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Mixed Elastohydrodynamic Lubrication in Micro Wedge-Platform Thrust Bearing Involving Adsorbed Layer

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1. Introduction

Micro thrust bearings have important application values in supporting the loads and reducing the friction and wear in micro rotating machinery. With the developments of micro machines and micro robots, they have received intensive attention [1-4]. As rotating mechanical components, they are required to be lubricated normally by gas or the fluids with short-chain-length molecules. The surface clearances in them are intrinsically very small and often no more than 100 nm. There are unavoidably the physically adsorbed layer on the bearing surface the thickness of which is determined by the fluid-bearing surface interaction and can often reach the 1nm scale [5, 6]. In this circumstance, the effect of the physically adsorbed layer in the bearing can not be ignored [7]. Thus, classical hydrodynamic theories [8], which ignore the physically adsorbed layer and assume the fluid in the whole contact as continuum, are no longer applicable for micro thrust bearings.

A traditional dint to study micro bearings is to use molecular dynamics simulation (MDS) to simulate the flow behavior in the bearing or to use MDS to simulate the flow of the fluid molecule layers adjacent to the bearing surfaces and use the continuum fluid model to simulate the flow of the fluid between the two adsorbed layers [9-12]. For the bearing (length and width) sizes on the 1nm or 10nm scales, these approaches may be usable. However, for bigger bearing sizes they are difficult to implement because of the over huge cost of the computational source. New analytical approaches for them must be sought for solving this existing problem.

In recent years, Zhang derived the closed-form explicit multiscale flow equations for the two-dimensional flow problem by considering the molecular-scale non-continuum adsorbed layer flow and the intermediate continuum fluid film flow [13]. The advantage of his equations is to give fast solutions to the engineering micro channel flow with a normal computer.

In micro thrust bearings, the surface roughness of the bearing surface even on the molecular-size scale may be comparable to the bearing clearance, and the surface roughness effect should be considered. When both the magnitude of the surface roughness and the wavelength of the waviness of the rough surface are on the same scale with the bearing clearance, the Stokes roughness should be present and the Navier-Stokes equation should be used to account for the vortex flow [14, 15].

In a micro thrust bearing, although the film pressures might be as low as on the scales of 1 MPa or 10 MPa, the magnitudes of the elastic deformations of the bearing surfaces caused by them may be comparable to the bearing clearance. Thus, the effect of the surface elastic deformation may be significant in a micro thrust bearing.

The present paper analytically addresses the performance of the micro wedge-platform thrust bearing when the effects of the surface roughness, the surface elastic deformation and the physically adsorbed layer on the bearing surface are incorporated. Besides the film pressure and film thickness distributions and the carried load of the bearing, the results of the friction coefficient of the bearing are presented, calculated from the multiscale flow theory.

2. Micro Wedge-Platform Thrust Bearing in Mixed Lubrication

A micro thrust bearing can be equivalently treated as the bearing formed by the stationary rough elastic pad and the smooth rigid plate moving the speed u as shown in Fig. 1. Here, on the stationary bearing surface is imposed the sinusoidal surface roughness. On both the bearing surfaces there are the physically adsorbed layer formed owing to the fluid-bearing surface interaction. The bearing is in the so-called "mixed lubrication" where the magnitude of the surface elastic deformation under the film pressures, the surface roughness and the thickness of the adsorbed layer all are comparable to the bearing clearance, which is on the scales of 100 nm or 10 nm or even less. In the present study, although the magnitude of the surface roughness (R_z) may be on the same scale with the bearing clearance, the wavelength of the waviness of the rough surface is 2 to 3 orders higher than the minimum bearing

