A Numerical Study of Friction in Isothermal EHD Rolling-Sliding Sphere-Plane Contacts With Spinning

This paper presents a study of the spinning influence on film thickness and friction in EHL circular contacts under isothermal and fully flooded conditions. Pressure and film thickness profiles are computed with an original full-system finite element approach. Friction was thereafter investigated with the help of a classical Ree–Eyring model to calculate the longitudinal and transverse shear stresses. An analysis of both the velocity and shear stress distributions at every point of the contact surfaces has allowed explaining the fall of the longitudinal friction coefficient due to the occurrence of opposite shear stresses over the contact area. Moreover in the transverse direction spinning favors large shear stresses of opposite signs, decreasing the fluid viscosity by non-Newtonian effects. These effects have direct and coupled consequences on the friction reduction that is observed in the presence of spinning. They are expected to further decrease friction in real situations due to shear heating. DOI: 10.1115/1.4001104

Keywords: elastohydrodynamic lubrication, point contact, spinning, film thickness, traction, non-Newtonian behavior, complex kinematics, shear stress distribution

1 Introduction

In numerous mechanical systems, for instance in the flange-roller end contacts in roller bearings or in variable traction drive systems, a spinning motion is superimposed upon rolling and sliding. This additional kinematic component produces specific frictional effects. After a first experimental study detailed in Ref. [1], we propose here a numerical investigation of the effect of spinning on friction based on the analysis of shear stress distributions. This introductory section will focus on both a literature review and the specific definition of the contact kinematics due to spinning.

1.1 Literature Review. Relatively few works on spinning contacts have been published by the Elastohydrodynamic Lubrication (EHL) community. In the early time of numerical developments, Snidle and Achard [2] simulated a sphere spinning in a contact groove under hydrodynamic and pure spinning conditions. Later, Dowson et al. [3–6] published several studies on this topic. The first paper dealt with a ball on plane contact under pure spin. In the last one, rolling, sliding, and spinning motions were imposed in elliptical contacts. More recently, Zou et al. [7] and Yang and Cui [8] also studied similar operating conditions. These papers focused on pressure and film thickness predictions, and none of them considered the spinning effect on friction. The main conclusion that arose from these works is that pressure and film thickness distributions lose their symmetry when spinning is involved. Film thickness also tends to decrease with spinning and appears more influenced than pressure. Finally, some authors calculated friction in spinning EHL contacts [9–11]. They all showed that friction decreases when spinning increases, but without giving any explanations. A similar trend on friction has been underlined in a previous experimental work of the authors [1].
The aim of the current paper is to link the local effects of spinning (i.e., the local kinematics) to the macroscopic measurement of friction. The local kinematics are detailed in Sec. 1.2.

1.2 Contact Kinematics Due to Spin. The contact between a spherical-end pin (solid B) and a plane disk (solid D) is represented schematically in Fig. 1. CB and CD represent two points situated on the axis of rotation of solids B and D, respectively. The origin is located at the center of the disk contact area. This geometrical configuration is chosen to be the same as the one studied experimentally in Ref. [1].

To highlight the spinning effect, the velocity components of points A_D and A_B, respectively, on the surface of the plane and the spherical-end solid have to be expressed. These velocities are function of the rotational speed of each solid (ω_D, for solid D and ω_B for solid B) and the position of the axis supporting the rotational speed. Moreover both velocities (on surfaces D and B) depend on the space variables, meaning that, in contrast with classical rolling/sliding analysis, the local velocities are not rigorously the same all over the rubbing surfaces.

- The local velocity of a given point A_D of solid D is expressed as follows:

\[ V_D^B(A_D) = V_B^D(C_D) + A_D C_D \times \omega_D \]

where 0 represents a fixed frame, with

\[ V_B^D(C_D) = 0, \quad \omega_D = \begin{pmatrix} 0 \\ -\omega_D \\ 0 \end{pmatrix} \]

and

\[ A_D C_D = A_D O \times O C_D = \begin{pmatrix} -x \\ 0 \\ -z \end{pmatrix} + \begin{pmatrix} 0 \\ 0 \\ R_D \end{pmatrix} \]

where O represents the origin of the fixed frame 0.

