ANALYSIS OF THE MULTIPHASE LUBRICATING OIL EFFECT ON THE PERFORMANCE OF THE TILTING-PAD JOURNAL BEARING

by

Yuchuan ZHU[a,b*], Zhengyi JIANG[a,c], Ling YAN[d], Yan LI[d], Fangfang AI[d], and Shangshu LI[e]

[a] School of Materials and Metallurgy, University of Science and Technology Liaoning, Anshan, China
[b] School of Innovation and Entrepreneurship, University of Science and Technology Liaoning, Anshan, China
[c] School of Mechanical, Materials and Mechatronic Engineering, University of Wollongong, Wollongong, Australia
[d] State Key Laboratory of Metal Material for Marine Equipment and Application, Liaoning, Anshan, China
[e] School of Mechanical Engineering and Automation, University of Science and Technology Liaoning, Anshan, China

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As heavy industry develops, large amounts of tilting-pad journal bearings are widely used in advanced technology and key equipment. So, it has become a hot research direction to ensure the stable operation of tilting-pad journal bearings by using multiphase lubricating oil. The aim of the present research was to clarify whether using the multiphase lubricating oil has a positive effect on the performance of the bearings. The approach is based on computational multiphase fluid dynamics and finite-element method. Reynolds averaged equations of multiphase flow was applied to computation for improving the accuracy. The change of loading capacity of oil film was studied with CFD simulation under particles added to the lubricating oil. The results indicate that the bearing capacity of bearing increases when the particle content, diameter, and density increase. The performance of bearing becomes better when the multiphase lubricating oil is applied in the oil film of bearing. The implications of these results are that the development of multiphase lubricating oil has important practical significances.

Key words: *solid particles, multiphase flow, lubricating oil, tilting-pad journal bearing, performance of bearing*

Introduction

In the long history of research on sliding bearing, the sliding bearing with fixed bearing bush can hardly guarantee normal running performance under high speed and heavy load operating environment due to its own structural limitations. As a result, the tilting-pad journal bearing (TPJB) has been widely used in high speed rotating machinery with good vibration resistance and stability.

* Corresponding author, e-mail: zhu_yuchuan@126.com

At present, most researches on TPJB mainly focus on the improvement of the support loss characteristics by changing the bearing structure, so as to improve the dynamic stability of the bearing systems.

As early as 1983, some researchers demonstrated an effective method that improved the running stability of the tilting-pad journal bearing by changing elasticity of fulcrum. Brockwell *et al.* [1] analyzed how the different modes and elasticity of the fulcrum of tile affect the integral rigidity and damping coefficient of the tilting-pad journal bearing. Yan *et al.* [2] analyzed the dynamic model of TPJB system with n tiles, considering the dynamic model of $n + 2$ DoF when the tiles swing independently and $2n + 2$ DoF when the tiles swing and move elastic. Some results were obtained that selecting the appropriate value of the damping coefficient can improve effectively the dynamic stability of tilting pad bearing system.

It is an important research direction of the bearings to optimize the structure of tilting tile bearing. However, the characteristics of the lubricating oil is another important research direction in the research of TPJB field. Many researchers have done a lot of works in this field. Kang *et al.* [3] analyzed the influence of fluid inertia and compressibility on oil film dynamic parameters by using numerical calculation method. Cookson *et al.* [4] used numerical calculation to analyze the influence of eccentric shaft on oil film dynamic characteristics. Chen *et al.* [5] analyzed the influence of different oil supply modes on oil film dynamic characteristics by the same approach. Dang *et al.* [6] found the load direction has effects on both the static and dynamic characteristics of the bearing flow. In recent years, it is found that the turbulence appearing in the flow of TPJB have a positive effect on the performance of bearing, the turbulent flow mixing increases the diffusion of thermal energy and makes more uniform the temperature profile across the film.

