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Influence on Journal Bearing Considering Wall-Slip in EHL

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Abstract. Based on the line contact Elastohydrodynamic model and the limiting shear stress model of the journal bearing, the limiting shear stress at the contact interface of bearing, shaft and lubricant is discussed. And the effects of limiting shear stress on the film thickness, the pressure profile and friction coefficient are analyzed. The effect of wall-slip on elastohydrodynamic lubrication of journal bearings is discussed when the shear stress at the interface of solids and liquids reaches its limiting value. The results show that different wall-slip types will produce different lubrication states and the wall-slip of shaft interface will increase the oil film thickness in the sliding region. On the contrary, the wall-slip of the bearing interface will decrease the oil film thickness. The friction coefficient will decrease when the wall-slip occurs and the degree is greater at the interface of shaft than bearing.

1. Introduction

A considerable number of research effort has been devoted to the EHL analysis of interfacial wall-slip [1-2]. Zhang Zhahui et al [3-4] concluded that there are three kinds of wall-slip conditions in the line contact Elastohydrodynamic lubrication model and the lubrication equation of the ideal visco-plastic lubricant considering wall-slip. Jonas Stahl et al [5] utilized the limiting shear stress model indicating that the interfacial velocity of lubricant is determined by the limiting shear stress. And the influence of the wall-slip on the film pressure and film thickness can be obtained by taking the lubricant velocity represented by the limiting shear stress into the Reynolds equation.Zhang Yongbin et al [6-7] explored the influence of different sliding ratios on wall-slip under line contact.

Although a great deal of research on wall-slip was carried out, there are few studies on the effect of limiting shear stress value and distribution at the contact interface on lubricant. Therefore, in order to analyze the wall-slip in the journal bearing comprehensively, this paper uses the line contact Elastohydrodynamic model and the limiting shear stress model of the journal bearing to study the influence of the different value and distribution of the limiting shear stress on the oil film thickness, pressure profile and friction coefficient at the contact interface between the bearing and the lubricant.

2. Theory

Figure 1 shows the line contact lubrication model of the journal bearing. The region of A and B represent the bearing interfaces treated by the surface technology. The limiting shear stress at the interface between each region and the lubricant can be different value. In addition, the fluid is assumed to be Newtonian fluid.

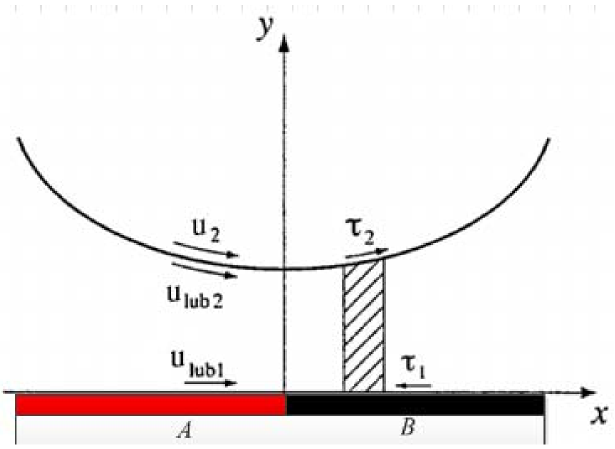

Figure 1. The model of journal bearing

Reynolds Equation. The classical Reynolds equation is based on assumption of no-slip between the lubricant and the contacting surfaces. Two new variables representing the lubricant velocities, Ulub1 and Ulub2, should be introduced considering wall-slip. Consequently, there is a slightly modified equation according to

$$\frac{\partial}{\partial x}\left(\frac{\rho h^3}{\eta}\frac{\partial p}{\partial x}\right)-6\frac{\partial}{\partial x}\left(\left(U_{\text{lub1}}+U_{\text{lub2}}\right)\rho h\right)=0 \tag{1}$$

