Effect of groove texture on the dynamic characteristics of water-lubricated thrust bearing

Huihui Feng[1], Liping Peng[1]

[1]*Scholl of Mechanical and Electrical Engineering, Hohai University, Changzhou, Jiangsu 213022, China*

Abstract. In this study, the effects of groove texture on the dynamic characteristics of water-lubricated thrust bearing are theoretically investigated. The turbulent Reynolds equation and its perturbation equations for water-film lubrication are derived and solved by using finite difference method. Dynamic characteristics including the stiffness and damping coefficients of the bearing are calculated. The effects of rotary speed, film clearance and geometrical parameters including groove texture depth and circumferential angle on the dynamic characteristics of the bearing have been investigated.

1 Introduction

Water-lubricated bearings have attracted many attentions due to their low viscosity, low power loss at high rotary speed, good damping as well as long lift-time. Those bearings find application in water pumps, hydroelectric turbines and high-speed spindles [1]. However, despite of the low friction of the bearing at high rotary speed, the low viscosity also introduces a lower load capacity compared with the bearing lubricated by oil.

The application of surface texture has been shown to improve the performances of fully lubricated hydrodynamic bearing by many researchers [2,3]. The texture introduces micro-cavity effect which generates the additional film pressure of the bearing [2,3]. Tala-ighil et al. [4] pointed out that an appropriate arrangement of the textured area can improve the hydrodynamic load capacity and decrease the friction torque. Shi and Ni [5] found that with the increase of the groove number and reduction of groove size, the load capacity increased. As a result, in order to improve the load capacity of the water-lubricated bearing, the surface texture is introduced in this research.

In the previous study, the mathematical models and numerical solving methods were investigated to analyze the effect of rotary speed, shape of texture, texture depth and other structural parameters on the load capacity, friction of the bearings with surface texture. as the analysis of the bearings with surface texture were investigated [5-8]. Zhou et al. [6] analyzed the effects of the texturing parameters on the load capacity based on the Reynolds equation and proposed an optimal design method for the texture. However, the majority of the researches mainly focused on the static performances of the bearings with surface texture. So far, the dynamic characteristics of the bearings with texture have not been sufficiently investigated.

This paper aims to study the effect of groove texture on the stiffness and damping coefficients of hydrodynamic water-lubricated thrust bearing. The steady Reynolds equation considering turbulence is adopted; the linear perturbation method is used to derive its perturbation equations for water-film lubrication. And then the steady and perturbation equations are solved by finite difference method to obtain the steady and perturbed pressure distributions. Dynamic characteristics including the stiffness and damping coefficients of the bearing are calculated by integration of the pressure distribution. Finally, the effects of rotary speed and various geometrical parameters, such as groove texture depth, circumferential angle on the dynamic coefficients will be investigated.

Nomenclature	
c	Damping coefficient, N/(ms)
h_0	Film clearance, μm
h_{st0}	Maximum depth of a groove, μm
k	Stiffness, N/m
l_{st}	Width of a single groove
p	Dimensional pressure, Pa
G_θ, G_r	Turbulent coefficients
H	Dimensionless film thickness
H_{st}	Dimensionless depth of a groove
P	Dimensionless pressure
P_0	Reference pressure, Pa
μ	Viscosity, Pa·s
μ_0	Reference viscosity, Pa·s
θ	Angular coordinate
θ_{st}	Groove angle
Ω	Rotary speed, rpm

2 Mathematical Model

The schematic view of the groove-textured water-lubricated thrust bearing is shown in Figure. 1. The water was supplied through the inner side of the film and outflowed through the outer side of the film.

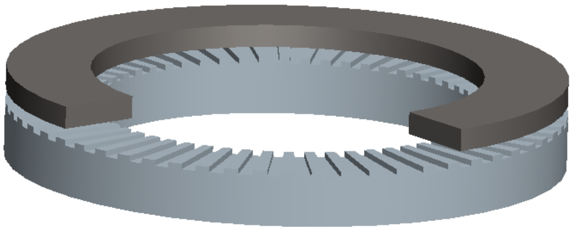

Figure 1. Schematic view of the groove-textured thrust bearing.

