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Research on the oil film pressure of floating ring bearing in the steady dynamic state

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ABSTRACT

Based on the theory of hydrodynamic lubrication, the inner and outer oil film pressure and capacity of floating ring bearings (FRBs) in the steady dynamic state were researched. According to the lubrication mechanism of FRB, dynamic load Reynolds equation about the inner and outer oil film pressure was deduced at first, then film pressure distribution at the finite-length and Reynolds boundary conditions was solved and simulated by using finite difference method and MATLAB. Moreover, the relationship of film pressure and load-carrying capacity vary with the eccentricity and the ratio of width to diameter were also studied. Research indicates that film pressure of FRB changes periodically with time in the steady dynamic state. The higher the eccentricity or the ratio of width to diameter, the greater the film pressure and the load-carrying capacity.

KEYWORDS

Bearing design; Steady state; Reynolds equation; Oil file thickness; Oil film stiffness.

INTRODUCTION

As the support of the rotating part, floating ring bearings (FRB) has a double lubricant film which can improve the lubrication quality, reduce the relative velocity between the journal and the bearing, and reduce friction and temperature of the bearing chamber. At the same time, FRB can relieve film vibration caused by high speed rotation. Compared with plain sliding bearings and rolling bearings, FRB is more suitable for high speed, high temperature and other harsh environments. Therefore, the FRB are widely used in various rotating systems.

Based on the various advantages and widespread application of FRBs, many domestic and international scholars did a lot of research on oil film feature of FRB. Kiyoshi Hatakenaka et al. considered the impact of the axial film rupture and cavitation phenomena on the FRB oil film pressure, and studied the characteristics of oil film pressure[1]. Andres et al. studied the thermal effects of FRB, and noted that the bearing clearance will be affected by thermal effects, and thus affect the floating ring speed as well as its inner and outer film pressure distribution[2]. Chun have studied the lubrication systems of turbocharger, and compared oil film pressure distribution of non-bubble-containing lubricant with the bubble-containing lubricant[3]. Guorong Wang et al. used the infinite length bearing theory and the infinite short bearing theory to simplify the Reynolds equation, and got the relationship between FRB oil film capacity and structure parameters[4]. Zibo Ye et al. analyzed the relationships among work mechanism, structure parameters and properties of internal combustion engine turbocharger FRB in the steady static state[5]. Zhaohui Kang et al. studied the mechanism of the floating ring, pointed out that the location of maximum oil film pressure of squeeze film system will move to at the junction of the positive and negative pressure area of cavitation oil film system with the increasing of eccentricity[6]. Zhi Wang et al. used infinitely long bearing theory and integration methods to study the characteristics of the inner and outer oil film pressure of the FRB under the state of synchronous precession[7]. Based on the FRB of a gas turbine and hydrodynamic lubrication, the Reynolds equation for the inner and outer oil film in the steady dynamic state was established. The finite length bearings theory and Reynolds boundary conditions are used to get the pressure distribution of the inner and outer film through computer simulation. This paper analysis the relationship between film pressure and capacity changes with eccentricity and width-diameter ratio, and provides a reference for the further design of FRB.

MATERIALS AND METHODOLOGY

The basic hydrodynamic lubrication Reynolds equations of sliding bearing in the steady dynamic state

Hydrodynamic lubrication of journal bearing depends on relative motion of the journal and bearings, which make viscous fluid flow from the big to the small end clearance used to withstand the loads and avoid contact between bearing and journal, so it can reduce frictional resistance and protect working surface. With the assumption of no change in pressure along the direction of the oil film thickness, lubricating fluid is Newtonian fluid, lubrication fluids is incompressible (ρ is constant) and changes of viscosity are not considered (μ is constant), under the state of journal and bearing are rotating at the same time, the Reynolds equation for fluid in bearing clearance can be written as follows[8]:

$$\begin{aligned}&\frac{\partial}{\partial x}\left(\frac{h^3}{\mu}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(\frac{h^3}{\mu}\frac{\partial p}{\partial z}\right)=6\left(U_j+U_b\right)\frac{\partial h}{\partial x}\\&+6h\frac{\partial(U_j+U_b)}{\partial x}+12V_j\end{aligned} \quad (1)$$

