An Investigation into Grease Behavior in Thermal EHL Circular Contacts

YANSHUANG WANG and BOYUAN YANG
Henan University of Science and Technology
Luoyang 471003, China

INTRODUCTION

Grease is widely used in many applications; for example, in electric motors, gears, worm gears and high-speed bearings of a grinding machine, etc. Traction data are very important for the dynamic behavior design and failure analysis of large bearings and gears lubricated by grease. Although the EHL theory of oil lubrication is well documented (Fang, et al. (1); Bair, et al. (2)), the EHL theory of grease lubrication is not well established because of the complexity of its rheological properties. Kauzlarich and Greenwood (3) published the first theoretical analysis of EHL with grease in 1972. They formulated the Reynolds equation with the Herschel-Bulkley model and examined the validity of this model. A 2D lubrication equation with the Bingham model was derived by Yang and Qian (4). Dong and Qian (5) obtained a numerical solution of an EHL problem with the Bauer model. However, the Bingham, Herschel-Bulkley, and Bauer models, which are the most commonly used rheological models in grease analysis, are adaptable to low shear rates from 0 to 1000 s\(^{-1}\), but these models become invalid for higher shear rates from 10\(^4\) to 10\(^6\) s\(^{-1}\) found under elastohydrodynamic contacts (Gérard and René (6)). In most EHL analyses, the temperature effects on the viscosity have been ignored, assuming an isothermal process. However, in the case of a high-speed rolling bearing lubricated with grease, both the temperature effects on the viscosity and the non-Newtonian effects become significant, and thermal non-Newtonian EHL analysis is needed for prediction of a more accurate shear stress and traction force. Traction measurements in combined rolling-sliding speeds at high contact pressure are often used to find the mean rheological properties of elastohydrodynamic oil films (Shifeng and Cheng (7); Gupta, et al. (8)), but are not developed for grease films. In the present paper, the behavior of two lithium greases, No. 7007 and 7018, are experimentally investigated under elastohydrodynamic conditions on a single-point contact apparatus. Thereafter, the thermal elastohydrodynamic lubrication problems of point contacts with the Evans-Johnson model on grease are investigated.

KEY WORDS

Lubricating Grease; Rheological Parameters; Traction; Elastohydrodynamic Lubrication

TEST RIG AND ITS PERFORMANCE

The traction experiment was carried out on an improved ball-disk test rig designed and built by Yang, et al. (9), as shown in Fig. 1. The details about the test rig and the method to measure the traction coefficients can be referred to in the literature (Yang et al. (9)). The tests are run at ambient temperature with a gas-pressure grease feed device by which the grease is fed continuously from a pressurized piston and cylinder reservoir and a controlled fresh grease layer is continuously deposited on each of the contacts. In order to eliminate the lateral sliding and spin, the rigid traction measurement system, which consists of a resonance force sensitive quartz sensor and a rigid bracket jointed by a frictionless hinge, is designed and made as shown in Fig. 2. Thus, the traction force on the ball specimen is only in the rolling direction. When the normal load \(W\) and the rolling speed \(v = (v_1 + v_2)/2\) are kept constant (where \(v_1\) and \(v_2\) are the linear velocities for the surfaces of ball and disc specimens, respectively), the curves describing the traction behavior of the lubricants can be generated in the form of traction coefficient \(\mu\) against slide-to-roll ratio \(S = (v_1 - v_2)/v\) over all the operating conditions by this traction test rig. The point contact apparatus can also be used to determine the rheological properties of the elastohydrodynamic grease film at high pressure and high shear strain rate through the measurements of the traction data.

The performance of the test rig is as follows: 1) The range of rolling velocity is 10 to 50 m/s, and the velocities can be changed steplessly. 2) Slide-to-roll ratio \(S\) is in the range of 0.0 to 0.2. 3) The maximum Hertzian contact pressure is from 0.8 GPa to 1.5 GPa. 4) The ball and the disc specimens are both made of GCr15 and their average surface roughness \(\sigma\) is less than 0.02 \(\mu\)m.
Fig. 1—Traction test rig.

