INFLUENCE OF FIT CLEARANCE ON THE STABILITY OF “THREE OIL FILM-ROTOR” STRUCTURE

Wang, Y.^{*,#}; Yang, H. Y.^{*}; Chen, G.^{*} & Jia, Y. J.^{**}

^{*} School of Mechanical Engineering, Xihua University, Chengdu 610000, China

^{**} Pipechina Southwest Pipeline Company, Chengdu 610000, China

E-Mail: wangyu@mail.xhu.edu.cn (# Corresponding author)

Abstract

During the rotation, the stability of the shaft changes with the fit clearance between the bearing and the journal. Taking the “three oil film-rotor” structure as the object, this paper systematically explores the variation in oil film pressure, oil film thickness, and shaft axis orbit, when the shaft is subject to transient impact load and oil film cavitation, based on the Reynolds equation under three-dimensional (3D) conditions. The results show that: The intermediate oil film of the “three oil film-rotor” structure had the greatest impact on the carrying capacity of the system; the most significant variation in oil film thickness was observed, when the pressure on the intermediate oil film reached the peak; the fit clearance between the bearing and the journal suppressed the surface pressure on oil film, and intensified the deformation of oil film. The research findings lay the theoretical basis for how the fit clearance of the lubrication system affects system stability.

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Key Words: “Three Oil Film-Rotor” Structure, Fit Clearance, Oil Film Pressure, Oil Film Thickness, Axis Orbit

1. INTRODUCTION

The bearing is an important and popular part in various machinery. For high-speed rotating machinery, the journal bearing has been widely used as supporting components. Lubricating the bearing can reduce the resistance loss and improve the cooling effect of high-speed rotating machinery. Thus, bearing lubrication is often adopted to enhance the stability and safety of mechanical rotation [1-4].

The stability of the mechanical rotation system is critical to the entire transmission system. Fit clearance is commonly used between the journal and the bearing to store lubricating oil and to compensate for various errors. The size of fit clearance directly bears on the degree of movement of the shaft. To promote the application of journal bearing, it is of far-reaching significance to examine how the fit clearance affects the stability of the rotor system.

Recent years has seen abundant research on the variation in various parameters of the oil film in the rotor system. In particular, researchers have been constantly improving the calculation method for the dynamic properties of the oil film in the sliding bearing, creating many ways to evaluate the stability of the rotor-lubrication system. Taking the entire rotor system as a nonlinear system, Zhang and Zhu provided a numerical solution for the nonlinear analysis of the oil film-rotor system and defined the stability margin of the rotor system [5, 6]. Some scholars treated the rotor-lubrication system as a linear system and judged whether the whole system is stable by the critical speed or the relationship between internal and external resistances. Peng et al. argued that the instability of the sliding oil film in the rotor does not always tend to be unstable with the increase of speed but has a stable area in the change process [7, 8].

During the rotation of the oil film, there will inevitably be a cavitation zone. The simplest way to determine the cavitation zone is the Sommerfeld’s solution to Reynolds equation, which assumes that the cavitation zone falls in the negative pressure zone [9-11]. Based on this method,

some scholars averaged the two phases of oil and gas into one phase and used the improved Reynolds equation to analyse the cavitation zone in the oil film [12, 13]. As for the unbalanced response of the rotating shaft induced by cavitation, the general research method is to judge the stability change of the system based on the instantaneous change of the shaft axis orbit, especially the oscillation amplitude and convergence speed [14-16].

The above studies have explored the stability of the oil film-rotor system in various ways. Most of them attempt to calculate the whirling motion, thickness, and pressure of the oil film. However, relatively few scholars have investigated the variation in shaft stability induced by fit clearance. This research discovers that fit clearance has a great impact on oil film pressure, oil film thickness, and shaft axis orbit, and thus severely affects the service life of the shaft. The discovery highlights the urgency to mitigate the influence of the fit clearance on the oil film of bearing. The research findings enrich the evidence on the stability of the “three oil film-rotor” structure and provide a guide for the application of the structure.

