Thermodynamics and Stability Analysis of High-Speed Hydrodynamic Journal Bearing

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ABSTRACT: The thermodynamic model of the high-speed hydrodynamic journal bearing is established to analyze the influence of the temperature-viscosity effect on the oil film in aspects of dynamic characteristics and stability. The coupling governing equations of the bearing including the Reynolds equation, energy equation and viscosity-temperature equation are solved to obtain the pressure distribution, temperature distribution and viscosity distribution, according which the stiffness coefficients and damping coefficients are acquired. Then the stability analysis of the bearing is processed based on Routh-Hurwitz stability criteria. The result indicates that when temperature-viscosity effect is taken into account, the dynamic characteristics, threshold speed and threshold mass show distinct decrease, and the degree of which is positive relative to the rotating speed, the eccentricity and the aspect ratio.

Keywords: journal bearing; temperature-viscosity effect; dynamic characteristics; stability analysis

1 INTRODUCTION

With the development of modern industry, the demands for rotor-support system with high speed, low friction power loss and good stability is higher, so hydrodynamic journal bearings are widely applied for its excellent adaptability, high accuracy and long lifespan. As the research is further promoted, the thermal effect of the lubrication oil is discovered to have significant influence on the performance of hydrodynamic journal bearings.

Wand Ying-jia investigated the static characteristics of infinitely wide journal bearings theoretically under the coexistence state of fluid with laminar flow and turbulence considering the thermal effect of lubrication oil, and gave some meaningful conclusions after comparing the solutions with that are under pour laminar flow state and without taking the thermal effect into account [1]. Simos T E analyzed the characteristics of textured journal bearings using Finite Difference Method, obtained more realistic results by considering the thermal effect and studied the thermal effects in the hydrodynamic behavior of textured journal bearings [2]. Zhang Zhenshan proposed a THD model of plain journal bearings which involved the synthetic solution of the generalized Reynolds equation, three-dimensional energy equation and the heat conduction equations of the solids, and provided a series of results to show that the thermal boundary conditions have considerable effects on the THD analysis of plain journal bearings [3]. Arab Solghar A. investigated the thermo-hydrodynamic characteristics of journal bearings with two axial grooves by means of computational fluid dynamics technique, and the governing equations are discretized through using hybrid scheme and solved simultaneously through employing the SIMPLEX algorithm [4]. Li Yuansheng calculated the oil film pressure and temperature distribution of a journal bearing using the finite element method, obtained the dynamic coefficients, and concluded the effects of the film temperature on the dynamic performance [5].

In this article, dynamic characteristics and stability of high speed hydrodynamic journal bearings were investigated through considering the temperature-viscosity effect of the lubrication oil. The coupling governing equations was solved by using Finite Difference Method based on some suitable boundary conditions, and the influence of the temperature-viscosity effect of the oil film in different operating conditions is analyzed through a series of comparison results.

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2 THE BASIC STRUCTURE AND GOVERNING EQUATIONS

The basic structure of high speed hydrodynamic journal bearings is shown in Figure 1:

![Bearing structure diagram](image)

Figure 1. Bearing structure diagram.

2.1 Reynolds equation and boundary conditions

Assuming that the lubrication oil is incompressible Newton fluid and the axis of the rotor is in parallel with the axis of the bearing, the dimensionless dynamic Reynolds equation is described as Equation 1:

\[
\frac{\partial}{\partial \varphi} \left( \frac{\partial p}{\mu \partial \varphi} \right) + \left( \frac{d}{T} \right)^2 \frac{\partial}{\partial \lambda} \left( \frac{\partial p}{\mu \partial \lambda} \right) = \frac{\varepsilon \varepsilon}{\partial T} + 6(\varepsilon \cos \varphi + \varepsilon \sin \varphi) \tag{1}
\]