The movement trajectory of axis relative to bearing is a closed and stable curve in the steady dynamic state, meanwhile, bearing capacity is produced by the rotation effect the wedge effect) and squeeze effect of the lubricant film. In Eqs. (1), V_j reflected the squeeze effect, represents the journal center move to squeeze film to form pressure, and $V_j = c\frac{d\varepsilon}{dt}\cos\varphi + c\varepsilon\frac{d\Phi}{dt}\sin\varphi$, $U_j = R_j\omega_j + c\frac{d\varepsilon}{dt}\sin\varphi - c\varepsilon\frac{d\Phi}{dt}\cos\varphi$, $U_b = R_b\omega_b + c\frac{d\varepsilon}{dt}\sin\varphi - c\varepsilon\frac{d\Phi}{dt}\cos\varphi$, If $x = R_j\varphi$, put V_j, U_j, U_b, x into Eqs. (1), because $\frac{c}{R_j}$, $\frac{c\varepsilon}{R_j}$, $\frac{h}{R_j}$ are very little, items which contain the coefficients can be omitted. The general Reynolds equation of plain journal bearing can be obtained as follows through simplifying Eqs. (1):

$$\frac{\partial}{\partial x}\left(\frac{h^3}{\mu}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(\frac{h^3}{\mu}\frac{\partial p}{\partial z}\right)=6\left(\omega_j+\frac{R_b}{R_j}\omega_b\right)\frac{\partial h}{\partial \varphi}$$
$$+12\left(c\frac{d\varepsilon}{dt}\cos\varphi+c\varepsilon\frac{d\Phi}{dt}\sin\varphi\right) \tag{2}$$

Reynolds equations for the outer and inner oil film of FRB in the steady dynamic state

As can be seen from Figure 1, the FRB is a kind of double film lubricated bearings. The double film is separated by floating rings. The lubricant film between journal surface and the inner surface of the floating ring is inner film. The lubricant film between the outer surface of floating ring and inner surface of bearing is outer film. During the operation of the journal, the floating ring rotates at a constant speed with the rotation of the journal.

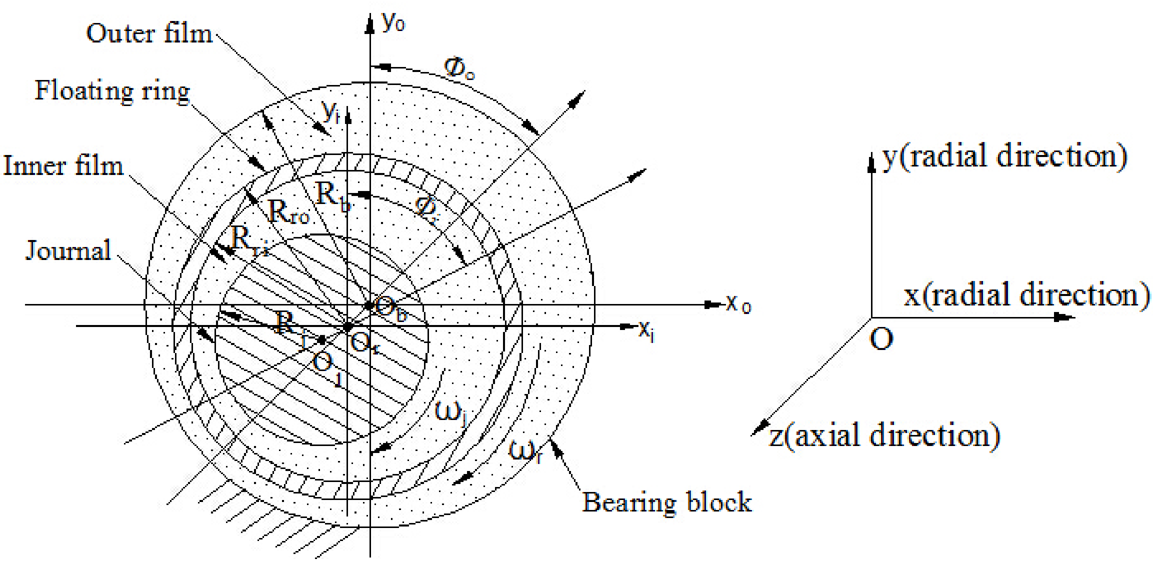

Figure 1 : Floating-ring bearing section

The Reynolds equation for the outer oil film of FRB

For the Reynolds equation of the outer oil film, the speed on the right side in the case of bearing fixed only involves the speed of floating ring. Corresponding to the lubrication theory and Eqs. (2), the Reynolds equation for the outer oil film lubrication can be written as follows:

