is the key point in the design process. Research on the main transmission mechanism of the traction machine at home and abroad is mostly concentrated on its vibration characteristics and component failure analysis, and there are few studies on safety based on its dynamic load-bearing capacity. This paper takes the worm pair traction machine as the research object and provides a preliminary quality inspection method based on its dynamic load-bearing capacity in the absence of experiments to achieve the purpose of lowering design costs while ensuring load-bearing safety.
2. The working principle and main parameters of the worm pair traction machine in cargo lift

Set the related technical parameters of traction machine according to the actual work requirements.

2.1. Working principle of the main transmission mechanism of traction machine

The driving relationship of the worm pair traction machine is shown in figure 1. Traction machine movement mainly by the motor to provide the original force, through the reducer to achieve the rated speed, by the brake and other equipment to achieve the start and stop functions of reliable operation. The traction wire rope is connected to the lift platform through one end of the traction wheel, and the other end is connected to the counterweight device. The gravity of the lift platform and the counterweight device causes the traction wire rope to be compressed in the rope groove of traction wheel. When the motor rotates, due to the friction between the rope groove of traction wheel and the traction wire rope, the traction wire rope drives the lift platform to move relative to the counterweight, and the lift platform runs up and down the rail in the hoist-way[3].

Figure 1. The main transmission mechanism of traction machine in cargo lift.

2.2. Main parameters

The research object of this paper is the worm pair traction machine in cargo lift[4-5].

(1) The weight of the lift platform and the traction wire rope \( G(\text{kg}) : 500 \);  
(2) Rated load \( Q(\text{kg}) : 1000 \);  
(3) The rated speed of the lift platform \( v(\text{m/s}) : 1 \);  
(4) Height of lift \( H(\text{mm}) : 10200 \);  
(5) Counterweight weight \( W(\text{kg}) : W = G + KQ \); \( K \) is the traction machine balance coefficient, and the balance factor is 0.5; Therefore, \( W = 1000 \);  
(6) Total efficiency of traction machine \( \eta \) : For the worm gear traction machine, the total efficiency is generally 0.5 to 0.55, so the total efficiency sets 0.55;  
(7) Worm axial tooth Angle \( \alpha(\text{deg}) : 20 \);  
(8) Surface heat dissipation coefficient \( a_w : 18 \);  
(9) Operating environment temperature of the traction machine \( t_o(\text{°C}) : 20 \);  
(10) Number of traction wire ropes: 5.

3. Dynamics simulation analysis

Establish a 3D model of the main transmission mechanism of traction machine, then build a multi-body dynamics model with the help of a virtual prototype and perform dynamics calculations[6-9]. The dynamics model is shown in Figure 2.
Add the speed of 970 r/min to the drive, and add the maximum load torque of 2010.5 N·m to the load side.

3.1. Analysis for worm gear teeth contact force
Select two extreme operating conditions which are the maximum load rise and the maximum load drop to compare. Iterative calculation of 1000 steps, through the result post-processing to obtain the worm gear teeth contact force in the case of maximum load in the rise and the maximum load drop, as shown in Figure 3 and Figure 4.

![Figure 3. Worm gear teeth contact force in the case of maximum load in the drop.](image)

![Figure 4. Worm gear teeth contact force in the case of maximum load in the rise.](image)

It can be seen from the Figure 3 and Figure 4 that when the cargo lift in the case of maximum load in the drop, its maximum contact force is 3692.9 N. While the cargo lift in the case of maximum load in the rise, its maximum contact force is 11543.7 N. It can be assumed that, the power required for the rise is about 3 times the amount of power required for the drop.
3.2. Analysis for worm gear angular velocity

The worm input angular velocity is 970 r/min, which is 5820 deg/s. The theoretical worm output angular velocity is 190.8 deg/s. The simulation results are as Figure 5:

![Figure 5. Output angular velocity of worm gear.](image)

As shown in Figure 5, the maximum angular velocity of worm gear is 213.585 deg/s and the average angular velocity is 190.531 deg/s. The simulation data is in line with the theoretical value and meets the design requirements.

