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Non-Newtonian thermal Effects in Elastohydrodynamic Lubrication between the Two Rolling Systems

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To analyze complicated phenomena on the fluid hydrodynamic and the elastic deformation between sliding body surfaces, an analysis to the elastohydrodynamic lubrication of sliding contacts has been developed taking into account the thermal and non-Newtonian effects. The computational technique handled the simultaneous solution of the non-Newtonian hydrodynamic effects, elasticity, the load, the viscosity variation, and temperatures rise. The results included the lubricant pressure profile, film thickness, velocity, shear stress, and temperature distribution, and the sliding frictional force on the surface at various slip conditions. These factors showed a great influence on the behavior resulted in the film shape and pressure distribution. Especially, Non-Newtonian effects and temperature rise by the sliding friction force acted as important roles in the lubrication performance.

Keywords : Elastohydrodynamic, Non-Newtonian, Friction, Contact, Elasticity

1. INTRODUCTION

The tribological phenomenon in the machine elements frequently occurs as elastohydrodynamic (EHD) contacts such as rolling and journal bearings, gears, and cams. The study of EHD in these contacts has attracted as reasons for an important role in system performance and life. Therefore, the surface geometrical profiles of the conjunctions in the field of EHD accomplish on the functional relationship between traction and the contact slide. In recent studies of the line contacts, complete solutions have been developed on the traction by numerical considering the full EHD problem [1]. However, due to numerical difficulties associated with wide range applications, a number of approximation methods of traction calculation have been reported as various results by numerical scheme adjustment, pressure and temperature distribution in contact, film thickness, and lubricant properties [2]. This paper is aimed to computational efficiency and numerical stability for wide application of design calculations through the extended literature review. It has performed an analysis of thermal and non-Newtonian effect with bodies in line contact. Also, it is tried to give both reasonable accurate traction and its usefulness as a common design tool to predict the traction under several slip and heavy load conditions.

2. THEORETICAL FORMULATION

In the presented thermal EHD line contact problem, the approximation methods can be associated with coupled solution of thermal Reynolds equation and energy equation with a converged pressure distribution, temperature rise and temperature distribution, and film thickness in contact conjunction. The margins and text layout are as follows:

2.1 Lubrication model and viscosity

Since the lubricant in EHD conjunctions is sheared under severe pressure, a non-Newtonian model (nonlinear viscous Ree-Eyring model [3]) for the lubricant in the macro-line contacting environment was considered as $F(\tau)=\tau_0\eta^{-1}\sinh(\tau/\tau_0)$

Viscosity is a determining factor in pressure and film thickness generation in EHD. The lubricant viscosity on both pressure temperature proposed by Roelands [4] is employed as the following form:

$$\eta(p,T)=\mu_0\exp\left\{[\ln\mu_0+9.67]\left[\left((T-138)/(T_0-138)\right)^{-\omega}\left(1+5.1\times10^{-9}p\right)^{\psi}-1\right]\right\} \quad (1)$$

where $\psi=\dfrac{\alpha}{5.1\times10^{-9}[\ln\mu_0+9.67]}$ and $\omega=\dfrac{\beta(T_0-138)}{\ln\mu_0+9.67}$.

2.2 Reynolds Equation

From the equilibrium equation in x direction and a boundary condition $u(x,0)=U_1$, the velocity profile can be written as:

$$u(x,z)=\tau_0\left[\cosh(\tau/\tau_0)F_1(x,z)+\sinh(\tau/\tau_0)F_2(x,z)\right]+U_1 \quad (2)$$

where

$$F_1=\int_0^z\frac{1}{\eta}\sinh\left(\frac{1}{\tau_0}\frac{dp}{dx}z\right)dz \text{ and } F_2=\int_0^z\frac{1}{\eta}\cosh\left(\frac{1}{\tau_0}\frac{dp}{dx}z\right)dz$$

The shear stress on the surface in the Eq. (2), τ, can be evaluated utilizing another boundary condition $u(x,h)=U_2$. Reynolds equation for non-Newtonian behavior can be obtained as:

$$\frac{d}{dx}\left\{\tau_0\left[\cosh\left(\frac{\tau}{\tau_0}\right)F_3(x,z)+\sinh\left(\frac{\tau}{\tau_0}\right)F_4(x,z)\right]\right\}=-U_1\frac{dh}{dx} \quad (3)$$

where $F_3(x,z)=\int_0^h\{F_1(x,z)\}dz$ and $F_4(x,z)=\int_0^h\{F_2(x,z)\}dz$.

When approaches infinity, non-Newtonian behavior on the pressure distribution decreases and reduces to the conventional Reynolds form. Boundary conditions for Reynolds equation are applied as $p|_{x=x_i}=0$ and $p|_{x=x_o}=dp/dx|_{x=x_o}=0$.