The velocity field at any point A_D can then be written as

\[ V_D^B(A_D) = \begin{pmatrix} U_D \\ V_D \\ W_D \end{pmatrix} = \begin{pmatrix} \omega_B R_D \\ 0 \\ 0 \end{pmatrix} + \begin{pmatrix} -z \omega_D \\ 0 \\ x \omega_D \end{pmatrix} \]

(1)

- The local speed of a given point A_B of solid B depends on the tilting angle λ, defined in Fig. 1 as the angle between the y axis and the axis of rotation of solid B. The local speed at A_B is expressed as

\[ V_B^D(A_B) = V_B^D(C_B) + A_B C_B \times \omega_B \]

with

\[ V_B^D(C_B) = 0, \quad \omega_B = \begin{pmatrix} 0 \\ \omega_B \cos \lambda \\ \omega_B \sin \lambda \end{pmatrix} \]

and

\[ A_B C_B = \begin{pmatrix} -x \\ R_B + h_0 - h(x,z) \\ -z \end{pmatrix} \]

and then, with \( h_0 - h(x,z) \ll R_B \),

\[ V_B^D(A_B) = \begin{pmatrix} U_B \\ V_B \\ W_B \end{pmatrix} = \begin{pmatrix} \omega_B \sin \lambda \cdot R_B \\ 0 \\ 0 \end{pmatrix} + \begin{pmatrix} z \omega_B \cos \lambda \\ x \omega_B \sin \lambda \\ -x \omega_B \cos \lambda \end{pmatrix} \]

(2)

For the two solids, the velocity vector over the surfaces can be divided into three components. First, one component is constant over the surfaces: \( U_D = \omega_B R_D \) for solid D and \( U_B = \omega_B R_B \) for solid B. The average of these two components represents the classical mean entrainment speed at the contact center in the \( x \)-direction: \( U_e = (U_D + U_B)/2 \).

Then, two other components are variable over the surface. Some are oriented along the main/rolling direction (longitudinal component), and others along the transverse direction. These three components can be represented by the sketches in Fig. 2.

The effect of such a particular contact kinematics will be studied in two steps. First, a Newtonian isothermal model for film thickness and pressure calculation will be presented, and the effect of spinning will be pointed out.
This model will then be extended to friction calculation and friction results will be discussed. For this, a non-Newtonian lubricant model is used and thermal effects are still disregarded. The intention here was only to point out the main mechanisms controlling film thickness and friction. Therefore, the focus will be brought on the local modifications that spinning introduces in both velocity and shear stress distributions over the contact area.

2 Film Thickness and Pressure Calculations in a Spinning Contact

The numerical model developed to calculate pressure and film thickness will be detailed in Sec. 2.1. Then, the results will be shown and analyzed in Sec. 2.2.

2.1 Calculation Process. The calculation of film thickness and pressure distributions is done by an original process called full-system finite element approach. The main idea is to solve simultaneously the two physics involved in EHL—hydrodynamics and linear elasticity—using a finite element analysis. This method will not be detailed here (see Refs. [12,13] for details), but the equations to be solved will be recalled.

Hertz theory for a ball-on-disk dry contact predicts a contact radius \( a \) and a maximum pressure \( p_H \) such that

\[
a = \left( \frac{3LR_b(E_b(1 - \nu_b^2) + E_D(1 - \nu_D^2))}{4E_bE_D} \right)^{1/3} \quad \text{and} \quad p_H = \frac{3L}{2\pi a^2}
\]

where \( R_b \) is the ball radius, \( L \) is the applied normal load, and \( E_b \) and \( E_D \) are the Young modulus and Poisson ratio of the spherical-end and plane disk solid, respectively.

In the present EHL model, the elastic deformation due to contact pressure is computed over an equivalent cubic solid with an edge length of \( 60a \), as represented in Fig. 3. By using equivalent material properties for this solid (expressed later in this section), the total deformation of both solids \( B \) and \( D \) are calculated. The hydrodynamic problem (Reynolds equation) is solved over a part of the upper surface of the cube (a square of \( 6a \) edge length). These dimensions have been established as the smallest ones above which the accuracy of the results remains unchanged.

