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Variation of surface dimple in point contact thermal EHL under ZEV condition

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ABSTRACT

This paper focuses on the variation of the surface dimple in point contact thermal elastohydrodynamic lubrication (EHL) under zero entrainment velocity (ZEV) condition theoretically by employing Newtonian and Ree–Eyring fluid models. From higher surface velocity to lower value, both fluid models predict that the depth of dimple increases before lowering again while the Newtonian central pressure keeps rising and the Ree–Eyring one increases before showing a constant value. Moreover, there are differences in the distributions of shear stress, temperature rise, maximum temperature rise and traction coefficient. Finally, the influence of load on the dimple is also investigated. For the same load, the Newtonian fluid model predicts a deeper and slimmer centralized dimple.

1. Introduction

According to the early EHL experiments [1,2] and theories [3], the typical EHL film shape or thickness of point contact is characterized by flat thickness in contact region and a decrease in the oil film at the exit zone. And the contour map of film thickness looks like a horseshoe. However, the point contact EHL film demonstrates a dimple instead of the parallel film thickness under some special conditions. For example, in the experiment of Chiu and Sibley [4] and Kaneta et al. [5], their results showed that a dimple occurred when the glass disc was moving fast while the steel ball was stationary. Besides Kaneta et al. [5] found that the dimple vanished on the inverse kinematic condition. This unusual phenomenon was quantitatively explained by Yang et al. [6] using the "temperature–viscosity wedge" mechanism which was first proposed by Cameron and named as "viscosity wedge" [7]. In the subsequent research of Kaneta and Yang [8], two or three stable dimples could be generated between a glass disk and a 3-in. diameter steel ball under lower or higher sliding speeds. Later Čermák performed a non-steady numerical solution and the dimples occurred by impacting the glass with the steel ball [9]. Afterwards, Yang et al. [10] found that steady dimples connected with a shallower furrow occurred in elliptical glass–steel contact under pure sliding conditions. Wang et al. [11] investigated the dimple in thermal EHL of elliptical contact with arbitrary entrainment under pure sliding condition. More recently, Guo et al. [12] observed an inlet dimple together with a wedge-shaped film in point EHL experiment under pure disk sliding condition. They attributed the phenomenon to the limiting shear strength of lubricant. In 2014, Fu et al. [13] observed a tri-dimple EHL film shape in pure ball sliding experiments. This phenomenon was explained using the localized adsorption layer formed on the ball surface.

Another situation which creates a dimple in the Hertzian contact zone is high slip ratio or ZEV. In 2000, Yagi et al. [14] reported a dimple shape in the conjunction between a steel ball and a sapphire disk under high slip ratio. For the mechanism of oil film establishment under this severe condition, there are two kinds of qualitatively reasonable explanations. One is immobile boundary film effect [15], the other is the "temperature–viscosity wedge" [6]. In the complete numerical calculations of Yang et al. [6] and Guo et al. [16], it was found that the film formation under ZEV was mainly ascribed to the effect of "temperature–viscosity wedge".

Recently one of the authors observed smaller centralized ZEV dimples with quite low surface velocity in ball–disk optical interferometric experiments. Aiming at explaining such "small dimples" quantitatively, numerical simulation of line contact has been performed by the authors [17]. Newtonian and Ree–Eyring fluid models were employed in the simulations and it was found that the Ree–Eyring model successfully predicted the small centralized dimples at lower surface velocities while the calculation based the Newtonian model diverged at the same kinematic condition. How the variation of the surface dimple in point

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Nomenclature

a'	radius of Hertzian contact circle (m)
a, b	surfaces a and b
c, c_a, c_b	specific heat of lubricant and solids (J/(kg K))
d	thickness of thermal layers in solid a and b
$\overline{d}$	dimensionless thickness of thermal layers in solid a and b, d/a'
E'	reduced elastic modulus (Pa)
h	film thickness (m)
h_0	reference parameter for the dimensionless oil film (m)
h_{00}	rigid central film thickness (m)
$\overline{h}$	dimensionless film thickness for calculation, $hR_x/(a')^2$
H	dimensionless film thickness for results, $(h/R_x)\times 10^5$
k, k_a, k_b	thermal conductivity of lubricant and solids (W/(m K))
nt	time step
p	film pressure (Pa)
p_H	maximum Hertzian pressure, $3w/(2\pi a'^2)$ (Pa)
P	dimensionless pressure, p/p_H
R_a, R_b	radius of steel ball (m)
R_x, R_y	equivalent radii in x and y directions (m)
t	time (s)
$\overline{t}$	dimensionless of time, $t/(a'/u_0)$
T	temperature (K)
T_0	ambient temperature (K)
$\overline{T}$	dimensionless temperature, T/T_0
$\overline{T}_{max}$	dimensionless maximum temperature
u, v	velocity in the x and y directions (m/s)
u_0	reference velocity (m/s)
$u_{a,b}$	velocities of surface a and b (m/s)
U,V	dimensionless velocity, u/u_0, v/u_0
U_0	dimensionless reference velocity, $\eta_0 u_0/E'R_x$
$U_{a,b}$	dimensionless velocities of solids a and b, $u_{a,b}/u_0$
w	applied load (N)
W	dimensionless applied load per unit length, $w/E'R_x^2$
x, y	horizontal coordinates (m)
X,Y	dimensionless horizontal coordinates, x/a', y/a'
z, z_a, z_b	vertical coordinates of film and solids (m)
Z	dimensionless vertical coordinates of film, z/h
Z_a, Z_b	vertical coordinates of film and solids, $z_{a,b}/a'$
α	Barus' pressure–viscosity coefficient (m^2/N)
β	thermal viscosity coefficient of lubricants (K^{-1})
η	viscosity of lubricant, (Pa s)
η_0	ambient viscosity of lubricant, (Pa s)
$\overline{\eta}$	dimensionless viscosity of lubricant, η/η_0
S	slide–roll ratio, $2(u_a-u_b)/(u_a+u_b)$
ρ, ρ_a, ρ_b	densities of lubricant and solids, (kg/m^3)
ρ_0	ambient density of lubricant, (kg/m^3)
$\overline{\rho}$	dimensionless density of lubricant, ρ/ρ_0
τ_0	characteristics shear stress for Ree–Eyring lubricant (Pa)
τ	shear stress (Pa)
$\overline{\tau}$	dimensionless shear stress, τ/p_H