2. NUMERICAL MODELLING

2.1 Governing equation

The Reynolds equation was adopted to calculate the distribution of oil film pressure. Without considering the effects of the volume force and area force in the oil, it was assumed that the pressure in the oil does not change along the film thickness direction, and the oil is an incompressible fluid. Then, the Reynolds equation can be constructed as:

$$\nabla_T \cdot \left(\frac{-\rho h^3}{12\mu} \nabla_{TP} + \frac{\rho h}{2} (v_a + v_b) \right) - \rho ((\nabla_T b \cdot v_b) - (\nabla_T a \cdot v_a)) = 0 \quad (1)$$

where, ρ is the density (kg/m^3); h is the thickness of the lubricant (m); μ is the dynamic viscosity of the fluid ($\text{Pa}\cdot\text{s}$); p is the pressure (Pa); a is the inner diameter of the film (m); v_a is the tangential speed on the inner diameter (m/s); b is the outer diameter of the film (m); v_b is the tangential speed on the outer diameter (m/s).

During the modeling, the rotating journal was regarded as a solid wall surface. Since the pressure was assumed to be constant along the thickness direction of the whole oil film, the pressure distribution on the lubricating surface was calculated with the tangent projection of the gradient operator ∇_T . Setting $\rho((\nabla_T b \cdot v_b) - (\nabla_T a \cdot v_a))$ to zero, the equation can be simplified as:

$$\nabla_T \cdot \left(\frac{-\rho h^3}{12\mu} \nabla_{TP} + \frac{\rho h}{2} (v_a + v_b) \right) = 0 \quad (2)$$

The wedge-shaped gap between the journal and the shaft is the main geometric condition for the formation of dynamic pressure by the oil film. In the cylindrical bearing, there exist a convergent wedge-shaped gap and an open wedge-shaped gap along the rotation direction. One hypothesis [17] believes that a complete oil film exists in each wedge-shaped gap, i.e., there is a full-peripheral oil film (Sommerfeld boundary condition); the other hypothesis holds that only the convergent wedge-shaped gap has a complete oil film, i.e., there is a half-peripheral oil film (semi-Sommerfeld boundary condition). The Sommerfeld boundary condition is adopted in this research. Under this boundary condition, negative pressure will appear in the calculation result on oil film pressure. In practice, the negative pressure should be ignored, and replaced with atmospheric pressure. This theory is based on the Jakobsson–Floberg–Olsson (JFO) cavitation theory. Considering the formation of cavitation, the fluid flow in the journal bearing can be divided into two zones:

(1) The entire oil film zone: the pressure will change in this area, but the pressure change is limited by the cavitation pressure.

(2) The cavitation zone: the air and the fluid mix in this zone; the pressure in this zone is assumed as the cavitation pressure.

2.2 Model construction

The “three oil film-rotor” structure was modelled as follows. The shaft was meshed into 150 grids by edge distribution. Fig. 1 shows the meshing results of the oil films No. 1-3.

Figure 1: Finite-element model.

Then, a uniform load was applied on the shaft to approximate the uneven mass distribution of the shaft. Besides, a transient impact load was imposed on the shaft to simulate the external impact on the shaft during rotation. Table I lists the material parameters used in the calculation and the relevant parameters of the bearing journal.

Table I: Model parameters.

Name	Symbol	Value
Rotor length	L	395 [mm]
Rotor diameter	D	26.09 [mm]
Young's modulus of rotor	E	2.05 E11 [Pa]
Density	ρ	7,800 [kg/m ³]
Poisson's ratio	ν	0.3
Bearing length	L_j	30 [mm]
Initial fit clearance	C	0.05 [mm]
Viscosity of lubricating oil	μ	0.019 [Pa's]
Angular speed	ω	1,200 [rpm]

3. RESULTS ANALYSIS

3.1 Influence of fit clearance on oil film pressure

The fit clearance at the journal was set to 0.05, 0.1, and 0.15 mm, respectively. Then, the pressure distribution on each oil film was calculated under each of the three fit clearances at the time of 0 s, 0.4 s, 0.8 s, 1.2 s, 1.6 s, and 2 s, in turn. Except for the calculation of the initial state, an external instantaneous impact load of 200 N was applied on any other example at 0.2 s. The impact load lasted 0.1 s (Fig. 2).

