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A numerical and experimental study on the interface friction of ball-on-disc test under high temperature

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ABSTRACT

This paper investigates both experimentally and theoretically the effect of lubricant on the interface stresses under high temperature. A statistical formula was developed to predict the Stribeck curves by considering asperity-lubricant interactions at the contact interface. This enables us to decouple the interface friction stresses contributed by lubricant and asperity solid-solid contact sliding. It was found that the temperature increase at the interface can considerably alter the lubricant performance because of the effect of lubricant additive and that the conventional assumption of constant friction coefficient for asperity contact is unreasonable. As the temperature increased, the boundary layer formed by lubricant molecules dissolved while a soft solid layer was formed by lubricant additives at the contact interface. The finding was confirmed by the effect of lubricant performance on the experimentally measured Stribeck curves. It was also found that in the boundary lubrication regime, friction is sliding-speed-dependent due to the formation-destruction cycle of the soft solid layer formed by lubricant additives when the lubricant temperature increases.

1. Introduction

Lubricant plays a critical role in tribology systems such as in bearings and metal forming processes. Due to the microscale asperity-lubricant and asperity-asperity interactions, the interface frictional stress consists of two components in a contact sliding system, the solid-solid friction on the asperity contact area and the lubricant shear traction in the rest of the contact region. Thus Stribeck curves [1] have been widely used to investigate such interface friction stresses because they can cover different lubrication regimes, i.e., boundary, mixed and hydrodynamic lubrication.

Extensive efforts have been made to construct such Stribeck curves by either experimental or numerical means. An experimental measurement of a Stribeck curve can be achieved, as the applied load and friction stress can be directly obtained by using a ball-on-disc technique. A numerical approach, however, is much more complicated since it requires to deal with the microscale asperity–lubricant interaction and the multi-scale deformation at the contact interface. Mixed elastohydrodynamic lubrication (EHL) models [2–5] are usually required to calculate the interface hydrodynamic pressure and friction stress. To simplify the analysis, Johnson et al. [3] developed a mixed EHL model by using constant scaling factors. Following their global sharing concept for mixed EHL, Gelinck and Schipper [6] calculated the Stribeck curve for lubricated elastic rough line contact by using the Newtonian behaviour of lubricant. Subsequently, Lu et al. [7] compared the theoretically predicted Stribeck curve with their experimental measurements by using the constant scaling method. Some further works on the Stribeck curves can also be found in the literature [8,9]. Apart from statistical calculations, deterministic EHL [10–14] models have also been used to obtain Stribeck curves. Martini et al. [15] and Wang et al. [16] calculated the Stribeck curves under various sliding speeds. In these works, the nonlinear elastic-viscous behaviour of lubricant was used to calculate the lubricant shear stresses. Zhu et al. [17] calculated the Stribeck curves for lubricated counter formal contacts of rough surfaces by considering the temperature effect.

As discussed above, the lubricant shear traction and solid-solid contact friction stresses are equally important, as both of them contribute to the interface friction. While the former is dependent on the effective lubricant viscosity which is dependent on the pressure and temperature, relative sliding speed and average thickness of the lubricant film, the latter largely relies on the strength of the boundary layer formed by lubricant molecule or additives on the solid contact area. The formation and strength of the boundary layer are significantly affected by the operation conditions (e.g., relative sliding speed and lubricant temperature).

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Nomenclature

A	contact area ratio
E	Young's modulus
E'	equivalent Young's modulus
E^*	reduced Modulus
F_N	applied load in the ball-on-disc test
F_t	measured total friction force
γ	surface direction pattern
ϕ_x	average pressure flow factor
ϕ_y	average pressure flow factor
ϕ_s	average shear flow factors
h	nominal separation
H	dimensionless nominal separation
h_a	asperity height
h_T	average lubricant film thickness
H_t	dimensionless average lubricant film thickness
H_{tc}	percolation threshold
μ_a	friction coefficient for the solid-solid contact sliding
N_a	asperity density
p_a	average asperity contact pressure
$\bar{p}_a$	average asperity contact pressure over the whole real contact area
p_f	average hydrodynamic pressure
R_a	average asperity radius
ρ	lubricant density
ρ_0	bulk lubricant density
S_0	constant for the temperature effect on the lubricant viscosity
σ	standard deviation of asperity heights
T	lubricant temperature in Kelvin
T_0	lubricant reference temperature in Kelvin
τ_a	asperity friction stress
τ_f	lubricant shear traction
u_1	ball surface speed
u_2	disc surface speed
υ	Poisson's ratios
w	elastic deflection
η	effective lubricant viscosity
η_0	lubricant viscosity at the reference temperature and atmosphere pressure

