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Analysis of transient elastohydrodynamic lubrication of point contact subjected to sinusoidal dynamic loads

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Abstract. Vibration is inevitable from all machine elements operations since the load, surface velocities or contact radii change in time. In the case of a ball bearing application, the load changes as the ball elements rotate around the track. This change will be more rapidly when the external load changes. The vibrations induce a sinusoidal dynamic load working on the contact. On the other hand, the imperfect waviness on the contacting surfaces causes the vibration effects on the film thickness to become more pronounced. This paper numerically investigates the influence of the sinusoidal varying loads due to vibrations on the film thickness and pressure profile of the isothermal, elastohydrodynamically lubricated point contact with a wavy surface. For that purpose, a single contact between ball element and inner raceway is modeled as a contact between a spherical ball and a flat plate. The results reveal that the external sinusoidal dynamic load induces the modulations on the film thickness and pressure profile. The waviness on one contacting surface causes the changes in the EHL behaviors to become more pronounced.

1. Introduction

A lubricated contact between non-conformal surfaces in machine elements such as gears, rolling bearings, and cam and followers are commonly identified as elastohydrodynamic lubrication (EHL) regime in which the contact generates a very high pressure over a very small contact area. The behaviours of EHL, such as the pressure and film thickness distribution, explain the important considerations in the analyses of machine elements contacts. The analysis of the generated pressure in the EHL contacts is crucial to investigate the subsurface stress tensor and to predict the failure of initial crack whereas the study of the film thickness formed between the lubricated contacts improves the performance of machine elements. However, it is generally known that machine elements operate almost always under time-dependent conditions. In the case of ball bearing applications, the contact between rolling elements and raceway in the loaded zone changes in time due to varying orbital position of the rolling elements during rotation. This such condition causes a harmonic change in the load applied to the contact and is now known as vibration. The change in the external load applied to the contact will cause the vibration to become more pronounced. In addition, manufacturing processes of ball bearing components cannot guarantee that the surfaces will be perfectly smooth, there will be imperfections on the surfaces, such as waviness. This imperfect surfaces may cause the film thickness changes and the pressure as well. Hence, a transient effect needs to consider during the analysis of the EHL contact.

A realistic condition of the contacting surfaces of EHL point contact has been numerically considered in the past four decades, for example, indentations [1, 2], ridges [3], and waviness [4]. The results observed that the imperfect surfaces induce a change in the film thickness and pressure profile. However, the solvers were still in the assumption of the constant load conditions. Furthermore, Wijnant [5] presented the sinusoidal dynamic load effects in the field of EHL point contact and the results revealed that although the surfaces are perfectly smooth, the film thickness shape and pressure...
profile in the contact area are no longer flattened as shown in the case of the static load condition for the smooth contacting surfaces. In this paper, the combination of the sinusoidal dynamic load effect due to the harmonic change in the external load as well as the waviness effect on the contacting surface is considered by accounting for both squeeze effect in the Reynolds equation and the sinusoidally varying loads in the equation of motion.

2. Governing Equations

A contact of ball-raceway in a ball bearing application can be modelled as a contact between a spherical ball and a flat plate. First, the spherical ball subjected to the static load as an initial condition. Then, the load alters as the sinusoidal dynamic loads along the $z$-direction in the coordinate system. The surface of the inner raceway is wavy whereas the surface of the ball is assumed to be smooth.

In order to analyze the behavior of the transient EHL point contact, some equations must be solved simultaneously, i.e. Reynolds, film thickness, lubricant properties, and load balance equations. The dimensionless Reynolds equation for the transient EHL point contact can be written as:

$$\frac{\partial}{\partial x} \left( \varepsilon \frac{\partial P}{\partial x} \right) + \frac{\partial}{\partial y} \left( \varepsilon \frac{\partial P}{\partial y} \right) - \frac{\partial (\varepsilon H)}{\partial y} - \frac{\partial (\varepsilon H)}{\partial y} = 0$$

(1)

