Influence of velocity slippage on nonlinear shaft center trajectory of journal bearing

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Abstract. Reynolds equation considering boundary slippage was used to calculate the oil film pressure, and Euler method was used to establish the motion equation of the axis in the coordinate system to solve the acceleration, velocity and position coordinates of the axis. Due to the influence of velocity slippage, the oil film pressure increases with the increase of slippage length, thus increasing the oil film force, the vibration of the axis and aggravating the instability of the axis, but decreasing the rotational speed of vortex; the shaft center trajectory gradually moves down to reduce the maximum external bearing load. When the length-diameter ratio becomes large, the effect of slippage length on shaft center trajectory is gradually reduced.

1. Introduction

With the improvement of research technology, it has been confirmed that under high speed and ultra-high speed working conditions, the shear stress will exceed the limit shear stress and take place slippage on the solid-fluid surface. Guo F [1] et al. found that the slip theory model can be divided into slip length model and ultimate shear stress model. Shaft center trajectory, a more intuitive way to reflect the instantaneous motion and stability of the spindle, is always a decisive role in service life and reliable performance of hydrodynamic lubricated journal bearing. Ma Jinkui [2] proposed a method to calculate the position of nonlinear the shaft center trajectory.

In terms of solving the transient flow field, K.P.Gertzos [3] et al. calculated the pressure distribution of sliding bearing under dynamic load by adopting dynamic grid technology. Adatep H [4] studied the tribological properties of grooveless and micro-grooved bearings under dynamic loads. By conducting experiment on test bed and numerical calculation through various numerical methods, and obtain a large number of calculation results of friction, friction coefficient, axial trajectory and film thickness. Lin Qinyin [5] et al. studied the influence of groove on bearing performance at high speed by using the Shaft center trajectory at different groove depths. Krupka [6] et al. performed a study on the effect of surface texture on lubrication film formation of hydrodynamic lubricated contact under transient conditions. Medina [7] et al. research the transient characteristics of dynamic pressure sliding bearing with surface texture. Zhang YB [8] analyzed that the fluid film slippage at the stationary contact surface in the fluid inlet zone can generate the hydrodynamic load-carrying capacity between two parallel plane surfaces. Chen Dongjiu [9] extended the research of hydrostatic bearing to the micro scale, and further applied the dynamic characteristic coefficient to the modeling of the shaft center trajectory of the hydraulic spindle.
and analyzed the motion state of the spindle axis while solving the dynamic characteristic coefficient. But research of transient characteristics considering slip is still lacking.

In this research, slip length model was selected. Due to the oil film force varies with the mesh density, the optimal mesh grid number is selected considering the calculation accuracy and time. The slip length is set to 0, 5, 10, 15 and 20μm respectively to obtain the shaft center trajectory. And by using control variable method, the influence of slip length, rotational speed, length-to-diameter ratios, external load and clearance ratio on the shaft center trajectory is being calculated and analyzed.

2. Calculation model of bearing

2.1. Geometric model

When the external load is applied, the rotor axis and the bearing center no longer coincide, as show in Figure.1

![Figure 1. geometric model of bearing](image)

2.2. The Reynolds equation with boundary slippage

Considering the influence of slippage velocity, the Reynolds equation is modified with the velocity of the contact surface between bearing bush and oil film as the boundary condition. The dimensionless Reynolds equation is given as:

\[
\frac{\partial}{\partial \phi} \left( H^2 \left( 1 + 6 \frac{B}{H} \frac{\partial P}{\partial \phi} \right) \right) + \left( \frac{r L}{H} \right)^2 \frac{\partial}{\partial \lambda} \left( H^2 \left( 1 + 6 \frac{B}{H} \frac{\partial P}{\partial \lambda} \right) \right) = 6 \frac{\partial H}{\partial \phi} + 12 \frac{\partial H}{\partial \tau}
\]

(1)

The oil film thickness changes with time can be expressed as:

\[
\frac{\partial H}{\partial \tau} = \dot{X} \sin(\phi) + \dot{Y} \cos(\phi)
\]

(2)

