The Effect of Journal Misalignment on the Lubrication Performance of Partially Textured Journal Bearing with Elastic Deformation

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Abstract. In order to analyse the effects of journal misalignment and elastic deformation on the lubrication performance of partially textured journal bearing, an elastic-hydrodynamic lubrication (EHL) model was specifically established based on the steady-state Reynolds equation. Through numerical calculation, the effects of elastic deformation, degree of misalignment and misalignment angle on load capacity and overturning moment were investigated. The results show that when the degree of misalignment is large, the elastic deformation cannot be ignored. With the increase of degree of misalignment, the load capacity and overturning moment are increased. With the increase of misalignment angle, the bearing performance variations are complicated. When the misalignment angle is equal to 90°, there are two pressure peaks in the film pressure. The load capacity is the lowest, and overturning moment is the largest. The optimized partial texturing is still beneficial to the performances of bearing when journal becomes misaligned, which is attributed to the additional hydrodynamic lift.

1. Introduction

Journal bearings are key parts of rotor systems, which are widely used in the fields of pump, motor and compressor. Their lubrication performance directly affects the reliability of the whole system. Surface texturing is considered to be an effective approach to enhance the tribological performance of sliding contact elements [1, 2]. Therefore, researchers introduce surface texture into bearings to improve their performance. For example, Brizmer and Kligerman [3] performed a dimensionless parametric study on textured journal bearings. They found that full texturing is detrimental and the partial texturing can improve load capacity at low eccentricity. Gong et al. [4] built an EHL model to analyze the performance of micro-groove bearings, and found that the effects of the micro-grooves on the performance of bearings are little for the smaller Young’s modulus. Gropper [5] et al. gave a brief overview of the key findings and some guidelines on texture design. All of above studies were carried out under the condition of journal alignment. In practice, the shaft becomes misaligned due to manufacturing tolerances, assembly errors and asymmetric bearing load, etc. Sun and Gui [6] pointed out that misalignment affects the bearing performance. Gu [7] et al. studied the performance of bearing with misalignments and textures. Their results showed that along with the increased deflection angles, the effect of texturing on the tribological performance of journal bearing would turn beneficial into harmful. Xiang [8] et al. found that the misalignment direction angle will result in the decreased
load capacity of the micro-grooved bearing. Manser [9] et al. investigated the textured surface and journal misalignment on the performance of hydrodynamic journal bearing. However, when journal misalignment and elastic deformation are considered, whether the optimized texturing parameters can improve the bearing performance has not been reported.

In this research, an EHL model of partially textured journal bearing is developed and validated. Based on the model, the influence of journal misalignment and elastic deformation on the performance of partially textured bearings is analyzed.

2. The model

2.1. Film thickness

The geometry of a misaligned journal bearing is shown in Fig. 1 (a). The film thickness in journal bearing considering journal misalignment can be expressed as

\[ h_0 = c + e_0 \cos(\theta - \varphi_0) + e' \left( \frac{z}{L} - \frac{1}{2} \right) \cos(\theta - \alpha - \varphi_0) \] (1)

where \( c \) is radial clearance, \( e_0 \) is the eccentricity at the bearing axial mid-plane, \( \varphi_0 \) is the attitude angle, \( e' \) is the magnitude of the projected journal axis on the bearing mid-plane, \( \alpha \) is the misalignment angle between the line of centers and the rear center of the misaligned journal, \( L \) is the bearing width, \( \theta \) is circumferential angle.

The dimensionless film thickness considering journal misalignment can be described as

\[ \bar{h}_0 = 1 + e_0 \cos(\theta - \varphi_0) + e' \left( \frac{y}{L} - \frac{1}{2} \right) \cos(\theta - \alpha - \varphi_0) \] (2)

Where \( e_0 \) is eccentricity ratio, \( e' \) is the misalignment eccentricity ratio presented by [10]

\[ e' = \frac{e'}{c} = D_m e'_{\text{max}} \] (3)

Where \( D_m \) is dimensionless degree of misalignment, \( e'_{\text{max}} \) is the maximum possible value of \( e' \).

As shown in Fig. 1 (b), The geometric parameters of the dimple include radius \( r_p \), dimple depth \( h_p \), and area density \( S_p \), \( S_p = \pi \cdot r_p^2 / (L_1 \cdot L_2) \). The angle \( \theta_1 \) and \( \theta_2 \) are the angular position of texture.

![Figure 1. (a) A misaligned journal bearing, (b) texturing parameters for a journal bearing.](image-url)
In the Winkler elastic foundation model [11], the deformable object is modeled as a set of spring elements. The elastic deformation can be therefore expressed as

$$\Delta = \frac{pt}{E}$$  \hspace{1cm} (4)

Where \( p \) is pressure, \( t \) is the thickness of bearing bush, \( E \) is a combined elastic modulus [11].

The film thickness of a partially textured bearing considering journal misalignment and elastic deformation can be described as

$$h = \begin{cases} h_0 + h_p + \Delta & (x, y) \in \Omega \\ h_0 + \Delta & (x, y) \notin \Omega \end{cases}$$  \hspace{1cm} (5)

Where \( \Omega \) is the area occupied by texture.

2.2. Lubrication model

The pressure distribution of oil film for a bearing operated at steady-state conditions can be expressed as the following equation

$$\frac{\partial}{\partial x} \left( \frac{h^3}{12\eta} \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial y} \left( \frac{h^3}{12\eta} \frac{\partial p}{\partial y} \right) = \frac{1}{2} U \frac{\partial h}{\partial x}$$  \hspace{1cm} (6)

Where \( \eta \) is dynamic viscosity, \( p \) is hydrodynamic pressure, \( U \) is the velocity of journal surface.

