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Analysis of Properties of Thrust Bearing in Ship Propulsion System

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Abstract: Thrust bearing is a key component of the propulsion system of a ship. It transfers the propulsive forces from the propeller to the ship's hull, allowing the propeller to push the ship ahead. The performance of a thrust bearing pad is critical. When the thrust bearing becomes damaged, it can cause the ship to lose power and can also affect its operational safety. For this paper, the distribution of the pressure field of a thrust pad was calculated with numerical method, applying Reynolds equation. Thrust bearing properties for loads were analyzed, given variations in outlet thickness of the pad and variations between the load and the slope of the pad. It was noticed that the distribution of pressure was uneven. As a result, increases of both the outlet thickness and the slope coefficient of the pad were able to improve load bearing capability.

Keywords: thrust bearing; thrust pad; lubrication properties; ship propulsion shaft; geometrical factors.
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1 Introduction

In the ship building industry and marine transportation, the size of ships continues to become larger, necessitating similar increase in propulsive power. The thrust bearing of ship's propulsion shaft, a key part of ship power-train, operates in more and more rough conditions. Unstable running speed, heavy propulsion force, higher film pressure and big machinery deformation have all become problems. The various faults of the thrust bearing are produced: fore and aft vibration along transmission shaft, harsh noise, high lubricant temperature, thrust disc or tile destroy heavily. The faults are associated with the operating conditions of the thrust bearing. The lubrication condition and style are the key factor leading to faults of thrust bearing performance. Several researchers reported their investigation results. Performance of different pad geometries have been proposed by Fuller and Rowe. Lin (2000) takes pad surface roughness into account to research dynamic stiffness and damping characteristics of the thrust bearing. Xie (1992) carries on the analysis on the thermal effect on hydrodynamic lubrication performance with thrust bearing in water turbo-generator. Influence of an axial thrust bearing defects on the dynamic behavior of an elastic shaft is reported by Berger *et al.* (2000). Other relative researches were reported in reference(Glavatskih SB *et al*. 2002; Wang *et al*., 1999; Oshimoto S. and Kohno, 2001). In this paper, the distribution of lubrication pressure field is obtained by solving Reynolds equation. The relationships among the bearing load, the outlet film thickness and the slope of the thrust bearing pad are analyzed respectively. After analysis on the calculation results, some constructive conclusions are gotten.

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2 Model and equations

Fig.1 represents the hydrodynamic thrust bearing model of a ship propulsion shaft. It contains one thrust disc and several thrust pads. The thrust disc is fixed to the propulsion shaft and rotating with it. The thrust pads are mounted on the bearing seat case. When the shaft is running at a rotating speed ω, the hydrodynamic lubrication film is obtained and carries loads. The propulsion force is transferred from the thrust disc to the thrust pad through the lubrication pressure film field and lets the ship navigating. The film pressure of the thrust bearing is governed by the Reynolds equation and associated with the running parameters and geometric dimensions of the thrust bearing.

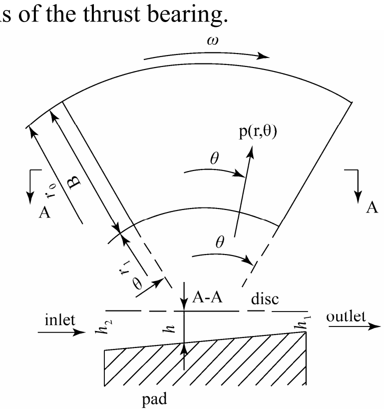

Fig.1 Thrust bearing scheme

The Reynolds lubrication equation(Wen SZ, 1990) is given in cylindrical coordinates as following,

$$\frac{\partial}{\partial r}\left(\frac{rh^3}{\mu}\frac{\partial p}{\partial r}\right)+\frac{\partial}{r\partial\theta}\left(\frac{h^3}{\mu}\frac{\partial p}{\partial\theta}\right)=6r\omega\frac{\partial h}{\partial\theta} \quad (1)$$

where, p is the pressure of lubrication oil film, h is the thickness of oil film, $h=h_1+kr\sin(\theta_1-\theta)$, h_1 is the minimum film thickness at the angle θ_1 at the outlet of the thrust bearing, k is the slope coefficient of the thrust bearing, μ is the oil dynamic viscosity, r is the radial coordinate of thrust bearing, value of r varies between outer radius r_0 and inner radius r_i, θ is the angular coordinate of a thrust bearing, the value of the angle θ changes between outlet angle θ_1 and inlet angle θ_2, ω is the rotating speed of the shaft.

