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Influence of thermo-mechanical properties of coatings on friction in elastohydrodynamic lubricated contacts

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ABSTRACT

This paper presents a numerical investigation of the influence of thermo-mechanical properties of coatings on friction in elastohydrodynamic contacts. In a previous work by the author, it was shown that thermal properties of coatings had a significant influence on friction. In fact, under high sliding speeds, friction was found to increase with the thermal inertia of coatings. The current work reveals that mechanical properties of coatings also have a significant impact on friction. Friction actually exhibits an increase with the rigidity of the coating. Furthermore, a combination of soft coatings with low thermal inertia is shown to maximize friction reduction while hard coatings with high thermal inertia maximize friction increase. These effects are found to increase with the coating thickness.

1. Introduction

The influence of surface coatings on the performance of elastohydrodynamic lubricated (EHL) contacts has been a subject of interest for the tribological community over the last few decades. From the earliest works on the topic, the focus was on the effect of mechanical properties of coatings on pressure, film thickness and stress distribution within the solid components of these contacts. The early numerical models for coated EHL contacts assumed rigid substrates and elastic coatings such as the work of Bennett and Higginson [1] or Elsharkawy and Hamrock [2,3] for the case of isothermal line contacts or also the work of Jin [4,5] for isothermal circular contacts. One of the first numerical models to account for the elasticity of the substrates in the elastic deformation calculation for the solid components is that of Elsharkawy et al. [6] followed by that of Liu et al. [7,8] or also more recently that of Wang et al. [9]. All of the aforementioned works assumed isothermal operating conditions and focused on the effects of the coating's rigidity and its thickness on lubrication performance. It was found that stiffer (or harder) coatings lead to increased central contact pressure and pressure spike and reduced contact area. The reverse effect was observed with soft coatings. It was also found that all these effects increase with the thickness of the coating layer. In addition to considering the effects of the mechanical properties of coatings on pressure and film thickness, Fujino et al. [10] and Chu et al. [11] also looked at the influence on stress distribution within the contacting solid elements. The former considered standard rolling conditions whereas the latter assumed a pure squeeze motion. Both works reached the same conclusion that for an enhanced fatigue life of the components, thicker and softer coatings are preferable as they lead to lower maximum stresses within the solids. This being said, surface coatings for elastohydrodynamic lubrication applications have always been selected based on their mechanical properties so as to reduce contact pressures and stresses and increase film thickness with the purpose of attaining an improved fatigue life for the corresponding components.

Interest in selecting coatings so as to control friction in EHL contacts has only appeared recently after several experimental works reported reduced friction in Diamond-Like-Carbon (DLC) coated EHL contacts such as Evans et al. [12] or Kalin et al. [13,14]. Originally, it was thought that the observed friction reduction is a consequence of boundary slip at the lubricant–solid interfaces. However, Björling et al. [15] reported friction reduction in their measurements done on DLC coated contacts with operating conditions under which boundary slip is very unlikely to occur. This led them to assume that the root cause for friction reduction might be thermal effects within the lubricating film. This assumption was later verified in [16] by validating the observed friction reduction measurements against numerical results. The employed numerical model did not include any boundary slip effects. It is based on the finite element full-system approach for thermal elastohydrodynamic lubrication developed by Habchi et al. [17]. Details of the incorporation of the surface coatings in the finite element model developed by the author for thermal non-Newtonian EHL contacts can be found in [18]. In this work, the author showed that friction in EHL contacts may be controlled by a suitable choice of surface coatings based on the thermal properties of their material. It was found that low thermal inertia coatings

