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NEW CRITERIA INVESTIGATION ON THE LUBRICATION STATES TRANSITION FOR WATER LUBRICATED PLAIN JOURNAL BEARING

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The aim of this paper is to investigate a new criteria for the lubrication regimes transition of plain journal bearings lubricated by water. Mixed lubrication model with micro-asperities contacts has been discussed in details. Mimetic algorithm is employed to obtain numerical solutions. Relationships between the asperity contact load ratios and the lubrication states transition with different external loads, different rotating speeds, different radial clearances and different elastic moduli parameters are obtained. This manuscript lay a solid foundation for the further investigation of new criteria of lubrication states transition.

Keywords: water lubricated bearing, friction coefficient, asperity contact load ratio, lubrication states transition, new criterion

1. Introduction

In the startup and stopping process of the shaft, lubrication state of the water lubricated bearing is in a dynamic fluctuation state under different operating conditions. For low speed and heavy external load, the bearing works in boundary lubrication regime (BL) or mixed lubrication regime (ML); for high speed and light external load, the bearing is in hydrodynamic lubrication regime (HL). Therefore, lubrication states transition occurs in the process. And the parameter which reflects the lubrication mechanism on the lubrication interface is the asperity contact load ratio wasp, i.e. the micro-asperities contacts effect. To some extent, friction coefficient is closely related to the asperity contact load ratio on the lubrication interface.

For a specific bearing, the author analyzes its pressure distribution and lubrication areas for different external loads. Fig. 1 and Fig. 2 present the pressure contour and lubrication areas of the bearing under different operating conditions.

With the deterioration of operating conditions and lubrication states, in these cases, i.e. the increases of external load, the film pressure increases nonlinearly. The pressure gradients on the axial and circumferential direction are more and more large. Also, the horizontal force component decreases while the vertical force component increases gradually, the corresponding attitude angle decreases and eventually approaches the 90 ° limit value. Furthermore, with the increases of external load, the efficient lubrication area decreases sharply and the in the lubrication region, the area carrying external load is rather not even. Therefore, the lubrication condition and the operating stability is worse and worse. This circumstance illustrates that the lubrication state is in a dynamic fluctuation state under different operating conditions. Lubrication states transition occurs in the process. Correspondingly, the asperity contact load ratio is always in a process of dynamic oscillation. When the asperity contact load ratio wasp approaches to 1, the interface mainly presents the dry friction features.

When the asperity contact load ratio wasp approaches to 0, the two contact interfaces are completely be separated by the lubricant. $0<w_{asp}<1$, the lubrication state on the interface is complicated, it is the mixture of boundary and hydrodynamic lubrication. Therefore, critical values for asperity contact load ratio wasp exists that can divide the lubrication regimes into different states: BL, ML and HL. Depend on the critical value, different wasp, different surface interaction mechanism occurs, certainly corresponding to distinct wear and friction characteristics. Therefore, it is very necessary to investigate the asperity contact load ratio of water lubricated plain journal bearing. A critical point is that, depending on lubrication regime, different film thickness ratio lambda, different surface interaction mechanisms occur, certainly leading to distinct wear and friction responses.

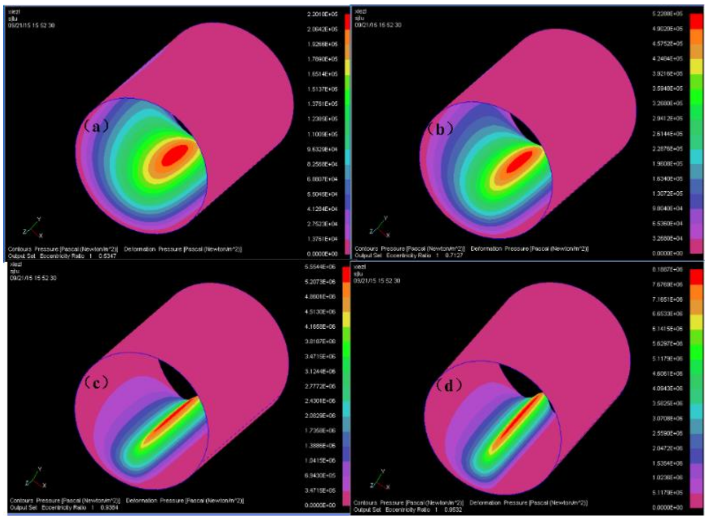

Figure 1: Pressure contour of the bearing under different lubrication conditions (XYZ).