If assumed the variables as, $B=r_0-r_i$ is breadth on the radial direction, r_0 and r_i are the outer and inner radius of thrust bearing respectively, $\bar{\mu}=\mu/\mu_0$, μ_0 is the virgin dynamic viscosity on the normal condition, $\bar{r}=(r-r_i)/B$, $\bar{r}_i=r_i/B$, $\delta=h_2-h_1$, h_1 and h_2 are the outlet and the inlet film thickness of the thrust bearing on the circumference direction, $h=\bar{h}\delta$, $p=\dfrac{\delta^2}{\mu_0 B^2\omega}\bar{p}$.

Then the Eq.(1) is expressed in a non-dimension form as following,

$$\frac{\partial}{\partial\bar{r}}\left[\frac{(\bar{r}+\bar{r}_i)\bar{h}^3}{\bar{\mu}}\frac{\partial\bar{p}}{\partial\bar{r}}\right]+\frac{\partial}{(\bar{r}+\bar{r}_i)\partial\theta}\left(\frac{\bar{h}^3}{\bar{\mu}}\frac{\partial\bar{p}}{\partial\theta}\right)=6(\bar{r}+\bar{r}_i)\frac{\partial\bar{h}}{\partial\theta} \quad (2)$$

In order to calculate the bearing load, the pressure field is obtained by numerical method to solve the Reynolds equation. First, the non-dimension lubrication area of the thrust pad is divided into $m\times n$ grids by linear space on the radius and circumference direction coordinates respectively. Then the pressure at every grid point is gotten by Gauss-Seidel numerical integral method with part differential Reynolds equation. The integral boundary condition of lubrication is,

$$\bar{p}\big|_{\bar{r}=\bar{r}_i}=\bar{p}\big|_{\bar{r}=\bar{r}_o}=0\,,\quad \bar{p}\big|_{\theta=\theta_{1i}}=\bar{p}\big|_{\theta=\theta_{2o}}=0$$

After the pressure field is worked out, the non-dimension load W of each thrust pad is integrated by numerical integral calculation of the pressure field in the lubrication area of the pad. The integration formula of load W of a pad is expressed as following,

$$W=\int_{\theta_2}^{\theta_1}\int_{\bar{r}_i}^{\bar{r}_o}\bar{p}\bar{r}\mathrm{d}\bar{r}\mathrm{d}\theta \quad (3)$$

3 Calculation and discussion

3.1 Data of thrust bearing

The thrust bearing of the ship propulsion shaft has seven pads. Its outer radius is r_o=0.23 m and inner radius is r_i=0.11 m. The oil dynamic viscosity is μ_0=0.045 P. The outlet thickness h_1 and the slope coefficient k of the pad are changed in the calculation procedure, which values are given according to the operation conditions of the thrust pads. The rotating speed of the ship propulsion shaft is $\omega=375$ r/min. The other data is given in the calculation procedure.

3.2 Distribution of pressure field

Given the slope coefficient k=0.02 and h_1=5 μm, the lubrication pressure field of the thrust bearing pad is worked out with the Reynolds equation. The calculation result of pressure field is showin in Fig.2 and Fig.3. Abscissa x is the non-dimensional circumferential direction of θ, ordinate y is the non-dimensional radial direction of r. The contours of pressure represent the distribution state of lubrication force. According to the figures, it can be observed that the pressure field is unsymmetrical and distributes uneven, the pressure in the intermediate area is much higher than that in surrounding area. Based on the distribution of pressure field, the supporting point of the pad is located at the bigger oil pressure area. This lets the pad balance with the outer forces and makes the thrust operate in good conditions. This avoids the bearing faults to produce and makes the lubrication condition of the pads improved greatly.