5) The contact zone is in a state of full EHD lubrication and the surface roughness has no contribution to the traction force between the ball and the disk. (The minimum film thickness \( h_{\text{min}} \) computed by the Hamrock-Dowson formula is above 0.1 \( \mu \text{m} \), thus the roughness parameter \( \lambda = \frac{h_{\text{min}}}{\sigma} > 5 \).

The lateral sliding speed and spin at the contact center is practically nil.

The physical properties of the base oil of the two lithium greases are as follows: dynamic viscosity in ambient temperature \( \eta_0 = 0.05 \text{ Pa·s} \); viscosity-pressure coefficient \( \alpha = 1.80 \times 10^{-8} \text{ Pa}^{-1} \); temperature-viscosity coefficient \( \beta = 0.032 \text{ /°C} \); thermal conductivity of film \( k_f = 0.0966 \text{ N/s·°C} \).

ANALYTICAL EXPRESSION OF SHEAR STRESS IN CIRCULAR CONTACT

To specify the lubricant properties that govern the traction behavior, it is necessary to postulate a rheological model that relates the shear stresses to the shear strain rates. In this paper, the authors adopt the model developed by Evans and Johnson (11), which is an extended form of the model proposed by Johnson and Tevaarwerk (12). An important feature of the traction experiments is not covered by the TJ viscoelastic rheological model (Eq. [1a]). With constant viscosity, the hyperbolic sine function implies that the shear stress increases without limit when the shear rate is increased. The fact that traction measurements have demonstrated that the mean shear stress rarely exceeds one-tenth of the mean pressure led Smith (13) to propose that, at high pressures, the lubricant behaved like a solid with a “plastic” yield strength. Strong evidence for the existence of a limiting stress as a property of a fluid has been provided by the experiments of Bair and Winer (14, 15). In order to accommodate the existence of a limiting shear stress, Evans and Johnson retain both \( \tau_0 \) and \( \tau_L \) as independent fluid properties since their physical basis is different. Thus, a four-parameter model, which is in unidirectional shear, can be obtained as follows (Evans and Johnson (11)):

\[
\dot{\gamma} = \frac{1}{G_e} \frac{d\tau}{dt} + \frac{\tau_0}{\eta_{eq}} \sinh \left( \frac{\tau}{\tau_0} \right) \quad \text{for } \tau < \tau_L \quad [1a]
\]
\[
\tau = \tau_L \quad \text{for } \tau \geq \tau_L \quad [1b]
\]

with the four fluid parameters: the effective elastic shear modulus \( G_e \) of fluid plus ball and disc, the viscosity \( \eta_{eq} \), the reference Eyring stress \( \tau_0 \), and the limiting stress \( \tau_L \). The shear stress at an operating condition can be evaluated by determining these fluid parameters and solving Eq. [1].

A schematic of the contact geometry between the ball and disc in full EHL regime is shown in Fig. 3 together with the coordinate system. With no spin and lateral slide, many two-dimensional shear planes are considered for any ordinate \( y \).

The following assumptions are made:
1. Shear stress is considered only in the parallel region, and the flow caused by pressure gradient is ignored.
2. The base oil film thickness, which is assumed to be uniform in the Hertzian contact circle, is computed in this paper since the grease film thickness is close to its base oil film (Gérard and René (6)), and its value is the central film thickness according to Hamrock-Dowson’s equation (10):

\[
\frac{h_{iso}}{R} = 2.69 \left( \frac{\eta_0 \nu}{ER} \right)^{0.67} (aE)^{0.53} \left( \frac{W}{ER^2} \right)^{-0.067} \times (1 - 0.61e^{-0.73k}) \]  

[2]

Considering the inlet shear heating, the isothermal film thickness \( h_{iso} \) computed above is modified by a thermal reduction factor, \( \phi_f \), which was proposed by Gupta, et al. (8):

\[
h = \phi_fh_{iso}
\]

\[
\phi_f = \frac{1 - 13.2(\frac{D}{L})^{0.42}}{1 + 0.213(1 + 2.235(\frac{D}{L}))^{0.66}} \]  

[3]

where the thermal loading parameter \( L = \eta_0 \nu^2 \beta/k_f \).