$$\frac{\partial}{\partial x}\left(\frac{h_o^3}{\mu}\frac{\partial p_o}{\partial x}\right)+\frac{\partial}{\partial z}\left(\frac{h_o^3}{\mu}\frac{\partial p_o}{\partial z}\right)=6\omega_r\frac{\partial h_o}{\partial \varphi_o}$$
$$+12\left(c_o\frac{d\varepsilon_o}{dt}\cos\varphi_o+c_o\varepsilon_o\frac{d\Phi_o}{dt}\sin\varphi_o\right) \tag{3}$$

The Reynolds equation for the inner oil film of FRB

For the Reynolds equation of the inner oil film, the speed on the right side refers to the speed of floating rings and journal. Corresponding to Eqs. (2) and the specific size of floating rings, the Reynolds equation for the inner oil film lubrication can be written as follows:

$$\frac{\partial}{\partial x}\left(\frac{h_i^3}{\mu}\frac{\partial p_i}{\partial x}\right)+\frac{\partial}{\partial z}\left(\frac{h_i^3}{\mu}\frac{\partial p_i}{\partial z}\right)=6\left(\frac{R_{ri}}{R_j}\omega_r+\omega_j\right)\frac{\partial h_i}{\partial \varphi_i}$$
$$+12\left(c_i\frac{d\varepsilon_i}{dt}\cos\varphi_i+c_i\varepsilon_i\frac{d\Phi_i}{dt}\sin\varphi_i\right) \tag{4}$$

THE DIMENSIONLESS CALCULATION MODEL AND THE DETERMINATION OF PRESSURE BOUNDARY CONDITIONS

The Dimensionless outer and inner oil film and its discussion and analysis

The dimensionless Reynolds equation for the inner and outer oil film is the most compact form of the problem, which was easy to be solved by numerical analysis of computer programming.

Outer oil film

For the numerical calculation, Eqs. (5) is the dimensionless form of Eqs. (3), which can be given by introducing the following dimensionless variables:

$x = R_{ro}\varphi_o$, $H_o = h_o / c_o = 1 + \varepsilon_o \cos\varphi_o$, $\lambda_o = z/(l/2)$, $\psi_o = c_o / R_{ro}$, $e_o = c_o\varepsilon_o$

$$\begin{aligned}&\frac{\psi_o{}^2}{\mu}\frac{\partial}{\partial\varphi_o}\left(H_o{}^3\frac{\partial p_o}{\partial\varphi_o}\right)+\frac{\psi_o{}^2}{\mu}\left(\frac{d_{ro}}{l}\right)^2\frac{\partial}{\partial\lambda_o}\left(H_o\frac{\partial p_o}{\partial\lambda_o}\right)=\\&-6\varepsilon_o\sin\varphi_o\left(\omega_r-2\frac{d\Phi_o}{dt}\right)+12\frac{d\varepsilon_o}{dt}\cos\varphi_o\end{aligned} \quad \textbf{(5)}$$

① When $\omega_r - 2\frac{d\Phi_o}{dt} = 0$, it can be considered that the instantaneous pressure of oil film is only generated by extrusion effect between the bearing and the outer of floating rings. If $P_o = p_o\psi_o{}^2/(12\mu\frac{d\varepsilon_o}{dt})$, the dimensionless Reynolds equation can be obtained as:

$$\frac{\partial}{\partial\varphi_o}\left(H_o{}^3\frac{\partial P_o}{\partial\varphi_o}\right)+\left(\frac{d_{ro}}{l}\right)^2\frac{\partial}{\partial\lambda_o}\left(H_o\frac{\partial P_o}{\partial\lambda_o}\right)=\cos\varphi_o \quad \textbf{(6)}$$