2.3. Calculation model of film force

For bearings of limited width, the fluid film forces are given by:

\[
F_e = \frac{F_e}{\eta_0 \omega L} = \int_0^{2\pi} \left( \int_{-0.5}^{0.5} P \, d\lambda \right) \sin \phi \, d\phi
\]

(3)

\[
F_\eta = \frac{F_\eta}{\eta_0 \omega L} = \int_0^{2\pi} \left( \int_{-0.5}^{0.5} P \, d\lambda \right) \cos \phi \, d\phi
\]

(4)

2.4. Nonlinear trajectory equation

The equation of equilibrium force is established according to Newton’s second law. \( \gamma = \frac{\eta L}{M \omega_0 \omega} \), \( \delta = \frac{g}{\omega_0 \omega^2} \). The equilibrium force equation can be written as the following relations:

\[
\ddot{X} = \gamma F_e + \dot{q}_x
\]

(5)
\[ \ddot{Y} = \gamma F + \bar{q}_y + \delta \]  

Shaft center trajectory is determined by the axis position parameters, assuming that the time interval between two points is $\Delta \tau = \pi/200$, the Euler method is used to calculate the next position parameters.

3. Computational nonlinear shaft center trajectory

3.1. Basic parameter

| Parameter             | Symbol | Unit | Value  |
|-----------------------|--------|------|--------|
| Bearing Length        | L      | m    | 0.024  |
| Bearing diameter      | D      | m    | 0.06   |
| Radius clearance      | C      | m    | 1.0×10^{-4} |
| The quality of rotor  | M      | Kg   | 10     |
| Lubricating of viscosity | $\eta$ | Pa·s | 0.019  |
| Lubricating of density | $\rho$ | kg/m³ | 880    |

3.2. Nonlinear shaft center trajectory calculation steps

When calculating the oil film force, the axis always satisfies the balance equation of the force. Since the axis does not reach stability, the acceleration of the axis is always changing. Axis position parameters $X, Y, \dot{X}, \dot{Y}$ are continuously modified until the equilibrium condition is reached.

3.3. Mesh-density

By setting different initial positions for calculation comparison, it can be seen that the initial position parameter does not affect the final calculation result, but the four parameters ($X, Y, \dot{X}, \dot{Y}$) cannot be equal to 0 at the same time. Number of divisions have great influence on the accuracy and time of calculation. It can be seen from the equilibrium force equation and calculation procedure that the change of oil film force will have an impact on the location parameters of the axis, so it is reasonable to analyze only the change of oil film force. The changes of horizontal oil film force and vertical oil film force with time are calculated, as shown in Figure 2 and Figure 3.

![Figure 2. The horizontal oil film force](image1)

![Figure 3. The vertical oil film force](image2)

It can be seen from the figure that the instability of oil film force increases with the increase of mesh density. When the number of grids is $400 \times 150$, the change of oil film force is significantly increased, the convergence is slow, and the calculation time is significantly longer. However, too few meshes will decrease the calculation accuracy. So, the fluid model is defined with $100 \times 60(\varphi, \lambda)$ divisions.

3.4. Result

3.4.1. Calculation examples under specific working conditions. The characterizes parameters are calculated at the speed of 9000 rpm, the slip length is 10μm and the vertical external load is 400N.
According to the variation diagram of oil film force with time in Figure. 2 and Figure. 3, when the axis is stable at a certain position, the horizontal force at this point is 0, and the vertical force keeps balance with the gravity. Figure. 4 and Figure. 5 show that the velocity and acceleration are both 0. The oil film pressure distribution is shown in Figure 6.

3.4.2. The effect of slippage length. The slippage is defined with 0, 5, 10, 15, 20μm respectively. Nonlinear shaft trajectory with different slippage length is shown in figure 8. With the increase of slippage length, the shaft center trajectory fluctuation of the shaft increases, and the final equilibrium position of the axis is far away the center of the clearance circle.