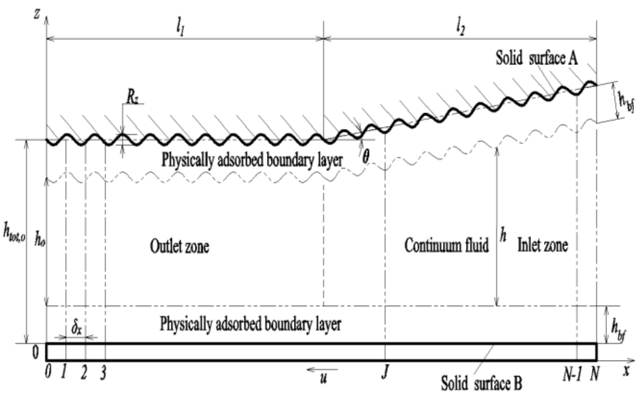


Fig. 1 The studied hydrodynamic lubricated micro wedge-platform thrust bearing involving the effects of the surface roughness, the surface elastic deformation and the physically adsorbed layer on the bearing surface

clearance so that the effect of the Stokes roughness is negligible and the flow can still be assumed as in laminar [14]. The lubricating fluid is often simple fluids like water or the fluids with short-chain-length molecules; Under the entrainment of the moving surface, the physically adsorbed fluid molecule layers should be considered as essentially in flow.

In Fig. 1, the thickness of the adsorbed layer is h_{bf}, the continuum fluid film thickness is h, the continuum fluid film thickness on the exit of the bearing is h_0, the surface separation on the exit of the bearing is $h_{tot,0}$, the widths of the outlet and inlet zones of the bearing are respectively l_1 and l_2, and the wedge angle of the bearing is θ. The order numbers of the discretized points evenly distributed in the whole lubricated area are also shown in Fig. 1.

3. Mathematical Analysis

Zhang's multiscale flow equations [13] were used to describe the molecular-scale non-continuum flow of the adsorbed layer and the flow of the intermediate continuum fluid film. The analysis is based on the following assumptions:

a. The physical properties of the two adsorbed layers are the same;

b. The fluid is Newtonian;

c. No interfacial slippage occurs on any interface;

d. The flow is isothermal;

e. The side leakage is negligible.

For the used low sliding speed $u = 1.0\times10^{-4}$ m/s, assumptions b-d stand; If the bearing length (in the third coordinate direction) is much larger than the bearing width ($l_1 + l_2$), assumption e stands.

The detailed analysis for the hydrodynamic lubrication in the thrust bearing has been shown by Zhu and Zhang [16]. Here, that analysis is borrowed, except the surface separation in the bearing is expressed as:

$$h_{tot} = h_{00} + f(x) + R_z \sin(\omega x + \phi)/2 - \\ - \frac{2}{\pi E_v} \int_0^{l_1+l_2} p(s) \ln(x-s)^2 \, ds, \tag{1}$$

where E_v is the equivalent Young's modulus of elasticity of two bearing surfaces, h_{00} is constant, p is the film pressure, R_z is the maximum height of the surface roughness, ω is the frequency of the waviness of the rough surface, φ is the phase angle of the rough surface, x is the coordinate in the flow direction shown in Fig. 1, s is the integral variable, and

$$f(x) = \begin{cases} 0 & , \text{ for } 0 \le x \le l_1 \\ (x - l_1)\tan\theta & , \text{ for } l_1 < x \le l_1 + l_2 \end{cases}. \tag{2}$$

The calculations of the shear stresses τ_a and τ_b respectively on the stationary and moving bearing surfaces in the present sandwich film lubricated area have been shown by Xiao and Zhang [17]. Once τ_a and τ_b are calculated, the friction coefficients on the stationary and moving bearing surfaces in the present study are respectively numerically calculated as:

$$f_a = \frac{\sum_{J=1}^{N} \tau_{a,J} \cdot \delta_x}{w}, \quad f_b = \frac{\sum_{J=1}^{N} \tau_{b,J} \cdot \delta_x}{w}, \tag{3}$$

where w is the load per unit contact length carried by the bearing, δ_x is the distance between the neighboring discretized points, N is the maximum order number of the discretized point, and the subscript "J" is the order number of the discretized point.

4. Numerical Calculation and Parameter Formulation

In the present study, the numerical calculation and the numerical solution procedure are the same with those shown by Zhu and Zhang [16] and here not repeated.

The fluid piezo-viscous effect and the fluid compressibility under pressure were considered. The dependence of the fluid viscosity and density on film pressure has been formulated by Zhu and Zhang [16]. The weak, medium and strong fluid-bearing surface interactions were respectively considered. The values of the characteristic parameters for them were presented by Zhu and Zhang [16].