② When $\omega_r - 2\frac{d\Phi_o}{dt} \neq 0$, if can be considered that the instantaneous pressure of oil film is simultaneously generated by the rotation effect and extrusion effect to withstand external loads. If $\omega^* = \omega_r - 2\frac{d\Phi_o}{dt}$, $P_o = \frac{p_o\psi_o{}^2}{6\mu\omega^*}$, $q_o = \frac{2}{\omega^*}\frac{d\varepsilon_o}{dt}$, the dimensionless Reynolds equation can be obtained as:

$$\begin{aligned}&\frac{\partial}{\partial\varphi_o}\left(H_o{}^3\frac{\partial P_o}{\partial\varphi_o}\right)+\left(\frac{d_{ro}}{l}\right)^2\frac{\partial}{\partial\lambda_o}\left(H_o\frac{\partial P_o}{\partial\lambda_o}\right)=\\&-\varepsilon_o\sin\varphi_o+q_o\cos\varphi_o\end{aligned} \quad \textbf{(7)}$$

Inner oil film

For the numerical calculation, Eqs. (8) is the dimensionless form of Eqs. (4), which can be given by introducing the following dimensionless variables:

$x = R_j\varphi_i$, $H_i = h_i / c_i = 1 + \varepsilon_i \cos\varphi_i$, $\lambda_i = z/(l/2)$, $\psi_i = \frac{c_i}{R_j}$, $e_i = c_i\varepsilon_i$

$$\begin{aligned}&\frac{\psi_i{}^2}{\mu}\frac{\partial}{\partial\varphi_i}\left(H_i{}^3\frac{\partial p_i}{\partial\varphi_i}\right)+\frac{\psi_i{}^2}{\mu}\left(\frac{d_j}{l}\right)^2\frac{\partial}{\partial\lambda_i}\left(H_i\frac{\partial p_i}{\partial\lambda_i}\right)=\\&-6\varepsilon_o\sin\varphi_o\left(\frac{R_{ri}}{R_j}\omega_r+\omega_j-2\frac{d\Phi_i}{dt}\right)+12\frac{d\varepsilon_i}{dt}\cos\varphi_i\end{aligned} \quad \textbf{(8)}$$

①When $\frac{R_{ri}}{R_j}\omega_r+\omega_j-2\frac{d\Phi_i}{dt}=0\frac{n!}{r!(n-r)!}$, it can be considered that the instantaneous pressure of oil film is only generated by the squeeze effect between the inner of floating rings and journal. If $P_i=p_i\psi_i^2/(12\mu\frac{d\varepsilon_i}{dt})$, the instantaneous dimensionless Reynolds equation can be obtained as:

$$\frac{\partial}{\partial\varphi_i}\left(H_i^3\frac{\partial P_i}{\partial\varphi_i}\right)+\left(\frac{d_j}{l}\right)^2\frac{\partial}{\partial\lambda_i}\left(H_i\frac{\partial P_i}{\partial\lambda_i}\right)=\cos\varphi_i \tag{9}$$

②When $\frac{R_{ri}}{R_j}\omega_r+\omega_j-2\frac{d\Phi_i}{dt}\neq 0$, the instantaneous floating rings inner film pressure is generated by rotation effect and extrusion effect at the same time to withstand external load. If $\omega^*=\omega_r-2\frac{d\Phi_o}{dt}$; $q_i=\frac{2}{\omega^{\otimes}}\frac{d\varepsilon_i}{dt}$; $P_i=\frac{p_i\psi_i^2}{6\mu\omega^{\otimes}}$, the instantaneous dimensionless Reynolds equation can be obtained as:

$$\frac{\partial}{\partial\varphi_i}\left(H_i^3\frac{\partial P_i}{\partial\varphi_i}\right)+\left(\frac{d_j}{l}\right)^2\frac{\partial}{\partial\lambda_i}\left(H_i\frac{\partial P_i}{\partial\lambda_i}\right)= -\varepsilon_i\sin\varphi_i+q_i\cos\varphi_i \tag{10}$$

