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Low degree of freedom approach for predicting friction in elastohydrodynamically lubricated contacts

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ABSTRACT

A low degree of freedom, semi-analytical model for rapid estimation of the friction coefficient in elastohydrodynamically lubricated contacts was developed and tested. Its estimates are based on the shear rate dependent Carreau equation for the apparent viscosity, together with the hydrodynamic pressure and the temperature of the lubricant. To validate the approach, the model's predictions were compared to experimental coefficient of friction measurements acquired using a ball-on-disc test device at various applied loads, entrainment velocities, and slide to roll ratios. The model's predictions were in good agreement with the experimental results, showing that it is suitable for use in multibody dynamics analyses where rapid computation of elastohydrodynamic friction is required to minimize computing time and resource consumption.

1. Introduction

The continuing development of engineering machines has provided many challenges in the field of tribology. Many machine elements require lubrication to reduce friction and wear. Consequently, there is great interest in studying the static, dynamic, and vibrational behavior of lubricants and components in order to improve their function and efficiency. Friction and the risk of failure due to various contact types have been investigated in a range of applications such as the contacts between the roller and the raceway in rolling element bearings, between gear teeth in gear transmissions, and between the follower and the camshaft in cam mechanisms. All of these systems operate in the elastohydrodynamic lubrication (EHL) regime, where the lubricant pressure is high enough to significantly change the geometry of the lubricated conjunction through elastic deflection of the mating surfaces.

EHL has been studied for several decades and many models of varying complexity have been developed to predict variables such as the minimum film thickness, maximum pressure and friction. However, accurate modeling of the lubricant's rheological behavior in systems operating under EHL regimes continues to be challenging due to the thinness of the oil film and the relatively large elastic deformation of the mating surfaces, see e.g. [28].

Under EHL conditions, the relationship between shear stress and shear rate is not always linear. In many studies, this relationship has been modeled using the Eyring sinh law. One early example is the work [22] of Tevaarwerk and Johnson, who used this model to estimate friction in EHL contacts. Evans and Johnson [11] also applied the Eyring sinh law and used it to identify four traction regimes with different nonlinear behaviors. A more recent example of its use was reported by Jacod et al. [19], who proposed a generalized traction curve for isothermal EHL contact. Although Eyring friction models remain popular, multiple studies have shown that they are inadequate for describing shear thinning behavior, see e.g. [4,21].

Recently, Liu et al. [26] implemented a finite difference discretization of the Reynolds equation, which was coupled with a rheological model that was not based on the Eyring sinh law. Instead, the rheology was described using the free volume model for compressibility together with the Carreau expression for the shear thinning response. This approach yielded a traction curve for low sliding velocities and relatively low loads. Subsequently, Habchi and co-workers [15] successfully predicted film thicknesses and pressure distributions at higher loads using a finite element model. This approach has also been used by Habchi et al. [14] with a non-Newtonian rheology model, which takes the limiting shear stress into account, based on lubricant transport properties to assess EHL friction. In the same paper, they defined four traction regimes that may be encountered in EHL contacts (linear, non-linear, plateau and thermoviscous), and linked them to four dimensionless numbers. In their approach, the Murnaghan

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Nomenclature

a_v	Thermal expansivity [K^{-1}]
B_F	Fragility parameter in the new viscosity equation [-]
C_f	Coefficient of friction [-]
F	Load [N]
G	Dimensionless material parameter [-]
g	Thermodynamic interaction parameter K_l[-]
K_l	Thermal conductivity of lubricant [W/mK]
k_0	Isothermal bulk modulus at $p=0$ [Pa]
k_{00}	k_0 at zero absolute temperature [Pa]
k_0'	Pressure rate of change of isothermal bulk modulus at $p=0$ [-]
n	Power law exponent [-]
p	Pressure [Pa]
SRR	Slide to roll ratio, $SRR=(U_A-U_B)/(U_e)$
T	Temperature [K]
U	Dimensionless speed parameter [-]
U_e	Entrainment speed [$m \cdot s^{-1}$], $U_e=0.5(U_A+U_B)$
V	Volume [m^3]
V_0	Volume at $p=0$ [m^3]
V_R	Volume at reference state, T_R, $p=0$ [m^3]
W	Dimensionless load parameter [-]
β_k	Temperature coefficient of k_0 [K^{-1}]
$\dot{\gamma}$	Shear rate [s^{-1}]
η	Apparent viscosity [$Pa \cdot s$]
Λ	Limiting stress pressure coefficient [-]
λ_R	Relaxation time at TR and ambient pressure [s]
μ	Limiting low-shear viscosity [$Pa \cdot s$]
μ_R	Low shear viscosity at TR and ambient pressure [$Pa \cdot s$]
μ_∞	Viscosity extrapolated to infinite temperature [$Pa \cdot s$]
τ	Shear stress [Pa]
φ	Dimensionless viscosity scaling parameter [-]
φ_∞	Viscosity scaling parameter for unbounded viscosity [-]