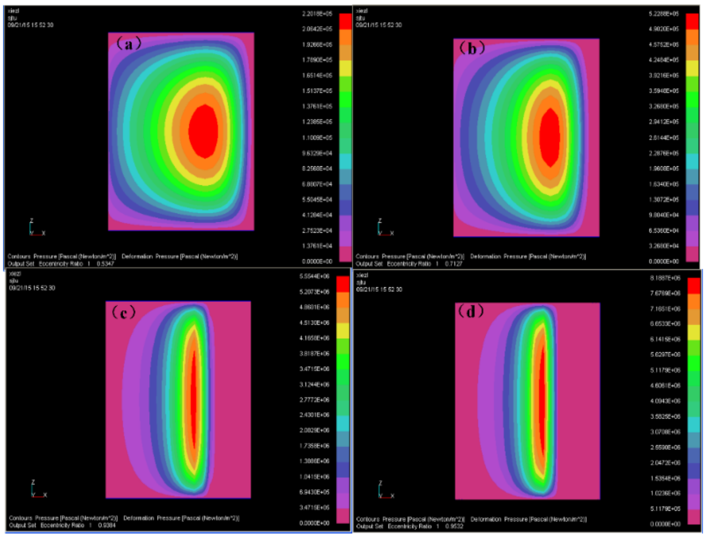

Figure 2: Lubrication area of the bearing under different lubrication conditions (XZ plane).

For the asperity contact load ratio wasp, Zhu[1, 2] investigates the lubrication states transition through a set of deterministic solutions and obtains the results of asperity contact load ratio. Define boundary lubrication as having more than 90% of load supported by the asperity contact, its corresponding λ ratio is generally smaller than 0.01-0.05. Karupannasamy[3] calculates the coefficient of friction under local contact conditions to better evaluate the formability of the sheet metal product[4]. Ren[5] exploits a 3D plasto-elasto-hydrodynamic lubrication(PEHL) model to investigate the effects of surface irregularities(single asperity or dent, it can be considered as basic elements of complicated surface roughness). Bosman[6] considers that the transition can be predicted by a maximum temperature occurring at the asperity level interface using a asperity based contact model. In view of the recent existing researches, former researchers have done much beneficial work. Little attention has been focused on the asperity contact load ratios wasp. Therefore, it will be studied in the present work.

The aim of this paper is to investigate a new criteria for the lubrication regimes transition of plain journal bearings lubricated by water. Mixed lubrication model with micro-asperities contacts has been discussed. Mimetic algorithm is employed to obtain numerical solutions. Relationships between the asperity contact load ratios and the lubrication states transition with different external loads, different rotating speeds and different radial clearances are obtained. This manuscript lay a solid foundation for the further investigation of new criteria of lubrication states transition.

2. Mathematical analysis

2.1 Mixed lubrication model with micro-asperities contacts

From Fig. 3 we can see the schematic of plain journal bearing with stochastic roughness surface model in the coordinate system. In the rectangular coordinate system with origin at the bearing center, the X-axis is in the static load direction and Z-axis is defined according to the right-hand rule. Y-axis is also the axial direction.