Most researchers focus on the structure and flow field analysis of TPJB. However, the change of bearing performance when solid nanoparticles are mixed in lubricating oil has not been focused. Particles, sediments, debris generated by friction and wear may also be mixed with lubricants in the process of bearing operation, so it is essential to study the characteristics of multiphase lubricating oil. Meanwhile, some researchers suggest that the addition of various solid suspension additives in the production of lubricants can improve the lubrication performance of fixed tile sliding bearing lubricating oil [7]. Tenne *et al.* [8] focused on enhancing the anti-friction and anti-wear properties of lubricants by incorporating inorganic nanoparticles. However, the lubrication effect of such lubricating oil in tilting-pad journal bearing is still unclear.

Based on the CFD theory, this study analyzes the influence of the average diameter, density and concentration of solid particles in lubricating oil on the operation performance of TPJB using numerical simulation approach. It analyzes how the presence of solid particles in lubricating oil affects the operation performance of TPJB from different aspects, which provides a theoretical basis for the preparation of tilting tile bearing oil in practice.

Methods

Multiphase flow theory basis for fluid calculation

Multiphase flow theory is widely used in mechanical engineering design. The lubricating oil flow of the tilting-pad journal bearing can be simplified into a viscous solid-liquid flow model between cylinders. In the field of solid-liquid two-phase flow, there are generally two research methods: one is to study the fluid as a continuous medium and the particle as a discrete system, to discuss particle dynamics, particle`s motion trail, *etc.* The other is to study the fluid and solid particles both as continuous medium, assume that they have continuous velocity and equivalent transport properties in space. The later one is adopted in this study to treat lubricating oil and solid particles both as continuous medium and consider the influence of solid particles on lubricating oil film. The control equation of liquid flow can be expressed:

$$\frac{\partial V_{\mathrm{fr}}^*}{\partial r^*}+\frac{1}{r^*}\frac{\partial V_{\mathrm{f}\theta}^*}{\partial \theta}+\frac{\partial V_{\mathrm{fz}}^*}{\partial z^*}+\frac{V_{\mathrm{fr}}^*}{r^*}=0 \tag{1a}$$

$$\frac{\partial V_{\mathrm{fr}}^*}{\partial t^*}+V_{\mathrm{fr}}^*\frac{\partial V_{\mathrm{fr}}^*}{\partial r^*}+\frac{V_{\mathrm{f}\theta}^*}{r^*}\frac{\partial V_{\mathrm{fr}}^*}{\partial \theta^*}+V_{\mathrm{fz}}^*\frac{\partial V_{\mathrm{fr}}^*}{\partial z^*}-\frac{V_{\mathrm{f}\theta}^{*2}}{r^*}=$$

$$=\nu\left(\frac{\partial^2 V_{\mathrm{fr}}^*}{\partial r^{*2}}+\frac{1}{r^*}\frac{\partial V_{\mathrm{fr}}^*}{\partial r^*}+\frac{1}{r^{*2}}\frac{\partial^2 V_{\mathrm{fr}}^*}{\partial \theta^{*2}}+\frac{\partial^2 V_{\mathrm{fr}}^*}{\partial z^{*2}}-\frac{2}{r^{*2}}\frac{\partial V_{\mathrm{f}\theta}^*}{\partial \theta^*}-\frac{V_{\mathrm{fr}}^*}{r^{*2}}\right)-\frac{1}{\rho_{\mathrm{f}}}\frac{\partial p^*}{\partial r^*}-\frac{nF_{\mathrm{r}}^*}{\rho_{\mathrm{f}}} \tag{1b}$$

$$\frac{\partial V_{\mathrm{f}\theta}^*}{\partial t^*}+V_{\mathrm{fr}}^*\frac{\partial V_{\mathrm{f}\theta}^*}{\partial r^*}+\frac{V_{\mathrm{f}\theta}^*}{r^*}\frac{\partial V_{\mathrm{f}\theta}^*}{\partial \theta^*}+V_{\mathrm{fz}}^*\frac{\partial V_{\mathrm{f}\theta}^*}{\partial z^*}+\frac{V_{\mathrm{fr}}^*V_{\mathrm{f}\theta}^*}{r^*}=$$