As shown in Fig. 2, the surface stress of the shaft was concentrated between oil films. The largest surface pressure appeared on oil film No. 2. Calculation shows that the three oil films at different positions of the bearing witnessed similar change trends of oil film thickness and

pressure, with only numerical differences in the two parameters. Therefore, only the calculation results of oil film No. 1 were extracted at different fit clearances (Figs. 3 to 5).

Figure 2: Surface stress of the shaft and surface pressure of oil films in the “three oil film-rotor” structure.

Figure 3: Time variation in the pressure of oil film No. 1 at the fit clearance of 0.05 mm (unit: MPa).

As shown in Fig. 3, when the shaft was static, an initial pressure was formed on the surface of the oil film under the action of gravity; the pressure was relatively small, and symmetrically distributed on the film surface. After the shaft started to rotate, the positive pressure zone and the negative pressure zone were concentrated at the contact between the shaft and the journal, and the surface pressure of the oil film stabilized at 0.4 MPa.

Figure 4: Time variation in the pressure of oil film No. 1 at the fit clearance of 0.1 mm (unit: MPa).

As shown in Fig. 4, the surface pressure of the oil film stabilized at 0.3 MPa at the fit clearance of 0.1 mm, down by 0.1 MPa from the stabilized surface pressure at the fit clearance of 0.05 mm. Compared with that at 0.05 mm, the positive and negative pressure zones on the oil film expanded at 0.1 mm.

Figure 5: Time variation in the pressure of oil film No. 1 at the fit clearance of 0.15 mm (unit: MPa).

As can be seen from Figs. 3 to 5, when the fit clearance was 0.05 mm, the oil film pressure was symmetrically distributed in the radial direction, owing to the weight of the shaft in the initial state; after the shaft started to rotate, the pressure on the oil film increased and then remained stable; with the growth in fit clearance, the positive and negative pressure zones on the oil film gradually moved to the two sides, and the pressure in the negative pressure zone was the atmospheric pressure. It can be seen from the figures that the negative pressure zone expanded, while the oil film surface pressure declined.

Figure 6: Distribution of oil film pressure at different fit clearances.

As shown in Fig. 6, the pressure on the oil film surface was unevenly distributed along the circumferential and axial directions, while the positive and negative pressure zones were symmetric on the surface of the oil film; when the fit clearance was 0.05 mm, the oil film surface pressure stood at 0.08 MPa; after the fit clearance increased to 0.1 mm, the oil film surface pressure dropped to 0.015 MPa; the smaller the fit clearance, the larger the oil film surface pressure; in this case, a very narrow oil wedge appeared between the shaft and the journal; a very high pressure would appear near the oil wedge, pushing up the damage of the shaft.

Figure 7: Variation in oil film surface pressure at 0.2-0.3 s (unit: MPa).

As shown in Fig. 7, under the impact load, the three oil films differed in the distribution and magnitude of pressure. The highest surface pressure of 0.6 MPa appeared on oil film No. 2, while the lowest surface pressure of 0.2 MPa was observed on oil film No. 3. After the impact load was over, the surface pressure decreased by 0.2 MPa on oil film No. 1, declined by 0.3 MPa on oil film No. 2, and remained unchanged on oil film No. 3. This means the pressure on the surface of the intermediate oil film will increase when the rotor shaft is subject to an external impact load. In the actual operation of the shaft, the stress on bearing at the middle of the “three oil film-rotor” system changes the most significantly, under the instantaneous impact load. In other words, when the shaft is subject to instantaneous impact load, the intermediate oil film is most likely to be damaged.

