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Lubrication Performance Analysis of Tilting Pad Journal Bearing with Dynamic and Static Pressure

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Abstract. The application of bearing technology in the field of intelligent equipment and special robots continues to promote technological innovation and industrial upgrading, such as the improvement of bearing materials and the development of intelligent bearings, which helps to improve the performance and operation efficiency of equipment. In this paper, the model of dynamic and static pressure tilting pad journal bearing is established, and the influence of eccentricity and preload on the static characteristics of dynamic and static pressure tilting pad journal bearing is studied. The results show that along the rotation direction of the journal, the oil film thickness decreases gradually, the static pressure of the pad increases gradually, and the lower pad is the main bearing pad; With the increase of eccentricity and preload coefficient, the bearing load coefficient and friction coefficient also increase.

Keywords. Sliding bearing, tilting pad, dynamic and static pressure

1. Introduction

Bearings are key moving parts in intelligent equipment and special robots. Bearing selection can be considered in the design and optimization process to improve equipment performance and reliability. Compared with the traditional radial sliding bearing, the tilting pad radial sliding bearing has multiple bearing tiles, each tile can independently swing around the fulcrum, and has high stability [1-3]. Edoardo et al. [4] established a thermohydrodynamic model to predict the static parameters such as stiffness and damping coefficient and minimum oil film thickness of tilting pad journal bearings by training neural networks. Benti et al. [5] studied the influence of the number of tiles on the dynamic characteristics of the tilting pad journal bearing, and compared the experimental and numerical results. In this paper, a dynamic and static pressure tilting pad journal sliding bearing model is established. The static pressure support can effectively avoid the collision between the journal and the bearing bush, and prolong the service life of the bearing. The structure of the oil cavity makes the bearing produce dynamic pressure effect and improve the stiffness of the oil film [6-7]. Wei et al. [8] established a four-shallow-cavity hybrid bearing model, and studied the static performance indexes such as bearing capacity, minimum film thickness and temperature

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rise of the bearing. Ding et al. [9] established a model of hydrodynamic and hydrostatic bearings with deep and shallow cavities, and studied the multi-objective optimization design of bearing structure based on genetic algorithm.

At present, scholars have studied more on the lubrication performance of hydrodynamic and hydrostatic bearings and tilting pad bearing, but less on the hydrodynamic and hydrostatic bearings combined with tilting pad structure. In this paper, the influence of eccentricity and preload on the static characteristics of dynamic and static pressure tilting pad bearing is studied by numerical analysis.

2. Theoretical model of dynamic and static pressure tilting pad radial sliding bearing

Tilting pad journal bearings usually have two distribution modes of load between pads and load on pads. Hydrodynamic and hydrostatic bearings usually adopt the distribution of load between pads. The dynamic and static pressure tilting pad bearing studied in this paper adopts the form of inter-pad load. On the basis of considering the tilting pad structure, the static pressure cavity is added to each pad, and the influence of eccentricity and preload on the static characteristics of the bearing is studied. Figure 1 is the structure diagram of dynamic and static pressure tilting tile, the bearing has three tiles uniformly distributed in the circumferential direction, where O_b is the bearing center, O_j is the journal center, r is the journal radius, and R is the bearing radius.

The hydrostatic chamber of the bearing is set at the center of the bearing bush, that is, symmetrical along the circumferential and axial midline of the bearing. The structure of the bearing bush is shown in Figure 2. An oil inlet hole is machined in the middle of the hinge by EDM to supply oil to the oil cavity. The oil inlet hole is used as a capillary restrictor to adjust the pressure of each oil chamber to adapt to the change of different external loads. In the design process of dynamic and static pressure bearings, it is necessary to pay attention to the size of static pressure. If the static pressure is too large, the dynamic pressure will be destroyed, and if the static pressure is too small, it will not be able to provide sufficient support for the system.