2.1.1 The Hydrodynamic Problem. Assuming fully flooded conditions, laminar flow, Newtonian rheology, smooth surfaces, and isothermal, steady-state regime, the classical Reynolds equation can be written as a function of pressure \( p \) as

\[
\frac{\partial}{\partial x} \left[ \frac{p h^3}{12 \eta} \frac{\partial p}{\partial x} \right] + \frac{\partial}{\partial z} \left[ \frac{p h^3}{12 \eta} \frac{\partial p}{\partial z} \right] - \frac{\partial (U_m \rho h)}{\partial x} - \frac{\partial (W_m \rho h)}{\partial z} = 0
\]

with the following:

(a) \( h \) the film thickness, expressed as

\[
h(x, z) = h_0 + \frac{x^2}{2R_b} + \frac{z^2}{2R_b} - u_2(x, z)
\]

where \( u_2(x, z) \) is the displacement of the equivalent solid in the \( y \)-direction

(b) \( \rho \) the density, varying with pressure according to the Dowson–Higginson relationship

\[
\rho(p) = \rho_R \exp \left( \frac{(\ln(\eta_R) + 9.67)}{(1 + \left( \frac{p}{\rho_R} \right)^{0.59})} \right)
\]

where \( \eta_R \) is the reference viscosity, \( P_R = 1.96 \times 10^8 \) Pa, and \( \eta_R = a P_R / (\ln(\eta_R) + 9.67) \)

(c) \( \eta \) the viscosity, varying according to Roelands equation as

\[
\eta(p) = \eta_R \exp \left( \ln(\eta_R) + 9.67 \right) \left( 1 + \left( \frac{p}{\rho_R} \right)^{0.59} \right)
\]

(d) \( U_m \) and \( W_m \) being the mean entrainment velocity (non-uniformly spread across the contact region due to the spinning motion)

\[
U_m(x, z) = \frac{U_b(x, z) + U_D(x, z)}{2}
\]

\[
W_m(x, z) = \frac{W_b(x, z) + W_D(x, z)}{2}
\]

Zero pressure boundary conditions are applied at the edges of the contact domain. Due to the diverging surfaces at the contact exit, negative pressures may arise (cavitation zone). To avoid this unrealistic solution, a penalty method is used enforcing the negative pressures to zero, as described in Ref. [14].

2.1.2 Elastic Deformation. As presented earlier in this paper, the elastic deformation is calculated for an equivalent solid with Young modulus \( E_{eq} \) and Poisson ratio \( \nu_{eq} \) being a composition of both solids \( B \) and \( D \) characteristics [12]. When solids \( B \) and \( D \) are both made of the same material, the equivalent solid characteristics are simply

\[
E_{eq} = \frac{E_B}{2} = \frac{E_D}{2}
\]

\[
\nu_{eq} = \nu_B = \nu_D
\]

The equations solved in the cubic volume represented in Fig. 3 are

\[
\sigma_{ij} = \frac{E_{eq}}{1 + \nu_{eq}} \left( \varepsilon_{ij} + \frac{\nu_{eq}}{1 - 2\nu_{eq}} \varepsilon_k \delta_{ij} \right)
\]

with \( \sigma_{ij} \) representing the nine components of the stress tensor, \( \varepsilon_{ij} = 1/2 \delta_{ij} u_{xj} + \delta_{ij} u_{xi} \) representing the strain matrix, and \( (u_1, u_2, u_3) \) being the displacements in the three directions of space \((x_1, x_2, x_3) = (x, y, z)\).

In agreement with Habchi et al. [12] who first developed the same numerical approach applied to a similar geometry, zero displacement boundary condition is applied to the bottom surface of the cubic structure (Fig. 3). In the contact region (where \( p \) satisfies...
the Reynolds equation), the condition \( \sigma_{zz}(=\sigma_{xx}) = -p \) is assumed. Zero stress boundary condition is applied elsewhere.

