Numerical Analysis Based on Oil Film Characteristics of Sliding Bearing

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Abstract. Taking the finite-length sliding bearing as the research object, the Reynolds equation is solved by the finite difference method and the successive over-relaxation iteration method to obtain the oil film pressure. It is concluded that the dimensionless oil film characteristics are related to bearing eccentricity and length diameter ratio. In order to study the oil film characteristics of sliding bearing, the bearing eccentricity and length diameter ratio are taken as the research objects, and the oil film thickness, load component, attitude angle and end discharge flow are solved by numerical analysis. It is concluded that with the increase of eccentricity, the bearing capacity of oil film increases, the attitude angle decreases and the end discharge flow increases with the case of a certain length diameter ratio.

1. Introduction

Bearings are widely used in mechanical engineering as supporting elements of rotating parts, and its main function is to reduce friction force of rotating parts[1]. Sliding bearing is widely used because of its high rotation accuracy, high dynamic stiffness, high damping and long life[2-3].

There are many types of sliding bearings, which can be divided into gas lubrication, fluid lubrication, grease lubrication and solid lubrication according to the type of lubrication. The application research on hydrodynamic lubrication sliding bearings is the most extensive. Some scholars have analyzed and solved the equation[4-6]. Reynolds[7] proposed the Reynolds equation of fluid lubrication theory for the first time by combining viscous fluid motion equation and fluid continuous motion equation. This equation is the theoretical basis of fluid lubrication bearing research, and the solution of the equation is the core problem of studying the oil film characteristics of sliding bearing. Li Yuansheng[8] used finite element method and finite difference method to analyze transient response of rotor and pressure distribution of oil film flow field, and discussed the influence of clearance width, bearing length and lubricating oil viscosity on dynamic characteristic coefficient of dynamic sliding bearing.

In this paper, the Reynolds equation is discretized based on the finite difference method, and the successive over-relaxation iteration method is used to solve the internal oil film characteristics of the sliding bearing. Through numerical analysis, the oil film pressure, oil film thickness, oil film bearing capacity and end discharge flow rate of sliding bearing are obtained, and the characteristic analysis of sliding bearing oil film is studied.
2. Numerical analysis of dynamic characteristic coefficient of sliding bearing

2.1. Reynolds equation

According to the fluid lubrication theory of sliding bearing, the radial oil film pressure of sliding bearing obeys the generalized hydrodynamic Reynolds equation:

$$\frac{1}{R_b^2} \frac{\partial}{\partial \varphi} \left( \rho h^3 \frac{\partial p}{\partial \varphi} \right) + \frac{\partial}{\partial z} \left( \rho h^3 \frac{\partial p}{\partial z} \right) = \left( \frac{v_r + v_b}{2R_b} \right) \frac{\partial (\rho h)}{\partial \varphi} + \frac{\partial (\rho h)}{\partial t}$$  \hspace{2cm} (1)

Where $\mu$-lubricating oil dynamic viscosity; $p$-oil film pressure; $h$-oil film thickness; $\rho$-oil film density; $\varphi$-sliding bearing circumferential coordinate; $z$-sliding axial coordinate;

When analyzing and calculating the sliding bearing, it is often carried out in a dimensionless form to facilitate the calculation. Introduce dimensionless numbers:

$$c = \frac{c}{L_b}, \rho = \rho, \mu = \frac{\mu}{2\mu}, p = \frac{p}{p}, \lambda = \frac{\lambda}{2R_b}, \varepsilon = \frac{\varepsilon}{c}$$  \hspace{2cm} (2)

Substituting the dimensionless equation (2) into the equation (1), the simplified dimensionless Reynolds equation is shown in the following equation:

$$\frac{\partial}{\partial \varphi} \left( h^3 \frac{\partial \bar{p}}{\partial \varphi} \right) + \frac{\partial}{\partial z} \left( h^3 \frac{\partial \bar{p}}{\partial z} \right) = -3\varepsilon \sin \varphi$$  \hspace{2cm} (3)