The boundary conditions of the inner and outer film pressure

For cylindrical bearings, there are three different assumptions about pressure boundary conditions. Sommerfeld boundary conditions point out that the whole film has a complete gap, the pressure is a periodic function along the axial direction. Semi-Sommerfeld boundary conditions hold that complete oil film can only exists in gap shrinkage area, but the oil film is broken in the open area. Reynolds boundary conditions are up to the actual situation, which deem that the edge of complete film depends on the conditions that $p=0$ and $\partial p/\partial\varphi=0$,but when the boundary conditions are $z=\pm l/2$ in the axial direction, $p=0$, in which the bearing center as coordinates plane and the largest oil clearance as start measurement angle. On this basis, the Reynolds boundary conditions are considered to be the inner and outer film pressure boundary conditions. Making the boundary conditions dimensionless, the dimensionless pressure boundary conditions of outer and inner film are given as follows:

$$\text{Outer film boundary:}\begin{cases}0\le\varphi_o\le 2\pi,-1\le\lambda_o\le 1\\ \text{both sides of the bearing: }\lambda_o=\pm1,\ P_o=0\\ \text{starting eage: }\varphi_o=0,\ P_o=0\\ \text{termination eage: }P_o=0,\ \frac{\partial P_o}{\partial\varphi_o}=0\end{cases}$$

$$\text{Inner film boundary:}\begin{cases}0\le\varphi_i\le 2\pi,-1\le\lambda_i\le 1\\ \text{both sides of the bearing: }\lambda_i=\pm1,\ P_i=0\\ \text{starting eage: }\varphi_i=0,\ P_i=0\\ \text{termination eage: }P_i=0,\ \frac{\partial P_i}{\partial\varphi_i}=0\end{cases}$$

DIMENSIONLESS OIL FILM CAPACITY AND ACTION ANGLE

Oil film capacity support the journal or floating rings, which reduce operating friction and ensure sufficient lubrication. The action line of inner film capacity through the journal center; the action line of outer film capacity through the center of floating ring. F_{Px} and F_{Vx} are the parallel component and the perpendicular component of the dimensionless capacity of the film in the offset line, respectively ($x=o,i$, o denotes outer film and i denotes inner film). They can be expressed as

$$F_{Px} = \int_{-1}^{1}\int_{\varphi_1}^{\varphi_2} P_x \cos\varphi_x d\varphi_x d\lambda_x \quad F_{Vx} = \int_{-1}^{1}\int_{\varphi_1}^{\varphi_2} P_x \sin\varphi_x d\varphi_x d\lambda_x$$

Thus, the bearing capacity is $F_x = \left(F_{Px}^{\ 2} + F_{Vx}^{\ 2}\right)^{1/2}$, and the action angle of oil film capacity is $\varphi_x = \begin{cases} \pi - \arcsin(F_{Vx} / F_x), F_{Px} \geq 0 \\ \arcsin(F_{Vx} / F_x), F_{Px} < 0 \end{cases}$.

FINITE DIFFERENCE METHOD TO SOLVING THE DISTRIBUTION OF THE INNER AND OUTER FILM PRESSURE OF FRB

The solution of the dimensionless pressure value of the inner and outer film are similar, taking outer film lubrication equation for example. Eqs. (6) is a special transient phenomenon, which only produce extrusion film pressure. So, the more representative Eqs. (7) is to be used. Now, Eqs. (7) to do the following decomposition:

$$\frac{\partial}{\partial\varphi_o}\left(H_o^{\ 3}\frac{\partial P_o}{\partial\varphi_o}\right) + \left(\frac{d_{ro}}{l}\right)^2 \frac{\partial}{\partial\lambda_o}\left(H_o\frac{\partial P_o}{\partial\lambda_o}\right) = -\varepsilon_o \sin\varphi_o \tag{11}$$

$$\frac{\partial}{\partial\varphi_o}\left(H_o^{\ 3}\frac{\partial P_o}{\partial\varphi_o}\right) + \left(\frac{d_{ro}}{l}\right)^2 \frac{\partial}{\partial\lambda_o}\left(H_o\frac{\partial P_o}{\partial\lambda_o}\right) = \cos\varphi_o \tag{12}$$