Where the lubricant velocities either are equal to the surface velocities when there is no-slip at solid-liquid interface, Ulub1= U1, Ulub2 =U2, or equal to their value when slip occurs, Ulub1=U^s_{lub1}, Ulub2 =U^s_{lub2} , is calculated from the shear stress equations according to

$$\tau_{L1}=\tau_1=-\frac{h}{2}\frac{\partial p}{\partial x}+\eta\frac{U_2-U^s_{\text{lub1}}}{h}\Rightarrow U^s_{\text{lub1}}=U_2-\frac{h}{\eta}(\tau_{L1}+\frac{h}{2}\frac{\partial p}{\partial x}) \tag{2}$$

$$\tau_{L2}=\tau_2=\frac{h}{2}\frac{\partial p}{\partial x}+\eta\frac{U^s_{\text{lub2}}-U_1}{h}\Rightarrow U^s_{\text{lub2}}=\frac{h}{\eta}(\tau_{L2}-\frac{h}{2}\frac{\partial p}{\partial x}) \tag{3}$$

Where τLi , the limiting shear stress between lubricant and interfaces, is given by

$$\tau_{Li}=\tau_{0i}+\gamma_i P(i=1,2) \tag{4}$$

Where τ0i is the initial limiting shear stress, and γi is the limiting shear stress proportionality coefficient, and i corresponds to the surface of bearing and the surface of shaft respectively.

Film Thickness Equation. The film thickness equation, describing the gap between bearing and shaft, consists of two parts, the nominal distance between the undeformed surfaces and the elastic deformations of the surfaces, reads

$$h\left(x\right)=h_0+\frac{1}{2R}x^2-\frac{4}{\pi E'}\int_{-\infty}^{\infty}p(x')\ln(\frac{\left|x-x'\right|}{b})dx' \tag{5}$$

Force Balance Equation. Since full film conditions are assumed, the entire load per width unit is carried by the lubricant film and the calculation is simply an integration of the lubricant film pressure. Consequently, the force balance equation reads

$$w = \int_{-\infty}^{\infty} p(x)dx \tag{6}$$

Density and Viscosity. The density and viscosity, characteristic parameters of lubricant, are both pressure and temperature dependent, due to the isothermal conditions assumed in this work, and the models of density and viscosity used are only pressure dependent.

$$\eta = h_0 \exp\left\{\left(\ln \eta_0 + 9.67\right)\left[(1+5.1\times10^{-9} p)^z \text{-}1\right]\right\} \tag{7}$$

$$\rho = \rho_0 \left(1 + \frac{0.6\times10^{-9} p}{1+1.7\times10^{-9} p}\right) \tag{8}$$

Where η0, ρ0 corresponds to viscosity and density of lubricant at ambient pressure respectively.

3. Results and Discussion

Some preliminary results, using the wall-slip model, are presented and discussed here. The input parameters are taken from Table 1.

Table 1. Input parameters for the studied load case

Parameter	Value	Dimension
U	5.0×10^{-13}	[-]
W	1.0×10^{-5}	[-]
A	2.19×10^{-8}	[Pa^{-1}]
E_1、E_2	2.06×10^{11}	[Pa]
υ_1、υ_2	0.3	[-]
η_0	0.08	[Ns/m^2]
γ_1、γ_2	0.01	[-]

3.1. Influence of different types on lubricant film pressure and thickness

According to the different types of slippage, the wall-slip region is divided into four parts, shown in fig. 2(a). Figure 2(b.1-7) show seven types of slippage, where "+" and "-" represent that there is wall-slip and no-slip in corresponding region respectively.

The results with the wall-slip model were compared with no-slip analysis, and the film thickness and pressure variations from these analysis are presented in Fig. 3. From the results shown in Fig. 3, it can be seen that there is obvious difference between wall-slip and no-slip analysis. Figs. 3(b.1-3) show that film thickness decreases when slippage only occurs at the interface between lubricant and bearing, meanwhile corresponding film pressure decreases. On the contrary, Figs. 3(b.4,b.6) show the film thickness haves a significant increase when slippage only occurs at the interface between lubricant and shaft, meanwhile corresponding film pressure increases. This phenomenon can be explained by studying the lubricant flow continuity in combination with the velocity variation. Since the velocity decreases, the film thickness is forced to increase in order to maintain the lubricant flow continuity when wall-slip occurs at the shaft interface. Corresponding reason explains the decreases of film thickness when wall-slip occurs at the bearing interface.