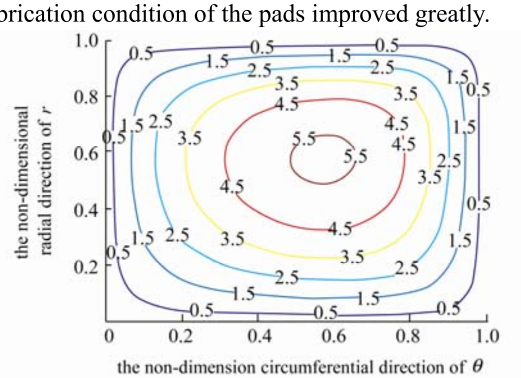

Fig.2 Contour of pressure field

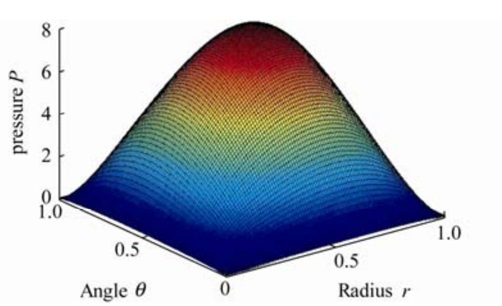

Fig.3 3-D distribution of pressure field

3.3 Relationship between non-dimensional load W and outlet film thickness h_1

When the other thrust bearing parameters keep unchangeable, and the slope coefficient k of the thrust pad is given values 0.03, 0.02, 0.01 respectively according to the running conditions of the pads, the relationship between the non-dimension load W and outlet film thickness h_1 of the thrust pad is investigated. Fig.4 shows the calculation results. It can be obtained based on the results that when the outlet film thickness h_1 is becoming bigger gradually, the non-dimension load W will decrease correspondingly. But the decrease of W is not linear change with the thickness h_1. The bearing load capability W is decreasing quickly when the thickness h_1 is among small value domain, while it is changing slowly when the thickness h_1 is among big value domain. Because the thickness h_1 is associated with the bearing clearance closely, the loading-capability of a bearing is decided by the bearing clearance directly. When the loading-capability of a bearing increases, the bearing clearance must be adjusted and become smaller.

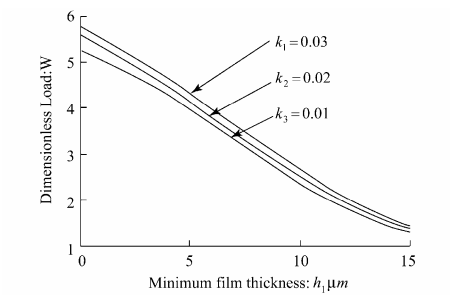

Fig. 4 Relationship between W and h_1

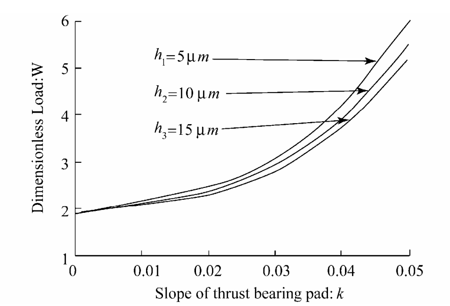

Fig.5 Relationship between W and k

3.4 Relationship between non-dimension load W and slope coefficient k of the pad

The value of the outlet film thickness h_1 of thrust bearing pad is given in the solving process of Reynolds equation while other parameters keep constant. Assumed the thickness h_1 value is given as 5 μm, 10 μm, 15 μm respectively, the non-dimension load W is calculated by numerical integration method based on the above theory. The results of calculation are shown in the Fig.4. After analyzing the Fig.5, it can be observed that the non-dimension load W is associated with the change of the slope coefficient k of the thrust bearing pad. When the outlet film thickness h_1 is 5 μm, 10 μm, 15 μm respectively, the non-dimension bearing load W becomes bigger correspondingly with the increase of the slope coefficient k of the pad. But the relationship between the load W and the slope k is different with the thickness h_1. By comparison with the non-dimension load W on the condition of h_1=10 μm, the non-dimension load W on the condition of h_1=5 μm changes more sharply than on the condition of h_1=15 μm. It is induced that the slope coefficient k plays an important role in the performance of the pad thrust bearing.

4 Conclusions

The loading-capability of a thrust bearing is influenced by the distribution of pressure field and the parameters of the thrust pad. Based on the governed Reynolds equation, the distribution and non-dimension load are worked out by numerical integral method. After analyzing the results of calculation, the following conclusions may be drawn:

1）The oil pressure field is unsymmetrical and distributes uneven. The pressure in the intermediate area is much higher than that in surrounding area，the most is more than 5.5 MPa.

2）The thickness of outlet oil and the slope coefficient of the thrust pads influenced the loading-capability of a bearing greatly. With the increase of the slope coefficient and decrease of the outlet thickness of the pad, the loading-capability of a bearing improves correspondingly.

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