Here:

\[
\tilde{p} = \frac{p - \mu \cdot r}{\mu}, \quad \varphi = r \cdot \varphi, \quad \lambda = \frac{z}{1/2} (-1 \leq \lambda \leq 1),
\]

\[
p_0 = \frac{2 \mu}{\varepsilon}, \quad \varepsilon = e^r, \quad \varepsilon = \frac{\varepsilon}{r}, \quad \varphi = \frac{\varepsilon}{e}.
\]

The Reynolds boundary conditions used here is stated as Equation 2:

\[
\begin{align*}
\lambda &= \pm 1, \quad \varphi = 0, \quad \varphi = 0, \quad \varphi = 0, \quad \varphi = 0, \quad \varphi = 0 \tag{2}
\end{align*}
\]

\[
\begin{align*}
\frac{\partial p}{\partial \varphi} &= 0, \quad p = 0
\end{align*}
\]

2.2 Energy equation and the boundary conditions

Energy equation of the oil film in thermal insulation flow condition is stated as Equation 3 [7]:

\[
A \left( \frac{p}{h} \right) + \frac{A}{3} \left( \frac{\partial T}{\partial \varphi} \right)^2 \left( \frac{\partial T}{\partial \lambda} \right)^2 = \frac{\partial}{\partial \varphi} \left[ \frac{\partial p}{\partial \varphi} \right] + \left( \frac{d}{T} \right)^2 \frac{\partial}{\partial \lambda} \left( \frac{\partial p}{\partial \lambda} \right) \tag{3}
\]

Here:

\[
A = 6 \mu_0 \Omega r^2 \left( T_r \rho c, e^2 \right), \quad T_r = \mu_0 \Omega \left( \rho c, e^2 \right), \quad T = T_r T
\]

The boundary condition at the starting position of the oil film used here is as Equation 4:

\[
\varphi = 0, T = T_r, \mu = \mu_0. \tag{4}
\]

2.3 Viscosity-temperature equation

Reynolds viscosity-temperature is applied here and stated as Equation 5:

\[
\mu = \mu_0 e^{-\alpha(T - T_0)} \tag{5}
\]

Where \(\alpha\) is viscosity-temperature susceptibility.

2.4 Oil film thickness equation

The oil film thickness equation is given as Equation 6:

\[
\tilde{h} = \frac{h}{c} = 1 + \varepsilon \cos \varphi \tag{6}
\]

Here: \(\varepsilon = e \).

3 STABILITY ANALYSIS OF THE JOURNAL HYDRO-DYNAMIC BEARING

Finite difference method is used to solve the above governing equations, and programs in C language are written to perform the iteration calculation, then the pressure value, the temperature value and viscosity value of each node can be acquired. Next disturbance equations are solved to acquire the disturbance pressure distributions, according which the dimensionless stiffness and damping coefficients \(k_x, k_y, k_x, k_y, k_x, k_y, b_x, b_y, b_z\), and \(\bar{b}_x, \bar{b}_y, \bar{b}_z\), can be computed with integral method. Assuming the mass of the rotor which is \(2m\) dimensionless mass of the rotor is \(2M\), for the one-mass symmetric stiffness rotor, the necessary condition for this system is given as Equation 11 based on Routh-Hurwitz stability criterion:

\[
\bar{b}_x + \bar{b}_y > 0 \tag{7}
\]

\[
M \left( \bar{k}_{xx} \bar{k}_{yy} + \bar{k}_{xy} \bar{k}_{yx} - \bar{k}_{xy} \bar{k}_{yx} \right) - \bar{k}_{xx} \bar{k}_{yy} - \bar{k}_{xy} \bar{k}_{yx} > 0 \tag{8}
\]

The threshold speed of the oil film is expressed as Equation 8:

\[
\Omega_{st} = \frac{\mu \bar{F}_{eq}}{m \varphi^3 \varphi^2} \tag{9}
\]