The boundary conditions of two decomposed equations is equal to the Eqs. (7). With the assumption that the solution of Eqs. (11) is P_1 and the solution of Eqs. (12) is P_2, the $P_1 + q_o P_2$ must be the solution of Eqs. (7) according to the linear superposition principle. That is to say, the Eqs. (11) is the oil film pressure generated by the rotation effect, and the Eqs. (12) is the oil film pressure produced by extrusion effect. In fact, the film pressure is the superposition of rotation effect and squeeze effect according to a certain condition, which is varied with the impact of q_o. However, q_o is changing with time within one period, so the actual pressure of the oil film are also periodic changes within one rotation period. Now, the finite difference method is utilized to solve the above Eqs. (11) and (12).

Utilizing the finite difference method to solve the distribution of oil film pressure, which expand the oil film alone the plane, divide it into a number of grids and utilize the pressure value of each node to compose each order difference quotient which is approximately replace the derivatives of Reynolds equation. Eventually, the equations turned into a set of algebraic equations, and solved the pressure on each node value. The obtained discrete pressure values approximately expressed the distribution of the oil film pressure[11]. Set wide-diameter ratio of outer film is 0.5 and eccentricity is 0.5. According to the Finite Difference Method, the outer film is divided into $n \times m = 30 \times 40$ units shown in Figure 2. What's more, MATLAB is programmed to solve the pressure distribution of outer film and perform the associated numerical analysis.

ANALYSIS OF NUMERICAL SIMULATION RESULTS

Analysis of the dimensionless oil film pressure

The Figure below shows the distribution of dimensionless outer oil film pressure which was obtained by utilizing MATLAB simulated. When pure rotation effect is only considered, the oil film pressure is small and the distribution is an approximate parabolic distribution. Dimensionless oil film pressure increases to the maximum pressure value within a certain period, then decline sharply, and drop to zero in the area of slightly larger than 180° shown in Figure 3 and Figure 6. When pure extrusion effect is only considered, the oil film pressure is symmetrical distribution along the eccentric line, and the oil film pressure of the extrusion effect is bigger than the rotation effect shown in Figure 4 and Figure 6. In the steady dynamic state, the oil film pressure in the actual process is the superposition of the two volume, and the superposed value is associated with the axial movement, which is related to the value of q_o. Assume that $q_o = 1$ in an instant, then the superposed total pressure of the oil film at the instant is shown in Figure 5 and Figure 7.