The research results show that the oil film under the fit clearance of 0.05 mm can provide a relatively large support force during high load operation, but the shaft also receives a large reaction force from the oil film and a large stress on itself; the oil film stress is relatively small under the fit clearance of 0.15 mm during low load operation, resulting in a relatively long service life of the shaft.

3.2 Influence of fit clearance on oil film thickness

Based on the above model, the authors continued to analyse the influence of the fit clearance on the thickness distribution of the oil films. Figs. 8 to 10 present the thickness variation of oil film No. 1 under the fit clearance of 0.05, 0.1, and 0.15 mm, respectively, within the calculation time of 2 s.

Figure 8: Time variation in the thickness of oil film No. 1 under the fit clearance of 0.05 mm (unit: μm).

As shown in Fig. 8, at the initial moment, the oil film thickness was unevenly distributed in the direction of gravity; after the shaft started to rotate, the oil film thickness peaked at $65\ \mu\text{m}$, and minimized at $35\ \mu\text{m}$, suggesting that the shaft rotation does not affect the thickness distribution of the oil film.

Figure 9: Time variation in the thickness of oil film No. 2 under the fit clearance of 0.1 mm (unit: μm).

As shown in Fig. 9, the initial thickness of the oil film peaked at $100\ \mu\text{m}$ under the fit clearance of 0.1 mm, up by $50\ \mu\text{m}$ from that under the fit clearance of 0.05 mm; after the shaft started to rotate, the peak thickness of the oil film was $160\ \mu\text{m}$, an increase of $95\ \mu\text{m}$, while the minimum thickness was $40\ \mu\text{m}$, an increase of $5\ \mu\text{m}$.

Figure 10: Time variation in the thickness of oil film No. 1 under the fit clearance of 0.15 mm (unit: μm).

As can be seen from Figs. 8 to 10, the thickness distribution of oil film No. 1 changed as the fit clearance varied between the three values. In the “three oil film-rotor” system, the thickness of the oil film was unevenly distributed along the circumferential direction of the oil film. Since the fit clearance is adopted at the shaft journal, the shaft became eccentric in the direction of gravity at the initial moment, and the shaft axis moved downward along the gravity direction, making the thickness distribution of the oil film uneven in that direction. After the shaft started to rotate, the oil film thickness distribution changed, and decreased at the position of a large oil film pressure. Once the shaft rotation stabilized, the oil film thickness no longer changed.

Figure 11: Oil film thickness distribution under different fit clearances.

As shown in Fig. 11, the oil film was thick in the middle and thin on the ends in the circumferential direction of the entire journal; with the growth in fit clearance, the peak thickness of the oil film continued to increase; the thickness of the oil film changed with the oil film surface pressure: the film was thin under a high pressure.

Figure 12: Thickness variation in oil films from 0.2 s to 0.3 s (unit: μm).

As shown in Fig. 12, when the external impact load disappeared at 0.3 s, the peak thickness of oil film No. 1 decreased by $5\ \mu\text{m}$, that of oil film No. 2 decreased by $10\ \mu\text{m}$, and that of oil film No. 3 decreased by $5\ \mu\text{m}$. Oil film No. 2 witnessed the most significant thickness change. When the shaft was subject to external impact load, oil film No. 2 was easy to oscillate, adding to the risk of shaft operations.

The research results show that the thickness of oil film No. 2 changes most drastically, when the “three oil film-rotor” system is subject to an impact load; this oil film plays a critical role in the stability of the shaft. If the shaft rotation needs to be highly stable, the fit clearance between the shaft and the journal should be minimized.