In which:

\[
\bar{F}_{eq} = \frac{\bar{k}_{xx} \bar{b}_x + \bar{k}_{xy} \bar{b}_y - \bar{k}_{yx} \bar{b}_x - \bar{k}_{yy} \bar{b}_y}{\bar{b}_x + \bar{b}_y}
\]
\[ \gamma^2 = \left( \frac{\omega}{\Omega} \right)^2 = \frac{(\bar{k}_{yy} - \bar{k}_{xx})(\bar{k}_{yy} - \bar{k}_{yx})}{\bar{b}_{yy} \bar{b}_{yy} - \bar{b}_{xy} \bar{b}_{yx}} \] (10)

Nomenclature:
- \( r \) — radius of bearing (mm);
- \( l \) — width of bearing (mm);
- \( e \) — eccentricity (mm);
- \( \varepsilon \) — eccentricity ratio;
- \( h \) — thickness of oil film (mm);
- \( \varphi \) — angle measured from the position of the thickest oil film (rad);
- \( n \) — whirling frequency on threshold speed \( (s^{-1}) \);
- \( \Omega \) — journal rotational speed \( (s^{-1}) \);
- \( \psi \) — relative clearance;
- \( \rho \) — density of lubrication oil \( (Kg/m^3) \);
- \( \mu \) — viscosity of lubrication oil \( (Pa \cdot s) \);
- \( \mu_0 \) — initial viscosity of lubrication oil \( (Pa \cdot s) \);
- \( \alpha \) — temperature-viscosity susceptibility \( (K^{-1}) \);
- \( k \) — stiffness coefficient;
- \( b \) — damping coefficient;
- \( m \) — mass (kg);
- \( M \) — dimensionless mass (kg);
- \( k_{eq} \) — equivalent stiffness of the oil film \( (N/m) \);
- \( \gamma_\mu \) — threshold whirl ratio.

4 AN EXAMPLE FOR ANALYZING THE PERFORMANCE OF A HYDRODYNAMIC JOURNAL BEARING

The statistics in Table 1 indicates the structure parameters and operation parameters of the bearing, as shown in Figure 1.

<table>
<thead>
<tr>
<th>Table 1. Bearing structure parameters and operation parameters.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Radius of bearing</td>
</tr>
<tr>
<td>Width of bearing</td>
</tr>
<tr>
<td>Radial clearance</td>
</tr>
<tr>
<td>Initial temperature</td>
</tr>
<tr>
<td>Temperature-viscosity susceptibility</td>
</tr>
<tr>
<td>Density of lubrication oil</td>
</tr>
<tr>
<td>Specific heat</td>
</tr>
<tr>
<td>Mass of the rotor</td>
</tr>
</tbody>
</table>

4.1 Temperature distribution analysis

Figure 2 shows the temperature distribution of the oil film when the rotational speed is 25,000r/min, the eccentricity ratio is 0.7 and the aspect ratio is 1.0. It can be seen that the temperature increase steadily in the circumferential direction from the start to the fracture edge, and peaks at the fracture edge. In the axial direction, the temperature in both ends shows slightly higher than the temperature in the middle part, but this trend is far less distinct than that of the circumferential direction.

4.2 Dynamic characteristics analysis

Figure 3 and Figure 4 show the upward trend of the dimensionless stiffness coefficients \( \bar{k}_{yy} \) and \( \bar{k}_{yx} \) with the eccentricity ratio rising up. When temperature-viscosity is taken into account, the stiffness coefficients decrease distinctly compared with the results of the isothermal method, and the gaps between them grow wider with the eccentricity ratio rising up. The faster the rotor spins, the greater the decrease is.
Figure 5 and Figure 6 show the increase of the dimensionless damping coefficients $\bar{b}_{xy}$ and $\bar{b}_{yx}$. Similar to the stiffness coefficients, when temperature-viscosity is considered, the damping coefficients decrease distinctly, and the differences between the results of the THD method and the isothermal method becomes greater with the eccentricity ratio rising and with the rotor accelerating.