$$=\nu\left(\frac{\partial^2 V_{\mathrm{f}\theta}^*}{\partial r^{*2}}+\frac{1}{r^*}\frac{\partial V_{\mathrm{f}\theta}^*}{\partial r^*}+\frac{1}{r^{*2}}\frac{\partial^2 V_{\mathrm{f}\theta}^*}{\partial \theta^{*2}}+\frac{\partial^2 V_{\mathrm{f}\theta}^*}{\partial z^{*2}}-\frac{V_{\mathrm{f}\theta}^*}{r^{*2}}+\frac{2}{r^{*2}}\frac{\partial V_{\mathrm{fr}}^*}{\partial \theta^*}\right)-\frac{1}{r^*}\frac{\partial p^*}{\partial \theta^*}-\frac{nF_{\theta}^*}{\rho_{\mathrm{f}}} \tag{1c}$$

$$\frac{\partial V_{\mathrm{fz}}^*}{\partial t^*}+V_{\mathrm{fr}}^*\frac{\partial V_{\mathrm{fz}}^*}{\partial r^*}+\frac{V_{\mathrm{f}\theta}^*}{r^*}\frac{\partial V_{\mathrm{fz}}^*}{\partial \theta^*}+V_{\mathrm{fz}}^*\frac{\partial V_{\mathrm{fz}}^*}{\partial z^*}=$$

$$=\nu\left(\frac{\partial^2 V_{\mathrm{fz}}^*}{\partial r^{*2}}+\frac{1}{r^*}\frac{\partial V_{\mathrm{fz}}^*}{\partial r^*}+\frac{1}{r^{*2}}\frac{\partial^2 V_{\mathrm{fz}}^*}{\partial \theta^{*2}}+\frac{\partial^2 V_{\mathrm{fz}}^*}{\partial z^{*2}}\right)-\frac{1}{\rho_{\mathrm{f}}}\frac{\partial p^*}{\partial z^*}-\frac{nF_{\mathrm{z}}^*}{\rho_{\mathrm{f}}} \tag{1d}$$

Considering the existence of solid particles in lubricating oil, the motion control equations of solid particles can be expressed:

$$\frac{\partial \alpha_{\mathrm{p}}}{\partial t^*}+\frac{\partial V_{\mathrm{pr}}^*\alpha_{\mathrm{p}}}{\partial r^*}+\frac{1}{r^*}\frac{\partial V_{\mathrm{p}\theta}^*\alpha_{\mathrm{p}}}{\partial \theta^*}+\frac{\partial V_{\mathrm{pz}}^*\alpha_{\mathrm{p}}}{\partial Z^*}+\frac{\alpha_{\mathrm{p}}V_{\mathrm{pr}}^*}{r^*}=0 \tag{2a}$$

$$\frac{\partial V_{\mathrm{pr}}^*}{\partial t^*}+V_{\mathrm{pr}}^*\frac{\partial V_{\mathrm{pr}}^*}{\partial r^*}+\frac{V_{\mathrm{p}\theta}^*}{r^*}\frac{V_{\mathrm{pr}}^*}{\partial \theta^*}+V_{\mathrm{pz}}^*\frac{\partial V_{\mathrm{pr}}^*}{\partial z^*}-\frac{V_{\mathrm{p}\theta}^{*2}}{r^*}=-\frac{1}{\rho_{\mathrm{p}}}\frac{\partial p^*}{\partial r^*}+\frac{nF_{\mathrm{r}}^*}{\alpha_{\mathrm{p}}\rho_{\mathrm{p}}} \tag{2b}$$