2.1.3 Load Equilibrium. The external load exerted on the contact is totally supported by the lubricant film. Therefore, the equilibrium of forces requires that the total pressure generated in the contact domain \( S \) balances the external applied load \( L \) as follows:

\[
\int \int_S p(x,y) \, dS = L
\]

This equation is satisfied by adjusting \( h_{op} \), the constant parameter of the film thickness equation.

Thus, knowing the geometry, the materials and fluid characteristics, the load and velocities applied to the contact, then the unknowns (the pressure field inside the contact, \( h_{on} \) and the deformation of the surfaces, and thus the film thickness) can be determined by solving this complete system of equations.

2.2 Film Thickness and Pressure Results. The calculations are done with realistic input values, comparable to our previous experimental study [1]. All the characteristics are summarized in Table 1.

Generally speaking, a convenient way to represent the velocity components is to use the mean entrainment speed \( U_e \) (only valid at the contact center) and the slide-to-roll ratio, noted SRR, and defined as \( \text{SRR} = (U_{R0} - U_{O0})/U_e \).

In this section, SRR is set to 0. Note that for each set of chosen \( \lambda, U_e \), and SRR both rotational speed of solid \( D \) (\( \omega_D \)) and solid \( B \) (\( \omega_B \)) can be determined, and thus all the velocity field, as detailed previously.

Obtaining film thickness being a preliminary but necessary step before friction prediction, it is interesting to compare the numerical results given by the model described above with other predictions from literature. A quantitative analysis is proposed in Table 2 where central and minimum film thicknesses predicted under pure rolling conditions, respectively, by Hamrock–Dowson [15] and Chevalier [16] relationships are compared with numerical solutions obtained for \( \lambda = 90 \) deg in the present work. The very good agreement obtained for different entrainment velocities allows validating the numerical method initially developed in the frame of EHD rolling/sliding point contacts [12,13] and extended here in the case of large-size EHD conjunctions with rolling/sliding/spinning motion.

The central and minimum film thickness results, for different values of the entrainment speed, as a function of the tilting angle \( \lambda \), are shown in Fig. 4. To accurately interpret the results, it should be recalled that the spinning increases when the angle \( \lambda \) decreases (the zero angle position corresponds to almost pure spinning, which can be considered as the “drilling kinematics”). The curves in Fig. 4 have then to be observed from right to left to be interpreted from almost no spinning to almost pure spinning. This figure shows that the spinning influences the film thickness only for very low values of the \( \lambda \) angle. Beyond \( \lambda = 2 \) deg for the central thickness and \( \lambda = 4 \) deg for the minimum film thickness, the contact behaves as if no spinning was applied. Furthermore, the minimum film thickness decreases more than the central film thickness and the entrainment speed seems to increase the spinning effect.

To better understand this film thickness decrease, the ratio between the current film thickness (for different values of \( \lambda \)) and the film thickness at \( \lambda = 90 \) deg (thus without spinning caused by the spherical-end solid rotation) is plotted in Fig. 5. This figure shows that the entrainment speed has almost no effect on this film thickness ratio. As already seen in Fig. 4, the film thickness decreases under a low value of the tilting angle \( \lambda \). The minimum thickness is much more affected by spinning than the central film thickness, at higher angles.

<table>
<thead>
<tr>
<th>( U_e ) (m/s)</th>
<th>( h_{c, H&amp;D} ) (µm)</th>
<th>( h_{c, Chev} ) (µm)</th>
<th>( h_{c, H&amp;D} ) (µm)</th>
<th>( h_{c, Chev} ) (µm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.25</td>
<td>0.083</td>
<td>0.24</td>
<td>0.086</td>
</tr>
<tr>
<td>2</td>
<td>0.40</td>
<td>0.16</td>
<td>0.40</td>
<td>0.16</td>
</tr>
<tr>
<td>3</td>
<td>0.52</td>
<td>0.23</td>
<td>0.53</td>
<td>0.23</td>
</tr>
</tbody>
</table>

Fig. 4 Central and minimum film thickness versus tilting angle

Fig. 5 Tilting angle influence on \( h_{c, H&D} \), the rolling+spinning over rolling film thickness ratio
In Fig. 6, the film thickness profile is plotted along the line \( x = 0 \), thus along the line perpendicular to the rolling direction in the middle of the contact. The conditions are the same as before, for two different entrainment speeds \( U_e = 1 \) m/s and 3 m/s, \( \lambda \) varying from 90 deg to 0.5 deg.