3. The pressure distribution in circular EHL contact area is the Eq. [6] with \( \phi \)

4. The temperature rise of the surface of the ball and disc is given as follows:

\[
\Delta \theta_s = \theta_s - \theta_0 = \frac{1}{\sqrt{\pi \rho _{alon} \kappa _{alon} S}} \int_a^\infty \frac{-k_f \left( \frac{\eta_0 \nu}{E} \right) \left[ z \eta_0 \theta_0 \right]}{\sqrt{x - \xi}} d\xi, \quad y \in [-a, a] \]  

[7]

5. The temperature rise within the grease film was calculated by solving the energy equation shown by Eq. [8]:

\[
\rho cv \frac{\partial \Delta \theta_f}{\partial x} = k_f \frac{\partial^2 \Delta \theta_f}{\partial z^2} + \frac{\tau dV}{h} \]  

[8]

Assuming that the supply grease temperature \( \theta_0 \) is equal to the bulk temperature of ball and disc, the temperature within the oil film is given by: \( \theta_f = \theta_0 + \Delta \theta_0 + \Delta \theta_f \).

**DETERMINATIONS OF RHEOLOGICAL PARAMETERS**

**Viscosity**

It is assumed that the viscosity parameter-pressure-temperature relationship of grease is the same as the viscosity-pressure-temperature relationship originally proposed by Roelands for lubricating oil (Houpert (16)).

\[
\eta_{eq} = \eta_0 \exp \left\{ \frac{[\ln(\eta_0) + 9.67]}{[1 - 1 + 5.1 \times 10^{-9} \rho ]^{2}} \times \left( \frac{[\theta_f - 138]}{[\theta_0 - 138]} \right)^{-0.67} \right\} \]  

[9a]

where \( \eta_{eq} \) is the effective equilibrium viscosity. Roelands’ parameter \( Z \) and \( S_0 \) are expressed as follows by means of the pressure-viscosity coefficient of grease \( a \) and viscosity \( \eta_0 \) at ambient pressure and supply grease temperature.

\[
Z = \frac{\alpha}{5.1 \times 10^{-9} [\ln(\eta_0) + 9.67]} \]  

[9b]

\[
S_0 = \frac{\beta (\theta_0 - 138)}{\ln(\eta_0) + 9.67} \]  

[9c]

The temperature within the grease film \( \theta_f \) is an average value with respect to \( z \).

It is also possible to define an equivalent viscosity-pressure coefficient \( \alpha^* \), representative of Roelands’ viscosity-pressure relationship (Houpert (16)), namely:

\[
\eta_{eq} = \eta_0 \exp(\alpha^* \rho) \]  

[10a]

\[
\alpha^* = \frac{[\ln(\eta_0) + 9.67]^{[\theta_f - 138]/[\theta_0 - 138]} - [1 - 5.1 \times 10^{-9} \rho]^{Z - 1}}{[\ln(\eta_0) + 9.67]^{2}} \]  

[10b]
Eyring Stress $\tau_e$

For the non-linear ascending region of a traction curve, the elastic effect can be neglected. The shear stress $\tau$ at the local point in the Hertzian contact can be expressed by the following equation:

$$\tau = \tau_0 \sinh^{-1}\left(\frac{\eta_0 \gamma'}{\tau_0}\right) = \tau_0 \sinh^{-1}\left(\frac{\eta_0 \gamma'}{\tau_0} \exp[\alpha^* p_0^{1/2} - r^2]\right)$$  \[11\]

where $r$ is the dimensionless radius from the center of the contact given by:

$$r^2 = \left(\frac{x - a}{\eta}\right)^2 + \left(\frac{y}{\eta}\right)^2$$  \[12\]

Integrating Eq. [11] over the Hertzian contact, the mean shear stress $\tau_m$ is then expressed by the following equation:

$$\tau_m = \int_0^1 \tau_0 \sinh^{-1}\left(\frac{\eta_0 \gamma'}{\tau_0} \exp[\alpha^* p_0^{1/2} - r^2]\right) 2\pi dr$$  \[13\]