2.2. Reynold equation solving

The five-point difference method is used to discretize equation (3), and the corresponding difference equation form at point $i$ is obtained:

$$p_{i,k} = A_i p_{i+1,k} + B_i p_{i-1,k} + C_i (p_{i,k+1} + p_{i,k-1}) + D_i$$  \hspace{2cm} (4)

The coefficients are:

$$A_i = \frac{\bar{h}_{i+\frac{1}{2}}}{E_i \Delta \varphi^2}, B_i = \frac{\bar{h}_{i-\frac{1}{2}}}{E_i \Delta \varphi^2}, C_i = \frac{\bar{h}_i}{\lambda^2 E_i \Delta z}, D_i = \frac{3\varepsilon \sin \varphi_i}{E_i}, E_i = \frac{\bar{h}_{i+\frac{1}{2}} + \bar{h}_{i-\frac{1}{2}}}{\Delta \varphi^2} + \frac{2\bar{h}_i}{\lambda^2 \Delta z}$$  \hspace{2cm} (5)

For the solution of equation (4), the successive over-relaxation iteration method can be used to analyze and obtain the pressure value of each node in the fluid medium region. In this paper, MATLAB programming is used to solve the oil film pressure value of the sliding bearing.

2.3. Calculation of oil film bearing capacity and end discharge

The dimensionless bearing capacity of the bearing in the radial and tangential directions can be obtained through numerical integration calculation of Equation (6):

$$\bar{F}_z = -\int_0^\varphi \left( \int_1 \rho d\lambda \right) \sin \varphi d\varphi$$

$$\bar{F}_\eta = -\int_0^\varphi \left( \int_1 \rho d\lambda \right) \cos \varphi d\varphi$$  \hspace{2cm} (6)

The resultant force of dimensionless bearing capacity is:

$$\bar{F} = \sqrt{\bar{F}_z^2 + \bar{F}_\eta^2}$$  \hspace{2cm} (7)

The angle between the connecting line between the center of the shaft diameter and the center of the bearing bush and the line of the load force is the deflection angle:

$$\theta = \arctan \left( \frac{\bar{F}_z}{\bar{F}_\eta} \right)$$  \hspace{2cm} (8)

During the operation of the sliding bearing, part of the lubricant will flow in the bearing in the circumferential direction, while the other part will flow out from both ends. This amount of outflow is...
called end leakage. In order to ensure the balance of the bearing system, the oil volume needs to be replenished in time. Therefore, this part of the compensated flow rate is called the end drain flow rate. Then the dimensionless end discharge flow expression is as follows:

$$\bar{Q} = \int_0^{2\pi} H \frac{\partial P}{\partial \lambda} \, d\varphi = \sum_{i=1}^{m} \left[ 1 + \varepsilon \cos((i-1)\Delta \varphi) \right]^{3/2} \delta \frac{P_{ik} - P_{ik+1}}{\Delta \lambda} \Delta \varphi$$

(9)

3. Case analysis

It can be seen from equation (4) that the distribution of dimensionless oil film pressure $P$ only depends on the aspect ratio $\lambda$, eccentricity $\varepsilon$, meshing degree $m$ and $n$. In this paper, the eccentricity is 0.477, the aspect ratio is 0.5, the circumferential grid $m$ is 60 and the axial grid $n$ is 40. A set of dimensionless oil film pressure distribution is obtained, and its three-dimensional distribution is shown in Figure 1.