Both of the high rotary speed and low viscosity of water lead to large Reynolds number. In consequence, the water film becomes high turbulent. For an isoviscous incompressible fluid, the dimensionless Reynolds equation governing the turbulent flow for the water-lubricated thrust bearing is given as [9]:

$$\frac{\partial}{\partial R}\left(\frac{G_x R H^3}{\bar{\mu}}\frac{\partial P}{\partial R}\right)+\frac{\partial}{\partial \theta}\left(\frac{G_y H^3}{R\bar{\mu}}\frac{\partial P}{\partial \theta}\right)=\frac{R}{2}\frac{\partial H}{\partial \theta}+R\frac{\partial H}{\partial \tau} \quad (1)$$

Where $R=r/r_{in}$, $H=h/h_0$, $P=p/P_0$, $P_0=\Omega r_0^2\mu_0/c^2$, $\bar{\mu}=\mu/\mu 0$. The details of the turbulent coefficients G_r, G_θ can be obtained as follows [10]:

$$G_r = \frac{1}{12 + 0.0043\left(\mathrm{Re}_h\right)^{0.96}}$$
$$G_\theta = \frac{1}{12 + 0.0136\left(\mathrm{Re}_h\right)^{0.9}} \quad (2)$$

At both of the supply and exit plane of the bearing, the pressure takes a constant value equal to the ambient pressure.

$$P_{in}=P_{out}=1 \quad (3)$$

The Reynolds cavitation boundary condition is adopted.

The bottom line studied here is parabola, so the film thickness is expressed as follows:

$$H=\begin{cases}1 & \text{film land}\\ 1+H_{st} & \text{in the groove}\end{cases} \quad (4)$$

$$H_{st}=\begin{cases}0 & if\ (X,Y\notin\Psi)\\ -\frac{4h_{st0}}{l_{st}^2 h_0}X^2+\frac{h_{st0}}{h_0} & if\ (X,Y\in\Psi)\end{cases} \quad (5)$$
$$\Psi: -\frac{l_{st}}{2}\le X\le\frac{l_{st}}{2}\ and\ r_i\le Y\le r_o$$

Where, H_{st} is the dimensionless film thickness at the groove, r_i and r_o are the inner radial and outer radial of the thrust bearing, respectively.

Under the assumption of small amplitude motion of the thrust bearing, the perturbed form of the pressure expression and film thickness expression are substituted into the Reynolds equation to derive the zeroth and first-order of the disturbed Reynolds equation:

$$\frac{\partial}{\partial R}\left(\frac{G_r R H_0^3}{\bar{\mu}}\frac{\partial P_\xi}{\partial R}\right)+\frac{\partial}{\partial \theta}\left(\frac{G_\theta H_0^3}{R\bar{\mu}}\frac{\partial P_\xi}{\partial \theta}\right)=F_\xi,\ (\xi=0,z,\dot{z})$$
$$F_0=\frac{R}{2}\frac{\partial H_0}{\partial \theta}$$
$$F_z=-\frac{\partial}{\partial R}\left(\frac{3G_r R H_0^2}{\bar{\mu}}\frac{\partial P_0}{\partial R}\right)-\frac{\partial}{\partial \theta}\left(\frac{3G_\theta H_0^2}{R\bar{\mu}}\frac{\partial P_0}{\partial \theta}\right)$$
$$F_{\dot{z}}=R \quad (6)$$

The finite difference method is used to solve the Reynolds equation. The stiffness K_{zz} and damping coefficient C_{zz} of the thrust bearing can be calculated by integrating the perturbed pressure across the fluid film.

$$K_{zz}=-\iint \alpha P_z R d\theta dR \quad (7)$$

$$C_{zz}=-\iint \alpha P_{\dot{z}} R d\theta dR \quad (8)$$

3 Calculation procedure

The Reynolds equation is numerically solved by finite difference method (FDM). A computational MATLAB code is developed to calculate the performance of the bearing. The calculation process includes two main steps: static performance calculation and dynamic characteristics calculation. At first, the steady Reynolds equation is numerically solved to obtain the steady pressure distribution. The steady Reynolds equation refers to the equation (6) when the subscript of ξ satisfies ξ=0. Secondly, once the steady pressure distribution is obtained, the right part of the perturbed Reynolds equation (6) can be determined when the static pressure distribution P_0 is obtained. The FDM method is also used to discretize the perturbed equation.

