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Tribological evaluation of a coated spur gear pair

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ABSTRACT

A numerical thermal elastohydrodynamic lubrication model is developed for a coated gear pair, in which the elastic deformation is calculated through the frequency response function of Green function and the discrete-convolute fast Fourier transform. Tribological performance of the gear pair is evaluated in terms of the pressure, the film thickness, the coefficient of friction, the frictional power loss, as well as the temperature rise during meshing. Effects of the elastic modulus and the coating thickness of the coating material are studied under various working conditions. Results provide tribological guidance for engineering practice of gear coating design.

1. Introduction

Coatings are widely used in surface engineering to prevent wear and fatigue, and to prevent corrosion and surface decoration [1]. Surface coatings, e.g. tungsten carbon carbide coating (WC/C), provide significant increase in carburized gear scuffing resistance and wear resistance [2]. Alanou et al. [3] experimentally showed that thin hard coatings may enhance the scuffing and micropitting performance of gears since they exhibit very high levels of surface hardness combined with a low traction coefficient against dry steel, but they also found poor adherence under some cases.

Back to twenty years ago Huang et al. carried out a series of experiments using a ball-on-disc testing machine, to investigate the sliding friction behavior of physically vapor-deposited TiN, CrN coatings under both dry and lubricated conditions and found plowing actions of the hard asperities of the coating surface and material transfer from the steel counterface to the coating surface played an important role in determining the friction behavior during the initial transient state and steady state respectively [4].

Holmberg et al. [5] reviewed the coating tribology and explained tribologically important properties in different zones of coated surface, for example shear strength, chemical reactivity and roughness should be controlled for the coating surface while the adhesion behavior and shear strength are the key properties for the interface between the coating and the substrate. Krantz et al. [6] conducted surface fatigue test of metal-containing, carbon-based coated gears and results show that compared with uncoated gears, the fatigue life owing to the coating display approximately a six-fold increase. Michalczewski et al. [7] studied the resistance to rolling contact fatigue of WC/C-coated gears based upon a back-to-back gear test rig using the FZG PT C/10/90 method, results indicate that the resistance to the pitting wear of coated gears depends not only on coating material but also on which gear is coated. Very recently Bobzin et al. [8] experimentally investigated the reduction of friction losses in powertrain by DLC coatings on highly loaded gears under severe rolling-sliding conditions, and found compared to uncoated gears friction losses in EHL were reduced by up to 25% using the industrial reference DLC-REF1 and 39% using ZrC_g at higher loads and circumferential speeds.

However, due to the lack of reliable mathematical models and design systems for surface engineering, the requirements in the surface layer for specific applications are usually determined on the basis of experience and empirical formula [9]. Theoretically, finite element method are applied extensively for solving coated contact problems. Mao et al. [10] developed a numerical technique including integral transform and finite element methods for solving contact problems in layered rough surfaces under both normal and tangential forces. Semi-analytical models (SAM) [11,12] which are based on the Boussinesq–Cerruti integral equations have been developed to simulate rough surfaces in contact and are successfully applied to coated contact problems [13].

Tribological performance of typical uncoated gear pairs are studied extensively in the past decades, with the development of the theory of elastohydrodynamic lubrication and the theory of

Abbreviations: TEHL, thermal elastohydrodynamic lubrication; LOA, the line of action; DC-FFT, Discrete convolution, fast Fourier transform; FRF, Frequency response function

