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Study on the influence of bearing carrying capacity on dynamic bearing under different viscosity

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Abstract. When the dynamic pressure bearing is used, the viscosity changes of the lubricating oil affect the bearing capacity of the dynamic pressure bearing. In order to obtain the bearing capacity of dynamic pressure bearing under different viscosity, the lubricating oil temperature, oil film pressure and viscosity control equation are established according to the structure of dynamic pressure bearing ,the finite difference method is used to analyze the oil pressure under different viscosity. The influence of viscosity on maximum oil film pressure is obtained, which provides a theoretical basis for the selection of viscosity of dynamic pressure bearing oil.

Key words. Viscosity, hydrodynamic bearing, energy equation, carrying capacity.

1. Introduction

When a dynamic pressure bearing is working, after the shearing of the shaft, the temperature of the lubricating oil rises, the viscosity decreases, and the pressure of the oil film becomes smaller. When the energy loss is low, the temperature rises very little and there is little change in viscosity, when the oil film pressure is calculated, it is allowed to adopt the mean viscosity method without considering the energy loss. For high speed and ultra high speed grinding machines with dynamic pressure bearings, as they are often at high speed and light load conditions, when the dynamic pressure bearings are working, high energy loss is caused by the shearing to the lubricating fluid, and the influence of the change of viscosity on the bearing capacity can not be ignored. In Literature[1] used the dynamics software FLUENT to study the Carrying capacity of dynamic bearing, found that with the increase of no matter

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input pressure or spindle speed, carrying capacity is linear increased. Literature[2] investigated the effects of the orifice parameter, the distance between two recess rows, and the number of recesses on these dynamic characteristics. Literature[3] indicated that tilting pad journal bearings without a seal tooth have lower power loss and pad temperature than tilting pad journal bearings with a seal tooth. Literature [4] made the lubrication analysis closer to the actual situation and usable to the journal bearing design, found that the oil viscosity-pressure relationship has a significant effect on the lubrication of misaligned journal bearings, and the surface roughness will affect the lubrication of misaligned journal bearings when the eccentricity ratio and angle of journal misalignment are all large.

The studies on the characteristics of dynamic pressure bearing overlook the difference of bearing capacity of bearing under different lubricating oil viscosity. This paper established a control model of dynamic pressure bearing, used the finite difference method to perform the discrete solution of the established equations, and studied the effect of temperature rise on oil film pressure caused by different viscosity of lubricating oil, which provides a theoretical basis for the choice of the viscosity of the lubricating oil of the dynamic pressure bearing.

2. The structure of dynamic pressure bearing

In high speed and ultra high speed grinding machines with dynamic pressure bearings, the simplified model of the dynamic pressure bearing is shown in Figure 1, the journal revolves around the center O_2 with a speed of ω, and brings the lubricating fluid into the smaller gap to form the dynamic pressure, the distance between the journal center O_2 and the bearing center O_1 is an eccentricity, expressed with e, ϕ is an angle of displacement, θ is the included angle O_1O_2 between and O_1K, $h_{\min}$ and $h_{\max}$represent the minimum and maximum oil film thickness respectively, and R_1 and R_2 represent the radius of bearing and journal respectively.

3. Control equation of dynamic pressure bearing

In the operation of the dynamic pressure bearing, the journal rotates at high speed, the temperature of the fluid is raised at the same time that the dynamic pressure is produced, as a result of the effect of friction, the increase of temperature makes the viscosity of the fluid lower, so as to influence the pressure distribution. Therefore, it is necessary to establish a temperature control model, a pressure control model and a viscosity control model.

3.1. Temperature control equation

To calculate the temperature distribution of oil film under different initial viscosity conditions, it is necessary to investigate the quantity relation of lubricating oil from the angle of energy conservation. The differential form of the energy equation

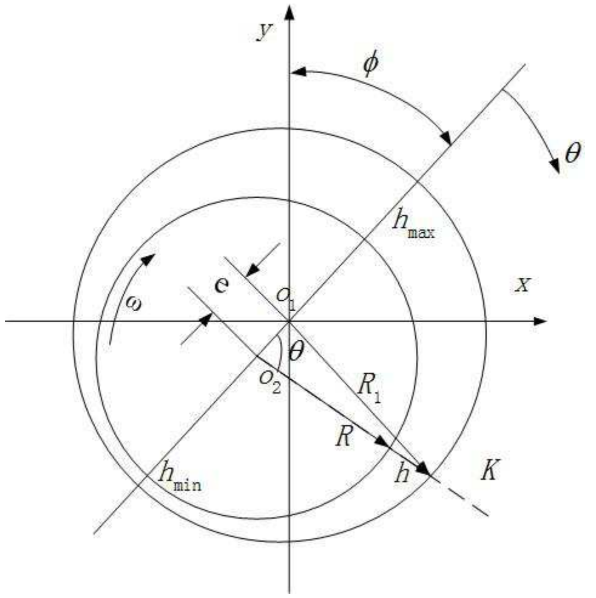

Fig. 1. The structure of dynamic pressure bearing

is as follows[5]:

$$\rho[v_x\frac{\partial(c_vT)}{\partial x}+v_y\frac{\partial(c_vT)}{\partial y}+v_z\frac{\partial(c_vT)}{\partial z}]=\mu((\frac{\partial(v_x)}{\partial y})^2+(\frac{\partial(v_z)}{\partial y})^2)+\frac{\partial}{\partial y}(k\frac{\partial T}{\partial y}) \quad (1)$$