![Figure 5. Dimensionless damping $\bar{b}_{xy}$](image)

![Figure 6. Dimensionless damping $\bar{b}_{yx}$](image)

It can be conveyed that when aspect ratio is invariable, the thermal effect of the lubrication oil makes the absolute value of the dynamic coefficients decrease, and the degrees of the decrease are positive relative to the eccentricity ratios and the rotating speeds.

![Figure 7. Dimensionless stiffness $\bar{k}_{xx}$ and $\bar{k}_{xy}$](image)

![Figure 8. Dimensionless damping $\bar{b}_{xx}$ and $\bar{b}_{xy}$](image)

Figure 7 illustrates the decrease of the absolute value of the dimensionless stiffness coefficients $\bar{k}_{xx}$ and $\bar{k}_{xy}$ with the rotor accelerating from 5,000 rpm to 25,000 rpm with the eccentricity being 0.5, and Figure 8 shows that of the dimensionless damping coefficients. As the aspect ratio rises from 0.5 to 1.0, the decrease shows a slight growth. These line charts demonstrate that the absolute value of dynamic coefficients is decreased with the rotor speeding up due to the influence of the temperature-viscosity effect, and the aspect ratio positively affects this decrease.

### 4.3 Stability analysis

Figure 9 illustrates the changing trend of the threshold speed of the oil film with the eccentricity ratio increasing from 0.1 to 0.7. When comparing the results of the THD method with that of the isothermal method, it can be seen that the threshold speed fall off for considering that the temperature-viscosity effect and the difference become greater as the eccentricity rising up. With the aspect ratio being 0.5, 0.8 and 1.0, the threshold speed increases distinctly and the difference for considering the temperature-viscosity effect also shows a remarkable growth. For example, as the eccentricity ratio being 0.6 and the aspect ratio being 1.0,
the threshold speed declines drop of 50% when the temperature-viscosity effect is taken account.

As shown in Figure 10, the threshold mass of the rotor increases with the eccentricity ratio increasing from 0.1 to 0.7, and the threshold mass decreases as the rotor speeds up due to temperature-viscosity effect. When the eccentricity ratio is greater than 0.6, all of the threshold masses increase extremely, and the difference converses when the eccentricity ratio is greater than 0.65.

Figure 11 describes the downward trend of the threshold mass with the eccentricity ratio being 0.5 and the rotor speeding up due to the temperature-viscosity effect of the lubrication oil. As the aspect ratio increases from 0.5 to 1.0, the rate of the decrease grows up slightly.

From these three line charts, it can be concluded that the decrease of the viscosity of the lubrication oil due to the temperature-viscosity effect causes a series of changes in aspect of Reynolds equation, pressure distribution, disturbance pressure distribution and the dynamic characteristics, based on which the stability performance decreases remarkably.

Figure 12 illustrates the drop of the threshold whirling frequency $\gamma_1$ with the eccentricity ratio growing from 0.1 to 0.9 and the aspect ratio being 1.0. Compared with the results of isothermal method, the threshold whirling frequency shows a decrease when temperature-viscosity effect is taken account and this decrease becomes more distinct with eccentricity ratio growing up and the rotational speed increasing.

Figure 13 presents the growth of the dimensionless equivalent stiffness of the oil film with eccentricity ratio growing from 0.1 to 0.9 and the aspect ratio being 1.0. When temperature-viscosity effect is considered, the growth becomes slower, and the faster the rotor spins, the slower the growth is.

5 CONCLUSION

Dynamic characteristics and stability of high speed hydrodynamic journal bearing was investigated in this article, and several conclusions can be drawn as follows:

(1) Dimensionless dynamic coefficients of the oil film decreases due to temperature viscosity effect, the difference becomes greater as the rotor speeds up, the eccentricity ratio increases and the aspect ratio rises up.

(2) When temperature viscosity effect is considered, the equivalent stiffness, the threshold speed, the mass and the whirl ratio of the rotor all shows distinct decrease. With the rotor speeding up, the eccentricity ratio is rising, and the aspect ratio is increasing, thus this decrease becomes gradually obvious.
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REFERENCES