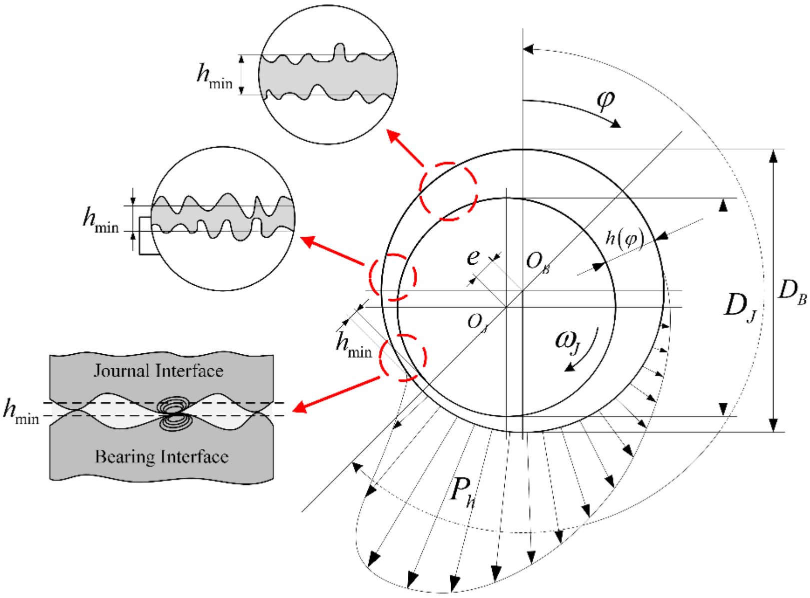

Figure 3: Schematic of water lubricated bearing witstochastic roughness surface modelial direction

Mixed lubrication model with consideration of micro-asperities contacts will be given directly:

$$\frac{\partial}{\partial x}\left(\phi_x \frac{\rho h^3}{\mu}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(\phi_z \frac{\rho h^3}{\mu}\frac{\partial p}{\partial z}\right)=6U\frac{\partial\left(\phi_c \rho h_T\right)}{\partial x}+6U\sigma_s\frac{\partial\left(\rho\phi_s\right)}{\partial x}+12\frac{\partial\left(\phi_c\rho h_T\right)}{\partial t}. \tag{1}$$

Film thickness model with macro deformation is obtained by:

$$h = h_0 + \delta h + \delta_1 + \delta_2 . \tag{2}$$

The specific formula and parameters illustrations can be found in reference[7, 8].

2.2 Load balance equation

The externally applied load is shared between fluid and asperity at various operating conditions. The magnitude of the resultant force of and equals to the external vertical load.

$$\vec{P}+\vec{F}_{fluid}+\vec{W}_{asp}=0 . \tag{3}$$

Asperity contact load ratio wasp, i.e. the fraction of load bared by micro-asperities, is given by following expression:

$$w_{asp} = \frac{p_{asp}}{p_{total}}. \tag{4}$$

Here, we only briefly describe the mixed lubrication model with consideration of micro-asperities and macro deformation. Mathematic model for friction coefficient and more other detailed information can be found in reference[7, 8].

3. Results and Discussion

From Table 1 we can obtain the basic parameters and operating conditions of the test bearing. The saturation water vapor pressure is 2340 Pa, dynamic viscosity of water is 0.001 Pa s. In the following part, effect of elastic modulus, external load, radial clearance will be discussed to investigate the asperity contact load ratio.

Table 1: Basic parameters of water lubricated bearing

Description	Symbol	Value	Unit
Bearing length	L	80	mm
Bearing diameter	D	62	mm
L/D ratio	L/D	1.30	--
Radial clearance	c	0.07	mm
Relative clearance	2c/D	1.13‰	--
Range of speed	N	0.001～10	m/s
Range of external load	F	8～620	N

3.1 Effect of elastic modulus

From Fig. 4 we can see the three-dimensional asperity contact load ratio contour distribution. X axis represents elastic modulus, Y axis represents velocity and Z represents asperity contact load ratio.

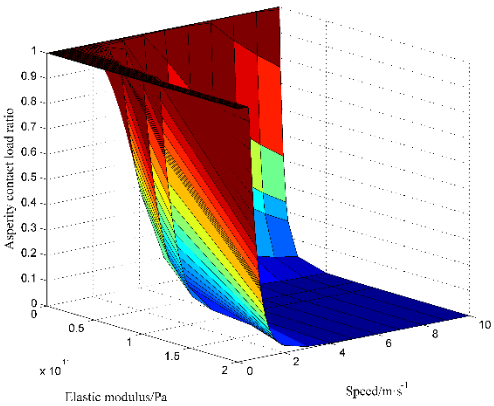

Figure 4: Asperity contact load ratio contour distribution versus speed and elastic modulus

On the whole point of view, increasing elastic modulus or decreasing velocity will vividly increases the asperity contact load ratio. As we all know, with the increase of rotating speed, the friction coefficient of the bearing will first decrease moderately in the low speed region, then decreases sharply in the medium speed region and increases mildly in the high speed region. Correspondingly, the asperity contact load ratio decreases moderately in the low speed region, then decreases sharply in the medium speed region and almost decreases to zero in the high speed region. This circumstances illustrates that the bearing works in the hydrodynamic lubrication regime in the high speed region.