5. Calculation Results

Exemplary calculations were made for the following input operational parameter values:

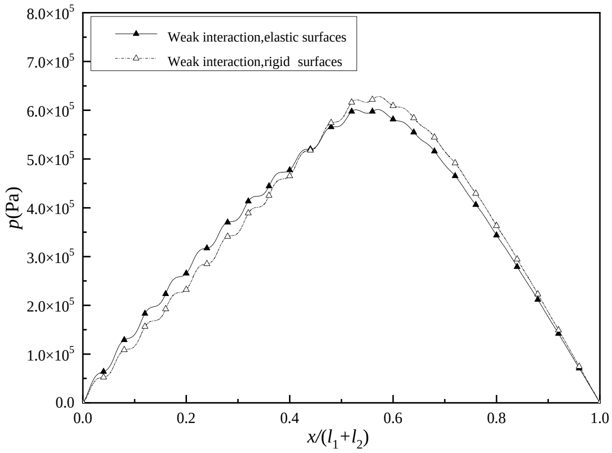
$D = 0.5$ nm, $\Delta_{n-2} / D = \Delta x / D = 0.15$, $\theta = 1.0\times10^{-4}$ rad,
$l_1 - l_2 = 100$ μm, $\beta = 4.0\times10^{-4}$ Pa^{-1}, $\omega = 6.28\times10^5$ rad/m,
$\phi = \pi$, $N = 1000$, $\alpha = 1.6\times10^{-8}$ m^{-2}/N, $\eta_a = 0.03$ Pa·s,
$E_v = 209$ GPa, $u = 1.0\times10^{-4}$ m/s,

where D is the fluid molecule diameter, η_a is the fluid ambient viscosity, α is the fluid viscosity-pressure index, β is the fluid density-pressure coefficient, Δx is the separation between the neighboring fluid molecules in the x coordinate direction in the adsorbed layer, and Δ_{n-2} is the separation between the neighboring fluid molecules across the adsorbed layer thickness just on the boundary between the adsorbed layer and the continuum fluid film [16].

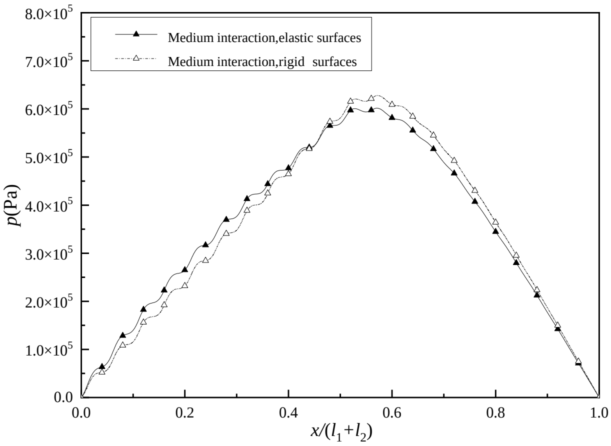
5.1. Film pressure distribution

Figs. 2, a-c show the film pressure distributions respectively for the elastically deformed and rigid bearing surfaces under different fluid-bearing surface interactions when $w = 70$ N/m and $R_z = 2$ nm. Since the load w is the same, it is reasonable that the film pressure distributions in these figures for different fluid-bearing surface interactions are similar for the same geometrical sizes of the bearing. However, it is noticed that for whichever fluid-bearing surface interaction the surface elastic deformation significantly reduces the maximum film pressure and considerably modifies the film pressure distribution. There are the significant film pressure ripples caused by the molecular-scale surface roughness, however the surface elastic deformation reduces the film pressure ripples.

Figs. 3, a-c show that for the given operating condition, whenever the bearing surfaces are elastically deformed or assumed as rigid, if the surface roughness (i.e. the value of R_z) is greater, the film pressures as well as the film pressure ripples are higher especially for the strong fluid-bearing surface interaction. However, this increasing