4 Results and discussion

To investigate the effect of groove texture on the dynamic characteristics of the water-lubricated thrust bearing, the parameters of rotary speed, texture depth, and texture circumferential angle are selected. The parameters of the bearing and groove are listed in Table 1.

Table 1. Parameters of the groove-textured bearing.

Item	Value
Inner Diameter [mm]	35
Outter Diameter [mm]	50
Viscosity [Pa s]	0.001
Density [kg/m^3]	1000
Number of grooves	60

4.1. Effect of rotary speed

Figure 2 shows the predicted stiffness of the bearing with groove texture in respect to the rotary speed. The stiffness represents the ratio of force to the displacement. The depth and circumferential angle of a groove are 15μm and 3°, respectively. The film clearance h_0 is 15μm. As shown in the figure, the stiffness increases significantly with the rotary speed. This is expected, as the hydrodynamic effect induced by the groove texture improved with the increased rotary speed.

For damping coefficient shown in Figure 3, it increases gradually with the rotary speed.

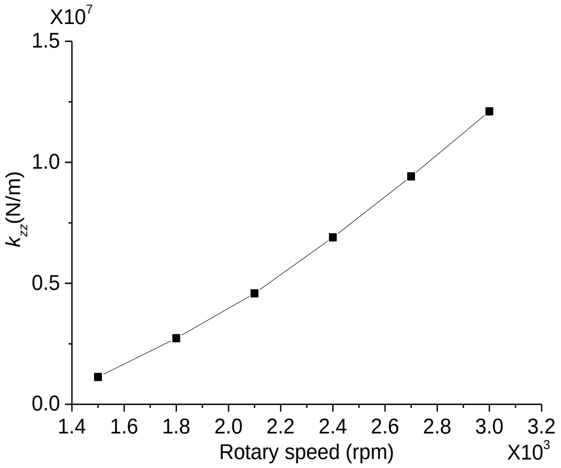

Figure 2. Stiffness vs rotary speed

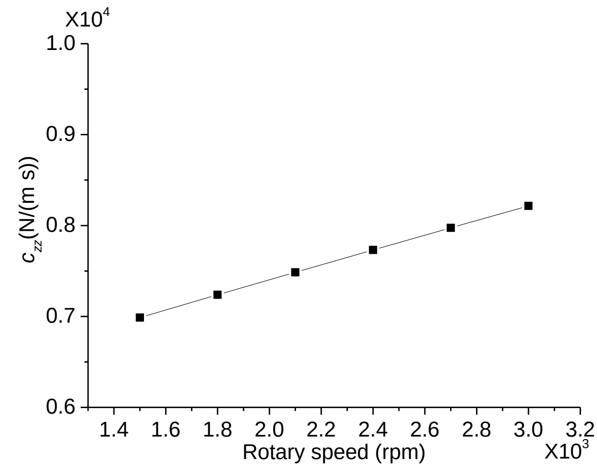

Figure 3. Damping coefficient vs rotary speed

4.2 Effect of film clearance

Figure 4 shows the effect of the film clearance on the stiffness of the bearing. The film clearance varies from 6μm to 21μm. The circumferential angle is 3°.

According to the results, the stiffness decreases dramatically in respect to the film clearance. When the film clearance increases to 21μm, the stiffness is quite small. The reason for this is probably that the larger the film clearance is, the weaker the hydrodynamic effect is.

