Research on the axial force and water film thickness of water-lubricated thrust bearing for mine high speed rescue pump

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Abstracts-Due to the limitation of mine well size and the requirement of rapid rescue for flooding, a mine high speed rescue pump with small volume and large head is developed and designed. According to the hydraulic characteristics and structure of rescue pump, the axial force under 6 different speeds and 4 different flow conditions are studied. The water lubricated thrust bearing is designed. The Reynolds equation is solved by finite difference method to solve the minimum thickness of water film under the above conditions. The results show that: the axial force increases with the rise of rotating speed; the smaller the axial force, the larger the water film thickness, the greater the change amplitude of water film thickness. The research work in this paper can provide guidance for the follow-up research of high speed and big power mine rescue pump.

1. Introduction

Mine high speed rescue pump uses frequency conversion technology to realize the high speed. After the speed increasing, the goal of high head, big flow, small volume and light weight could be achieved. At the same time, the frequency conversion speed regulation technology can adapt to the needs of large changes of water level in the mine[1-2].

Many scholars have studied the performance of water-lubricated bearings theoretically. LITWIN[3-4] studied the influence of surface roughness topography and design parameters on performances of water-lubricated polymer bearings, obtained that the transversal roughness along the sliding direction could enhance hydrodynamic capacity. CABRERA[5] used computational fluid dynamics(CFD) to analyze the behaviour of the bearings found that the relatively low film pressures caused significant rubber deflections were not high enough to produce lubricant viscosity changes. LIU[6] acquired that the rotational speed has no effect on the minimum nominal film thickness and eccentricity. LIANG[7] applied the fluid-solid two-way direct coupling to study the relationship between the axial displacement and the step parameters. OUYANG[8] proposed a performance evaluation model of water lubricated stern bearings based on entropy weight fuzzy comprehensive evaluation method. QIN[9]
measured the friction torque of water lubricated rubber bearings to study the relations between the friction features and the performance.

From the above researches, it can be seen that there are few studies on high speed and heavy load conditions. So, thrust bearing of mine high speed rescue pump is studied in the paper to improve the research field.

2. Theoretical Model

2.1 Generalized Reynolds Equation

Reynolds equation is the key for hydrodynamic lubrication. Under the relevant lubrication boundary conditions, the derivation of Reynolds equation is base on the continuity equation and momentum equation of viscous fluid mechanics[10].

Based on some assumptions such as Newtonian fluid, laminar flow, no slippage, combined with the continuity equation, the generalized Reynolds equation can be derived as follows:

\[
\frac{\partial}{\partial x} \left( \rho \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial y} \left( \rho \frac{\partial p}{\partial y} \right) = 6 \frac{\partial}{\partial x} \left( (U_1 + U_2) \rho h \right) + 6 \frac{\partial}{\partial y} \left( (V_1 + V_2) \rho h \right) + 12 \frac{\partial}{\partial t} (\rho h)
\]

\[\text{(1)}\]

where, \( \rho \) is the density of the liquid, \( P \) is the pressure, \( \mu \) is the viscosity of the liquid, \( h \) is the thickness of the liquid film, \( U_1, U_2 \) are the velocity of the liquid film surface along the X direction, \( V_1, V_2 \) are the velocity of the liquid film surface along the Y direction, \( t \) is the time.

After coordinate transformation and dimensionless processing

\[
\Lambda \left( \frac{1}{H^3} \frac{\partial^2 P}{\partial \Theta^2} + \frac{H^3}{4 \pi^2 R^2} \frac{\partial^2 P}{\partial \Theta^2} + \frac{3 H^2}{4 \pi^2 R^2} \frac{\partial P}{\partial \Theta} \frac{\partial H}{\partial \Theta} + \frac{3 H^2}{4 \pi^2 R^2} \frac{\partial^2 H}{\partial \Theta^2} + \frac{H^3}{R} \right) \frac{\partial P}{\partial R} = \Lambda \frac{1}{2 \pi} \frac{\partial H}{\partial \Theta}
\]

\[\text{(2)}\]

\( \Lambda \) is the bearing number and could be shown in:

\[
\Lambda = \frac{6 \mu \omega R_0^2}{h_c^2 p_0}
\]

Dimensionless treatment method can be expressed:

\( H = h/h_c; \ \Theta = \theta/2\pi; \ R=r/R_0; \ P=p/p_0. \)