For the convenience of calculation, we set:
$x = R\theta, y = c\bar{y}, z = 1/2\lambda, v_x = U\bar{v}_x, v_y = \psi U\bar{v}_y, v_z = U\bar{v}_z, \mu = \mu_0\bar{\mu}, \psi = c/R, \kappa = \kappa_0\bar{\kappa}, c_v = c_{v_0}\bar{c}_v, T = T_0\bar{T}, Take\ T_0 = UR\mu_0/(\rho c_{\nu_0}c^2)$For medium and high speed dynamic pressure bearings,so the formula of the energy equation can be further simplified to:

$$\rho[v_x\frac{\partial(c_vT)}{\partial\theta}+v_y\frac{\partial(c_vT)}{\partial y}+v_y\frac{\partial(c_vT)}{\partial z}]=\mu((\frac{\partial v_x}{\partial y})^2+(\frac{\partial v_z}{\partial y})^2) \quad (2)$$

We can calculated and obtained based on the related knowledge of Newton viscous fluid

$$v_x=\frac{1}{2\mu}\frac{\partial p}{\partial x}(y^2-hy)+U(\frac{h-y}{h}),v_z=\frac{1}{2\mu}\frac{\partial p}{\partial z}(y^2-hy)$$

We can obtain through the integration of formula (2) from 0 to h in the thickness direction:

$$\rho[\frac{hU}{2}-\frac{h^3}{12\mu}\frac{\partial p}{\partial x})\frac{\partial(c_vT)}{\partial x}+(-\frac{h^3}{12\mu}\frac{\partial p}{\partial x})\frac{\partial(c_vT)}{\partial z}]=\frac{\mu U^2}{h}+\frac{h^3}{12\mu}[(\frac{\partial p}{\partial x})^2+(\frac{\partial p}{\partial z})^2] \quad (3)$$

3.2. Pressure control equation

The pressure control equation of the dynamic pressure bearing is as follows:

$$\frac{\partial}{\partial x}(\frac{h^3}{\mu}\frac{\partial p}{\partial x})+\frac{\partial}{\partial z}(\frac{h^3}{\mu}\frac{\partial p}{\partial z})=6U\frac{\partial h}{\partial x} \tag{4}$$

3.3. Viscosity control equation

In dynamic pressure bearings at low and medium temperatures, the dynamic viscosity will decrease with the increase of temperature.This paper uses the Roelands equation which has considered the situation of viscosity temperature and viscous pressure[6]

$$\mu=\mu_0\exp\{(\ln\mu_0+9.67)[(1+5.1\times10^{-9}p)^{0.68}\times((T-138)/(T_0-138))^{-1.1}-1]\} \tag{5}$$

Where$\mu_0$is the viscosity(Pa?s) under the temperatureT_0,μ is the viscosity under the temperature T, p is the pressure,$p_0=6UR\mu_0/c^2, T=T_0\bar{T}, Take\ T_0=UR\mu_0/(\rho c_{\nu_0}c^2)$.

4. Numerical simulation of dynamic pressure bearing

4.1. boundary condition

4.1.1. Pressure boundary condition

$$p(\theta,\pm z/2)=p(\theta_{out},z)=p(\theta_{in},z)=(\partial p/\partial\theta)_{out}=0$$

4.1.2. Temperature boundary condition

$$(\partial T/\partial z)_{\bar{z}=0}=0, T(\theta_{in},z)=(T(\theta_{in},z)+T(\theta_{out},z))/2$$

4.2. Numerical calculation process

In iterative computation, the above equations (3), (4) and(5)need to be solved jointly. Steps of the solution process Firstly, calculate the pressure distribution[7], and then calculate the temperature distribution according to the initial pressure, after that replace the initial pressure and temperature into the viscosity temperature equation to calculate the viscosity distribution.Then we calculate the pressure distribution according to the calculated viscosity distribution. Finally, we use the initial convergence condition to judge whether the pressure is convergent. If we do not converge, we will calculate the temperature distribution, viscosity distribution and pressure distribution again, until the pressure convergence condition is satisfied.

4.3. Structural parameters and operating parameters

The structural parameters are shown in Table 1, and the running parameters are shown in Table 2.

