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On the compressive heating/cooling mechanism in thermal elastohydrodynamic lubricated contacts

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ABSTRACT

The importance of compressive heating/cooling occurring within lubricating films in thermal elastohydrodynamic lubricated contacts has not been given sufficient attention in the literature. This paper presents a numerical investigation of this mechanism and attempts to quantify its importance as compared to shear heating under pure-rolling and rolling–sliding conditions. It is found that even under pure-rolling, compressive heating/cooling remains in most cases less important than shear heating or at best, of the same order. Under rolling–sliding conditions, as soon as the slightest sliding occurs, heat generation is governed by shear heating. The dependence of compressive heating/cooling on operating conditions is also examined. Finally, the impact of this mechanism on the lubricating performance of these contacts is considered under various operating conditions.

1. Introduction

Heat generation within lubricating films of elastohydrodynamic lubricated (EHL) contacts can be a consequence of two separate mechanisms: lubricant compression/decompression and lubricant shear. The former is a consequence of a pressure build-up at the inlet of EHL contacts which leads to a compression of the lubricant accompanied by a generation of heat. But, at the exit of the contact, the pressure drop is associated to a decompression of the lubricant leading to the formation of a “heat sink” within the lubricating film. This combined heating/cooling mechanism by lubricant compression/decompression is often referred to as “compressive heating/cooling effect”. On the other hand, shear heating is a consequence of lubricant layers moving at different speeds and rubbing against each other across the lubricant film thickness, leading to frictional heat generation (shear heating). This is most pronounced under rolling–sliding or pure-sliding conditions where surface velocities of the contacting elements are different, but can also occur under pure-rolling conditions. In fact, under pure-rolling conditions, though the “Couette” component of the lubricant flow leads to a constant velocity distribution across the lubricant film thickness due to identical surface velocities of the contacting elements, the velocity profile itself is not constant, owing to the pressure-driven “Poiseuille” component of the flow [1]. This results in neighboring lubricant layers rubbing against each other and generating heat.

The interest in EHL thermal effects first appeared with the pioneering theoretical work of Cheng [2,3]. The first full numerical solution for the point contact problem was obtained by Zhu and Wen [4]. Since then, several authors proposed different methods to deal with this problem assuming a Newtonian or a non-Newtonian lubricant such as Kim and Sadeghi [5], Guo et al. [6], Kaneta et al. [7] or also Liu et al. [8] who solved the three-dimensional energy equation in order to determine the temperature variations throughout the lubricant film. An alternative method consists in reducing the three-dimensional heat transfer problem to a two-dimensional one by assuming a parabolic temperature distribution across the film thickness. This approach was used by many researchers; however, the parabolic temperature profile simplification leads to temperature predictions that are not accurate especially at the inlet of the contact as shown by Kazama et al. [9]. The reason lies in the occurrence of complex thermal convective effects which are associated with important reverse flows in this area. Salehizadeh and Saka [10] highlighted in their thermal non-Newtonian analysis of line contacts the importance of compressive heating in raising the lubricant's temperature in the inlet of the contact. Surprisingly, Kaneta et al. [11] found that compressive heating had an important impact on friction in EHL contacts at low levels of sliding (this issue will be discussed in further detail at a later stage). Sadeghi and co-authors [12,13] carried out a numerical analysis of thermal newtonian EHL point contacts and investigated the effect of compressive heating/cooling on temperature under pure-rolling conditions. They found that temperature increases in the inlet area due to shear heating and rises furthermore

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due to compressive heating but then falls in the exit region of the contact because of decompression cooling. They predicted a maximum temperature rise of about 10 °C and also found that in the presence of sliding the thermal response of the contact was governed by shear heating. In the experimental study of Reddyhoff et al. [14], the authors used an infrared temperature mapping technique with a ball-on-disk test-rig to investigate the effects of compressive heating/cooling on temperature rise in EHL contacts. Their tests were carried out both in "nominally pure-rolling", with the disk driving the ball, and in "forced pure-rolling" with the ball and disk being driven at the same surface speed using separate motors. However, being a work of experimental nature, the authors had very little access to what is actually happening within the lubricant film at the local level, so they could not really quantify the share of compressive heating/cooling in their temperature rise measurements as compared to that of shear heating.

From the earliest works listed above, it was recognized that heat generation within EHL conjunctions is a consequence of both lubricant compression and lubricant shear and both aspects were taken into consideration in the solution of the thermal part of the problem. Furthermore, most of these works acknowledge that the compressive/heating cooling effect is only significant under pure-rolling conditions and that, whenever sliding occurs, this effect becomes negligible compared to shear heating. However, none of these has examined this issue in more detail in an attempt to quantify the importance of compressive heating/cooling compared to shear heating and under which conditions the former became negligible compared to the latter. The current work provides a numerical investigation of compressive heating/cooling in EHL circular contacts. The advantage of a numerical approach (as compared to an experimental one such as that of Reddyhoff et al. [14]) is that the actual impact of compressive/heating cooling on lubrication performance may be quantified by simply switching this effect off and then back on in the numerical simulations; something that cannot be done in an experimental setup. In addition, the importance of compressive heating/cooling as compared to shear heating may also be exactly quantified under different operating conditions. The employed numerical model is the full-system finite element approach for the solution of the thermal EHL problem [15,16] introduced by the authors in recent years. This approach has been successfully used in predicting the lubricating performance of EHL contacts in terms of film thickness [15] as well as friction [17]. By "successfully" it is meant that a thorough validation against experiments has been attained. In this work, only circular contacts will be considered, however the findings and analysis are of a more general scope and may be extended to line or elliptical contacts. The outline of this paper is as follows: Section 2 provides a description of the selected lubricant and the dependence of its rheological and transport properties on pressure, temperature and shear stress. Then in Section 3, the main features of the employed numerical model are briefly recalled. Section 4.1 provides a quantification of compressive heating/cooling as well as shear heating and, most importantly, a magnitude comparison of these two effects within EHL contacts under a wide range of operating conditions under both pure-rolling and rolling–sliding regimes. In Section 4.2, the impact of compressive heating/cooling on the lubrication performance of circular EHL contacts is quantified and finally a general conclusion is offered in Section 5.