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Nomenclature

α^*	Pressure–viscosity coefficient (Pa^{-1})
μ_0	Ambient pressure and temperature lubricant's viscosity (Pa s)
υ_c	Coating's Poisson coefficient
υ_s	Substrate's Poisson coefficient
ρ_c	Coating's density (kg/m^3)
ρ_s	Substrate's density (kg/m^3)
ρ_0	Ambient pressure and temperature lubricant's density (kg/m^3)
τ	Shear stress (Pa)
a	Hertzian contact radius (m)
c_c	Coating's heat capacity (J/kg K)
c_s	Substrate's heat capacity (J/kg K)
c_0	Ambient pressure and temperature lubricant's heat capacity (J/kg K)
C	Volumetric heat capacity (J/m^3 K)
E_c	Coating's Young's modulus of elasticity (Pa)
E_s	Substrate's Young's modulus of elasticity (Pa)
F	Contact external applied load (N)
h	Lubricant film thickness (m)
H	Dimensionless lubricant film thickness
I	Thermal inertia (J/m^2 K $s^{1/2}$)
k_c	Coating's thermal conductivity (W/m K)
k_s	Substrate's thermal conductivity (W/m K)
k_0	Ambient pressure and temperature lubricant's thermal conductivity (W/m K)
p	Pressure (Pa)
P	Dimensionless pressure
p_h	Hertzian contact pressure (Pa)
R	Ball's radius (m)
SRR	Slide-to-roll ratio $=(u_b-u_p)/u_m$
t_c	Coating's thickness (m)
T	Temperature (K)
T_0	Ambient temperature (K)
u_b	Ball's surface velocity (m/s)
u_p	Plane's surface velocity (m/s)
u_m	Mean entrainment speed $=(u_b+u_p)/2$ (m/s)
x, y, z	Space coordinates (m)
X, Y, Z	Dimensionless space coordinates

Subscripts

c	Coating
s	Substrate
0	Value of parameter at ambient pressure and temperature

Dimensionless parameters

$H = hR/a^2$

$P = p/p_h$

$X = x/a$

$Y = y/a$

$$Z = \begin{cases} z/a : \text{substrates} \\ z/t_c : \text{coatings} \\ z/h : \text{lubricant} \end{cases}$$

can significantly reduce friction and high thermal inertia coatings increase it under high sliding speed conditions. These effects were shown to increase with the coating thickness which was later verified by the experiments of Björling et al. [19]. The origins of these findings were investigated in [20] and it was found that the underlying mechanisms are purely thermal. In fact, a low thermal inertia coating acts as an insulator, trapping the generated heat within the lubricant film and confining it to the central area of the contact. This leads to an overall temperature increase within the central part of the film. The temperature rise is associated with a viscosity decrease leading to reduced friction coefficients. The exact opposite takes place when high thermal inertia coatings are employed leading to increased friction. Most importantly, since temperature variations are restricted to the central area of the contact and do not propagate towards the inlet, friction variations were attained without any noticeable effect on film thickness.

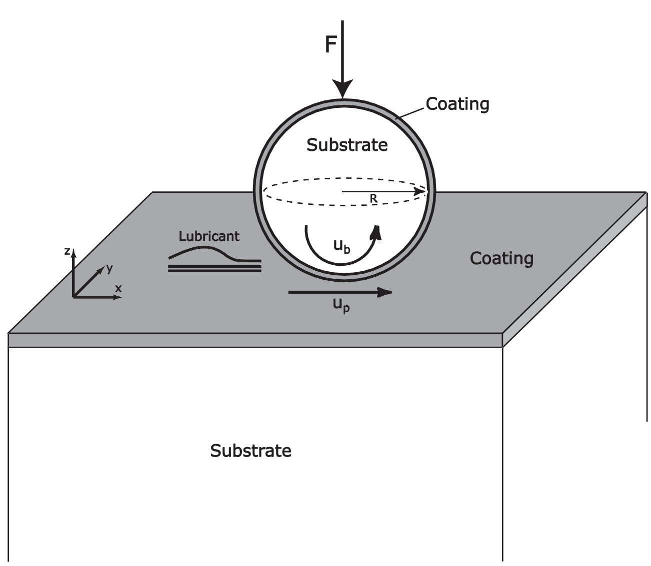

Fig. 1. Geometry of the coated TEHD circular contact.

In [20], only the influence of thermal properties of coatings on friction in EHL contacts was investigated along with the underlying heat transfer mechanisms. The current work investigates the possibility of controlling friction by a suitable choice of coatings based on the mechanical properties of their material or also the combination of mechanical and thermal (or thermo-mechanical) properties. The underlying physical mechanisms governing friction variations are also investigated in detail. Next, the numerical model employed in this work is briefly recalled.

2. Numerical model description

The numerical model employed in this work has been described in detail in [18]. Here, only its main features are recalled. The model is based on the finite element full-system approach introduced by Habchi et al. [17] for the solution of the thermal non-Newtonian EHL problem under steady-state considerations. The problem is reduced to that of an equivalent contact between a ball of radius R and a flat plane subject to an external applied load F as shown in Fig. 1. Solid surfaces are assumed to be smooth and moving at constant unidirectional velocities in the x-direction.

The numerical model entails two main sub-problems: EHL and thermal. The first consists in solving the 2D generalized Reynolds equation [21] applied to the fluid domain in the contact area, the