a



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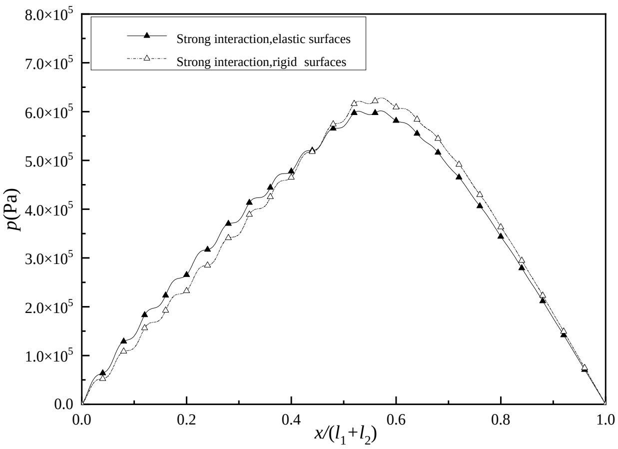
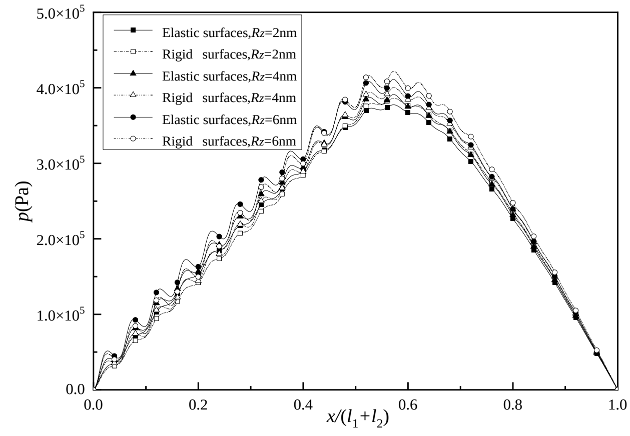
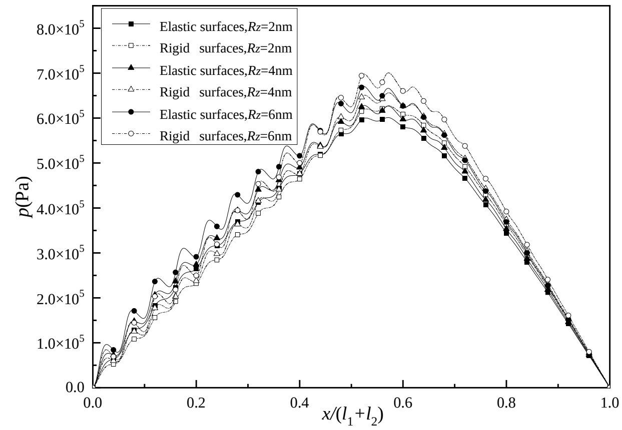


c

Fig. 2 Comparisons between the film pressure distributions respectively for the elastically deformed and rigid bearing surfaces under different fluid-bearing surface interactions when w = 70 N/m and R_z = 2 nm: a – for the weak interaction, b – for the medium interaction, c – for the strong interaction



a



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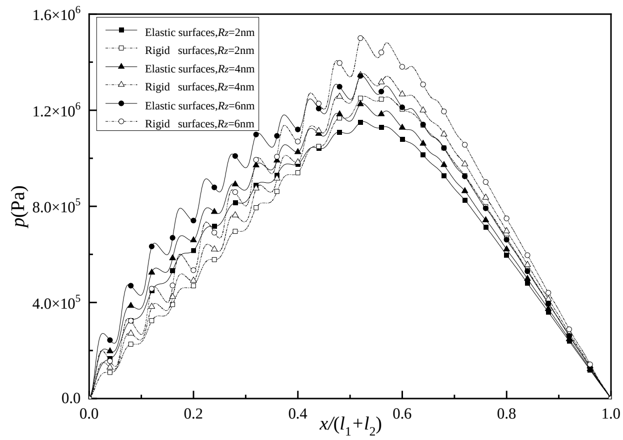