As shown in the Figure 5, the damping coefficient decreases from about 3×10^4N/(ms) to about 0.38×10^4N/(ms).

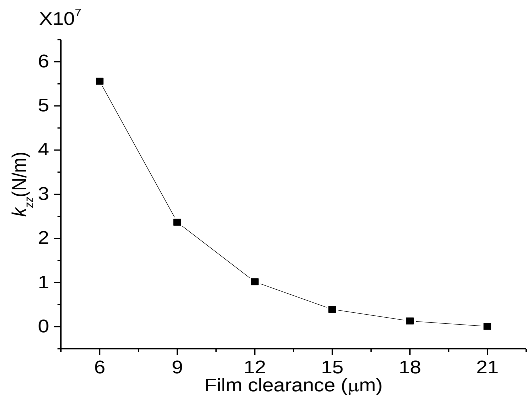

Figure 4. Stiffness vs film clearance

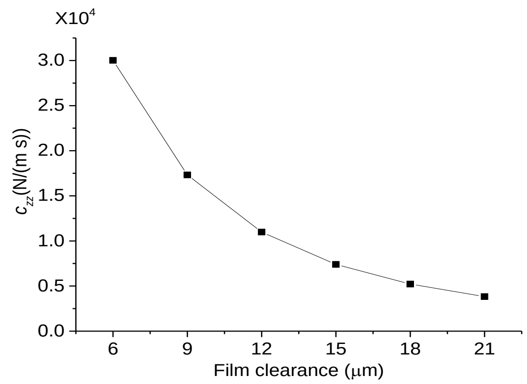

Figure 5. Damping coefficient vs film clearance

4.3 Effect of texture depth

Figures 6-7 illustrate the influence of the groove texture depth on the stiffness and damping coefficient of the bearing. The depth varies from 5μm to 25μm. The rotary speed is 20000rpm. The film clearance and circumferential angle of a groove are 15μm and 3°, respectively.

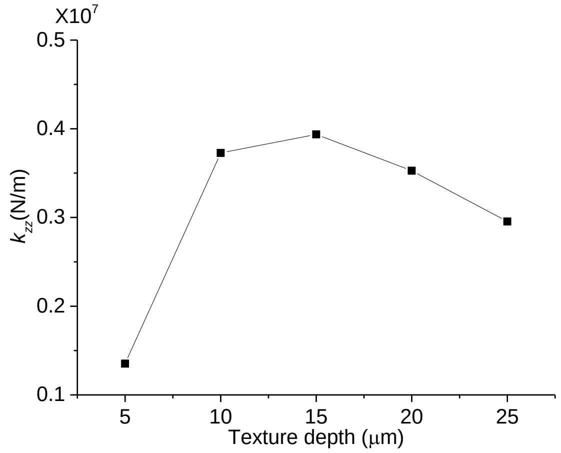

Figure 6. Stiffness vs texture depth

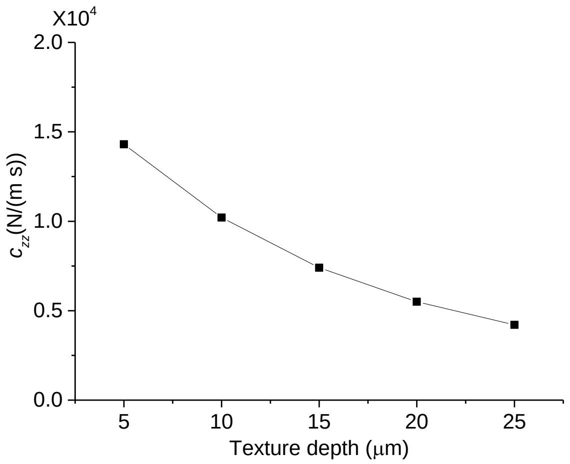

Figure 7. Damping coefficient vs texture depth

According to the results, the stiffness increases first and then decreases with the increasing texture depth. The reason for this is probably that the hydrodynamic effect increases first with the increasing texture depth; however, when the texture depth exceeds the film thickness, the drop of the pressure begins to be significant when the water flows from film land to texture. When the texture depth increases from 5μm to 10μm, there exists a significant increase for the stiffness; however, when the stiffness begins to decrease, the variation trend is slow. There exists a peak value when the texture depth is 15μm.

