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Elastohydrodynamic simulation of Rayleigh step bearings in thin film hydrodynamic lubrication

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ABSTRACT

An elastohydrodynamic numerical simulation is conducted for one-dimensional Rayleigh step bearings. The numerical model incorporates the piezoviscous effect of the lubricant and the elastic deformation of the bounding surfaces to solve the one-dimensional Reynolds equation. It is found that a small elastic deformation of less than 200 nm is responsible for film formation in thin film hydrodynamic lubrication. As the film thickness decreases, the divergent shape in the step zone causes delay in pressure growth, resulting in considerable reduction of load capacity while the convergent shape in the land zone improves slightly load capacity in some cases. Rayleigh step bearings have higher load capacity than fixed plane bearings because of the remained step shape.

1. Introduction

Recently, significant attention has been given to understanding the flow in thin films produced in lubricated contact areas. The use of low viscosity lubricants has been desirable because they allow high efficiency operation of machinery under high loads and high speeds. As a result, the film in the lubricated area becomes increasingly thinner. The lubrication regime during operation thus expands toward more severe conditions.

For the elastohydrodynamic (EHD) condition found in non-conformal contacts, it is well recognised that the piezoviscous effect and elastic deformation are responsible for the film formation because of the significant pressure of several gigapascals [1]. The established EHD numerical model that incorporates both these effects can predict the film thickness down to 1 nm [2]. On the other hand, in the case of conformal contacts, understanding the flow in thin films has been lacking because most studies have focused on the thicker film regime under high sliding speeds.

In a previous study [3], the authors focused on hydrodynamic lubrication in thin films to conduct an EHD simulation of one-dimensional fixed slider plane bearings. It was found that the small elastic deformation produced by low nominal pressures of less than 5 MPa caused a significant pressure reduction in thin films. In the present study, an elastohydrodynamic numerical simulation is conducted for one-dimensional Rayleigh step bearings [4] with thin film hydrodynamic lubrication. The obtained numerical solutions are compared with the isoviscous and rigid (IR) theory to discuss the influence of the small elastic deformation occurring in thin films under low nominal pressures.

2. Background

In the non-conformal contacts found in rolling bearings, cam-tappet systems, and traction drives, the load is supported in a small area, which causes a significant increase in pressure, up to several gigapascals. Under the EHD condition, the piezoviscous effect of the lubricant and the elastic deformations of the surfaces play important roles in the film formation. Dowson and Higginson [5,6] numerically showed that the film profile has a specific feature with a flat shape around the centre and a constriction shape located at the outlet because a significant increase in pressure causes elastic deformations of the surfaces. Arranged equations for the central and minimum film thicknesses were suggested based on a numerical simulation for convenient estimation [7]. Johnson [8] showed that the EHD condition could be classified into four regimes depending on the magnitudes of the piezoviscous effect and elastic deformation: isoviscous and rigid (IR), piezoviscous and rigid (PR), isoviscous and elastic (IE), and piezoviscous and elastic (PE).

Measurements of the film thickness have been effectively conducted under the EHD condition found in non-conformal contacts. Optical interferometry [9] is a powerful method for precisely obtaining the shape and thickness of the film. The limitation of the minimum film thickness was overcome by using the spacer layer imaging method (SLIM) [10]. A spectrometer was

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Nomenclature

E	elastic modulus of surface (Pa)
E'	equivalent elastic modulus of bounding surfaces (Pa)
F	dimensionless friction $F=-fh_0/(\eta_0 lu)$
H	dimensionless film thickness, $H=h/h_0$
H_1	dimensionless initial film thickness, $H=h_1/h_0$
K	convergence ratio, $K=(h_1-h_0)/h_0$
L_1	step ratio, $L_1=l_1/l$
L_2	land ratio, $L_2=l_2/l$
P	dimensionless pressure, $P=h_0^2p/(6\eta_0 lu)$
Q	dimensionless mass flow rate, $Q=q/(\rho_0 h_0 u)$
Q_c	dimensionless Couette mass flow rate, $Q_c=q_c/(\rho_0 h_0 u)$
Q_p	dimensionless Poiseuille mass flow rate, $Q_p=q_p/(\rho_0 h_0 u)$
S	dimensionless coordinate in direction of surface motion, $S=s/l$
X	dimensionless coordinate in direction of surface motion, $X=x/l$
X_0	dimensionless coordinate of no-pressure location, $X_0=x_0/l$
W	dimensionless applied load, $W=h_0^2\, w/(6\eta_0 l^2 u)$
f	friction (N)
h	film thickness (m)
h_0	initial film thickness at land zone (m)
h_1	initial film thickness at step zone (m)
l	width of pad, $l=l_0+l_1$ (m)
l_0	width of land zone (m)
l_1	width of step zone (m)
p	pressure of fluid film (Pa)
q	mass flow rate, $q=q_c+q_q$ (kg/ms)
q_c	Couette mass flow rate (kg/ms)
q_p	Poiseuille mass flow rate (kg/ms)
s	coordinate in direction of surface motion (m)
u	sliding speed of moving surface (m)
x	coordinate in direction of surface motion (m)
x_0	no-pressure location (m)
w	applied load (N/m)
α	pressure–viscosity coefficient (Pa^{-1})
δ	elastic displacement of bounding surfaces, $\delta=\delta_0-\delta_c$ (m)
δ_0	elastic displacement of bounding surfaces (m)
δ_c	elastic displacement of bounding surfaces at $x=\infty$ (m)
$\bar{\delta}$	dimensionless elastic displacement of bounding surfaces (m)
ρ_0	density at $p=0$ (kg/m^3)
ρ	density (kg/m^3)
$\bar{\rho}$	dimensionless density
σ	dimensionless parameter, $\sigma=24\eta_0 l^2 u/(\pi E' h_0^3)$
η_0	viscosity at $p=0$ (Pa s)
η	viscosity (Pa s)
$\bar{\eta}$	dimensionless viscosity
μ	friction coefficient, $\mu=-f/w$
μ^*	dimensionless friction coefficient, $\mu^*=(l/h_0)\mu=F/(6W)$
ν	Poisson's ratio