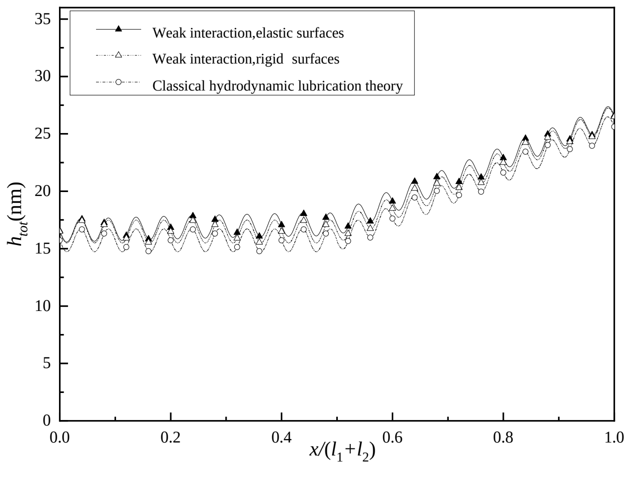
c

Fig. 3 Surface roughness effects on the film pressure distributions respectively for the elastically deformed and rigid bearing surfaces under different fluid-bearing surface interactions when $h_{tot,0}$ = 20 nm: a – for the weak interaction, b – for the medium interaction, c – for the strong interaction

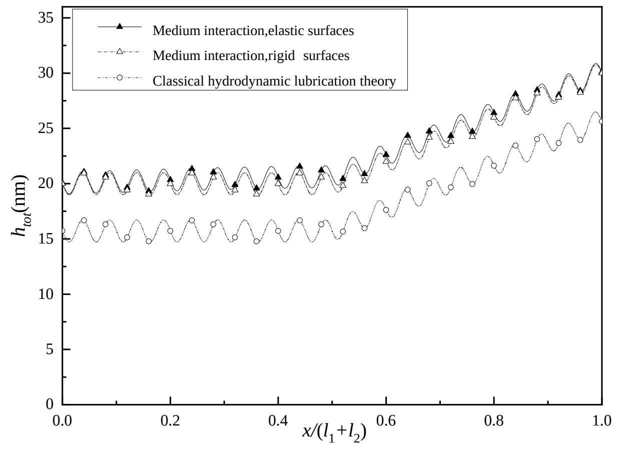
effect of the surface roughness on the film pressure is weaker for the elastically deformed bearing surfaces than for the rigid bearing surfaces, especially for the strong fluid-bearing surface interaction. For a given value of $h_{tot,o}$, the reductions of the maximum film pressure and the film pressure ripples owing to the surface elastic deformation are greater when the fluid-bearing surface interaction is stronger and the surface roughness is higher.

5.2. Film thickness distribution

Figs. 4, a-c show that for the given load, the surface elastic deformation modifies the total film thickness distribution and increases the total lubricating film thick-



a



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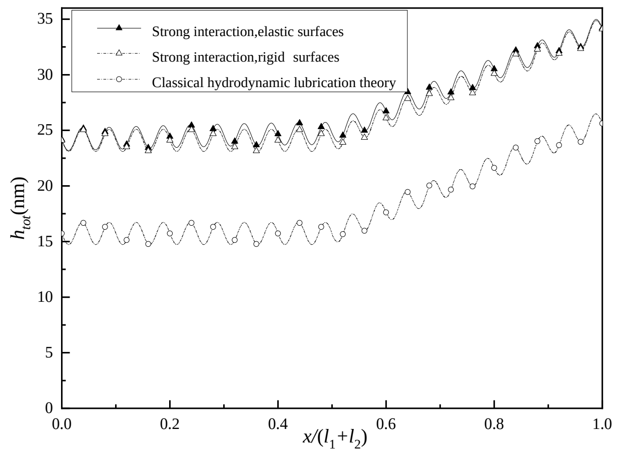


c

Fig. 4 Comparisons between the film thickness distributions respectively for elastically deformed and rigid bearing surfaces under different fluid-bearing surface interactions when w = 70 N/m and R_z = 2 nm: a – for the weak interaction, b – for the medium interaction, c – for the strong interaction

nesses compared to the results for the rigid bearing surfaces. The rough surface appears to be some flattened with the reduced surface roughness under the rippled film pressures. In the present mixed lubrication with the rough surface, when the fluid-bearing surface interaction is stronger,