The boundary conditions are $P(X_a,Y,T) = P(X_b,Y,T) = P(X,Y_a,T) = P(X,Y_b,T) = 0$, where $X_a, X_b, Y_a$, and $Y_b$ are the boundaries of the domain. It should be noted that the cavitation occurs in the contact, and hence the condition of $P(X_a,Y,T) \geq 0$ must be satisfied. $\varepsilon$ is defined according to:

$$\varepsilon = \frac{\beta H}{\bar{\eta}}$$

(2)

where $\beta$, $\bar{\eta}$, and $H$ denote the dimensionless density, dimensionless viscosity and dimensionless film thickness, respectively. $\lambda$ can be calculated as:

$$\lambda = \frac{6\eta_0 \mu_0 R_0^3}{\alpha^2 \pi h_0}$$

(3)

In this paper, the density and the viscosity of the lubricant are assumed to depend on the pressure, and hence the density-pressure relation proposed by Dowson and Higginson [6] as well as the Roelands pressure-viscosity relation [7] are employed. The dimensionless film thickness reads:

$$H(X,Y,T) = H_0(T) - \mathcal{R}(X,Y,T) + \frac{X^2}{2} + \frac{Y^2}{2} + \frac{2}{\pi} \int_{-\infty}^{+\infty} \int_{-\infty}^{+\infty} \frac{P(X',Y')dX'dY'}{\sqrt{(X-X')^2 + (Y-Y')^2}}$$

(4)

where $H_0(T)$ is an integration constant and determined by the dimensionless force balance equation:

$$\int_{-\infty}^{+\infty} \int_{-\infty}^{+\infty} P(X,Y,T) dX dY = \frac{2\pi}{3}$$

(5)

Under dynamic load condition, Equation (5) changes to be [5]:

$$\sum_{i=1}^{M} \sum_{j=1}^{N} P(X,Y,T) = \frac{2\pi}{3} (1 + \tilde{A} \sin(\Omega_s T))$$

(6)

where $\tilde{A}$ and $\Omega_s$ denote the dimensionless amplitude and dimensionless frequency of vibration, respectively. $\mathcal{R}(X,Y,T)$ in Equation (4) stands for the undeformed geometry of the lower body at dimensionless time $T$. This undeformed geometry can be calculated as [4]:

$$\mathcal{R}(X,Y,T) = \begin{cases} 0, & X \geq X_d \\ \tilde{A}_n \sin \left( 2\pi \frac{X - X_d}{\tilde{W}_n} \right), & X < X_d \end{cases}$$

(7)

where $\tilde{A}_n$ and $\tilde{W}_n$ denote the dimensionless amplitude and the dimensionless wavelength of the waviness (see Nomenclature), respectively. $X_d$ is the dimensionless location of the first wave passing into the contact and defined as:
3. Numerical Simulation

First, the stationary condition was solved using the initial static load applied to the contact. To solve the Reynolds equation for stationary problem, the pressure distribution of the Hertzian dry contact was employed as the initial pressure using Jacobi relaxation scheme, and along with the Half-space theory for solving the film thickness equation as well as the lubricant properties relations, the new pressure profile was obtained. Then, the film thickness, the density as well as the viscosity of the lubricant were updated based on the new pressure. Subsequently, the stationary result was used to solve the transient EHL point contact by accounting for the squeeze-term effect in the Reynolds equation and the sinusoidal dynamic load effect in the force balance equation. The same procedure as the stationary case was employed for each time step simulation. It should be noted that in order to ensure the whole waviness effect has been included in this investigation, the effect of the sinusoidal load was calculated after the waviness left the contact, i.e. \( X_d = 1.5 \).

4. Results and Discussion

4.1 Stationary result

Table 1. Different parameters and their values for stationary and sinusoidally varying loads cases

| Parameter                  | Value   |
|----------------------------|---------|
| Load, \( F \) (N)          | 20      |
| Rolling speed, \( u_m \) (m/s) | 1       |
| Slide-to-roll ratio, SRR  | 0       |
| Viscosity at ambient pressure, \( \eta_0 \) | 0.0835  |
| Viscosity index           | \( 1.95 \times 10^{-8} \) Pa\(^{-1} \) |
| Material parameter, \( G \) | 2279    |
| Dimensionless oscillation amplitude, \( \tilde{A} \) | 0.1     |
| Dimensionless excitation frequency, \( \Omega_d \) | \( 2\pi \) |