The two graphs in Fig. 6 show the same trends, but the magnitude is higher at \( U_e = 3 \) m/s than at \( U_e = 1 \) m/s. First, the results at \( \lambda = 90 \) deg and \( \lambda = 7 \) deg are almost the same. The profiles begin to change only when the angle \( \lambda \) falls below 4 deg. It can be noticed that the central film thickness is modified at a lower magnitude than the local minimum film thickness. Furthermore, due to the spinning effects on the kinematics, the central film thickness is modified at a lower magnitude than the local minimum film thickness. Therefore, the film thickness profile loses its symmetry; notice that the asymmetry in the central film thickness shape variations due to spinning showed in Figs. 4–6 qualitatively fit with previous works on rolling-spinning contacts. For instance, in the case of elliptical contacts of low ellipticity \( k = 2 \), Dowson et al. [3] found that \( h_m \) was much more affected by spinning than \( h_c \), and that an asymmetric film shape occurred when a substantial spinning velocity was applied. However, compared with the operating conditions simulated in the present study, their work was limited to the cases of moderate normal loads because of difficulties in achieving converged solutions at high values due to the iterative numerical approach they used.

Contrarily to film thickness, no major change was revealed in the pressure profile between the cases with and without spinning, and this conclusion was also reported in Ref. [3].

3 Friction Forces’ Calculation

Film thickness profiles have been obtained assuming a Newtonian behavior for the lubricant. This assumption holds because film thickness is mainly determined in the converging zone where the lubricant enters the contact. Pressure is moderate and shear rate is low (but increasing as soon as the thickness decreases).

However, the friction calculation can no longer be based on a Newtonian fluid rheology since unphysical results will arise. The reason is that friction, contrarily to film thickness, is mainly governed by the rheological behavior of the lubricant in the central area of the contact, where pressure and shear rate are relatively high. In this zone, the non-Newtonian behavior of the fluid cannot be neglected.

The numerical strategy proposed to compute friction is to start with the pressure and film thickness distributions found with the previous model and apply a non-Newtonian rheological model in the expression of the shear stresses in the fluid. The friction force (in both main/rolling and transverse directions) will be determined by integrating the shear stress over one of the two surfaces in contact. In this paper, we choose to analyze the friction force acting on the plane disk surface. Again, isothermal conditions and no slip at the solid walls are assumed.

3.1 Shear Stress and Friction Calculation. The classical Ree–Eyring non-Newtonian rheological model is used to account for the non-Newtonian behavior of the fluid. Under these assumptions, the shear rate in both main (\( x \)) and transverse (\( z \)) directions reads as

\[
\dot{\gamma}_x = \frac{\tau_{xx}(\tau_r)}{\tau_c},
\]

\[
\dot{\gamma}_z = \frac{\tau_{zz}(\tau_r)}{\tau_c}
\]

(10)

with

\[\tau_c = \sqrt{\dot{\gamma}_x^2 + \dot{\gamma}_z^2}\]

and

\[f(\tau_r) = \frac{\tau_0}{\eta} \sinh \left( \frac{\tau_r}{\tau_0} \right)\]

The Ree–Eyring reference shear stress \( \tau_0 \) has been measured independently on a ball-on-disk apparatus under the same Hertzian pressure and temperature: \( \tau_0 = 6.2 \) MPa.