2.2 Water Film Thickness Equation

\[
h = h_c + \delta_\theta \cdot R \sin(\theta - \theta_0) + \delta_\delta \cdot [\cos(\theta - \theta_0) - R_0] + D_b
\]

\[\text{(4)}\]

\[
h = ar^3 + br^2 + cr + d\theta^2 + e\theta^2 + f\theta + h_c
\]

\[\text{(5)}\]

Where, \( h_c \) is the standard water film thickness, \( R_0 \) is the radius of graphite bearing bush, \( \delta_\theta \) and \( \delta_\delta \) are the radial and axial inclination angles of bearing pad, \( \theta_0 \) is the initial angle of water film, and \( D_b \) is the elastic deformation of bearing bush. Formula (4) is dimensionless to obtain formula (5).

3. Theoretical Model

In the paper, the one-dimensional form of Reynolds equation is expanded and solved by numerical method. When the liquid film pressure distribution of thrust bearing is solved by the difference method, the film corresponding to a single thrust pad should be discretized by grid. The pressure value on each node of the grid constitutes the difference value and quotient value of each order. Based on the pressure value of each node, the corresponding performance parameter value is numerically integrated.

The fan-shaped liquid film of thrust bearing can be expanded into two-dimensional grids. The grid node position is represented by coordinates and numbered according to the circumference and
diameter direction. The liquid film circumferential direction $\theta$ is the abscissa, the diameter direction $r$ is the ordinate. The circular direction is divided into $m$ lattice, the step size of each lattice is: $\Delta \theta = \theta / m$. The diameter direction is divided into $n$ lattice, the step size of each lattice is: $\Delta r = r / n$.

According to the boundary conditions, Overrelaxation iterative method is used to give initial value $P_{ij}^{(1)}$ to each side node, and then solve the equation in turn. After first operation, the approximate pressure values obtained replace the initial pressure values of each node in turn to obtain the secondary approximate pressure distribution values $P_{ij}^{(2)}$ which is more accurate than the primary pressure value. After many iterations, the pressure distribution $P_{ij}^{(k)}$ with enough accuracy is finally obtained.

The value of $k$ is determined according to the relative convergence criterion as shown in formula (6). In the calculation, the maximum allowable relative error of $\zeta$ is $10^{-4}$.

$$\sum_{i} \sum_{j} \left| P_{ij}^{(i)} - P_{ij}^{(i-1)} \right| \leq \zeta$$

(6)

4. Calculation of Axial Force

GFQ200-300 (rated flow $Q_d$ is 200m$^3$/h; rated head $H_d$ is 300m) mine high speed rescue pump is selected as the research object. The model can be seen in Fig.1.

$$T = T_1 + T_2 + T_3$$

(7)

Where, $T_1$ is cover plate force which is leaded by the unsymmetry of the front and back cover; $T_2$ is the dynamic reaction force which leaded by force exerted on the impeller because of the flow in the impeller channel; $T_3$ is the rotor gravity.

$$T_1 = T_1' - T_1''$$

(8)

$$T_1' = \pi \rho g (R_m^2 - R_1^2) [H_t - \frac{\omega_n^2}{8g} (R_2^2 - R_m^2)]$$

(9)

$$T_1'' = \frac{1}{F_n} \frac{\xi_m}{2g} \rho \pi (R_m^2 - R_1^2) \left\{ 2g \left[ H_p - \frac{1}{8g} (u_z^2 - u_o^2) \right] \left( \frac{\xi_m}{F_n} + \frac{\xi_o}{F_o} \right) \right\}$$

(10)

Where, $T_1'$ is the initial cover force, $T_1''$ is the balance force generated in the balance chamber with the balance hole, $R_0$ is the impeller inlet radius, $R_2$ is the impeller outlet radius, $R_m$ is the impeller hub radius, $H_t$ is the theoretical head, $H_p$ is the impeller potential head, and $\omega_n$ is the angular velocity; $\xi_0$ is
resistance coefficient of balance hole; $F_{m}$ is flow area of sealing clearance; $F_{B}$ is total area of balance hole, $u_{2}$ is circumferential velocity at impeller outlet, $u_{m}$ is circumferential velocity at balance hole center. $\zeta_{m}$ is the resistance coefficient of seal clearance, where $\zeta_{m}=1.5 + \lambda \frac{L}{2b}$; $\lambda$ is 0.04-0.06, $b$ is unidirectional clearance; $L$ is clearance length.