In the present paper, parameters for stationary case, i.e. operating conditions and lubricant properties are shown in Table 1. This table also shows the sinusoidal dynamic load parameters for the case of dynamic load condition. Load of 20 N and rolling speed of 1 m/s result in the maximum Hertzian dry contact of 0.53 GPa with 0.135 mm radius contact. Slide-to-roll ratio \( SRR = 0 \) means both surfaces move with the same velocity of 1 m/s, and the sliding motion is absent in this condition, i.e. pure rolling condition. The numerical domain for this solver is \(-2.5 \leq X \leq 1.5\) and \(-2.0 \leq Y \leq 2.0\) with a single grid solution consists of \(129 \times 129\) points. The spatial mesh was \( \Delta X = \Delta Y = 0.03125 \) and the increment time was set to be equal to the spatial mesh, \( \Delta T = 0.03125 \). This corresponds to the 0.0042 milliseconds. Figure 1 shows the stationary result of film thickness for the smooth contacting surfaces by means of the reverse three-dimension plot of film thickness together with the centerline film thickness and pressure profile along \( X \)-direction. From the plots, it can be observed that all familiar features of EHL point contact for the medium load are displayed, such as the pressure distribution is almost semi-ellipsoid and the pressure spike occurs at the exit region as well as the well-known horse shoe shape in the contact. The central film thickness at \( Y = 0 \) is nearly flattened, and the minimum film thickness is found in the side lobes of the horse shoe shape near \( X = 0, Y = \pm 1 \).

In order to investigate the effect of wavy surface on the EHL behaviors, the quasi-transient analysis was performed by accounting for the squeeze-term in the Reynolds equation whereas the load is still...
assumed to be constant. Parameters of waviness were used in this simulation, i.e. $A_h = 0.05$ and $W_h = 2.0$ which give the waviness on the surface with 0.024 μm amplitude and 0.27 mm wavelength. This wavelength causes the waviness with one wave contacting the smooth surface. Figure 2 shows the influence of waviness on the film thickness and pressure profile. From the plots, it can be seen that the central film thickness shape is no longer flattened. However, the waviness on one contacting surface is almost completely deformed in the contact area, due to the high pressure in this area. The change in the pressure profile is also observed due to waviness in which the pressure is no longer semi-ellipsoid in the contact area.

4.2 Transient result

The effect of changes in the applied load due to external sinusoidal dynamic load on the film thickness and pressure profile can be clearly seen in Figure 3 for the case of smooth contacting surfaces. Using the dimensionless frequency excitation of $2\pi$ and dimensionless oscillation amplitude of 0.1, Figure 3 captures the centerline dimensionless pressure and film thickness at the different momentarily time steps. From these plots, one can observe that although the contacting surfaces are assumed to be perfectly smooth, there are modulations in the film thickness and pressure profile due to dynamic load. Moreover, since the external dynamic load applied to the contact changes sinusoidally, the changes in the film thickness as well as the pressure profile are periodic. This periodicity starts after the initially induced film thickness changes have propagated through the contact, for example, in this case the
periodicity start after $T \geq 2$. Taking a closer look at Figure 3, film thickness and pressure profile at $T = 2.0$ are coincident with those of at $T = 3.0$, and so do to the EHL behaviors at $T = 2.5$ which is similar to the EHL behaviors at $T = 3.5$. These phenomena are in good agreement with result shown in [5].

![Figure 3. Centerline dimensionless pressure (left labels) and film thickness (right labels) for $\Omega_e = 2\pi$ and $A = 0.1$ at different momentarily dimensionless time for the smooth contacting surfaces.](image)

To illustrate the combined effects of wavy surface and the sinusoidal dynamic load from external vibration applied to the lubricated contact, Figure 4 shows captures of the changes in the centreline film thickness and pressure profile along the X-direction at different times. Similar to the smooth contacting surfaces case, the film thickness and pressure profile modulate as the load applied to the contact changes sinusoidally. However, due to the waviness onto one surface, the modulations of film thickness and pressure profile are no longer periodic.

![Figure 4. Centerline dimensionless pressure (left labels) and film thickness (right labels) for $\Omega_e = 2\pi$ and $A = 0.1$ at different momentarily dimensionless time for the wavy contacting surface.](image)
5. Conclusion
In this work, a simple transient EHL point contact solver has been developed to investigate the influences of the waviness and sinusoidally varying load on the EHL behaviors. The simulation results reveal that the external dynamic load induces modulations on the film thickness and pressure profile in the lubricated contact. The waviness on one contacting surface causes the changes in the film thickness and pressure profile to become more pronounced.

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