Tribology International 73 (2014) 57–68

Contents lists available at ScienceDirect

Tribology International

journal homepage: www.elsevier.com/locate/triboint

A numerical model for the solution of thermal elastohydrodynamic lubrication in coated circular contacts

W. Habchi *

Lebanese American University, Department of Industrial and Mechanical Engineering, Byblos, Lebanon

ARTICLE INFO

Article history:
Received 28 November 2013
Received in revised form
2 January 2014
Accepted 5 January 2014
Available online 11 January 2014

Keywords:
Thermal elastohydrodynamic lubrication
Surface coatings
Finite elements
Circular contacts

ABSTRACT

This paper presents a finite element model for the solution of thermal elastohydrodynamic lubrication in coated circular contacts. The model is based on a full-system finite element resolution of the elastohydrodynamic and heat transfer equations. The effects of the coating's thermal and mechanical properties on lubrication performance are investigated. Two categories of surface coatings are considered based on thermal properties: high and low thermal inertia. It is found that low thermal inertia surface coatings act as insulators leading to a localized increase in the lubricant's temperature at the center of the contact. Therefore, friction can be significantly reduced while film thickness is barely affected. The opposite effect is observed for high thermal inertia coatings. These effects increase with coating's thickness.

1. Introduction

Surface coatings have been extensively used, for a long time now, to protect contacting surfaces in rotating machine elements from wear and fatigue. The addition of coatings to the surface of moving machine elements first came in the case of solid-to-solid dry contacts. In fact, a carefully selected thin surface coating with appropriate mechanical properties may be used to reduce the severity of the contact between surface asperities leading to reduced fatigue and wear and thus increased component life. Over the last few decades, the use of surface coatings has also extended to the case of lubricated machine elements. In particular, the elastohydrodynamic lubrication (EHL) of coated surfaces has gained increased attention as it was found from the earliest studies on the topic that surface coatings can significantly affect the lubricating performance of machine elements such as gears, bearings and cams which operate under elastohydrodynamic regime. One of the first attempts to model EHL contacts with coated surfaces was achieved by Bennett and Higginson [1] for the case of line contacts under isothermal Newtonian regime. Later, Elsharkawy and Hamrock [2,3] introduced a numerical model for the solution of the EHL contact of a rigid cylinder with a rigid foundation coated by an elastic layer also under isothermal Newtonian regime. These works studied the effects of the mechanical properties of the coatings and their thickness on pressure and film thickness distribution in EHL line contacts. It was found that hard coatings (coatings with a Young's modulus superior to that of the substrate), lead to higher pressure spikes than in the uncoated case where pressure spikes are also higher than the case of soft coatings (coatings with a Young's modulus inferior to that of the substrate). It was also found that the central pressure increases with the rigidity of the coating while the contact width is reduced. This effect was found to increase with the thickness of the coating. A central film thickness formula was even introduced for the case of coated line contacts lubricated by a non-Newtonian lubricant [4]. The solution of the three-dimensional case was out of reach until recently owing to limitations in computer power. One of the first studies dealing with circular EHL contacts with coated surfaces is that of Jin [5,6]. Most of the cases mentioned so far are limited to the case of rigid substrates bonded to elastic coatings. The incorporation of the elasticity of the substrate into the analysis was done recently by Elsharkawy et al. [7] using the differential deflection method introduced by Holmes et al. [8] to evaluate the elastic deformation of the contacting solids. Liu et al. [9,10] also incorporated the elasticity of the substrate in their analysis of coated EHL point contacts using the Discrete Convolution-Fast Fourier Transform method (DC-FFT) [11] to evaluate the elastic deflection of the contacting elements. The above mentioned studies on coated circular EHL contacts showed that coatings had the same effect on pressure and film thickness distribution as for the line contact case. That is, pressure spike height and central pressure increase with the rigidity of the coating while contact radius is reduced. As for line contacts, it was found that these effects increase with the coating's thickness.

All works mentioned so far assume isothermal conditions and focus on the effect of the rigidity of the coating and its thickness on pressure and film thickness in EHL contacts. The effect of the

* Tel.: +961 9547254.
E-mail address: wassim.habchi@lau.edu.lb

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http://dx.doi.org/10.1016/j.triboint.2014.01.002