For simplification, it can be approximately replaced by the following equation (Muraki and Dong [17]):

$$\tau_m = \frac{2\eta_0 \gamma'}{(\alpha^* p_0)} \left\{ 1 - \frac{\tau_0}{\eta_0 \gamma'} \left[ 1 + \ln\left(\frac{\eta_0 \gamma'}{\tau_0}\right)\right] \right\} + \tau_0 \left\{ 1 - \frac{1}{\alpha^* p_0} \ln\left(\frac{\eta_0 \gamma'}{\tau_0}\right) \right\} \ln\left(\frac{2\eta_0 \gamma'}{\tau_0}\right) + \frac{2}{3} \tau_0 \alpha^* p_0 \left[ 1 + \frac{1}{\alpha^* p_0} \ln\left(\frac{\eta_0 \gamma'}{\tau_0}\right) \right]$$  \[14\]

The nonlinear ascending region of a traction curve is employed to determine the parameters $\tau_0$ and $\alpha^*$ by applying Eq. [14] to the experimental traction curve with a least-squares method. It must be noticed that only the data satisfying the condition $\eta_0 \gamma' \exp(\alpha^* p) > \tau_0$ should be taken for the regression analysis with Eq. [14].

**Limiting Shear Stress $\tau_L$**

For a larger sliding velocity, traction force $F$ approaches a constant value $F_{\text{max}}$:

$$F_{\text{max}} = \mu \text{max} W = \pi a^2 \tilde{\tau}_L$$  \[15\]

where $\mu_{\text{max}}$ is the experimentally determined maximum value of the traction coefficient and $W$ is the normal load.

So, the mean limiting shear stress can be calculated by the following equation:

$$\tilde{\tau}_L = \frac{\mu_{\text{max}} W/\pi a^2}{\mu_{\text{max}}} = \frac{\tilde{\tau}_L}{\tilde{\mu}}$$  \[16\]

where $\tilde{\mu}$ is the mean Hertzian pressure.

**Shear Modulus**

The traction curve is a mixed response of both elastic and viscous behavior to the imposed small shear strain rate in the linear region, Eq. [1a] reduces to the classical Maxwell model of:

$$\dot{\gamma} = \frac{1}{\eta} \frac{d\tau}{dt} + \frac{\tau}{\eta_{\text{eq}}}$$  \[17\]

Even though the ball-disc machine traction test does not differentiate between viscous and elastic responses to the shear in the linear region, with Eq. [17] it is possible to extract the effective shear modulus from the linear part of the traction curves.

Because the traction curves from a ball-disc machine test provide only the average shear stresses, it is usual to take the average value for all the parameters. Then, according to Eq. [17], the average shear strain in circular contact can be expressed as:

$$\tilde{\gamma} = \frac{\tilde{\tau}}{\tilde{\mu}} + \frac{8\tilde{\gamma} a}{3\pi \eta_{\text{eq}} v}$$  \[18\]

where $\eta_{\text{eq}}$ is the mean effective equilibrium viscosity.

Since

$$\tilde{\gamma} = \frac{8\Delta v \mu}{3\pi v h} \tilde{\tau} = \frac{\mu \tilde{p}}{v}$$

therefore,

$$\tilde{\eta}_{\text{eq}} = \frac{m_h}{\tilde{\gamma}} \tilde{\eta}_{\text{app}} = \frac{\mu \tilde{p}}{h}$$

where $m = \frac{\mu}{\tilde{\gamma}}$, is the slope of the traction curve in the linear region.

**RESULTS**

The prediction values of traction force $F$ can be estimated by integrating $\tilde{\tau}$ over the contact circle shown in Fig. 3. The corresponding values of the traction coefficient can be obtained by
The average values of Eyring shear stress, limiting shear stress, and shear modulus of the two lithium greases were estimated by curve-fitting. An approximate thermal viscoelastic solution is present for the traction in the EHL circular contact, which describes the entire traction curve with sufficient accuracy. It is proved from the comparisons between model predictions and experimental observations that the Evans-Johnson model (11) cannot only be applied to oil but also extended to grease films to predict traction forces in elastohydrodynamic lubrication.

REFERENCES