Strength Analysis of Primary Pump Motor in Primary Reactor

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Abstract. The primary pump motor, as the driving source of nuclear reactor cooling loop, is the main device to ensure the safe and stable operation of nuclear power plant. Due to the difference in rotary inertia between the rotor and the rotating shaft, radial displacement, pressure and friction torque caused by centrifugal force during high-speed operation are different. The minimum fit tolerance, fit pressure safety factor and torsion strength safety factor are calculated by the strength check theory. Then, the finite element analysis of rotor stress of a 4500kW main pump motor is carried out and stress concentration is determined for rotor root and axial ventilation hole. Finally, the finite element analysis of stress field is carried out for end ring, copper bar, guard ring and pressure ring. It is found that there is obvious stress concentration at the top hole of the guard ring and the air duct of the pressure ring, and the bending stress of the copper bar and end ring is relatively high. Combined with the results of finite element analysis, the rotor end is improved, which can effectively improve the reliability of the end, increase the ventilation area of the end and save material cost.

1. Introduction

With the decrease of natural resources and the rapid economic development, the demand for nuclear power is more and more urgent. The driving motor is the main driving device for the circulation of primary coolant for high-temperature gas-cooled reactor reaction, the largest active component in nuclear power plant, and the main component to guarantee the circulation of coolant in nuclear power plant[1]. Therefore, the safety performance of its strength needs more consideration. These parts must be designed reasonably in addition to using high-strength materials[3].

2. Strength analysis of rotor

2.1 Calculation of matching tolerance

The fitting tolerance of rotor and shaft should be the deformation difference of radial displacement when under the condition of temperature rise. The force causing the radial displacement of the rotor consists of three parts: the centrifugal force of the yoke, the centrifugal force of the tooth and the centrifugal force of the copper bar. The rotor diagram is shown in figure 1.
Based on the strength check theory and the mechanical design principle, the formula for calculating the centrifugal force of copper bar and tooth can be deduced. Through the centrifugal force of the tooth and copper strip, the tangential normal stress generated in the inner circle of the punching plate can be calculated to obtain the tangential stress generated by the centrifugal force of the tooth and copper strip at the vent section.

When calculating the radial deformation of the rotor, the elastic modulus $E_p$ needs to be considered. Therefore, Centrifugal force such as tooth and copper bar will cause radial displacement $D_{tp}$ in the punching piece. Radial displacement (radius elongation) $D_m$ caused by centrifugal force at yoke. When the motor is running, the temperature will rise and the rotor will expand and deform. It is necessary to consider the radius displacement $D_{rt}$ caused by temperature rise in the inner circle of the core.

The radial displacement $S_m$ of the shaft can also be generated under the consideration of its own centrifugal force.

Therefore, it can be found that the core is affected by three radial displacements, which are inner circular radial displacements caused by the centrifugal forces at the tooth, copper bar and yoke. The radial displacement of the shaft also occurs under the action of centrifugal force. The deformation difference between the core and the shaft is calculated as follows:

$$\Delta R = D_{tp} + D_m + D_{rt} - S_m$$

Taking a 4500kW main pump motor as an example, the deformation difference between the rotor and rotor shaft during rated operation and overspeed operation can be obtained by using the above formula, as shown in table 1.

| $D_{tp}$,mm | $D_m$,mm | $D_{rt}$,mm | $S_m$,mm | $\Delta R$,mm |
|------|------|------|------|------|
| 0.1026 | 0.0643 | 0.0403 | 0.0053 | 0.2019 |
| 0.1131 | 0.0709 | 0.0403 | 0.0058 | 0.2185 |

According to table 1, the deformation difference between the rotor and the shaft is 0.2185mm when the motor is overspeed. The minimum matching tolerance should be slightly larger than this value, which is 0.22mm.

2.2 Compatible pressure calculation
The rotor and shaft adopt interference fit. When the motor is in different running states, the matching pressure is different. Under the premise of meeting the minimum matching tolerance, the motor meets the requirement of mechanical safety coefficient.

Combined with the main pump motor example, the safety coefficients of the rotor in three different situations are calculated respectively, as shown in table 2:
Table 2. Rotor safety factor for different working conditions

|                  | Static  | Rated speed | Over speed |
|------------------|---------|-------------|------------|
| resulting stress, Mpa | 116.8147 | 190.2094 | 218.4724 |
| Safety factor     | 3.5954  | 2.2081     | 1.9224     |

It can be seen from table 2 that high rotating speed produces high synthetic stress, but the synthetic stress of the example motor under the above three conditions does not exceed the strength limit, and the safety coefficient is within the reasonable range.

2.3 Check of the torsion strength

When the motor is loaded, it is equivalent to applying a damping moment at the extension end of the shaft. The damping moment overcomes the rotating moment of the motor, which will produce torsion on the motor rotor. The torsion moment must be smaller than the friction moment between the stamper and the shaft to guarantee the safe operation of the motor.

The safety coefficient of the motor is 4, which meets the strength requirement.

2.4 Rotor stress simulation

A 4500kW example motor is simulated by finite element method. In the modeling process, the plates, copper bars and axes all adopt plane strain elements. Taking the symmetry of the structure into consideration, 1/20 of the whole structure is taken to constrain the tangential displacement of the nodes at both ends of the section. The Frictional contact between the rotor and the shaft adopts frictional contact to simulate the hot sleeve, and the frictional contact between the copper bar and the washer also adopts frictional contact to transfer the centrifugal force of the copper bar. The temperature rise between the core and the shaft shall be 20, the minimum single side coordination tolerance shall be 0.22mm, and the maximum single side coordination tolerance shall be 0.25mm.