$$\frac{\partial V_{\mathrm{p}\theta}^*}{\partial t^*}+V_{\mathrm{pr}}^*\frac{\partial V_{\mathrm{p}\theta}^*}{\partial r^*}+\frac{V_{\mathrm{p}\theta}^*}{r^*}\frac{\partial V_{\mathrm{p}\theta}^*}{\partial \theta^*}+V_{\mathrm{pz}}^*\frac{\partial V_{\mathrm{p}\theta}^*}{\partial z^*}+\frac{V_{\mathrm{pr}}^*V_{\mathrm{p}\theta}^*}{r^*}=-\frac{1}{r^*\rho_{\mathrm{p}}}\frac{\partial p^*}{\partial \theta}+\frac{nF_{\theta}^*}{\rho_{\mathrm{p}}\alpha_{\mathrm{p}}} \tag{2c}$$

$$\frac{\partial V_{\mathrm{pz}}^*}{\partial t^*}+V_{\mathrm{pr}}^*\frac{\partial V_{\mathrm{pz}}^*}{\partial r^*}+\frac{V_{\mathrm{p}\theta}^*}{r^*}\frac{\partial V_{\mathrm{pz}}^*}{\partial \theta^*}+V_{\mathrm{pz}}^*\frac{\partial V_{\mathrm{pz}}^*}{\partial z^*}=-\frac{1}{\rho_{\mathrm{p}}}\frac{\partial p^*}{\partial z^*}+\frac{nF_{\mathrm{z}}^*}{\alpha_{\mathrm{p}}\rho_{\mathrm{p}}} \tag{2d}$$

In the liquid phase control equation, V_{fr}^*, $V_{\mathrm{f}\theta}^*$, and V_{fz}^* are the radial velocity, circumferential velocity, and axial velocity of the fluid phase in cylindrical co-ordinate system, respectively, t^* – the time variable, p^* – the pressure variable, ρ_{f} – the fluid density, n – the number of particle density, and g – the acceleration of gravity.

In the solid phase control equation, V_{pr}^*, $V_{p\theta}^*$, and $V_{pz}^*$are the radial velocity, circumferential velocity and axial velocity of the solid phase, respectively, in cylindrical co-ordinate system, p – the particle density, α_p – the volume fraction of the particle phase in multiphase mixture, and F^* – the interaction force between the liquid phase and particle phase, it can be expressed:

$$F_i^* = 3\pi\mu d(V_{fi}{}^* - V_{pi}{}^*) + \frac{\rho_f \pi (d)^3}{12}\frac{\partial}{\partial t^*}(V_{fi}{}^* - V_{pi}{}^*), \quad \text{for } i = r, \theta, z \tag{3}$$

where d is the diameter of particle, μ – the dynamic viscosity of the fluid, and ρ_f – the fluid density. In this equation it is found that the Stokes drag and added mass are relevant to the properties of the particle and oil. So this study studied the properties of particle effect on the performance of tilting pad journal bearing.

Reynolds averaged equation of liquid flow

The instant velocity V_f^* is expressed by the average velocity and the fluctuating velocity:

$$V_{fr}{}^* = \overline{V_{fr}{}^*} + v_{fr}^{'} \tag{4a}$$

$$V_{f\theta}{}^* = \overline{V_{f\theta}{}^*} + v_{f\theta}^{'} \tag{4b}$$

$$V_{fz}{}^* = \overline{V_{fz}{}^*} + v_{fz}^{'} \tag{4c}$$

where V_{fr}^*, $V_{f\theta}^*$, and V_{fz}^* are the radial average velocity, circumferential average velocity and axial average velocity of the fluid phase, respectively, $v_{fr}^{'}$, $v_{f\theta}^{'}$, and $v_{fz}^{'}$ are the radial fluctuating velocity, circumferential fluctuating velocity and axial fluctuating velocity of the fluid phase, respectively.

Equations (4a)-(4c) are substituted into the control equation of liquid flow. The time-averaged operation of the equations results in the final form of the control equation.