It should be noted that direct asperity solid–solid contact occurs when the boundary layer breaks down, and the solid friction stress will approach the dry solid friction. Numerical calculations of solid friction stress, however, always assume a constant friction coefficient for the asperity sliding contact by neglecting the existence of a boundary layer. Compared with numerical calculations, experimental measurements are much easier, as both applied force and total friction force can be accurately detected by a load cell. However, the measured forces always include the contributions from both the solid-solid contact part and lubricant contact part. It is difficult to distinguish their individual effects on the interface stresses. Thus, a feasible method is required to investigate the lubricant performance on the rough surface contact, and then identify the influence of the boundary layer on the interface friction stress.

This paper investigates the lubricant performance both experimentally and theoretically by considering the effects of sliding speed and lubricant temperature. The experimental Stribeck curves will be obtained by using the ball-on-disc technique under lubrication, which will enable us to understand the influence of lubricant performance on the interface friction stress. A theoretical formulation of the Stribeck curves corresponding to the experimental conditions (e.g., measured surface roughness, asperity density and lubricant viscosity) will also be performed with the aid of a statistical analysis. A full contact model will then be developed to assess the lubrication of ball-on-disc and to decouple the contributions of asperity solid–solid contact and lubricant film to the interface friction stress. Moreover, the mechanism of sliding-speed-dependent friction will be experimentally explored by using the ball-on-disc test under high temperature. It is expected that an in-depth understanding of lubricant performance under different contact sliding conditions will be achieved.

2. Experiment

2.1. Ball-on-disc setup

The measurement of the interface friction coefficient was conducted on a tribometer, CETR UMT100. Fig. 1 illustrates the testing configuration. The applied normal load was controlled by a vertically driven motor that can maintain a constant force during the test. The applied load and interface friction force were recorded by a load cell. The sliding speed at the contact zone was controlled by a rotation driving motor. To investigate the temperature effect on the lubricant performance, a heating chamber was manufactured and installed to control the lubricant temperature. To satisfy the full-flood condition for the ball-on-disc contact and avoid damaging the heating chamber caused by lubricant leaking, a liquid container was specifically manufactured and installed inside the heating chamber, as demonstrated in Figs. 1 and 2. In a test, the disc was fixed inside a liquid container. The ball surface was machined to the spherical radius of 0.07 m.

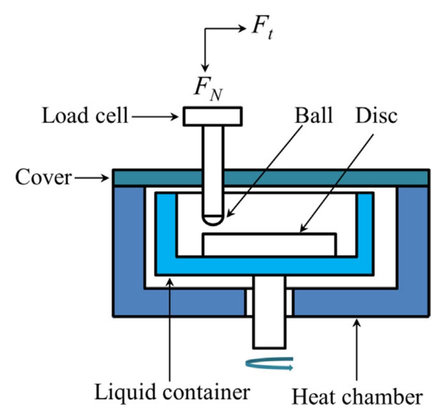

Fig. 1. Schematic configuration of the tribological tests.

2.2. Rough surface measurement

An accurate measurement of surface topography is important to the statistical modelling. Hence, the disc and ball surfaces used in experiments were measured with a Zygo NewView 700; lateral resolution of which is 1.375 µm. The profile of a rough surface, $h(x)$, was then extracted from the measured topography, and was in turn used to obtain the asperity density, average asperity radius etc. of the surface, following the spectrum theory [18]. The asperity density, N_a, average asperity radius, R_a, and standard deviation of asperity heights, σ, can be expressed in terms of three quantities, m_0, m_2 and m_4, known as spectral moments [18] determined by