3.3 Influence of fit clearance on axis orbit

The axis orbit was observed by changing the fit clearance between the shaft and the journal from 0.05 mm, 0.1 mm, to 0.15 mm. The shaft vibration from the start to the stabilization of rotation could be obtained from the acceleration response curves of the shaft axis in the axial

direction (y-direction) and the radial direction (z-direction). Thus, the variations of axis orbit, y-direction acceleration, and z-direction acceleration of the shaft within 2 s were calculated. To clearly illustrate the acceleration response of the shaft under impact load, the acceleration response curves in the first 0.3 s were extracted and plotted into Fig. 13.

Figure 13: Acceleration response curves of the shaft under three fit clearances.

As shown in Fig. 13, from 0 s to 0.1 s, the initial vibration produced in the startup of the shaft increased with the fit clearance; the first vibration of the shaft was very weak, which intensified with the growth in fit clearance. From 0.1 s to 0.2 s, the shaft rotation caused changes to oil film pressure; thus, the gas escaped from the oil film, leading to the second vibration of the shaft; the magnitude of the second vibration is positively correlated with the fit clearance. From 0.2 s to 0.3 s, the shaft underwent the three vibration due to the external impact load; with the increase in the fit clearance, the acceleration responses of the shaft weakened, and the vibration time shortened.

Shaft vibration could affect the axis orbit. To disclose the impacts of impact load and fit clearance on shaft axis, the authors calculated the shaft axis orbit, and presented it as Figs. 14 and 15.

Figure 14: Variation in shaft axis orbit under the fit clearance of 0.05 mm.

As shown in Fig. 14, starting from the start position, the axis was under the action of load at any moment, as the shaft continued to rotate. When the load at the journal changed, a new equilibrium position would appear for the axis. In this way, a series of equilibrium positions formed the axis orbit during shaft rotation. At 0.2 s, the shaft was subject to the impact load, causing changes to the equilibrium position of the axis; once the impact load ceased, the axis converged to a point under the joint effect of oil film pressure and gravity.

After the shaft started to move from the point (0, 0), the oil film pressure at the journal changed as the shaft rotated; under the joint action of shaft gravity and oil film pressure, the axis revolved around its equilibrium position.

Figure 15: Variation in shaft axis orbit under the three fit clearances.

As shown in Fig. 15, the impact load disrupted the original force equilibrium of the shaft, resulting in a shift of axis orbit. After the impact load disappeared, the shaft force resumed the equilibrium state, and the equilibrium position of the axis converged to a point.

On the influence of fit clearance on axis orbit, it can be seen that the fit clearance changed the convergence position and speed of the axis. Within the 2 s-long rotation, the axis converged at different positions under the three fit clearances. The smaller the fit clearance, the closer the axis converged to the initial position, the faster the axis convergence, and the better the convergence performance.

The research results show that, in the actual operations of the shaft, the shaft axis converges to a better position at a faster speed under a small fit clearance, which facilitates the rapid start of the shaft.

4. CONCLUSIONS

This paper explores how different fit clearances affect the properties of the oil films in the oil film-rotor system. The main conclusions are as follows:

(1) The change of fit clearance has a large impact on oil film pressure. The larger the fit clearance, the smaller the surface pressure of the oil film. Under the impact load, the intermediate oil film witnessed the sharpest pressure change, and boasted the largest surface pressure, suggesting that the external impact load can significantly enhance the surface pressure on the oil film.

(2) The increase of fit clearance will push up the peak thickness of the oil film. Under the impact load, the intermediate oil film saw the greatest peak thickness, which was distributed symmetrically along the circumferential direction; the greater the fit clearance, the larger the minimum thickness of the oil film, and the more violent the shaft vibration.

(3) Without any external force, the shaft axis would revolve around the rotation centre, under the joint action of the shaft gravity and oil film pressure; under the external impact load, the equilibrium position of spindle axis would change, and move along the direction of the impact load. After the impact load disappears, the axis would eventually converge to a point after a short period of oscillation. The larger the fit clearance, the slower the convergence of the axis, and the further the convergence position is from the initial position.

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