combined with SLIM to measure the thickness down to 1 nm with high accuracy [11]. When base oil is used without additives, the film thickness is in good agreement with the theoretical equation in the regime down to 1 nm [12].

In the conformal contacts found in journal bearings, thrust bearings, and mechanical face seals, the pressure generated in the film is lower than that under the EHD condition. The operational film thickness is greater than that under the EHD condition. Generally, the piezoviscous effect and elastic deformation have been given less attention than the EHD condition. For machine elements, the hydrodynamic lubrication theory [13–15] is used, which has well been established. However, when high sliding and load conditions are present, both of the effects appear even in conformal contacts. Needs [16] showed that the friction coefficient increased in the Striveck curve because of the piezoviscous effect for journal bearings. Previous studies have considered the elastic deformation based on the assumption of semi-infinite bodies for fixed slider plane bearings [17–23]. Other models of compliant bodies have been used, including a shell model for journal bearings [24–26] and a beam model for a pivoted bearing [27]. In the higher load case, the actual structural deformations of the bodies estimated by the finite element method have been incorporated into the Reynolds equation [28–32].

In the high sliding case, the heat generation in the film is pronounced. In conformal contacts, the influence of this heat generation on the viscosity variations [33] and heat expansion of the surfaces has been given more attention than the elastic deformation [34–42]. Ettles et al. [34,35] showed that the influence of the thermal expansion became greater than that of the elastic deformation when a large bearing was used. Cameron [36] and Cameron and Robinson [37–39] suggested that the tapered film shape produced by heat expansion was responsible for the pressure generation in parallel thrust bearings. Baudry et al. [40] suggested that the supporting method for the pad was responsible for its distortion. Bennet and Ettles [41] suggested that a cantilever thrust bearing could form a wedge in the leading side of the pad by utilising the elastic deformation of the pad.

As shown in the above studies, the piezoviscous effect and elastic deformation may appear under the hydrodynamic lubrication condition when a high load and sliding speed are applied. Under mild conditions with a low load and sliding speed, the well-established hydrodynamic lubrication theory has been believed to be adequate [13–15]. However, when the film thickness is small, the small elastic deformations that are produced even under a low nominal pressure become comparable to the thickness. As a result, these small elastic deformations can change the film profile. Carl [42] showed that the elastic deformation cannot be ignored under a nominal pressure of 7 MPa. Hemingway [43] indicated that the elastic deformation distribution depended on the location of the support ring for the pad and played an important role in film formation under a nominal pressure of 2 MPa. Nakamura et al. [44] showed by EHD simulations that the pressure generated at the inlet chamfer deformed elastically over the contact area in a parallel slide-way. The elastic deformation was less than 10 nm but the wedge shape in film produced by the elastic deformation was sufficient to support load. Kawabata et al. [45,46] conducted EHD simulations to show that the inlet profile of thrust bearings and circular bumps used in a scroll compressor played an significant role in pressure generation over the contact area in the same way as Nakamura et al. [44].

In a previous study [3], the authors conducted an EHD simulation of one-dimensional fixed slider plane bearings with thicknesses of less than 1 μm. It was found that the elastic deformation caused the disappearance of the convergence shape and produced a flat shape in the film. Such a film profile significantly reduced the pressure compared with the isoviscous and rigid theory. In the current study, the authors conducted an EHD simulation of Rayleigh step bearings.