contact is influenced by fluid rheology is unknown. Therefore, thermal EHL calculations for point contact are carried out using both Newtonian and Ree–Eyring flow models in this project.

2. Governing equations

The constitutive equations of the fluid models employed are as follows

$$\begin{cases} \frac{\partial u}{\partial z}=\frac{\tau}{\eta} & \text{Newtonian fluid} \\ \frac{\partial u}{\partial z}=\frac{\tau_0}{\eta}\sinh\left(\frac{\tau}{\tau_0}\right) & \text{Ree-Eyring fluid} \end{cases} \tag{1}$$

where τ_0 is essentially the limit of Newtonian response and is termed characteristics shear stress of Ree–Eyring fluid. Although the Ree–Eyring sinh law was criticized for its validity by some researchers, it is presently the most widely accepted model for shear-thinning in EHD lubricants at high pressure [18].

For point contact ZEV condition shown in Fig. 1, let the lower surface be surface a with a tangential velocity u_a and the upper surface is named surface b with an opposite velocity u_b. An infinite slide–roll ratio is produced. According to the work of Yang and Wen [19], the generalized transient point contact Reynolds equation for both Newtonian and non-Newtonian model can be written as

$$\frac{\partial}{\partial x}\left[\left(\frac{\rho}{\eta}\right)_e h^3\frac{\partial p}{\partial x}\right]+\frac{\partial}{\partial y}\left[\left(\frac{\rho}{\eta}\right)_e h^3\frac{\partial p}{\partial y}\right]=6u_a\frac{\partial}{\partial x}(\rho_a^* h)+6u_b\frac{\partial}{\partial x}(\rho_b^* h)+12\frac{\partial}{\partial t}(\rho_e h) \tag{2}$$

where $(\rho/\eta)_e=12(\eta_e\rho_e'/\eta_e'-\rho_e'')$, $\rho_a^*=2(\rho_e-\rho_e'\eta_e)$, $\rho_b^*=2\rho_e'\eta_e$, $\rho_e=(1/h)\int_0^h\rho dz$, $\rho_e'=(1/h^2)\int_0^h\rho\int_0^z 1/\eta dz' dz$, $\rho_e''=(1/h^3)\int_0^h\rho\int_0^z z'/\eta dz' dz$, $\eta_e=h/\int_0^h 1/\eta dz$, $\eta_e'=h^2/\int_0^h z/\eta dz$.

The boundary and cavitation conditions of the Reynolds equation are as follows

$$\begin{cases} p(x_{in},y,t)=p(x_{out},y,t)=p(x,\pm y_{out},t)=0 \\ p(x,y,t)\ge 0\ (x_{in}<x<x_{out},\ -y_{out}<y<y_{out}) \end{cases} \tag{3}$$

The film thickness equation is

$$h(x,y,t)=h_{00}(t)+\frac{x^2}{2R_x}+\frac{y^2}{2R_y}+\frac{2}{\pi E'}\int\int\frac{p(x',y',t)}{\sqrt{(x-x')^2+(y-y')^2}}dx'dy' \tag{4}$$

The load balance equation can be expressed as

$$\int\int p(x,y,t)dxdy=w \tag{5}$$

The Roelands [20] viscosity–temperature–pressure relation allowing the viscosity of lubricants vary in all direction is adopted

$$\eta=\eta_0\exp\left\{(\ln\eta_0+9.67)\left[-1+(1+5.1\times 10^{-9}p)^{Z_0}\left(\frac{T-138}{T_0-138}\right)^{-S_0}\right]\right\} \tag{6}$$

The density–temperature–pressure [21] relation is employed as

$$\rho=\rho_0\left[1+\frac{0.6\times 10^{-9}p}{1+1.7\times 10^{-9}p}-0.00065(T-T_0)\right] \tag{7}$$

In the present cases, the heat conduction in x and y directions is trivial compared with that in z direction so that it is ignored [22]. Therefore, without considering thermal radiation, the energy equation of the oil film can be written as

$$c\left[\rho u\frac{\partial T}{\partial x}+\rho v\frac{\partial T}{\partial y}+\rho\frac{\partial T}{\partial t}-q\frac{\partial T}{\partial z}\right]+\frac{T}{\rho}\frac{\partial\rho}{\partial T}\left(u\frac{\partial p}{\partial x}+v\frac{\partial p}{\partial y}+\frac{\partial p}{\partial t}\right)$$
$$=k\frac{\partial^2 T}{\partial z^2}+\eta\left[\left(\frac{\partial u}{\partial z}\right)^2+\left(\frac{\partial v}{\partial z}\right)^2\right] \tag{8}$$

When Eq. (8) is solved, the following boundary conditions need to be satisfied

$$\begin{cases} T(x_{in},y,z,t)=T_0 & (u(x_{in},y,z,t)\ge 0) \\ T(x_{out},y,z,t)=T_0 & (u(x_{out},y,z,t)\le 0) \end{cases} \tag{9}$$

where T_0 is the ambient temperature.