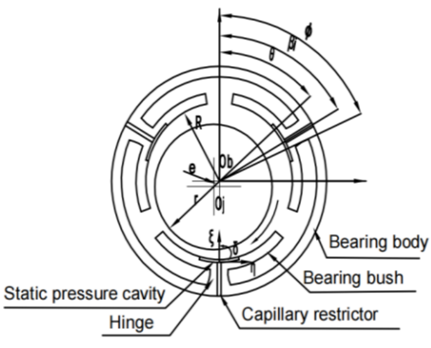


Figure 1. Bearing structure diagram

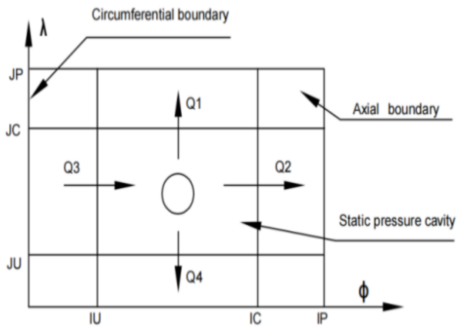


Figure 2. Static pressure cavity structure

Dynamic and static pressure tilting pad bearing is a kind of dynamic and static pressure bearing. Its calculation method is similar to that of static pressure bearing. The difference is that the bearing of tilting pad bearing will swing according to the change of load, and then adjust the pressure distribution. According to the dynamic and static pressure tilting pad structure diagram shown in Figure 1, the oil film thickness of the bearing and the expression of the Reynolds equation are as follows:

$$h = c + e\cos(\phi - \theta) + (c_1 - c)\cos(\phi - \beta_i) - \frac{\delta_i}{\psi}\sin(\phi - \beta_i) \tag{1}$$

$$\frac{1}{r^2}\frac{\partial}{\partial \phi}\left[h^3\frac{\partial h}{\partial \phi}\right] + \frac{\partial}{\partial \lambda}\left[h^3\frac{\partial h}{\partial \lambda}\right] = 6\omega\mu\frac{\partial h}{\partial \phi} \tag{2}$$

In the formula, c is the radius clearance of the bearing, e is the eccentricity of the bearing, ϕ is the angle measured from the Y positive half shaft, θ is the deviation angle, c_1 is the assembly clearance, β_i is the position angle of the fulcrum of the tile, δ_i is the swing angle of the tile, ψ is the clearance ratio.

The oil film force in the bearing rotor system is composed of static pressure and dynamic pressure. The static pressure is generated by the external oil supply flowing through the capillary restrictor to the static pressure oil chamber, and the static pressure chamber pressure is used as the boundary condition of the dynamic pressure to obtain the dynamic pressure. Figure 2. is the structure diagram of the hydrostatic oil cavity, which is meshed and the discrete equation is established.

The flow formula of capillary restrictor is:

$$Q_{in} = \frac{\pi d^4}{128\mu l}(P_s - P_i) \tag{3}$$

In the formula, d represents the diameter of the capillary, l represents the length of the capillary, μ represents the viscosity of the lubricating oil, P_s represents the oil supply pressure, and P_i represents the pressure of the static pressure chamber.

According to the schematic diagram of the static pressure oil chamber shown in Fig.2, the 1/2 step area of the static pressure oil chamber is selected as the control body, and the flow through the IC+1/2 and IU-1/2 sides of the control body is calculated.

$$Q_2 = \sum_{j=JU}^{JC} \Delta\lambda \left(\frac{Uh_{IC+\frac{1}{2},j}}{2} - \frac{h^3_{IC+\frac{1}{2},j}}{12\mu}\frac{P_{IC+1,j} - P_{IC,j}}{R\Delta\phi} \right) \tag{4}$$

$$Q_3 = \sum_{j=JU}^{JC} \Delta\lambda \left(\frac{Uh_{IU-\frac{1}{2},j}}{2} - \frac{h^3_{IU-\frac{1}{2},j}}{12\mu}\frac{P_{IU,j} - P_{IU-1,j}}{R\Delta\phi} \right) \tag{5}$$

In the formula, U represents the velocity of the moving plate, and the flow through the JC+1/2 and JU-1/2 sides is:

$$Q_1 = \sum_{i=IU}^{IC} \frac{h^3_{i,JU-\frac{1}{2}}}{12\mu}(P_{i,JU} - P_{i,JU-1}) \tag{6}$$

$$Q_4 = \sum_{i=IU}^{IC} \frac{h^3_{i,JC+\frac{1}{2}}}{12\mu}\left(P_{i,JC} - P_{i,JC+1}\right) \tag{7}$$

$$Q_{out} = \left(Q_1 + Q_2 + Q_4 - Q_3\right) / \frac{P_s c^3}{12\mu} \tag{8}$$

According to the flow balance through the control body, there are:

$$Q_{out} = Q_{in} \tag{9}$$

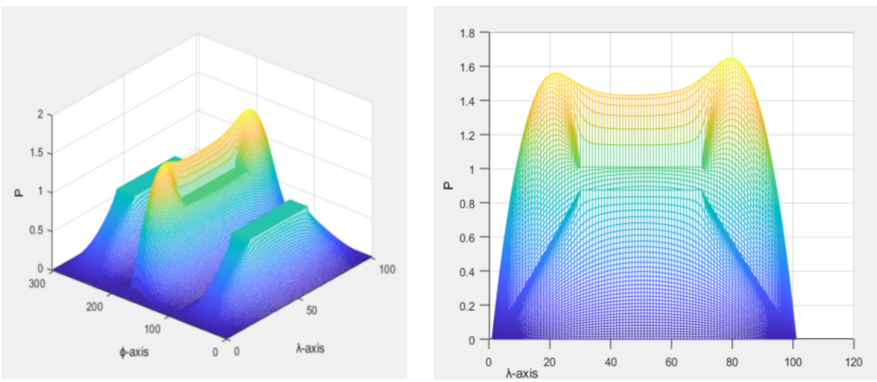
Based on the structural parameter data in Table 1, the lubrication performance of tilting pad journal bearing can be studied by using the finite difference method.