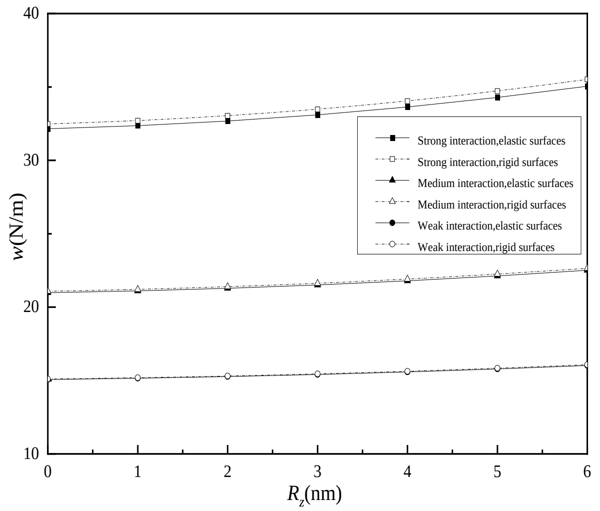
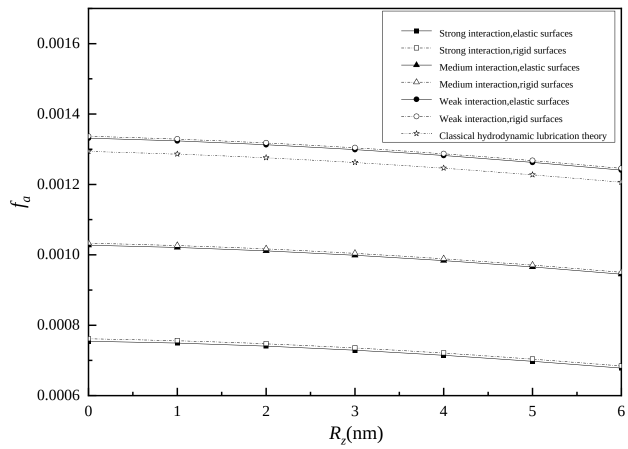
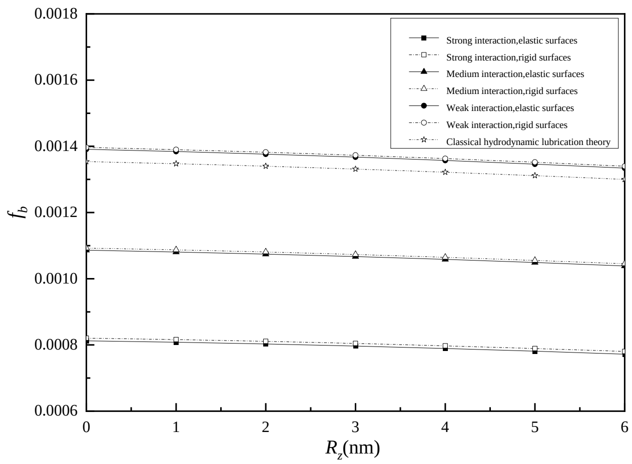


Fig. 5 Influence of the surface roughness on the carried load of the bearing respectively for the elastically deformed and rigid bearing surfaces under different fluid-bearing surface interactions when $h_{tot,0}$ = 30 nm



a



b

Fig. 6 Influences of the surface roughness on the friction coefficients (f_a and f_b) when $h_{tot,o}$ = 30 nm:
a – friction coefficient on the stationary surface,
b – friction coefficient on the moving surface

the total lubricating film thickness is higher for the given load. This is particularly the case for the strong fluid-bearing surface interaction. It indicates the great benefit of the strong fluid-bearing surface interaction in increasing the lubricating film thickness to separate the coupled surfaces to reduce the friction and wear in a hydrodynamic lubricated micro thrust bearing, the performance of which strongly depends on the effect of the physically adsorbed layer on the bearing surface.

5.3. Bearing load

Fig. 5 shows that for a given value of $h_{tot,0}$, the carried load of the bearing is increased with the increase of the surface roughness i.e. the value of R_z. This is particularly the case for the strong fluid-bearing surface interaction. For given values of $h_{tot,0}$ and R_z, the surface elastic deformation results in a bit reduction of the load-carrying capacity of the bearing especially for the strong fluid-bearing surface interaction. It is shown that even the molecular-scale surface roughness has a considerable influence on the load-carrying capacity of the bearing.