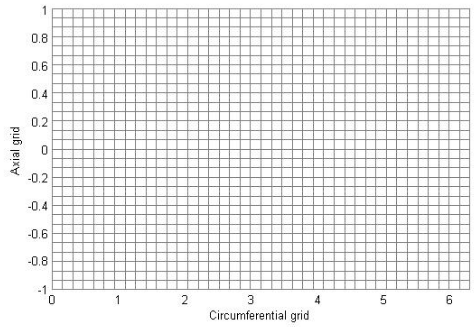

Figure 2 : Outer oil film meshing

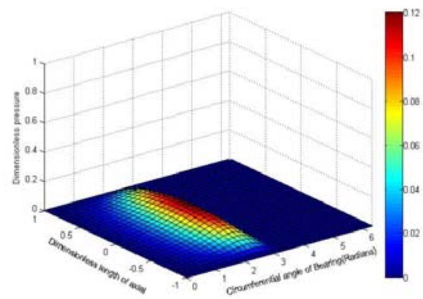

Figure 3 : Outer film pressure distribution of rotation effect

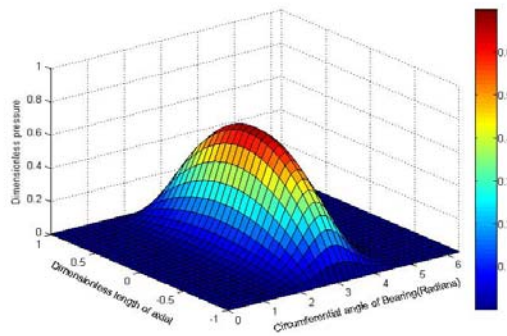

Figure 4 : Outer film pressure distribution of extrusion effect

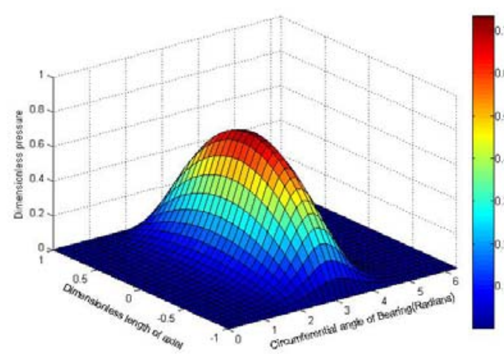

Figure 5 : Comprehensive oil film pressure distribution

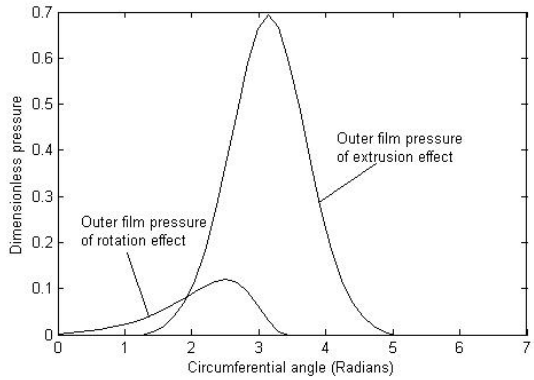

Figure 6 : Pressure distribution of bearing section

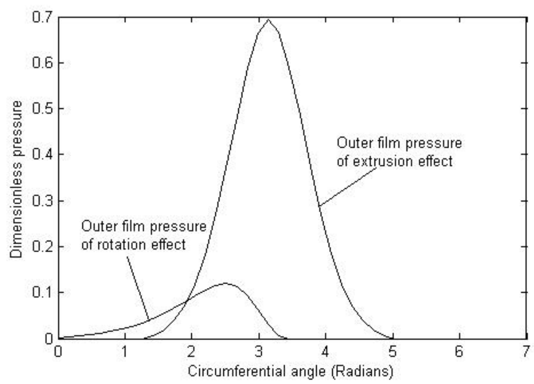

Figure 7 : Comprehensive pressure distribution of bearing section

Relationship between the dimensionless oil film pressure, bearing capacity and eccentricity, width-diameter ratio

In above Eqs. (5), the main factors affecting the dimensionless pressure are the eccentricity and width-diameter ratio. The ratio of bearing length and journal diameter, l/d , is called width-diameter ratio, which is one of the important parameters of radial sliding bearing. Smaller width-diameter ratio helps to improve the operation stability and increase the side discharge to lower the temperature, but the bearing capacity is reduced with the bearing width decreases. The impact of width-diameter ratio and eccentricity on the dimensionless oil film pressure and bearing capacity are shown in the following Figures. The oil film pressure increases with the increase of eccentricity (Figure 8) and width-diameter ratio (Figure 9). Bearing capacity increases with the increase of eccentricity (Figure 10), and width-diameter ratio (Figure 11).