Table 1. Dynamic and static pressure tilting pad journal bearing parameters

Name	Numerical value	Unit
Diameter of bearing	80	mm
Bearing width	50	mm
Bearing clearance	0.08	mm
Lube oil viscosity	0.015	Pa·s
Oil supply pressure	2	MPa
Throttle diameter	0.8	mm
Throttle length	10	mm
Bearing wrap angle	100	°

3. Calculation Results and Discussion

3.1. Dynamic and static pressure tilting pad radial sliding bearing pressure distribution



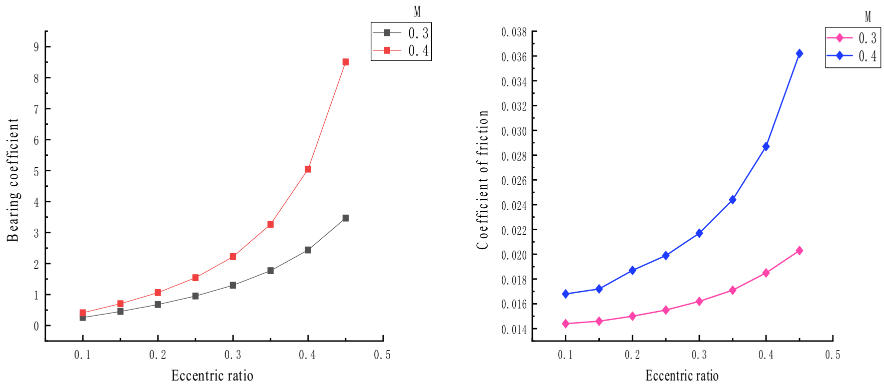
(a) Oil film pressure diagram **(b)** Oil film axial distribution pressure

Figure 3. Oil film pressure distribution

According to the preset parameter values of the hydrodynamic and hydrostatic tilting pad journal bearing shown in Table 1, the oil film pressure distribution of the hydrodynamic and hydrostatic tilting pad journal bearing shown in Figure 3 is calculated when the eccentricity is 0.3 and the initial deviation angle is 2 °. Along the rotation direction of the journal, the thickness of the oil film gradually decreases, and the static pressure of the pad gradually increases. The lower pad is the main bearing pad. It can be seen from the pressure distribution diagram that the lower pad has obvious dynamic pressure, and the oil film pressure of the pad is the largest.

3.2. The influence of eccentricity and preload on the static characteristics of tilting pad journal bearing with dynamic and static pressure

Figure 4(a). shows the influence of eccentricity and preload coefficient on bearing load coefficient. Figure 4(b). shows the influence of eccentricity and preload coefficient on the friction coefficient of bearing. It can be seen from the figure that eccentricity and preload coefficient have important influence on the static performance of bearing. Under the same preload coefficient, the bearing load coefficient and friction coefficient increase with the increase of eccentricity. At the same eccentricity, as the preload coefficient increases, the bearing coefficient and friction coefficient of the bearing will also increase, which is consistent with the conclusion of Reference [10].



(a) Effect of eccentricity on bearing coefficient (b) Effect of eccentricity on friction coefficient

Figure 4. Effect of eccentricity on static performance of bearing

4. Conclusion

In this paper, a dynamic and static pressure tilting pad journal bearing model is established to study the influence of eccentricity and preload coefficient on the lubrication performance of the bearing. The specific conclusions are as follows:

(1) The dynamic and static pressure tilting pad bearing adopts the distribution mode of inter-pad load. The oil film thickness at the lower pad is the smallest and the oil film pressure is the largest. The lower pad is the main bearing pad. It can be seen from the

pressure diagram that the lower pad has obvious dynamic pressure, and the dynamic pressure of the left pad and the right pad is not obvious.

(2) Under the same preload coefficient, with the increase of eccentricity, the bearing load coefficient and friction coefficient increase gradually. In the case of the same eccentricity, when the preload coefficient increases, the bearing coefficient and friction coefficient of the bearing also increase.

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