equation of state and a Vogel-like equation were employed to model the lubricant's rheological behavior. These models make it possible to perform detailed studies on the effect of shear thinning and piezoviscous parameters. The output of their model was shown to be in good agreement with empirical data from friction measurements, presented in the paper by Björling et al. [8], acquired using a ball-on-disc machine.

Since the lubricant's properties vary with e.g. pressure, temperature and shear rate, a rheological model describing this variation is essential for the mathematical modeling of EHL contact.

Barus [7] presented a linear expression for the relationship between pressure and viscosity based on experimental observations on marine glue. Later on, an exponential expression (Eq. (27-A)), mistakenly attributed to Barus, has been extensively used in the EHL community. This expression predicts the lubricant's viscosity rarely accurate at pressures of up to a few 100 MPa. These low pressures are where the greatest divergence from Arrhenius behavior occurs. Roelands [33] also presented an exponential pressure–viscosity relationship that for some fluids provides reasonably good results even at somewhat higher pressures, but it is generally not applicable when the pressure exceeds 1 GPa. Roelands' pressure–viscosity relation has become widely used in numerical simulations of EHL contact, see e.g. [25,37,38].

Friction is determined mainly by the viscosity in the central part of the contact, where the pressure is high, and it has been observed that using the exponential expression or Roelands' pressure–viscosity relationship leads to unrealistically high friction coefficients, see e.g. Habchi [13]. The film thickness is, however, governed by the conditions in the inlet of the EHL contact, where the pressure is relatively low. Rheological characterization by Bair [3] of a variety of lubricants, has highlighted the importance of selecting an appropriate model when simulating highly loaded EHL contacts. To reliably capture both the behavior of the viscosity and the density over the range of pressures typically found in EHL contacts, the rheological model must include a realistic representation of the lubricant's compressibility.

Fully deterministic models such as those discussed above, i.e. [14,23,26], can predict the film's thickness and pressure distributions with good accuracy and in high detail. However, they are not fast enough to be used to compute the film thickness and pressure over a whole engine cycle in the dynamically loaded EHL contact between the cam and the follower. A fast and reliable tool for estimating friction in EHL contacts would therefore be extremely useful in various industrial applications, particularly if it were compatible with Multibody Dynamic (MBD) software such as BEAST [36], a three dimensional MBD software package with a contact model for EHL line and point contacts that is based on numerical solution of the Reynolds equation, see e.g. Venner [38]. In an MBD software package, the contact behavior must be evaluated (in terms of stiffness and damping) repeatedly, so it is very important to be able to obtain acceptably accurate estimates at the lowest possible computational cost.

This paper presents a method for evaluating contact behavior under EHL conditions that requires much less computational time and resources than a fully deterministic model but retains sufficient accuracy to be used for determining the friction in bearings, gears, cam mechanisms and other typical tribological components. The methodology developed in this work is relatively similar to that presented by Wang and Poll [32], which is based on the Maxwell model. The present approach is based on the Tait equation of state (instead than the Maxwell model) and on a so-called shifted Carreau model that includes the compressibility of the lubricant. A simplified temperature distribution algorithm is implemented in the present approach, in order to consider the thermal effects. The model's feasibility was evaluated by comparing its predictions to the results of friction measurements, conducted using a ball-on-disc device at a range of loads, entrainment velocities and SRRs. The comparison showed that the model's predictions were in good agreement with the experimental data over a wide range of slide-to-roll ratios.

2. The model

This section describes the equations, variables and parameters used in our approach and the assumptions made to facilitate analysis.

2.1. Rheology

A low degree of freedom (LDOF) approach for engineering applications was developed to obtain rapid estimates of friction in EHL contacts. This approach is based on the shifted Carreau equation, which describes the apparent viscosity η as a function of shear rate $\dot{\gamma}$, pressure p and temperature T. More precisely:

$$\eta(\dot{\gamma},p,T)=\mu(p,T)\left[1+\left(\dot{\gamma}\lambda_R\frac{\mu(p,T)}{\mu_R}\frac{T_R}{T}\frac{V}{V_R}\right)^2\right]^{\frac{n-1}{2}}, \tag{1}$$