2. Selected lubricant and its properties

The lubricant selected for this work is a typical mineral oil (Shell T9) for which the rheological and transport properties have been thoroughly characterized in [17]. The variations of these properties with pressure, temperature and shear stress were measured and appropriate models were derived to represent these variations. The derived models are briefly recalled in the following. Subscripts 0 and R indicate, respectively, ambient conditions and a reference state which are taken to be the same in the current work ($p_0 = p_R = 0$ and $T_0 = T_R = 25\,^oC$). The Murnaghan [18] equation of state is used to model the density variations with pressure p and temperature T

$$\rho = \frac{\rho_R}{1 + a_V(T - T_R)} \times \left(1 + \frac{K_0'}{K_0} p\right)^{\frac{1}{K_0'}} \quad \text{with} K_0 = K_{00} \exp(-\beta_K T) \tag{1}$$

where $K_0' = 10.545$, $a_V = 7.734 \times 10^{-4}\ K^{-1}$, $K_{00} = 9.234$ GPa, $\rho_R = 875 Kg/m^3$ and $\beta_K = 6.090 \times 10^{-3}\ K^{-1}$ were obtained from experimental measurements. As for the viscosity dependence on temperature and pressure, a different and enhanced model is employed. In fact, in the current work, the improved Yasutomi correlation is used as proposed in [19]

$$\mu = \mu_g \exp\left[\frac{-2.303\ C_1 (T - T_g) F}{C_2 + (T - T_g) F}\right]$$
$$\text{with}: \quad T_g = T_{g0} + A_1 \ln(1 + A_2 p) \ \text{and} \ F = (1 + B_1 p)^{B_2} \tag{2}$$

Where $A_1 = 188.86\ °C$, $A_2 = 0.719 GPa^{-1}$, $B_1 = 8.2\ GPa^{-1}$, $B_2 = -0.5278$, $C_1 = 16.09$, $C_2 = 17.38\ °C$, $T_{g0} = -83.2\ °C$ and $\mu_g = 10^{\ 12}$ Pa·s. As for the shear dependence of viscosity, the single-Newtonian modified Carreau–Yasuda equation [20] is used to define the generalized viscosity η as a function of shear stress τ as follows:

$$\eta = \frac{\mu}{\left[1 + \left(\frac{\tau}{G}\right)^a\right]^{\frac{\frac{1}{n} - 1}{a}}} \tag{3}$$

where $G = 7.0$ MPa, $a = 5$ and $n = 0.35$. The lubricant was shown to exhibit a limiting shear stress behavior under high shear rates. The limiting value of the shear stress $\tau_L$was shown to vary linearly with pressure according to the following relationship:

$$\tau_L = \Lambda\ p \tag{4}$$

where the limiting-stress pressure coefficient $\Lambda = 0.083$. Finally, the dependence of the thermal properties of this lubricant on pressure and temperature is considered. First, its thermal conductivity k depends on temperature and pressure according to the following equation:

$$k = B_k + C_k \kappa^{-s} \quad \text{with} \quad \kappa = \left(\frac{V}{V_R}\right)\left[1 + A\left(\frac{T}{T_R}\right)\left(\frac{V}{V_R}\right)^3\right] \tag{5}$$

where $A = -0.101$, $B_k = 0.053$ W/m·K, $C_k = 0.026$ W/m·K, $s = 7.6$ and the term $V/V_R$is nothing else but ρ_R/ρ obtained directly from Eq. (1). As for the volumetric heat capacity $C = \rho\, c$ of this lubricant, it depends on temperature and pressure as follows:

$$C = C' + m\chi \quad \text{with} \quad \chi = \left(\frac{T}{T_R}\right)\left(\frac{V}{V_R}\right)^{-4} \tag{6}$$

where $C' = 1.17 \times 10^6\ J/m^3 \cdot K$, $m = 0.39 \times 10^6\ J/m^3 \cdot K$ and the term $V/V_R$is obtained from Eq. (1). It is noteworthy to mention that the above rheological models were derived from actual measured transport properties and used without any alteration of their corresponding parameters to predict film thickness and friction in EHL circular contacts under a wide range of operating conditions [17]. The predicted results showed excellent agreement with experiments, allowing the authors to establish a validated framework for the theoretical prediction of the lubricating performance of EHL contacts. The employed rheological models are (compared to the ones commonly used in the EHL literature) relatively complex with multiple parameters to be determined from