3.3. Analysis for traction wheel kinematic capacity

The design speed of the cargo lift is 1 m/s. Add a point on the traction wheel, whose coordinate is (295,250,727). That is to add a point on the bus line of the traction rope groove, and measure the speed of the traction machine. The linear velocity of the traction wheel is shown in the Figure 6:

![Figure 6. Output linear velocity of the traction wheel.](image)

The output linear velocity of the traction wheel is the running speed of the cargo lift. According to Figure 6, the maximum linear velocity of the simulation output is 1.10 m/s. The average linear velocity is 0.98 m/s. The simulation data is in line with the theoretical result and meets the design requirements.

In order to check the running stability of the cargo lift, the angular acceleration of the traction wheel connected with the lifting platform is calculated. The result is shown in Figure 7.
According to Figure 7, it can be found that the maximum angular acceleration of the cargo lift during the operation is 5.799E-6 deg/s², which occurs after the initial start of the Goods lifting device. In the whole operation process, the angular velocity fluctuation magnitude is small, the operation is relatively stable, the impact vibration is small, conforms to the design requirement.

4. Finite element analysis of worm pair

Based on the nonlinear finite element algorithm for friction-contact and combined with the dynamics computation results, the finite element calculation of the teeth contact strength and bending strength were calculated[10-15]. The results of finite element calculation are compared with those of theoretical calculation.

4.1. Check the teeth contact strength

The original theoretical formula of contact strength checking is derived from Hertz formula. The formula for contact stress $\sigma_{H}$ (MPa) is:

$$\sigma_{H} = \sqrt{\frac{KF_a}{L_o \rho_z}} Z_e$$  \hspace{1cm} (1)

Where:
- $F_a$, normal load on meshing tooth surface, and the unit is N;
- $L_o$, total length of contact line, and the unit is mm;
- $K$, load coefficient, $K = K_A K_B K_v$, where $K_A$ is the service coefficient and 1.2 is taken here; $K_B$ is the load distribution coefficient of the tooth, taking 1.3 at this time; $K_v$ is the dynamic load coefficient, taking 1.0 here;
- $Z_e$, elastic influence coefficient of the material.

After the above formula is deformed, the checking formula of the teeth contact strength can be obtained:

$$\sigma_{H} = 480 \sqrt{\frac{KT_z}{d \cdot m^2 \cdot z^2}} \leq [\sigma_{H}]$$  \hspace{1cm} (2)

Where:
- $\sigma_{H}$, the teeth contact stress of worm pair, and the unit is MPa;
- $[\sigma_{H}]$, the allowable contact stress, and the material of worm gear is tin bronze. Therefore, the main failure of worm gear is contact fatigue. In this case, $[\sigma_{H}] = 268$ MPa.

Through calculation, the theoretical result of the teeth contact stress is 236.56 MPa. Select transient dynamics module in Workbench for contact strength analysis. Select the tetrahedral meshing, and the basic partition size is set to 0.005 m. The final number of
elements is 103066. The constraint boundary of the worm pair is set as a fixed hinge, and the worm rotation torque obtained from the dynamics calculation is used as the driving load of the worm. A friction-contact model is established between the worm and worm gear teeth, and the friction factor is set as 0.15. The mating surfaces of worm teeth are used as the contact surfaces, and the worm gear teeth are used as the target surfaces. Finite element calculation model of worm pair shown in Figure 8.

![Finite element calculation model of worm pair.](image)

After calculation, the location where the maximum contact stress of the worm pair contact surfaces is located is shown in Figure 9:

![The location where the maximum contact stress of the worm pair contact surfaces.](image)

As shown in Figure 9, the maximum contact stress on the worm pair contact surfaces is 250.74 MPa, which is similar to the theoretical calculation and is within the allowable stress.