The expressions of the shear stresses are

\[\tau_{xx} = \frac{\partial}{\partial x} \left( \frac{2y - h}{2} \right) + \frac{\tau_c}{f(\tau_r)} \frac{U_h - U_d}{h}\]

\[\tau_{zz} = \frac{\partial}{\partial z} \left( \frac{2y - h}{2} \right) + \frac{\tau_c}{f(\tau_r)} \frac{W_h - W_d}{h}\]

(11)

Integrating the shear rate expressions leads to the system
systems of two nonlinear equations with two unknowns, this reason, the friction results will be plotted as a function of the ratio for varying tilting angles.

$$U_B - U_D = \int_{y=0}^{y=0} \left( \frac{\partial}{\partial x} \left( \frac{2y - h}{2} \right) + \frac{\tau_y}{\tau_e} \frac{U_B - U_D}{f(\tau_e)} \frac{dy}{h} \right) f(\tau_e)$$

$$W_B - W_D = \int_{y=0}^{y=0} \left( \frac{\partial}{\partial z} \left( \frac{2y - h}{2} \right) + \frac{\tau_y}{\tau_e} \frac{W_B - W_D}{f(\tau_e)} \frac{dy}{h} \right) f(\tau_e)$$

(12)

The shear stresses can then be obtained by solving the previous systems of two nonlinear equations with two unknowns, $\tau_{xy}$ and $\tau_{yz}$, using a Newton-like procedure.

The friction forces $F_x$ and $F_z$ in the x- and z-directions, respectively, acting on the lower surface (the plane disk) can be calculated as

$$F_x = \int \int \tau_{xy}(y=0)dS$$

$$F_z = \int \int \tau_{yz}(y=0)dS$$

(13)

In the following results, the longitudinal friction coefficient is calculated as $f_x = F_x/L$ and the transverse friction coefficient as $f_z = F_z/L$.

3.2 Friction Results. Friction greatly depends on sliding. For this reason, the friction results will be plotted as a function of the SRR. On the other hand, the mean entrainment speed is set to $U_e = 2$ m/s.

Figures 7 and 8 represent the longitudinal and the transverse friction coefficients, respectively, $f_x$ and $f_z$, as a function of SRR. Several curves are superimposed depending on the value of the tilting angle $\lambda$ that controls the spinning component.

It clearly appears in Fig. 7 that the decrease in the tilting angle $\lambda$ (thus the increase in spin) decreases the slope of the longitudinal friction curves for SRR values close to zero. With more spin, the longitudinal friction force is globally lower. Moreover, we can notice that the smaller the angle $\lambda$, the faster friction will decrease. This means that for low SRR ($\leq 10\%$) and low $\lambda$ ($\leq 7$ deg) values, there is an almost inverse proportional effect of tilting angle on friction.

The transverse friction evolution can be observed in Fig. 8. At $\lambda = 90$ deg, a sharp peak of friction can be observed around SRR equals zero. Then, when $\lambda$ decreases (thus when spinning increases), two remarks can be made. First, the maximum friction value increases slightly (in absolute value), and then reaches a plateau while $\lambda$ decreases. Second, the peak observed at $\lambda = 90$ deg becomes thicker, until the friction coefficient reaches the same maximum value regardless of sliding. Finally, it should also be noted that its value is two orders of magnitude smaller than the longitudinal friction coefficient ones.
4 Analysis and Discussion

Above the effects of spinning on film thickness and friction were presented. It is now possible to propose some explanations on the origin of these phenomena. For this purpose, the modifications occurring at the local scale in the presence of spinning will be introduced and discussed.

4.1 Local Velocities and Film Thicknesses. As it has been highlighted in Sec. 1.2, spinning involves additional terms (compared with the classical rolling-sliding kinematics) in the expression of the surface velocities. These terms are nonconstant speed components over the solid surfaces.

Two cases are compared here: one without spinning from the spherical-end solid rotation (λ=90 deg), called “Case 1,” and one with a large, but not extreme, additional spinning (λ=2 deg), called Case 2. Operating conditions are the same as those reported in Table 1: \( U_e \) is set to 3 m/s and SRR to 5%. This value for \( \lambda \) is clearly reduced. This explains the decrease in film thickness presented previously (see Fig. 6). The opposite mechanism applies at the position \( \{x=0, z=\pm a\} \), where the local entrainment velocity clearly increases with spinning, resulting locally in a thicker film thickness as mentioned previously. In the light of these explanations, the reason behind the loss of symmetry of the film thickness due to spinning (Fig. 6) appears obvious now.