Table 1. Structure parameters used for calculation of the bearing

name	**Code**	**unit**	**Numerical value**
Bearing learance	c	mm	0.09
Bearing width	b	mm	100
Radius of the neck	R	mm	60

Table 2. Operating parameters of the bearings used for calculation

name	**Code**	**unit**	**Numerical value**
Constant volume specific heat	c_v	J/kgK	1942
Fluid density	ρ	Kg/m^3	880.3
Dynamic viscosity	μ	Pa?s	0.05
Axial neck velocity	ω	rad/s	152.9
Conduction coefficient	K_0	W/mK	0.127
Oil intake temperature	T_0	?C	32
Eccentricity	ε		0.5

4.4. Program verification

Using the program to calculate??the circumferential temperature of the center section of the dynamic pressure bearing is compared with the experimental results of the literature [8], as shown in Figure.2, the calculated temperature distribution is close to the experimental results in Literature[8]??which proves that the model is accurate and credible.

5. Influence of viscosity on dimen-sionless oil film pressure

According to the analysis of equations (3), (4) and (5), during the operation, the maximum temperature rise and the dimensionless oil film pressure change of the dynamic pressure bearing are not only influenced by the viscosity of the lubricating oil, but also by the eccentricity, speed, bearing clearance and inlet temperature of the dynamic pressure bearing. In the study of the effect of viscosity on the bearing characteristics, this paper considers the role of the above factors comprehensively.

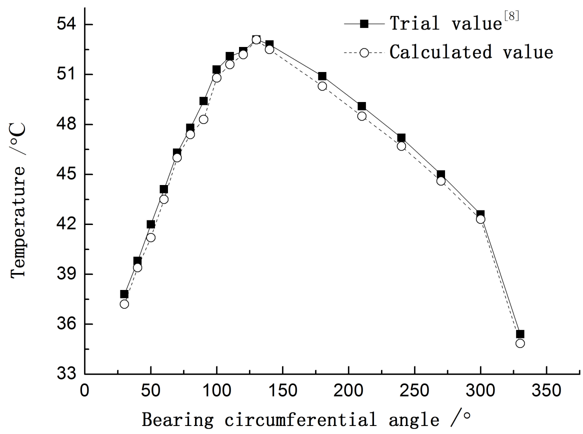

Fig. 2. Comparison between calculated values and experimental values

5.1. *Effect of eccentricity ratio under different viscosity*

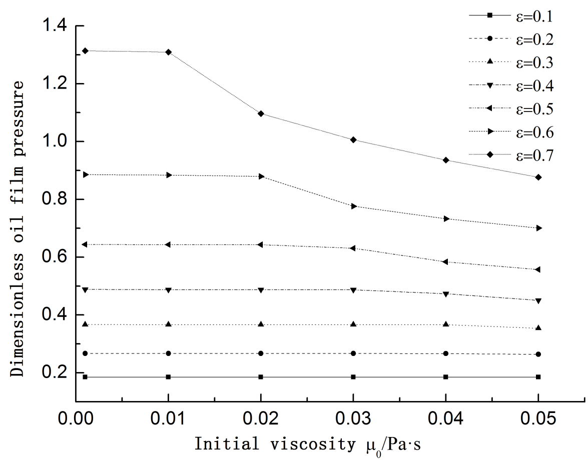

Fig. 3. Dimensionless oil film pressure curve under different viscosity

It can be seen from Figure 3 that, when the eccentricity ratio is less than 0.4, there is little difference between the maximum dimensionless oil film pressure at different initial viscosity; when the eccentricity ratio is equal to 0.7 and the initial viscosity is less than 0.01, the dimensionless oil film pressure is kept at about 1.3, however, when the initial viscosity is equal to 0.05Pa?s, the dimensionless oil film pressure is 0.88, down about 32.5%. This is because that, when the initial viscosity becomes larger, the friction heat becomes more, the greater the temperature rise will be, the rise of temperature leads to a drop in viscosity and a reduction in bearing capacity.

5.2. *Effect of axis speed under different viscosity*

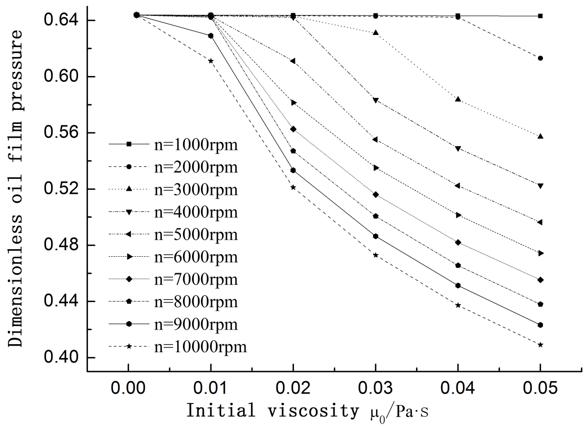

Fig. 4. The relation curve of the rotational speed and the dimensionless oil film pressure