Combined with the finite element calculation model and the actual force, the three operating states of the main pump driving motor under the maximum matching tolerance were simulated and calculated, respectively, the static state, rated speed (4000r/min) and over speed (4200r/min). The stress distribution diagram is shown in Figure 2.

![Stress distribution of the plates](image_url)

Figure 2. Stress distribution of the plates

It can be seen from the calculation results that the round hole and the tooth root of the rotor belong to the high stress area, and the calculation results of the three operating states are shown in table 3.

Table 3. Maximum stress values under different operating conditions

| Point of maximum stress | Tooth root (Mpa) | Round hole (Mpa) |
|-------------------------|------------------|-----------------|
| At rest                 | 267              | 424             |
| Rated speed: 4000r/min  | 335              | 510             |
| Over speed: 4200r/min   | 343              | 519             |

It can be seen from table 3 that the maximum stress on the circular hole site increases with the increase of rotating speed. For the main pump motor, the stress on the circular hole of the rotor's axial ventilation duct is greatest. This is due to the motor rated at runtime is in under the condition of high speed, the centrifugal force of the core is greater than the centrifugal force of rotor, the inner diameter of the core expansion deformation is bigger than expansion deformation of shaft diameter, and in order to ensure the core with the rotating shaft under the high speed remains interference fit, do not produce
out, must ensure that the minimum is slightly greater than the core fit tolerance and the axis of the distortion of the poor. However, increasing the minimum matching tolerance will inevitably lead to the pressure between the rotor and the rotating shaft, especially when there are axial ventilation holes, which will produce high stress area locally. In view of this problem, it is suggested to use better silicon steel sheet, such as 50WW270, to improve the strength limit of the stamping. Remove axial vents to ensure strength requirements, if cooling conditions permit.

3. End strength analysis

3.1 End stress simulation

The strength of the retainer ring and end ring of the main pump motor, including the retainer ring, end ring, copper bar, silicon steel sheet, compression ring and shaft, were modeled and analyzed by finite element method. In the process of building the model, all components adopt entity units and 1/20 of the overall structure is taken for analysis, and the tangential displacement on both sides is restrained. Retaining rings, silicon steel sheet, the contact area between pressure and axial frictional contact are adopted to simulate the thermal set of coordination, retaining rings and the contact area between ring clearance fit, the frictional contact simulation copper and silicon steel sheet interface frictional contact used to deliver the centrifugal force of a copper wire, copper bar and side ring, ring and clamping ring end face of the binding contact simulation welding connection.

Combined with the finite element calculation model and the actual force, finite element analysis is carried out on the operation of the motor at the maximum matching tolerance (4200r/min), and the stress distribution diagram of the guard ring, compression ring, end ring and copper bar is obtained respectively, as shown in Figure 3. The maximum stress value of each component is shown in table 4.

![Stress distribution](image)

Figure 3. Stress distribution

| Operating state | Protective ring (Mpa) | End ring (Mpa) | Copper bar (Mpa) | Compression ring (Mpa) |
|-----------------|----------------------|---------------|-----------------|-----------------------|
| 4200r/min       | 518                  | 151           | 296             | 475                   |

According to the stress distribution diagram of all parts at the end, the maximum stress of the guard ring occurs at the circular hole at the upper end, which is 518Mpa, and the maximum stress of the pressure ring appears at the ventilation duct, which is 475Mpa. All are within the allowed range of materials.
3.2 End Improvement

In the high speed rotating process, the end ring generates radial deformation under the action of centrifugal force. Due to the gap between the end ring and the guard ring, the deformation is not inhibited. The end ring will drive the end copper strip together to produce deformation along the outward direction of radius. In addition, due to the relatively small deformation of the copper bar in the groove restricted by the notch, the bending deformation of the copper bar at the pressure ring is generated, and the stress concentration area is generated locally, causing damage to the motor end.

To reduce the bending stress of the end ring and copper strip, the radial deformation of the end ring and copper strip should be suppressed. The contact surface of the guard ring and the end ring should be designed as a hot sleeve structure to fully restrain the radial deformation of the end ring and copper strip caused by centrifugal force. This structure can eliminate the welding connection between the guard ring and the pressure ring, increase the reliability, remove the ventilation hole on the guard ring and reduce the stress concentration area. At the same time, there is no need to use hot sleeve coordination between the guard ring and the shaft, and the part of the guard ring longer than the end ring can be completely subtracted to increase the ventilation area of the end, saving material cost. As shown figure 4.

![Figure 4. Improved structure of the end](image)

4. Conclusion

In view of the status of primary pump motor of nuclear reactor in nuclear power plant, the stress distribution of its rotor and end is analyzed according to the strength check theory. Taking a 4500kW motor as an example, finite element analysis is carried out to obtain the stress distribution of each component.

1. The working difference calculation of the rotor and the rotating shaft, the working pressure calculation and the torsion strength check are completed, and the minimum fitting tolerance and safety coefficient of the rotor and the rotating shaft are determined.

2. According to the results of strength check and finite element analysis, the stress distribution diagram of the rotor impeller is obtained, and the maximum stress concentration is found in the axial vents.

3. The stress distribution diagram of the guard ring, end ring, copper bar and the pressure ring is obtained by finite element analysis of the rotor end, and the maximum stress concentration is found at the guard ring and the pressure ring. Based on this, the end structure is designed to be improved.

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