Establishment of computational fluid domain model

The oil film flow field inside the three cavity liquid tilting-pad journal damping bearing is taken as the research object. The structural diagram of this type of bearing is shown in fig. 1. The bearing's outside diameter, D, is 80 mm, the bearing's width, L, is 80 mm, the minimum thickness of the oil film, h_0, is 0.03 mm, the oil inlet's diameter, d_0, is 3 mm, the axial width of the oil chamber is 60 mm, the thickness of the bearing bush, l_0, is 1 mm, the bearing bush locating pin's width is 1 mm, and the height of the bearing bush locating pin is 1.5 mm.

Considering the bearing area is mainly concentrated in the bottom of the bearing, only the oil cavity in the bearing area is calculated to save the computational time. Its loading region structural grid is shown in fig. 2.

The mesh dividing of flow field is an important part of CFD. The ICEM provides advanced geometry acquisition, mesh generation, and mesh optimization tools to meet the requirement for integrated mesh generation for today's sophisticated analyses. So ICEM was used in this study to divide meshes in fluid domain of multiphase flow field. Firstly, the grid blocks of each region are created. Secondly the block structure of each region is associated with the actual flow field model and the number of nodes in the key boundary region is limited. Finally, the grid quality is tested and is found that there is no negative volume and larger

Angle distortion. Since the quality of meshing is relatively ideal, the conclusion is relatively accurate when the structured meshes are used for simulation.

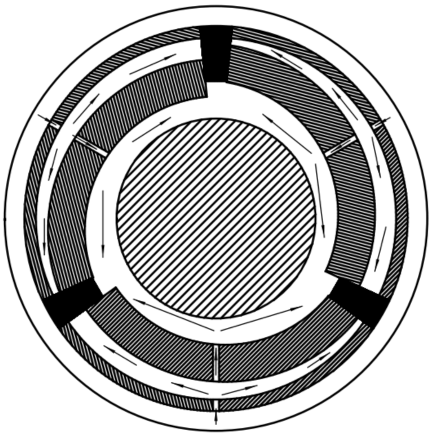

Figure 1. Simple structure diagram of bearing

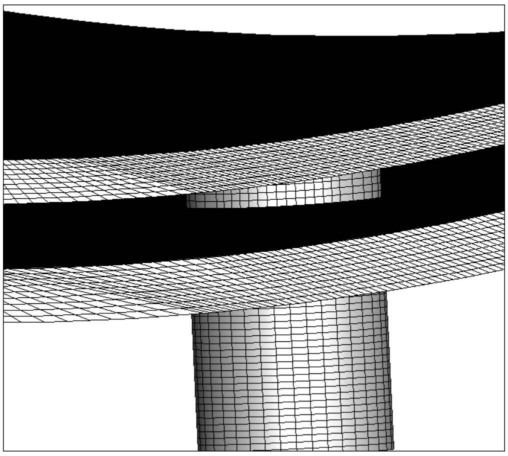

Figure 2. Structured grid diagram of partial flow field

Calculation model hypothesis

The calculation model assumes that the lubricating oil in the internal flow field of the sliding bearing is an incompressible fluid and the flow state is 3-D steady flow. There is no relative sliding between the multiphase lubrication fluid and the bearing, the heat generated is completely removed by the fluid, there is no heat transfer between the fluid and the wall. Considering that the multiphase oil film of bearing is in turbulent state [9], the turbulence model is used to solve the calculation.

Determination of the boundary conditions

This study focuses on the bearing area of the oil films. Thus the bearing area has one pressure inlet. The pressure is set at 7 MPa. The wall of the oil film in contact with the spindle is set as the rotating wall, and the rotation speed is from 0 to 20000 rpm. The interface between the oil film and the non-bearing area is set as symmetrical boundary condition. The boundary conditions are listed in tab. 1.