Nomenclature

η	lubricant's generalized Newtonian viscosity (Pa.s)
$\bar{\eta}$	dimensionless lubricant's generalized Newtonian viscosity
λ_c, μ_c	coating's Lamé parameters (Pa)
λ_s, μ_s	substrate's Lamé parameters (Pa)
μ	lubricant's viscosity (Pa.s)
μ_R	lubricant's viscosity at reference state (Pa.s)
μ_∞	lubricant's viscosity extrapolated to infinite temperature (Pa.s)
υ_c	coating's Poisson coefficient
$\upsilon_{c,eq}$	coating's equivalent Poisson coefficient
υ_s	substrate's Poisson coefficient
$\upsilon_{s,eq}$	substrate's equivalent Poisson coefficient
Λ	limiting stress–pressure coefficient
ρ	lubricant's density (kg/m^3)
$\bar{\rho}$	dimensionless lubricant's density
ρ_c	coating's density (kg/m^3)
ρ_s	substrate's density (kg/m^3)
ρ_R	lubricant's density at reference state (kg/m^3)
θ	modified dimensionless Hertzian pressure
τ	shear stress (Pa)
τ_L	limiting shear stress (Pa)
a	Hertzian contact radius (m)
c	lubricant's heat capacity (J/kg.K)
c_c	coating's heat capacity (J/kg.K)
c_s	substrate's heat capacity (J/kg.K)
C	lubricant's volumetric heat capacity (J/m^3.K)
E_c	coating's Young's modulus of elasticity (Pa)
$E_{c,eq}$	coating's equivalent Young's modulus of elasticity (Pa)
E_s	substrate's Young's modulus of elasticity (Pa)
$E_{s,eq}$	substrate's equivalent Young's modulus of elasticity (Pa)
F	contact external applied load (N)
G	lubricant effective shear modulus (Pa)
h	lubricant film thickness (m)
H	dimensionless lubricant film thickness
H_0	film thickness constant parameter
k	lubricant's thermal conductivity (W/m.K)
k_c	coating's thermal conductivity (W/m.K)
k_s	substrate's thermal conductivity (W/m.K)
p	pressure (Pa)
P	dimensionless pressure
p_h	Hertzian contact pressure (Pa)
p_0	ambient pressure (Pa)
p_R	reference 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 (°C)
T_0	ambient temperature (°C)
T_R	reference temperature (°C)
u, v, w	x, y and z-components of the solid's elastic deformation field (m)
U, V, W	dimensionless x, y and z-components of the solid's elastic deformation field
u_m	mean entrainment speed$=(u_b+u_p)/2$ (m/s)
u_b	Ball's surface velocity (m/s)
u_p	plane's surface velocity (m/s)
u_f, v_f	lubricant's velocity field x and y components (m/s)
V	volume (m^3)
V_R	volume at reference state (m^3)
x, y, z	space coordinates (m)
X, Y, Z	dimensionless space coordinates

Subscripts

b	ball
c	coating
p	plane
s	substrate

thermal properties of the coating on the lubrication performance of coated EHL contacts has earned very little attention. However, recent experimental works such as that of Evans et al. [12] or Björling et al. [13] reported significantly reduced friction (up to 50%) in DLC-coated EHL contacts. In [13], based on a simplified analytical estimation of the temperature increase in the lubricant film induced by the DLC surface coating, the authors suggested that the observed friction reduction might be a thermal phenomenon. This hypothesis was later verified in [14] by validating the friction measurements against numerical predictions obtained using the full-system finite element approach [15,16]. In [14], only thin 2 μm DLC coatings were considered which had very little influence on the elastohydrodynamic part of the solution but a significant effect on the thermal part. It was shown that this type of coating, which has a relatively low thermal inertia, prevents heat from being diffused from the center of the contact towards the peripheral area and the solids. Thus, a local increase in the temperature of the lubricant film at the center of the contact is observed. This local increase in temperature leads to reduced friction with very little effect on film thickness since the lubricant's temperature at the inlet of the contact is barely affected. In fact, the thermal inertia I of a given material is defined as $I=\sqrt{k\,\rho\,c}=\sqrt{k\,C}$, where C is the volumetric heat capacity of the material and k its thermal conductivity. Thus a high thermal inertia material usually has a high thermal conductivity and volumetric heat capacity. As such, it has a high ability to transport heat by conduction and advection, respectively. In fact, the thermal conductivity of a material represents its ability to transport heat by conduction while its volumetric heat capacity represents its ability to store and transport heat by advection (heat transfer by mass). On the other hand, a low thermal inertia material has a low ability to transport heat by conduction and advection, and thus it acts as an insulating material. The coating's thickness being very thin (2 μm), the effect of the coating was only included in the thermal part of the numerical model while in the EHL part uncoated surfaces were considered.

Based on the above, it is obvious that the lubrication performance of thermal elastohydrodynamic (TEHD) contacts can be controlled through an adequate choice of surface coatings that is not only based on their mechanical properties but also on their thermal properties. The latter have a significant impact on friction in coated EHD contacts. The author believes that this topic deserves being further investigated in detail. The current paper presents a numerical model for the solution of coated TEHD circular contacts incorporating surface coatings in both the EHL and thermal parts of the model. This model is used to run a comprehensive investigation of the effects of the mechanical properties as well as the thermal properties of the coating on the lubrication performance of TEHD circular contacts in terms of pressure, film thickness as well as friction. Next, the numerical model employed in this work is described in detail.