4.2. Check the bending strength of worm gear root

Because the shape of worm gear tooth is complex, it is difficult to calculate the bending stress of the tooth root accurately. Generally, worm gear is considered as helical cylindrical gear, and the following formula can be obtained:

\[
\sigma_F = \frac{KF_2}{b_2m_n} \gamma_{fa2} \gamma_{sa2} Y Y_{\beta}
\]

\[
= \frac{2KT_2}{b_2d_2m_n} \gamma_{fa2} \gamma_{sa2} \gamma_{\epsilon} Y_{\beta}
\]

(3)

where: \( \tilde{b}_2 \), the worm gear teeth arc length, \( \tilde{b}_2 = \frac{\pi d_2 \theta}{360 \cos \gamma} \). \( \theta \) is the worm gear tooth width angle which can be calculated at 100 °;

\( m_n \), normal modulus, \( m_n = m \cos \gamma \), and the unit is mm;

\( \gamma_{sa2} \), root stress correction coefficient;

\( Y_{\epsilon} \), the coincidence coefficient of bending fatigue strength. In this design, select \( Y_{\epsilon} \) as 0.667;
$Y_\beta$, helix angle influence coefficient, $Y_\beta = 1 - \frac{\gamma}{140^\circ}$.

After the above formula is deformed, the checking formula of the bending strength can be obtained:

$$\sigma_f = \frac{1.53KT}{d_1d_2m} Y_{\beta u2} Y_\beta \leq [\sigma_f] (4)$$

Where: $\sigma_f$, the bending stress of worm gear root; $Y_{\beta u2}$, worm gear profile coefficient. After checking the table, $Y_{\beta u2}$ is 2.279. $[\sigma_f]$, the allowable bending stress. After checking the table, $[\sigma_f]$ is 64 MPa.

By calculation, the bending stress of worm gear root was theoretically calculated to be 54.96 MPa.

The finite element calculation result of the location of the maximum bending stress of the worm gear root during the load-bearing motion are shown in Figure 10.

**Figure 10.** The location where the maximum bending stress of the worm gear root.

As shown in Figure 10, the finite element results are shown that the maximum bending stress is 59.81 MPa, which is within the allowable bending stress range.

### 4.3. Comparison of results

During the load-bearing process, the comparison between the finite element calculation results and the theoretical calculation results of the contact stress and bending stress of worm pair is shown in Table 1:

**Table 1.** Comparison of results.

|                        | Theoretical calculation (MPa) | Finite element calculation (MPa) | Allowable Stress (MPa) | Comparison (%) |
|------------------------|-------------------------------|---------------------------------|------------------------|----------------|
| The contact stress of worm pair | 236.56                        | 250.74                          | 268                    | 6              |
| The bending stress of worm gear root | 54.96                        | 59.81                           | 64                     | 8.8            |

Due to the consideration of the weight, moment of inertia and friction of the component, the actual working conditions are fully simulated. As shown in Table 1, whether it is the contact stress or bending stress, the finite element calculation results are larger than the theoretical calculation results. Among them, the finite element calculation result of the contact stress is 6% larger than that of theoretical calculation, and the finite element calculation result of the bending stress is 8.8% larger than that of theoretical calculation. This shows that the theoretical calculation has limitations for irregular forces in complex working conditions, and the finite element algorithm can reduce such limitations to a certain extent, which is closer to the results of actual working conditions. But no matter which algorithm, you can see that the results are within the allowable range, and it can be considered that the mechanical strength meets the design requirements.
5. The conclusion
(1) The dynamics calculation results of the main transmission mechanism of traction machine in a cargo lift show that the mechanism movement process is stable and there is no interference between the components. Its dynamics model and calculation results are reasonable, and the kinematic capacity can meet the work requirements.

(2) Combined with the calculation results of multi-body dynamics, the strength calculation of the worm pair of the main transmission mechanism of traction machine in a cargo lift was carried out based on the friction-contact nonlinear finite element method. By comparing the theoretical calculation results and the finite element calculation results, it is found that the finite element calculation results of contact stress and bending stress are larger than their theoretical calculation results. The errors are 6% and 8.8% respectively, and both meet the requirements of allowable strength. Its dynamic load-bearing capacity meets the design requirements.

(3) By analyzing the dynamic load-bearing capacity of the main transmission mechanism of traction machine in a cargo lift, this paper provides a method for preliminary safety prediction of the designed main transmission mechanism of traction machine in the absence of experiments, which can save the design costs to a certain extent.

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