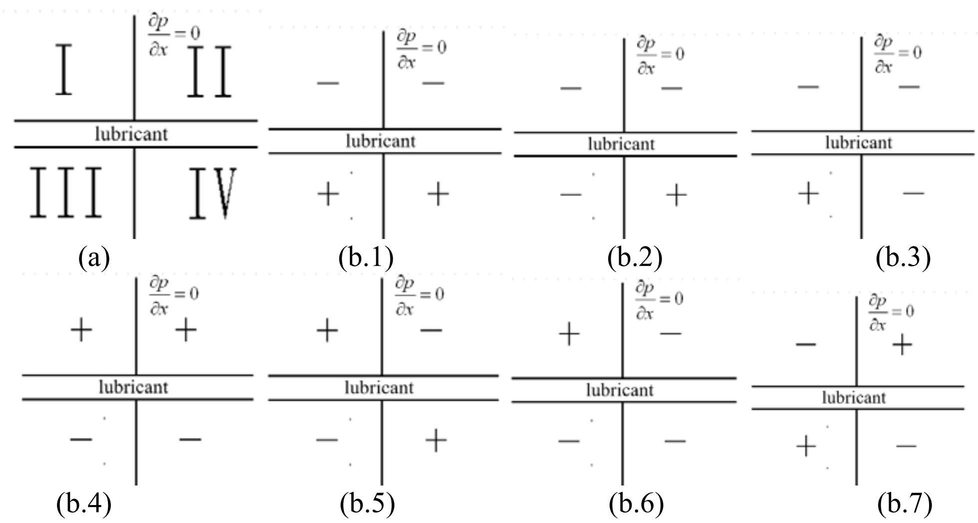

Figure 2. (a)Division of wall-slip region; (b.1-7) seven types of slippage and corresponding region