To ensure the reliability of rotor rotation, the trajectories should always be within the clearance circle. It may be clearly seen from figure 8 that, all 5 trajectories meet the requirements. According to the results of figure 9, with the increase of slip length, eccentricity significantly increases and the fluctuation of the shaft trajectory increases. Figure 10 implies that the pressure of dimensionless oil film increases and the convergence time increases with the increase of slip length.
3.4.3. The influence of speed on nonlinear shaft center trajectory. The influence of rotational speed on nonlinear shaft trajectory is analyzed. The nonlinear shaft center trajectories are presented in Figure 12.

![Figure 11. Eccentricity changes with time](image)

**Figure 11.** Eccentricity changes with time

**Figure 12.** Nonlinear trajectories with different rotation speed

As can be seen from figure 12, with the increase of speed, the vibration of the shaft increases and becomes more disorderly. The time to converge to the stable point becomes longer. When the speed increases to a certain value, the shaft center trajectory will no longer converge to a point. When slip length is 10μm, the figure 11 shows that the eccentricity ratio of the journal decreases with the increase of the speed, because the hydrodynamic effect is enhanced with the increase of the speed. It can also be known that the eccentricity will oscillate when the speed reaches 16800r/min. When the speed is exceeded, the nonlinear trajectory of the shaft will no longer be stable, an approximate elliptic trajectory will be formed instead. With the increase of slippage length, the rotational speed of the track vortex decreases gradually.

3.4.4. Influence of length-to-diameter ratios on nonlinear shaft center trajectory. Length-to-diameter ratios is an important parameter of journal bearing. In this paper, the length-to-diameter ratios is 0.4, 0.6, 0.8, 1.0 and 1.2 respectively. Figure 13 shows the nonlinear shaft trajectories at different length-to-diameter ratios.

![Figure 13. Trajectories with different length-to-diameter ratios](image)

**Figure 13.** Trajectories with different length-to-diameter ratios **Figure 14.** Eccentricity changes

The figure 13 shows that the axis gradually spirals above the center of the circle until it converges to a stable position, and the time to reach the equilibrium position significantly increases with the increase of length-to-diameter ratios. At the same time, the result in Figure 14 shows that eccentricity ratios decrease and the influence of slip length on the shaft center trajectory will decrease.
3.4.5. *Influence of external loads on nonlinear shaft center trajectories.* The comparison of nonlinear shaft center trajectory under four external load is illustrated in Figure 15.

![Figure 15. Nonlinear trajectories with different external load](image)

As shown in Figure 15, with the increase of the vertical load, the journal force decreases and the radial motion intensifies, the position of the stable also moves down gradually, and the time to reach the equilibrium position is shortened. By comparing different slippage lengths, the shaft center trajectory gradually moves down to reduce the maximum external bearing load with the increase of slippage length.

3.4.6. *Influence of clearance ratio.* The size of clearance will affect lubricity as well, thus affecting the stability of movement. Therefore, the range of the clearance ratios is subject to the slip length. When the slip length is 10μm, the shaft center trajectory is calculated as shown in figure 16.

![Figure 16. Nonlinear trajectories with different clearance ratio](image)

In figure 16, with the increase of clearance ratio, the bearing capacity of the oil film is reduced. Thus the eccentricity increased. As the clearance increases, vortexes increase during operation, which makes the oil film unstable and leads to the instability of the trajectory. At the same time, excessively thin oil film thickness cannot form a complete lubricating layer.

4. **Conclusion**

Under the conditions in this article:

1. Mesh-density has a great impact on the calculation accuracy and time. When the number of grids is more than 100×60, the oil film force will significantly increase.
2. Because of the influence of slip, the oil film pressure increases with the increase of slip length, which increases the oil film force, and the vibration of the shaft increases the instability of the shaft.
3. With the increase of rotating speed, the hydrodynamic effect is increased and the eccentricity rate is reduced. When the rotational speed reaches a certain value, it will not converge at a stable position, and vortex will occur. With the increase of slippage length, the rotational speed of vortex decreases.
4. The increase of length-to-diameter ratios will spiral and will increase the time to converge obviously. However, when the ratios of the bearing length and diameter become large, the effect of slippage length on shaft center trajectory is reduced gradually.
5. With the increase of the external load, the equilibrium position of the shaft moves down gradually, and moves down also with the increase of slippage length.

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