![Fig. 1 Dimensionless oil film pressure distribution](image)

3.1. Analysis of the influence of eccentricity on oil film thickness

In order to analyse the influence of the change of eccentricity on the thickness of the oil film when the aspect ratio is constant. In this paper, the aspect ratio is set to a fixed value of 0.5, and the eccentricity is set to 0.3, 0.4, 0.5 and 0.6 respectively. The oil film thickness values at the minimum and maximum oil film positions are obtained as shown in Table 1:

| $\varepsilon$ | Minimum oil film thickness | Maximum oil film thickness |
|--------------|---------------------------|---------------------------|
| 0.3          | 0.7                       | 1.3                       |
| 0.4          | 0.6                       | 1.4                       |
| 0.5          | 0.5                       | 1.5                       |
| 0.6          | 0.4                       | 1.6                       |

It can be seen from Table 1 that with the increase of eccentricity, the minimum oil film thickness decreases. The maximum oil film thickness value increases with the increase of eccentricity. However, the eccentricity cannot continue to increase because the oil film thickness value will be close to zero at this time, so that the bearing and the journal will be partly contacted, resulting in increased friction and wear.

3.2. The influence of eccentricity and aspect ratio on oil film bearing capacity and deflection angle

For the study of the bearing capacity of the oil film of the sliding bearing, this paper calculated the dimensionless oil film bearing capacity when the aspect ratio was 0.1, 0.4, 0.9, 1.5 and the eccentricity was [0.6, 0.95]. The calculation results are shown in Figure 5.

According to Figure 2, it can be concluded that the dimensionless oil film bearing capacity of the sliding bearing is directly proportional to the eccentricity. As long as the rigidity and strength of the bearing bush are large enough, the dimensionless bearing capacity will also tend to infinity. With the
increase of the aspect ratio, the bearing capacity of the dimensionless oil film of the bearing also increases. Because with the increase of the aspect ratio, the wedge effect formed by the gap between

![Figure 2](image2.png)

**Fig. 2** The influence of different aspect ratios and eccentricity on the bearing capacity of the oil film. The shaft diameter of the sliding bearing and the bearing shell is also more obvious, resulting in the dynamic pressure of the oil film is also increased, so the bearing capacity of the oil film is also increasing.

At the same time, the deflection angle under the same length diameter ratio and eccentricity is calculated. And the calculation results are shown in Figure 3. It can be seen that with the increase of eccentricity, the deflection angle gradually decreases. In the case of small eccentricity, the deflection angle decreases with the increase of length diameter ratio. In the case of large eccentricity, the deflection angle increases with the increase of length diameter ratio.

![Figure 3](image3.png)

**Fig. 3** The influence of the eccentricity of different aspect ratios on the deflection angle.

3.3. The influence of eccentricity and aspect ratio on the end leakage flow

![Figure 4](image4.png)

**Fig. 4** The influence of different aspect ratio eccentricity on the end leakage flow.
According to Figure 4, the end discharge increases with eccentricity from 0 to 0.9, and decreases with eccentricity from 0.9 to 1. When the eccentricity is close to 1, the shaft diameter and the bearing bush are in direct contact, and the oil pressure gradient is not established, so the end discharge flow rate shows a downward trend. When the eccentricity is constant, the end leakage flow of bearings with different aspect ratios decreases with the increase of the aspect ratio, and the general trend remains unchanged.

4. Conclusion
In this paper, the dimensionless Reynolds equation is discretized by the finite difference method, and the successive over-relaxation iteration method is used to solve the oil film pressure, and the three-dimensional distribution diagram of the oil film pressure is obtained by numerical analysis. The characteristics of the oil film of sliding bearing under different eccentricity and aspect ratio are analyzed. The conclusions that can be drawn are:

1. The dimensionless bearing capacity of finite-length sliding bearings increases with the increase of eccentricity, and at the same time increases with the increase of the ratio of length to diameter.
2. The deflection angle of a finite-length sliding bearing is inversely proportional to the aspect ratio when the eccentricity is small. At larger eccentricity, the aspect ratio is proportional to the deflection angle. But the deflection angle decreases as the eccentricity increases.
3. The general trend of the end leakage flow is increasing with the increase of the eccentricity. When the eccentricity becomes 1, the end leakage flow begins to decrease. It decreases with the increase of the aspect ratio.

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