Table 1. Boundary conditions of the simulation case

Boundary	Boundary type	Units	Value
Inflow	Pressure-inlet	[MPa]	2
End wall	Pressure-outlet	[MPa]	0.1013
Rotating wall	Moving wall	[rpm]	0-20000
The interface wall	Symmetry	–	–
The other walls	Wall	–	–

Model selection and parameter setting

In this study, the standard turbulence model is used to simulate the flow field. The dynamic viscosity of the lubricating oil is 0.027931 Pa·s and the density is 900 kg/m^3. The density of solid particles is 2250 kg/m^3 special graphite particles. The physical properties of lubricating oil are listed in tab. 2.

Table 2. Physical properties of lubricating oil

Physical property	Units	Value
Density	[kgm^{-3}]	900
Thermal conductivity	[Wm^{-1}K^{-1}]	0.135
Viscosity	[kgm^{-1}s^{-1}]	0.02792
Specific heat	[Jkg^{-1}K^{-1}]	2009.28

Results and discussion

Based on the multiphase flow theory, this study firstly observed the influence of the spindle speed on the bearing capacity of the new type of TPJB. The initial conditions of flow field are set as follows.

The pressure of the oil inlet, P, is 7 MPa, the side leakage pressure is standard atmospheric pressure, the dynamic viscosity of the lubricant, v, is 0.027931 Pa·s, and the density, ρ, is 900 kg/m^3. The two-equation turbulence model is applied in the calculation of the two phases turbulence system. The density of the solid particles in the lubricating oil, ρ_p, is 2250 kg/m^3, the average diameter of special graphite particles, d, is $11\cdot10^{-7}$ mm. The ratio of particle volume to lubricating oil volume, α_p, is 0.001%.

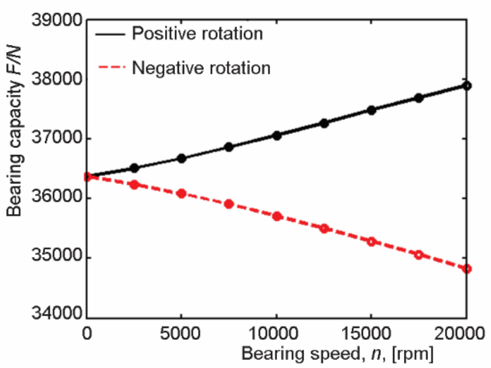

Figure 3. The effect of the bearing rotational speed on the loading capacity

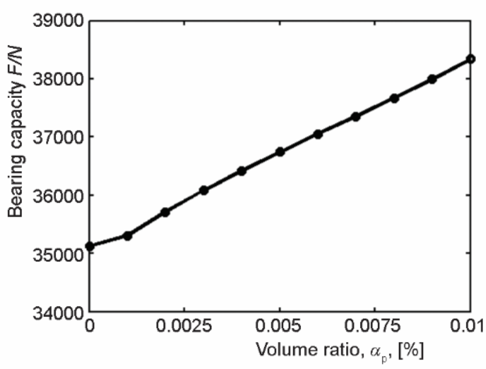

Figure 4.The effect of the particles concentration on the loading capacity

When the spindle speed changes, the influence of positive and negative speed increases on the oil film bearing capacity was observed as shown in fig. 3. It is found that when the spindle rotates in a positive direction, with the increase of spindle speed, the dynamic pressure effect is gradually obvious and the bearing capacity is effectively improved. Compared with ordinary static bearing, static bearing has obvious advantages. When the rotation direction of the main axis is reversed, because of the effect of the edge of the tile on the lubrication flow field, the lubrication flow field forms eddy current at the edge of the tile, and the eddy current becomes more prominent with the increase of the rotation speed, leading to a significant decrease in the bearing capacity. Hydrostatic sliding bearings can be avoided for this kind of situation. Therefore, this new type of dynamic and static damping bearing is suitable for high speed forward rotation. When high speed reverse is required, attention should be paid to the reduction of bearing capacity.