By introducing the following dimensionless parameters

$$\theta = \frac{x}{R}, \bar{y} = \frac{y}{L/2}, \bar{h} = \frac{h}{h_c}, \bar{p} = \frac{p}{p_{ref}}, \Lambda = \frac{6\eta \omega}{p_{ref} \left( \frac{R}{c} \right)^2} = \frac{6\eta UR}{p_{ref} c^2}$$

The dimensionless form of the Reynolds equation is

$$\frac{\partial}{\partial \theta} \left( \bar{h} \frac{\partial \bar{p}}{\partial \theta} \right) + \left( \frac{D}{L} \right)^2 \frac{\partial}{\partial \bar{y}} \left( \bar{h} \frac{\partial \bar{p}}{\partial \bar{y}} \right) = \Lambda \frac{\partial \bar{h}}{\partial \theta}$$  \hspace{1cm} (7)

2.3. Bearing performances

The oil film forces in the horizontal and vertical directions are

$$F_{oils} = -\int_0^L \int_0^{2\pi} pR \cos \theta d\theta dy$$  \hspace{1cm} (8)

$$F_{oix} = -\int_0^L \int_0^{2\pi} pR \cos \theta d\theta dy$$  \hspace{1cm} (9)

The load capacity of film is given by

$$F_{oil} = \sqrt{F_{oils}^2 + F_{oiz}^2}$$  \hspace{1cm} (10)
The load capacity of a journal bearing can be expressed by Sommerfeld number

\[ S = \frac{\eta UL}{\pi F_{oil}} \left( \frac{R}{c} \right)^2 \] (11)

Two components of the moment at circumferential and axial can be written as follow

\[ M_x = \int_0^L \int_0^{2\pi} p \left( y - \frac{L}{2} \right) R \cos \theta d\theta dy \] (12)

\[ M_z = \int_0^L \int_0^{2\pi} p \left( y - \frac{L}{2} \right) R \sin \theta d\theta dy \] (13)

The total misalignment moment is

\[ M = \sqrt{M_x^2 + M_z^2} \] (14)

3. Results and discussions

To validate the proposed calculation approach, film pressure has been calculated using the same parameters as in [6]. Table 1 shows that the present results are consistent with the ones in [6], so the correctness of the presented calculation approach is proven.

**Table 1. Magnitude of the projected and the maximum film pressures.**

| \( e'/\mu m \) | Present results/MPa | Results by [6]/MPa |
|----------------|----------------------|-------------------|
| 0              | 33.2                 | 33.06             |
| 4.6            | 38.58                | 39.6              |
| 11.52          | 415.09               | 415.35            |

The influence of misalignment and elastic deformation on partially textured bearing is analyzed using the bearing whose parameters are listed in Table 2.

**Table 2. Parameters for the numerical simulation**

| \( R/mm \) | \( L/mm \) | \( c/\mu m \) | \( \eta/\text{Pa}\cdot\text{s} \) | \( E/\text{GPa} \) | \( v \) | \( \varphi_0/^\circ \) | \( r_p/mm \) | \( h_p/\mu m \) | \( S_p/\% \) | \( \theta_1/^\circ \) | \( \theta_2/^\circ \) |
|-------------|-------------|----------------|----------------|----------------|------|----------------|-------------|----------------|-------------|----------------|----------------|
| 30          | 50          | 50             | 0.028          | 0.2            | 0.34 | 60             | 2           | 20             | 18.5        | 50            | 157            |

3.1. Effect of elastic deformation on the film pressure

The variation of load capacities as a function of eccentricity ratios are plotted in Fig 2. The result observed is that, when the eccentricity greater than 0.4, partial texturing is inefficient. Fig.3 shows that, when \( D_m \) great than 0.7, the elastic deformation cannot be ignored.
3.2. Effect of the degree of misalignment $D_m$

Fig. 4 shows the variation of main bearing performance parameters (the load capacity and overturning moment) as a function of the degree of misalignment ($\varepsilon_0=0.2$, $\alpha=90^\circ$). It can be seen that the main bearing parameters are significantly increased with increasing the degree of misalignment. The reasons are due to the smaller of oil film thickness and the larger asymmetric shape, which present a strong influence on hydrodynamic effect and moment respectively. Furthermore, partially textured bearing surface enhances the load capacity and reduces moment, compared to the smooth surface, which is attributed to the creation an additional hydrodynamic lift.

3.3. Effect of the misalignment angle $\alpha$

Fig. 5 shows the variation of the main bearing performance parameters versus the misalignment angle $\alpha$ under the condition $\varepsilon_0=0.2$, $D_m=0.8$. It can be observed that, as the misalignment angle increases, the load capacity decreases till the position $\alpha=60^\circ$ and then increases with increasing $\alpha$, while the moment increases till the position $\alpha=90^\circ$ and then decreases. The reason is as follow: when the misalignment angle $\alpha=90^\circ$, there are two peak values in the pressure axially near both ends, as shown
in Figure 6, which reduces the load capacity. Noteworthy, partially textured bearing surface also enhances the load capacity and reduces moment, compared to the smooth surface.

\[\text{Figure 5. Performance of journal bearing with the misalignment angle. (a) Sommer number, (b) Moment.}\]

\[\text{Figure 6. The film pressure distribution of misalignment bearings. (a) Smooth, (b) Partially textured}\]

4. Conclusion
An EHL model for misalignment journal bearing was developed and the effects of misalignment on bearing performances were analyzed. The following conclusions can be drawn from the results: Ignoring the elastic deformation of bearing bush may lead to error. When the journal misalignment, the optimized texturing still plays a beneficial role, which is attributed to the additional hydrodynamic lift. The larger the degree of misalignment, the more significant changes load capacity and moment are. The misalignment angle cause bearing performance degradation.

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