$$\xi_{B}$$ is the resistance coefficient of balance hole; $u_{m}$ is circumferential velocity at balance hole center. $\zeta_{m}$ is the resistance coefficient of seal clearance, where $\zeta_{m}=1.5 + \lambda \frac{L}{2b}$; $\lambda$ is 0.04-0.06, $b$ is unidirectional clearance; $L$ is clearance length.

| $R_s$ (mm) | $R_h$ (mm) | $R_m$ (mm) | $R_r$ (mm) | $b$ (mm) | $L$ (mm) | $R_B$ (mm) | $\xi_{B}$ | $q$ (m$^3$/h) |
|-----------|-----------|-----------|-----------|---------|--------|---------|--------|------------|
| 128       | 62.5      | 32.5      | 67.5      | 0.2     | 18     | 46      | 2      | 2.66       |

Tab.1 Geometric parameters of impeller

Geometric parameters of impeller can be seen in Tab.1: where, $R_s$ is sealing radius; $R_B$ is center distance of balance hole, $q$ is Leakage flow.

Fig.2 shows the axial force values of mine high speed rescue pump under different working conditions. Six speeds of 3000, 3600, 4200, 4800, 5400 and 6000 r/min and four flow rates of 0.8, 0.9, 1.0 and 1.1 $Q_d$ are selected for research. It can be seen that when the mine high speed rescue pump operating at different speeds, the resultant direction of axial force is different. Among them, at 3000 and 3600 r/min, the axial force direction of the unit is the same as the gravity direction with vertical downward. With the increase of rotating speed, the axial force decreases at the same corresponding flow rate; at the same speed, with the increase of flow rate, the axial force decreases. At four speeds of 4200, 4800, 5400 and 6000 r/min, the axial force direction of the unit is opposite to the gravity direction. At the same corresponding flow rate, the axial force increases with the increase of rotating speed. At the same speed, with the increase of flow rate, the axial force increases.

There are two thrust bearings fixed on both sides of the thrust disc. In this paper, the bearing near the pump inlet is the lower thrust bearing, and the bearing far away from the pump inlet is the upper thrust bearing. When the axial force direction is downward at 3000 and 3600 r/min, the pump unit is mainly supported by the lower thrust bearing. Under four working conditions of 4200, 4800, 5400 and 6000 r/min, the axial force direction is upward, the pump unit is mainly supported by the upper thrust bearing. It can be seen that the maximum axial force is 3417 N and the minimum axial force is 247 N. Considering the larger safety factor of 1.5, it is concluded that the thrust bearing should be designed at 5125 N to ensure the normal operation of the pump. The design of thrust bearing should not only meet the requirements of bearing axial force, but also satisfy the installation size inside the motor cavity. Considering the above factors, the geometric parameters of thrust bearing bush are as follow: internal diameter $D_{1b}$, 120 mm; external diameter $D_{2b}$, 225 mm; center angle $\theta_{b}$, 60°; incline angle $\beta_{b}$, 0.01°; thrust pads number $k_{b}$, 6.

5. Water film thickness

Fig.3 is minimum water film thickness of mine high speed rescue pump under different working conditions calculated by Matlab. The maximum water film thickness is 88um at 4200 r/min of 0.8$Q_d$, and the minimum water film thickness is 28um at 6000 r/min of 1.1$Q_d$. At 4200 r/min, compared with other rotating speeds, the difference of water film thickness with the change of flow rate is larger. Compared with the axial force, it can be inferred that the smaller the axial force is, the greater the water film thickness is, and the greater the change amplitude of the water film thickness is with the decrease of the axial force.

In the prototype experiment as shown in Fig.4, the water lubricated thrust bearing designed in the paper is used in the pump. The experiment result show that thrust bearing could meet the requirements of axial force. Fig.5 is the thrust bearing after experiment.
6. Conclusion

Taking high speed rescue pump GPQ200-300 as an example, the axial forces under six speeds and four flow conditions are calculated, and the Reynolds equation is solved by finite difference method. The maximum axial bearing capacity of thrust bearing is obtained by multiplying the value of axial force at various speeds and flow conditions by safety factor. The minimum water film thickness can be obtained by taking the bearing capacity into Matlab to solve Reynolds equation. The results show that the smaller the axial force, the larger the water film thickness, the greater the change amplitude of water film thickness. The research work in this paper can provide guidance for the follow-up research of high speed and big power mine rescue pump.

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