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Nomenclature

b_h	Half Hertzian contact width (m),$b_h=\sqrt{8FR/\pi E^*}$
B	Gear tooth width (m),$B=0.1$ m
$c, c_{a,b}$	Specific heat of the oil and the solids (J/(kgK))
D	Dimensionless elastic deformation
E_s	Elastic modulus of the substrate material (N/m^2),$E_s=2.0\times10^{11}$ Pa
E_c	Elastic modulus of the coating material (N/m^2),$E_c\in[1\sim8]$ E_s
E^*	Equivalent elastic modulus (N/m^2)
F	Load per tooth width (N/m)
h	Gap height between surfaces (m)
h_c	Coating thickness (m),$h_c\in[0,\ 160]$ μm
H	Dimensionless gap height $H=hR/b_h^2$
$k, k_{a,b}$	thermal conductivity of the oil and the solids (W/(mK))
m	the counterpart of radius in the frequency domain
N_1	Input speed (r/min)
Q_f	Total frictional power loss in the contact region (W/mm)
p	Pressure (Pa)
P	Dimensionless pressure, $P=p/p_h$
p_h	Hertzian maximum pressure (Pa),$p_h=2F/(\pi b_h)$
R	Equivalent radius (m),$R=(R_aR_b))/(R_a+R_b)$
R_a, R_b	Radius of the two surfaces, respectively (m)
t	Time (s)
$\bar{t}$	Dimensionless time $\bar{t}=tu_r/b_h$
T	Current temperature (K)
T_0	Ambient room temperature, oil inlet temperature (K)
$\overline{T}$	Dimensionless temperature $\overline{T}=T/T_0$
T_1	Input torque (Nm)
u_r	Rolling velocity (m/s)
U_a, U_b	Dimensionless velocity of the two solids
x	Coordinate of the domain along rolling direction $x\in[-4,\ 3]b_h$
X	Dimensionless coordinate of the domain along rolling direction $X=x/b_h$
X_a	Inlet value of the dimensionless domain $X_a=-4$
X_b	Outlet value of the dimensionless domain $X_b=3$
z	Coordinate along the direction across the film thickness (m)
Z	Dimensionless coordinate along the direction across the film $Z=z/h$
α	Viscosity-pressure index of the lubricant (p_a^{-1}),$\alpha=2.2\times10^{-8}$ Pa^{-1}
ε	Parameter in the Reynolds equation
η	Viscosity of the lubricant (Pas)
η_0	Ambient viscosity of the lubricant (Pas),$\eta_0=0.04$ Pas
η^*	Equivalent viscosity of the Eyring fluid (Pas)
$\bar{\eta}$	Dimensionless viscosity $\bar{\eta}=\eta/\eta_0$
$\rho, \rho_{a,b}$	Current density of the oil and the solids (kg/m^3)
ρ_0	Ambient density of the lubricant (kg/m^3)
$\bar{\rho}$	Dimensionless density of the lubricant $\bar{\rho}=\rho/\rho_0$
τ	Film shear stress (MPa)
$\bar{\tau}$	Dimensionless shear stress $\bar{\tau}=\tau/p_h$
τ_0	Eyring characteristic shear stress (MPa),$\tau_0=10$ MPa
ν_s, ν_c	Possion ratio of the substrate material and the coating, all equals 0.3
$\tilde{u}$	2D FRF of Green function
μ	The coefficient of friction
μ_c	Shear modulus of the coating, $\mu_c=E_c/[2(1+\nu_c)]$

mixed lubrication. Liu et al. [14–16] studied effects of tooth surface roughness, non-Newtonian fluid behavior, lubrication starvation, tooth dynamic load on gear tooth tribological behavior while Li et al. [17,18] focused on the fatigue and scuffing behavior of gears based upon a mixed lubrication model. However, lubricated contact of coated gear pairs are yet to be studied.

For a coated elastic half-space surface, an explicit Green's function is not available for the normal displacement in the space domain. Its frequency response function in the frequency domain has been derived by Nogi and Kato [19]. Then the FRF can be transferred into spatial domain to get the influence coefficients by using inverse fast Fourier transform (IFFT). Once the influence coefficients and the pressure distribution are obtained, the elastic deformation of coated surfaces can be calculated using the discrete convolution FFT method [20].

This work studies the tribological performance of a coated spur gear pair by developing a numerical coated thermal elastohydrodynamic lubrication model of coated gear tooth meshing. The generalized Reynolds equation [21] is applied to incorporate the effect of thermal environment and the non-Newtonian lubricant. Energy equations of the oil film as well as the tooth solids are proposed to solve the temperature distribution within the contact region. The DC-FFT method [20] is used to solve the surface elastic deformation where the influence coefficient is expressed in the frequency domain. Effect of working conditions and the coating properties on pressure, film thickness, surface traction, as well as the temperature are studied within the whole process of gear meshing.

2. Methodology

Spur gear lubricated contact problem can be simulated as line contact between two circles at each engaging position, as shown in Fig. 1. The proposed coated TEHL model is based upon the traditional uncoated EHL model the authors developed [22].

The coated EHL model mainly consists of the Reynolds equation which describes the fluid flow within the contact region, the film thickness equation which describes the gap between tooth interfaces, the force balance equation, and equations describing the viscosity-pressure relationship and the density–pressure relationship. The following equations are used to dimensionless the model:

$X=x/b_h,\ \ P=p/p_h,\ \ Z=z/h\quad H=hR/b_h^2,\ \ \overline{T}=T/T_0,\ \ \bar{t}=tu_r/b_h,\ \bar{\rho}=\rho/\rho_0,\ \ \bar{\eta}=\eta/\eta_0,\ \ \bar{\tau}=\tau/p_h$ where $X,\ Z, H, P, \overline{T}, \bar{t},\ \bar{\rho},\ \bar{\eta},\ \bar{\tau}$ are dimensionless forms of the distance from the Hertzian contact center along the rolling direction x, the distance from the bottom interface across the film thickness direction z, the film thickness h, the pressure p, the temperature T, the time t, the film density ρ, the film viscosity η, and the film shear stress τ, respectively. u_r is the rolling speed of the engaging surfaces, T_0 is the ambient temperature, ρ_0 and η_0 are the density and the viscosity of the fluid at ambient environment respectively. The operating temperature of the oil is the same as the ambient temperature.

b_h and p_h are the half Hertzian contact width and the maximum Hertzian pressure, respectively and can be expressed as

$$b_h=\sqrt{8FR/\pi E^*},\ \ p_h=2F/(\pi b_h) \tag{1}$$

where F is the load per tooth width (N/m), R is the equivalent radius $R=(R_aR_b))/(R_a+R_b)$, R_a and R_b are the radius of curvature of