5.4. Friction coefficient

Figs. 6, a-b show that for the given value of $h_{tot,0}$, the friction coefficients (f_a and f_b) respectively on the stationary and moving bearing surfaces are reduced with the increase of the surface roughness. The surface elastic deformation also results in a bit reduction of the friction coefficient of the bearing. If the fluid-bearing surface interaction is stronger, the friction coefficient of the bearing is lower. This again shows the great benefit of the strong fluid-bearing surface interaction in a micro thrust bearing i.e. heavily reducing the friction coefficient and pronouncedly conserving the energy cost in the bearing.

6. Conclusions

The mechanism of the load carrying and friction of a hydrodynamic lubricated micro wedge-platform thrust bearing was numerically investigated when the effects of the surface roughness, the surface elastic deformation and the physically adsorbed layer on the bearing surface were incorporated. The continuum fluid film is present in the whole lubricated area, but the effect of the physically adsorbed layer is significant because of the bearing clearance comparable to the thickness of the adsorbed layer. The flow in the bearing is multiscale and Zhang's multiscale flow equations [13] were used for solving the film pressure distribution in the bearing; The friction coefficient of the bearing was also correspondingly calculated. The used surface roughness is much smaller than the minimum bearing clearance, and the wavelength of the waviness of the rough surface is 2 to 3 orders greater than the minimum bearing clearance. In this condition, the Reynolds roughness is still valid and the flow can be taken as in laminar.

Exemplary calculations were made for the sliding speed $u = 1.0\times10^{-4}$ m/s. According to the calculation results, the conclusions are drawn as follows:

a. The effect of the surface roughness even on the 1nm scale is significant in the micro thrust bearing. It strongly depends on the fluid-bearing surface interaction owing to the effect of the adsorbed layer. For a given minimum bearing clearance, the increase of the surface roughness causes the higher film pressure ripples, raises the carried load of the bearing, but reduces the friction coefficient of the bearing; When the fluid-bearing surface interaction is stronger, these effects are more significant. For the elastically deformed bearing surfaces, the surface roughness effect is weaker than for the rigid bearing surfaces.

b. The effect of the surface elastic deformation is normally significant in the micro bearing. For a given load, it considerably modifies both the film pressure and film thickness distributions, reduces the maximum film pressure but increases the total lubricating film thicknesses. It also a bit reduces the friction coefficient of the bearing.

c. Because of the effect of the physically adsorbed layer, the strong fluid-bearing surface interaction has great application values in a hydrodynamic lubricated micro thrust bearing for the resulting much higher load-carrying capacity (for a given minimum bearing clearance) or much higher lubricating film thicknesses (for a given load) but much lower friction coefficient of the bearing.

d. In studying the performance of a hydrodynamic lubricated micro thrust bearing, the new analytical approach such as the multiscale hydrodynamic analysis should be used by simultaneously considering the effects of the surface roughness, the surface elastic deformation and the physically adsorbed layer on the bearing surface.

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MIXED ELASTOHYDRODYNAMIC LUBRICATION IN MICRO WEDGE-PLATFORM THRUST BEARING INVOLVING ADSORBED LAYER

S u m m a r y

The film pressure and film thickness distributions, carried load and friction coefficient of the hydrodynamic lubricated micro wedge-platform thrust bearing were numerically calculated when the effects of the surface roughness, the surface elastic deformation and the physically adsorbed layer on the bearing surface were incorporated. The bearing clearance is so small that the effect of the adsorbed layer is significant but in the whole lubricated area there is the continuum fluid film. The flow in the bearing is multiscale and the solution is sought from the multiscale approach. The obtained results show that for the given load, sliding speed and surface roughness, the surface elastic deformation modifies both the film pressure and film thickness distributions, it makes the total lubricating film thickness increased but reduces the maximum film pressure; For the given minimum bearing clearance and sliding speed, the increase of the surface roughness increases both the film pressure and carried load of the bearing but the surface elastic deformation reduces this surface roughness effect especially when the fluid-bearing surface interaction is strong; The surface elastic deformation, the increase of the surface roughness and the increase of the interaction strength between the fluid and the bearing surface all result in the reduction of the friction coefficient of the bearing.

Keywords: adsorbed layer, elastohydrodynamics, mixed lubrication, multiscale hydrodynamics, surface roughness, thrust bearing.

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