Under the same rotating speed, with the increase of elastic modulus, the asperity contact load ratio also increases. It shows almost the same changing role with the friction coefficient.

3.2 Effect of external load

From Fig. 5 we can see the three-dimensional asperity contact load ratio contour distribution. X axis represents external load, Y axis represents velocity and Z represents asperity contact load ratio.

Increasing external load or decreasing velocity will increases the asperity contact load ratio. In the low and medium velocity region, under the same rotating speed, with the increase of external load, the corresponding asperity contact load ratio also shows the same changing role with friction coefficient to some extend. This circumstance illustrates that asperity contact friction contributes the main part in the whole friction coefficient. Or we can say, the bearing actually works in the boundary or mixed lubrication regime. In the high speed region, with the increase of speed, the asperity contact load ratio decreases rapidly to zero. Under this circumstance, external load has very slight influences on the asperity contact load ratio.

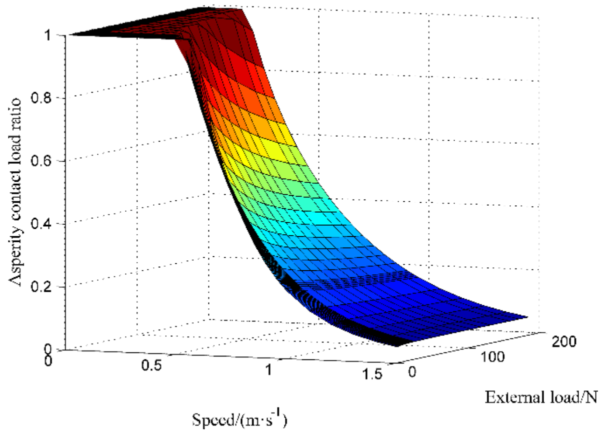

Figure 5: Asperity contact load ratio contour distribution versus speed and external load

3.3 Effect of radial clearance

From Fig. 6 we can the three-dimensional asperity contact load ratio contour distribution. X axis represents radial clearance, Y axis represents velocity and Z represents asperity contact load ratio.

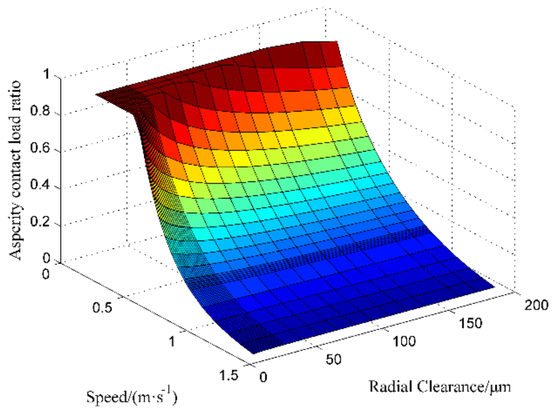

Figure 6: Asperity contact load ratio contour distribution versus speed and radial clearance

Increasing radial clearance or decreasing velocity will increases the asperity contact load ratio. In the low and medium speed region, the larger the radial clearance, the bigger the asperity contact load ratio. In the high speed region, the asperity contact ratio almost are the same and decreases rapidly to zero for three cases. From the whole trend, optimum radial clearance exists at which the friction coefficient and asperity contact load ratio can strike the optimal balance point. It is significant for the mechanical design of water lubricated bearings.

4. Conclusions

This study investigates the asperity contact load ratio wasp as well as the relationship between wasp and lubrication states transition. Large numbers of numerical analyses are performed to find the influence of external load, radial clearance and rotating speed on wasp. From the results, the following conclusions can be drawn:

1. With the increasing velocity, asperity contact load ratio first keeps constant in low velocity region, decreases until it strikes zero at medium velocity region and keeps zero in high velocity region;
2. For given operating conditions, models with the bigger elastic modulus, external load and radial clearance show the larger asperity contact load ratios;
3. Compared to external load and elastic modulus, surface roughness and radial clearance have more significant influences on the asperity contact load ratio.

These conclusions are useful for the determination of asperity contact load ratio as well as the transition of lubrication states. Research results are helpful for the analyses, design and optimization of such water lubricated journal bearings. Future efforts will include further validation of the model and new criteria for transition of lubrication states.

Acknowledgements

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