4.2 Local Shear Stresses and Friction. Due to spinning the change in the surface velocity distributions, i.e., the change in the local entrainment velocities, will also lead to local changes in the sliding field. Thus, the calculation of the shear stresses, mainly governed by the difference between the two surfaces’ local velocities, is now investigated.

The longitudinal \((\tau_{xy})\) and the transversal \((\tau_{yz})\) shear stresses, calculated on the spherical-end solid surface, are plotted in Figs. 10 and 11, respectively, as in Sec. 4.1 for two cases (Cases 1 and 2) previously defined.

In Case 1 in Fig. 10, the longitudinal shear stress presents the same shape as the pressure distribution. When spinning becomes significant (Case 2 in Fig. 10), the stress no longer follows the pressure distribution but it is now divided into two areas. Because a positive SRR is applied, the average speed of the spherical-end solid surface is higher than the plane surface speed. But due to the local velocities’ changes generated by spinning, an area appears where the spherical-end solid’s local velocity is lower than the plane disk’s local velocity (as suggested from Fig. 2), causing
negative shear stresses to arise. Increasing the SRR would move the limit between positive and negative shear stresses toward lower values of \( \frac{z}{a} \), and the negative area would become smaller. This explains why spinning influence is weak at high SRR as the shear stress becomes positive over the entire area like in Case 1. On the other hand spinning effect is very important at small SRR values due to the occurrence of two shear stress zones of opposite sign. When calculating the integral of the shear stress over the surface thus the friction force, Fig. 10 clearly shows that a lower value will be found in the presence of significant spinning since the negative part of the shear stress profile sets off the classical positive one.

Figure 11 represents the shear stress in the transverse direction \( \tau_{y} \). In Case 1, no spinning is introduced by the spherical surface, and the transverse shear stress is close to zero all over the surface (except near the pressure spike). It should be noted that even in this case, though negligible, a small spinning is globally introduced due to the plane disk motion. The local absolute values of the shear stress increase when spinning is introduced by the spherical surface rotation (Case 2 in Fig. 11). Once again, two distinct areas can be observed, with opposite shear stress signs. This leads to an average shear stress nearly equal to zero, and thus the transverse friction force is very low. However, the important local value of the shear stress would contribute to a viscosity decrease in the lubricant by shear-thinning effect. This is an additional explanation to the fact that friction decreases with spin. But this high local shear stress would also have an influence on the thermal dissipation. The local thermal effect cannot be measured, but it may reasonably be assumed that the additional shearing would generate more heat, and thus the temperature in the contact would increase. The final consequence of this temperature increase would again be a viscosity decrease in the lubricant, and thus a friction force decrease (Fig. 12).

5 Conclusion

A model for EHL contact with spinning is proposed in this paper. Pressure and film thickness profiles are computed with an original full-system finite element approach applied to a Newton-
ian fluid. This method (described in detail in Refs. [12,13]) is extended here to account for spinning effects. Shear stress distributions and friction are then calculated using as input the (Newtonian) pressure and film thickness distributions and a non-Newtonian rheological model. The approach, although not fully coupled, allows for the understanding of spinning effects on friction in lubricated EHD contacts.

From this work, the following conclusions can be drawn.

(1) Friction was investigated to calculate the longitudinal and transverse shear stresses in the contact area. This analysis has given the key to understand the mechanism linking spinning and friction (see Fig. 12). It is possible to establish that longitudinal friction (i.e., in the main direction of rolling/sliding) decreases as spinning increases. Longitudinal shear stress distributions show that, while the maximum stresses keep globally a constant value with the addition of spinning, the contact area is divided into two zones where the shear stress has opposite signs, explaining thus the lower longitudinal friction force (i.e., the integral of shear stress) than in the classical case. An indirect effect of spinning applies on the transverse shear stress. While the integral of the transverse shear stress keeps a constant value, with the addition of spinning, the local maximum stresses increase significantly. The increase in this transverse shear stress due to spinning is expected to have a direct effect by decreasing the fluid viscosity (by non-Newtonian effects). Moreover, not in this isothermal analysis, but in reality, the temperature increase due to the local increase in the shear stresses would further enhance the viscosity decrease. The indirect consequences of spinning via the transverse shear stress lead in both cases to a friction decrease.