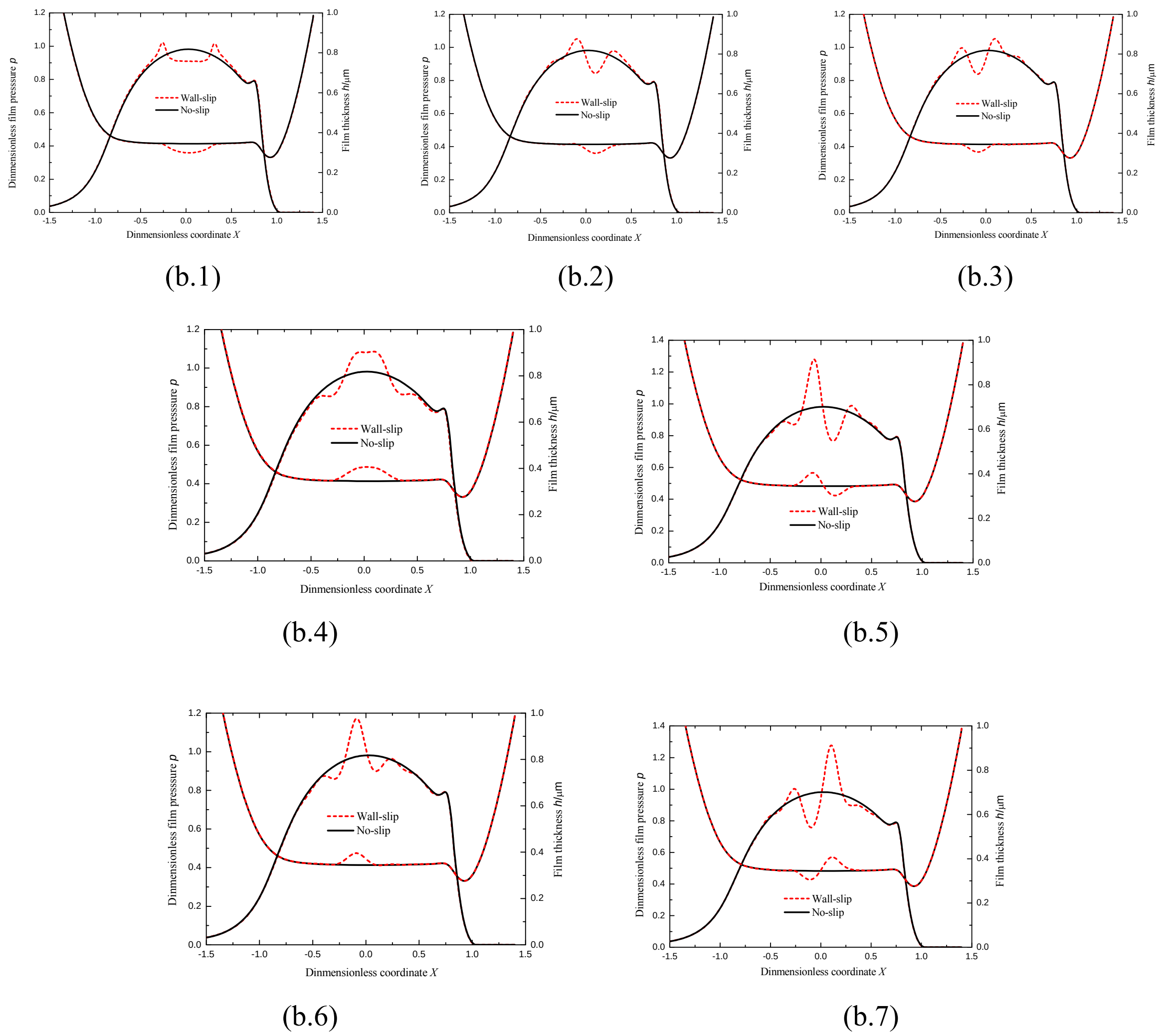

Figure 3. Film thickness and pressure distribution for a no-slip and a wall-slip Analysis (fig. 1-7 correspond to seven types of slippage)

3.2. Influence of different types on friction coefficient

There is a significant influence of wall-slip on frication coefficient in journal bearing, shown in Fig. 4. Fig. 4 illustrates the comparison between frication of no-slip and wall-slip. The friction coefficient will decrease when the wall-slip occurs greater at the interface of shaft than bearing. The phenomenon can be explained that no matter at which interface the wall-slip occurs, the shear rate caused by the velocity almost is same. But, the significant increase of pressure when wall-slip occurs at shaft interface arouses the increase of lubricant viscosity, which results in the increase of friction coefficient.

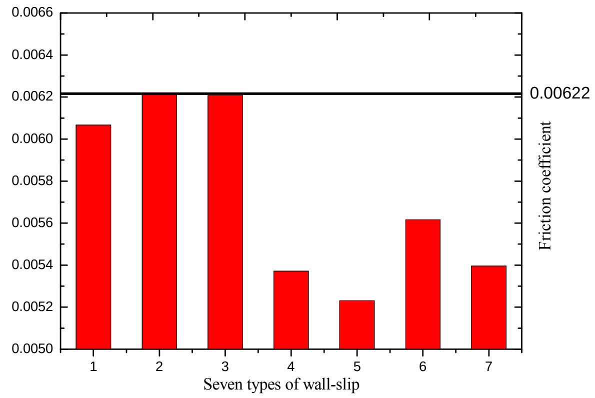

Figure 4. Frication coefficient of journal bearing considering slippage

4. Conclusion

Based on the limiting shear stress model, the effects of seven types of slippage on the pressure profile, the film thickness and friction coefficient in journal bearings were analyzed in this paper. The results indicate that there is significant influence of wall-slip on lubrication. When the slippage only occurs at the bearing interface, the oil film thickness and pressure will decrease. On the contrary, when the slippage only occurs at the shaft interface, the oil film thickness and the pressure will increase. The friction coefficient will decrease when the wall-slip occurs greater at the interface of shaft than bearing. Therefore, in order to reduce the friction coefficient of the journal bearing, we can use the method of surface modification to make Wall-slip only occur at the shaft interface.

Acknowledgments

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