It can be seen from Figure 4 that, when the rotational speed is equal to 1000rpm and the viscosity increases from 0.001 to 0.05, the dimensionless oil film pressure changes from 0.644 to 0.643; when the rotational speed is equal to 5000rpm and the viscosity increases from 0.001 to 0.05, the dimensionless oil film pressure changes from 0.644 to 0.496; when the rotational speed is equal to 5000rpm and the viscosity increases from 0.001 to 0.05, the dimensionless oil film pressure changes from 0.644 to 0.409. From the figure we can see that, when the rotational speed is slow, the viscosity has effect on the dimensionless oil film pressure, with the increase of rotational speed, the effect of viscosity on dimensionless oil film pressure is becoming more and more significant. This is because that, when the rotational speed becomes faster, the more obvious the heat of friction, the greater the temperature rise, which leads to the decrease of the dimensionless oil film pressure, and the greater the viscosity, the smaller the dimensionless oil film pressure.

5.3. *Influence of bearing clearance under different viscosity*

It can be seen from Figure 5 that, when the bearing clearance is equal to 110μm and the viscosity increases from 0.001 to 0.05, the oil film pressure changes from 0.267 to 0.243;

when the bearing clearance is equal to 50μm and the viscosity increases from 0.001 to 0.05, the oil film pressure changes from 0.267 to 0.180; from the figure we can see that, with the decrease of bearing clearance, the effect of viscosity on dimensionless oil film pressure is becoming more and more significant. This is because that, the smaller the bearing clearance, the smaller the oil film thickness, and the greater the viscosity, the higher the temperature rises, the smaller the dimensionless oil film pressure.

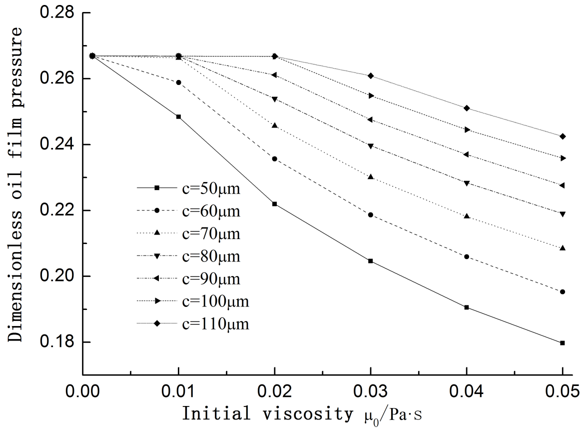

Fig. 5. The relation curve of the bearing clearance and the dimensionless oil film pressure

5.4. Influence of bearing clearance under different viscosity

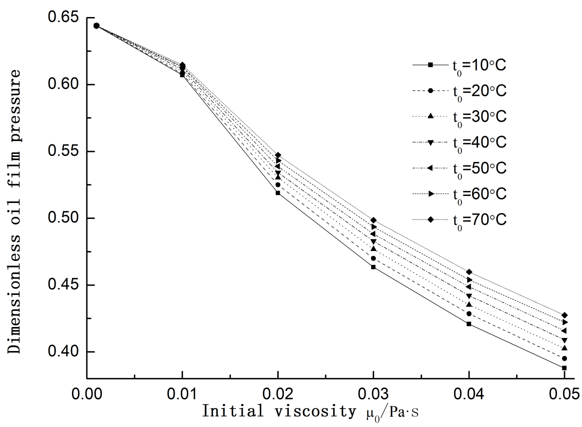

Fig. 6. The relation curve of the oil intake temperature and the dimensionless oil film pressure

It can be seen from Figure 6 that, when the oil intake temperature is equal to 10?C and the viscosity increases from 0.001 to 0.05, the dimensionless oil film pressure changes from 0.644 to 0.388; when the oil intake temperature is equal to 70?C and the viscosity increases from 0.001 to 0.05, the dimensionless oil film pressure changes from 0.644 to 0.428. From the figure we can see that, there is no obvious difference in the influence of viscosity on the dimensionless oil film pressure at different oil intake temperature. This is because that, the viscosity is mainly influenced by the temperature, the change of temperature is not great, and the change of dimensionless oil film pressure will not be very large, too.

6. Conclusions

The influence of viscosity on dimensionless oil film pressure:

Under different eccentricity ratios, the greater the viscosity, the lower the dimensionless oil film pressure;Under different rotational speeds, the greater the viscosity, the lower the dimensionless oil film pressure;Under different bearing clearances, the greater the viscosity, the lower the dimensionless oil film pressure;Under different oil intake temperature, the viscosity has little effect on the dimensionless oil film pressure.

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