(2) It has been found that, while the pressure profile is not significantly affected, the film thickness is changed by the introduction of spinning, in a way that it loses its classical symmetry. As a consequence of the local change in surface velocities, the minimum film thickness is found to be lower and lower as spinning increases.

(3) This simple analysis of film thickness and friction calculation in EHL spinning contacts leads to the explanations of the fundamental mechanisms of spinning. The next step to further investigate spinning effects on friction is to build a fully coupled model of pressure, film thickness, and friction calculation including thermal effects.

Acknowledgment

The authors wish to thank SKF for the permission to publish this paper and for the financial contribution.

Nomenclature and Notations

\( a \) = Hertzian contact radius (m)
\( E \) = Young’s modulus (Pa)
\( f_i \) = friction coefficient along the \( i \)-direction
\( F_i \) = friction force along the \( i \)-direction (N)
\( h \) = film thickness (m)
\( h_c \) = central film thickness (m)
\( h_m \) = minimum film thickness (m)
\( h_0 \) = constant in the film thickness equation (m)
\( L \) = normal load (N)
\( p \) = pressure (Pa)
\( P_R \) = reference pressure in the Roelands equation (Pa)
\( p_{H} \) = Hertzian contact pressure (Pa)
\( R_p \) = spherical-end solid radius of curvature (m)
\( SRR \) = classical slide-to-roll ratio (%)\( u_i \) = displacement of the equivalent elastic solid in the \( i \)-direction (m)

\( U_e \) = mean entrainment velocity at the contact center in the \( x \)-direction (m/s)
\( U_m \) = mean entrainment velocity in the \( x \)-direction (m/s)
\( U_{DO} \) = constant component of the velocity of any point on surface \( D \)
\( U_{BO} \) = constant component of the velocity of any point on surface \( B \)
\( U \) = speed component along the \( x \)-direction (m/s)
\( V \) = speed component along the \( y \)-direction (m/s)
\( W \) = speed component along the \( z \)-direction (m/s)
\( W_m \) = mean entrainment velocity in the \( z \)-direction (m/s)
\( z_R \) = exponent in the Roelands equation
\( V_0(A_B) \) = velocity vector of point \( A_B \) located on surface \( B \), with respect to the fixed frame \( 0 \)
\( V_0(A_D) \) = velocity vector of point \( A_D \) located on surface \( D \), with respect to the fixed frame \( 0 \)
\( \lambda \) = tilting angle (deg)
\( \omega \) = rotational speed (rad/s)
\( \nu \) = Poisson’s ratio
\( \rho \) = density (kg/m\(^3\))
\( \rho_R \) = reference density at ambient pressure (kg/m\(^3\))
\( \eta \) = viscosity (Pa s)
\( \eta_R \) = reference viscosity in the Roelands equation (Pa s)
\( \alpha \) = lubricant pressure viscosity coefficient according to Barus model (Pa\(^{-1}\))
\( \sigma_{ij} \) = stress tensor component in the equivalent solid (Pa)
\( e_{ij} \) = strain tensor component
\( \tau_0 \) = Ree–Eyring reference shear stress (Pa)
\( \tau_e \) = equivalent shear stress (Pa)
\( \tau_y \) = shear stress in the \( j \)-direction (Pa)
\( \tau_{yj} \) = shear rate in the \( j \)-direction (s\(^{-1}\))

Subscripts

\( B \) = denotes the spherical-end solid
\( D \) = denotes the plane disk solid
\( eq \) = relates to the equivalent solid in the elastoplastic problem

References


Downloaded From: http://tribology.asmedigitalcollection.asme.org/ on 09/28/2015 Terms of Use: http://www.asme.org/about-asme/terms-of-use