However, for the damping coefficient shown in Figure 7, it decreases gradually with the increased texture depth.

4.4 Effect of circumferential angle

The influences of circumferential angle on the dynamic characteristics of the bearing are illustrated in Figures. 8-9.

As shown in the figure, the circumferential angle is used to illustrate the texture width. Similar to the trend of the stiffness in respect to the texture depth, the stiffness increases first and then decreases with the circumferential angle. There is a peak value when the circumferential angle is equal to 4^o.

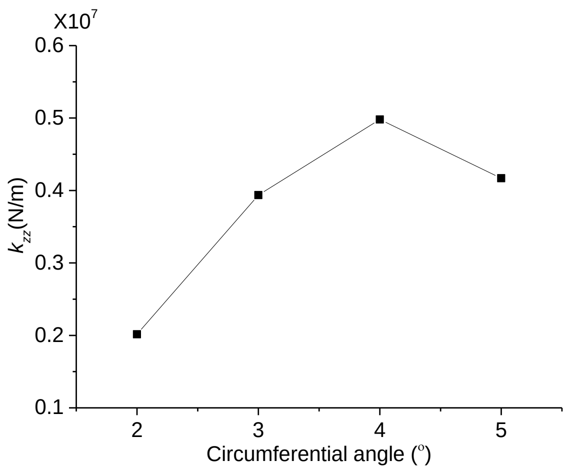

Figure 8. Stiffness vs circumferential angle

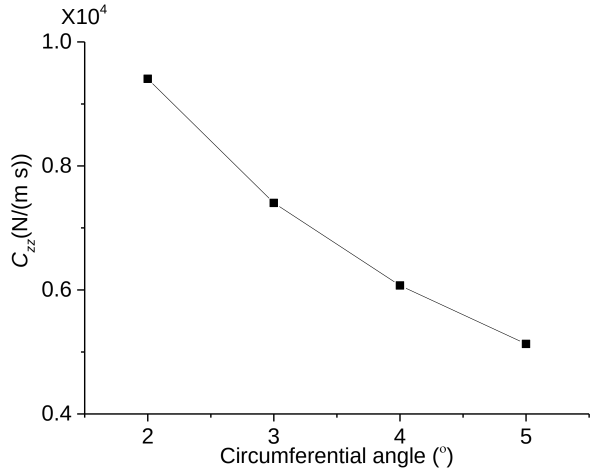

Figure 9. Damping coefficient vs circumferential angle

However, for the damping coefficient shown as in Figure 9, it still decreases in respect to the circumferential angle. With the circumferential angle increases from 2^o to 5^o, the damping coefficients decreases from 0.94×10^4N/(ms) to 0.52×10^4N/(ms).

5 Summary

The research carried out focused on the effect of groove texture on the stiffness and damping coefficients of water-lubricated thrust bearing. The turbulence is also included considering large Reynolds number lead by high rotary speed and low viscosity of water. The zeroth and first-order disturbed Reynolds equation to determine the dynamic characteristics and solved by FDM. The effects of rotary speed, film clearance, groove depth and circumferential angle on the dynamic characteristics of the bearing are analyzed. The main conclusions are summarized as follows:

(1) The stiffness and damping coefficients increase significantly with the increasing rotary speed due to the enhanced hydrodynamic effect.

(2) There exists an optimal texture depth and a circumferental angle for the stiffness of the bearing.

Acknowledgments

This work is supported by the National Natural Science Foundation of China (Grant NO. 51705131), the Fundamental Research Funds for the Central Universities (Grant NO.2016B14814), the Natural Science Foundation of Jiangsu Province (Grant NO. BK20160286) and the Changzhou Sci & Tech Program (Grant No. CJ20179022).

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