After that, the speed of the bearing spindle is set as 10000 rpm in the previous simulation conditions, and the ratio of the volume of solid particles in the lubricating oil to the volume fraction of oil fluid, α_p, is changed to monitor the bearing capacity, as shown in fig. 4. It is found that the bearing capacity of the bearing increases greatly with the increase of particle content gradually in the range of 0 to 0.01%. Considering that the addition of solid particles makes the flow field more stable and the vortex phenomenon is significantly reduced, adding appropriate solid particles to ordinary lubricating oil can improve the performance of lubricating oil.

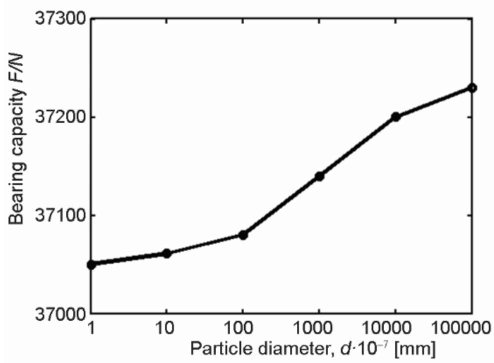

Figure 5. The effect of the particles diameter on the loading capacity

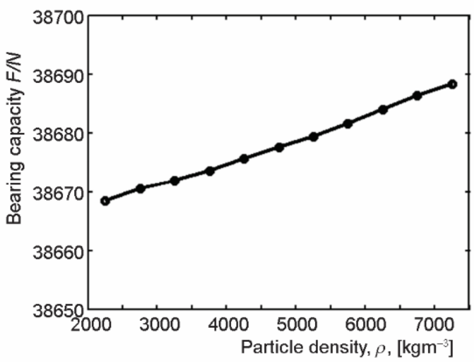

Figure 6. The effect of the particles density on the loading capacity

After considering the influence of particle content on bearing`s performance, it is found that the different diameters of solid particle in oil film have some remarkable influence on the bearing capacity. The setting of the basic conditions remains unchanged, the spindle speed, n, is set as 10000 rpm, and the volume fraction ratio of solid particles to lubricating oil, α_p, is 0.001%. The influence of the change of solid particle diameter on bearing capacity is monitored, as shown in fig. 5. It can be seen that when the particle diameter increases from 0.1 nm to 10 nm, the bearing capacity increases with the increase of diameter, but the rate of increase of bearing capacity is low. When the particle diameter continues to increase from 10 nm to 10000 nm, the carrying capacity is greatly improved, and the rate of carrying capacity increases is relatively high, and then the spindle speed is maintained at 10000 rpm, the particle diameter is 0.1 nm, the particle volume and lubricating oil volume fraction ratio, α_p, is 0.01%, and the oil inlet pressure is 7 MPa, the density of solid particles is changed to monitor the influence of different solid particle densities on bearing capacity, as shown in fig. 6. The results show that the bearing capacity increases with the increase of particle density. However, the bearing capacity does not change linearly with the particle density. In a word, the higher the density of solid particles added, the better the lubricating oil works. The higher density of solid particles would have some negative effect on the running bearing in the oil film, so the range of density of solid particles is a very important problem when the new multi-phases oil is produced.

Conclusion

This study studies the influence of solid particle physical properties on bearing capacity by adding solid particles into lubricating oil. On the basis of multiphase fluid dynamics the flow field of tilting-pad journal damping bearing was monitored, when the bearing was working under the high speed and heavy load operating environment. The results show that the lubricating oil added with solid particles can maintain better lubrication properties. With the increase of particle content and particle diameter, the bearing capacity can be increased. Changing the density particle within a certain range has some effect on the bearing capacity. So when the multiphase lubricating oil is produced, the density of particle is a key issue. This study provides a theoretical basis for the development of the new multiphase mixed lubricating oil. The results have some practical significance in the lubricating oil of tilting-pad journal bearing.

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