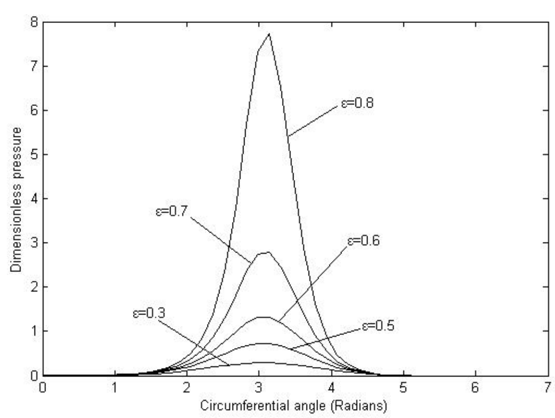

Figure 8 : The effect of eccentricity to pressure

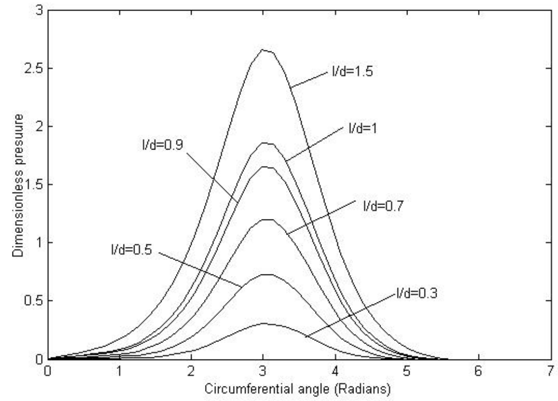

Figure 9 : The effect of width to diameter to pressure

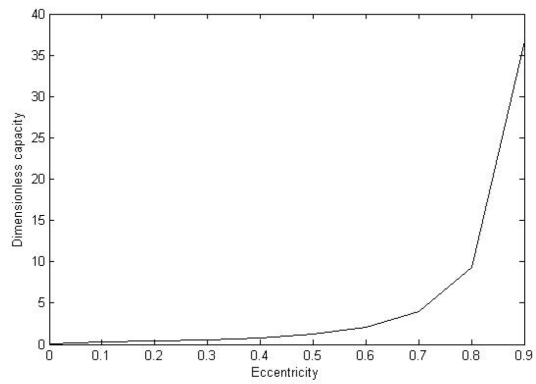

Figure 10 : Capacity varying with eccentricity

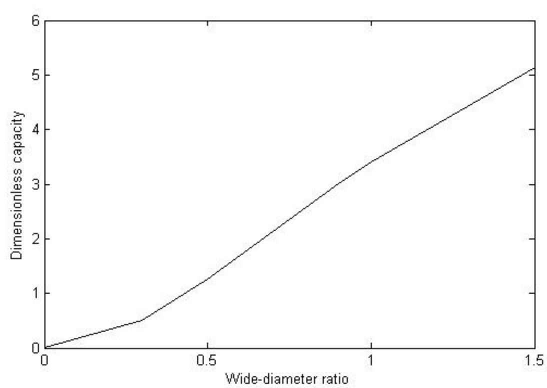

Figure 11 : Capacity varying with width to diamete

Analysis of the actual dimensional oil film pressure and bearing capacity

Compared with the dimensionless the oil film pressure and bearing capacity, the dimensional oil film pressure and bearing capacity should be multiplied by a corresponding coefficient, which is related to more factors. From the previous dimensionless process, the dimensional oil film pressure and bearing capacity should be $p = \frac{6P\mu\omega^*}{\psi^2}$ and $F = \frac{6F_x Rl\mu\omega^*}{\psi^2}$, respectively. The solving process of the inner film and outer film dimensionless pressure and bearing capacity are similar, but the coefficient multiplied to obtain the dimensional value are different, because it is concerned with the journal speed and floating ring speed. Furthermore, the inner film has greater bearing capacity than the outer film, because floating ring and journal have the same direction of rotation, the bearing capacity of the inner film is determined by the sum of the journal speed and floating ring speed, while the bearing capacity of the outer film only depends on the floating ring speed.

CONCLLUSION

This paper discusses the features of the inner and outer oil film pressure of floating ring bearing in the steady dynamic state through the finite difference method. Research indicates that the inner and outer oil film pressure at the steady dynamic state is superposed by the pressure of the rotation effect and squeeze effect. The oil film pressure generated by the rotation effect approximated parabolic distribution and its value is relatively small; the oil film pressure generated by the squeeze effect is symmetrical distribution and its value is relatively large. The oil film pressure and the capacity which are related to the eccentricity and the width-diameter ratio periodically varied with time in a rotating period. The higher the eccentricity or the width-diameter ratio is, the larger the film pressure and the capacity are.

NOTATION

R −radius U −linear velocity h −oil film thickness p −oil film pressure μ −dynamic viscosity of lubricant. ω − angular velocity ε −eccentricity Φ −attitude angle c −radial clearance P −dimensionless oil film pressure H −dimensionless oil film thickness l −bearing length ψ −relative clearance e −eccentricity λ −dimensionless bearing length d −diameter

SUBSCRIPTS

i −inner film o −outer film b −bearing j −journal r −floating-ring

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