2.3 Energy equation

Considering only conduction across the film thickness and velocity in the moving direction, the energy equation is:

$$\rho u c_p\frac{\partial T}{\partial x}=k\frac{\partial^2 T}{\partial z^2}+\tau\frac{\partial u}{\partial z}+\lambda uT\frac{dp}{dx} \quad (4)$$

Under the assumptions of semi-infinite bodies and perpendicular heat flow to surface, the boundary conditions can be expressed as [5]:

$$T|_{z=s}=T_0+\frac{k_f}{\sqrt{\pi\rho_i c_i k_i U_i}}\int_{x_{in}}^{x_{out}}(-1)^n\left.\frac{\partial T}{\partial z}\right|_{z=s}\frac{d\xi}{\sqrt{x-\xi}} \quad (5)$$

where $i=\begin{cases}\text{index 1, } n=2 & \text{at } s=0\\ \text{index 2, } n=1 & \text{at } s=h\end{cases}$

2.4 Pressure distribution and film thickness

The Hertzian pressure distribution is given by

$$p = p_{MAX}\left[1-(x/a)^2\right]^{0.5}, \text{ where } a = \left(8NR/\pi E'\right)^{0.5} \quad (6)$$

And a is the Hertzian half width and is related to the geometry, material properties and loading. The minimum film thickness formula for line contact was used to supply an initial of the central film thickness [6].

$$h_0 = 3.06U^{0.69}G^{0.56}W^{-0.1}R \quad (7)$$

The film shape in EHD is governed by the separation due to contact geometry and the surface deformation of each sliding surface. For the case of lubricated contact, if each of the sliding surfaces is modeled as an elastic half plane, The film shape including the deflection at the surface location x due to a pressure distribution within flat zone (x_1, x_2) is given by

$$h(x) = h_0 + \frac{x^2}{2R} + \frac{2}{\pi E'}\int_{x_{in}}^{x_{out}} \frac{p(\zeta)}{\sqrt{(x-\zeta)^2}}d\zeta \quad (8)$$

3. NUMERICAL PROCEDURE

The numerical process performed the simultaneous solution sets of Eq. (1) to Eq.(8). The governing equations were non-dimensionalized. The domain consisted of the contact region with thickness h_o, and inlet, and outlet regions. The domain was discretized using 2-D mesh and converted to a finite difference type. The input to initiate the pressure profile of Eq. (6) was supplied with an estimated minimum film thickness of Eq. (7). An initial temperature profile was assumed on the bounding surfaces. To satisfy the boundary condition, the inlet zone was extended up to $\bar{x} = -\infty$. The solution proceeds inside until outlet zone was finished at $\bar{x} = \infty$. First, viscosity relation (1) and velocity profile (2) were calculated. The Reynolds equation (3), and film shape (8) were solved iteratively until a converged solution was obtained. The new pressure profile and central film thickness were temporally accepted until temperature profile is determined. Next iteration in the Energy equation (4) was performed to obtain a converged solution. Eq. (5) was solved for roller surface temperatures over contact zone. These values were used into energy equation loop to calculate new temperature distributions and pressure, and film thickness re-distributions. This process was repeated till the convergence criteria ($\varepsilon <$ 0.001). Converged solution was obtained after a few iterations.

4. ANALYSIS

The design and test parameters of the considered systems were selected from previous works [1,2]. The computational procedure was used to iteratively evaluate the bearing performance in the thermal-contact mode and isothermal-contact mode at various slip conditions, respectively. The influence of non-Newtonian and thermal effects on EHD lubrication using the Eyring rheological law was compared with reference data.

In order to keep general stability in the convergence of the numerical method self-extracted convergent criteria was used to check whenever results change. The effects of self-extracted convergent criteria on the calculated film thickness and pressure distribution were presented for sample cases. The choice of inlet point was placed to $\bar{x}_{in} = -3$ as used in other methods. With relatively light loading conditions and low entraining velocities, the results were calculated for slip ratios ranging from nearly 0.0 to 1.0. From the results, the current calculation had reasonably consistent traction results below allowable error levels in the comparisons with an experimental result.

For the low entraining velocity case and light load condition, the pressure distribution has developed from the Hertz pressure, while the film thickness showed the minimum parallel gap in the Hertzian region. As it approached to high velocity, the EHD pressure distribution broke away significantly from the Hertz pressure, and the film gap is not as parallel. This film reduction is due to the non-Newtonian shearing act and thermal expansion that become stronger at higher entraining velocity. Under the heavy-load, the EHD pressure was very close to the Hertz pressure.

In the temperature distribution across moving direction, temperature rises occurred inside the contact zone for slip ratios. The temperature variations are relatively rough, and the temperature profile inside the contact zone followed the pattern of the pressure profile. The temperature profile showed changes at zone occurring frictional surface temperature rise in solid contact and at the places getting dominating shear heating, conduction heating in the fluidic action. The maximum mid-layer temperature in lubricant was significantly affected by the dependence of thermal conductivity in pressure. Temperature rise also increased with an increase in slip ratio as the same reason. The resulting change is not very significant for the slip ratios at light loading mode, while the change in higher slip ratios and loads was more significant due to the change in temperature inside the contact. Also, the effect of non-Newtonian of lubricant was deeper than Newtonian case at the constriction in the film thickness.

6. CONCLUSIONS

In the elastohydrodynamic lubrication, this approximate analysis handled the simultaneous solution for rheological effects, elasticity, the viscosity variation, and temperatures rise. The results included as follows: (1) The slip ratios and loads proportionally altered the magnitude of pressure peakedness and film thickness. (2) Both the viscosity variation and thermal expansion also had a great influence in the film shape and pressure distribution. (3) Throughout contact zone, non-Newtonian effects and temperature rise by the sliding frictional action acted as important roles. (4) Traction results were in good agreement with that of other approach.

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