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Finite deformation effects in soft elastohydrodynamic lubrication problems

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ABSTRACT

Soft elastohydrodynamic lubrication regime is typical for many elastomeric and biological contacts. As one or both contacting bodies are then highly compliant, relatively low contact pressures may lead to large deformations which are neglected in the classical EHL theory. In the paper, the related finite-deformation effects are studied for two representative soft-EHL problems. To this end, a fully-coupled nonlinear formulation has been developed which combines finite-strain elasticity for the solid and the Reynolds equation for the fluid, both treated using the finite element method with full account of all elastohydrodynamic couplings. Results of friction measurements are also reported and compared to theoretical predictions for lubricated contact of a rubber ball sliding against a steel disc under high loads.

1. Introduction

An increased interest in soft elastohydrodynamic (elastic-isoviscous) lubrication regime is recently observed which is due to numerous applications in technology (elastomeric seals, tyres, etc.), but also because this lubrication regime occurs in many biotribological systems (e.g., synovial joints, human skin contact, oral processing of food, etc.), see, e.g., [1–3].

Several aspects of lubricated soft contacts have already been studied experimentally, for instance, the influence of load and elastic properties [4], non-Newtonian effects [5], and surface wetting [5,6]. Experimental investigations of the roughness effects and of the transition from hydrodynamic to boundary lubrication can be found in [6–8], see also the related theoretical studies in [9,10].

Contrary to the more popular hard-EHL contacts operating in the elastic-piezoviscous regime, the pressure is relatively low in the soft-EHL contacts. Nevertheless, the elastohydrodynamic coupling is crucially important because one or both contacting bodies are highly compliant. This also means that relatively low contact pressures may lead to finite deformations of the contacting bodies. The corresponding effects have so far attracted little attention, and a study of those effects is pursued in this work.

Modelling of an EHL problem involves description of the fluid part, of the solid part and of the elastohydrodynamic coupling [11,12]. The fluid part is conveniently modelled using the classical Reynolds equation. In the classical EHL theory, the solid part is modelled using the linear elasticity framework. Furthermore, the elasticity problem is usually formulated for a half-space for which specialized solution techniques are available. While both assumptions (linear elasticity and half-space approximation) are fully adequate for hard-EHL problems, this is not necessarily so in the case of soft-EHL problems due to geometrical and material nonlinearities that are associated with the finite deformations and finite configuration changes.

The elastohydrodynamic coupling involves the solid-to-fluid coupling (lubricant film thickness depends on the deformation of the body) and the fluid-to-solid coupling (the hydrodynamic pressure and the shear stress are applied to the body as a surface traction). However, the Reynolds equation is formulated in an Eulerian frame on the contact boundary of the solid, and this introduces an additional coupling [13] due to the finite configuration changes (the domain on which the Reynolds equation is solved depends on the deformation of the solid).

A possible approach to modelling of the soft-EHL problems is to use the classical EHL theory, i.e., to neglect all the finite-deformation effects mentioned above. For instance, de Vicente et al. [7] applied the classical EHL solver to simulate an elastomeric point contact and derived a regression equation for the friction coefficient by fitting the corresponding numerical solutions. Their numerical solution was compared to experimental measurements, and a very good agreement was observed [7,14]. The experiments involved moderately large deformations as the ratio of the Hertzian contact radius to the ball radius was 0.17. Quite surprisingly, as shown in the present paper, the regression equation of [7] agrees very well with the predictions of the present fully nonlinear

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model also for much higher loads (and thus for much larger deformations) for which the ratio of the Hertzian contact radius to the ball radius exceeds 0.3. Despite the good agreement in terms of the friction coefficient, the local values of film thickness and hydrodynamic pressure do not exhibit such a good agreement. Furthermore, some notions, such as the central film thickness, are no longer well defined once finite configuration changes are involved.

The elastic half-space approximation is a feasible approach for point and line contacts. If more complex geometry is involved, computational techniques such as the finite element method are needed to reliably determine the contact pressure. In early works, the linear elasticity framework combined with the finite element method was adopted in the modelling of soft-EHL problems, for instance, in [15,16] in the context of reciprocating seals.

Finite deformation effects are partially taken into account in a more advanced approach in which the contact pressure is computed for a fully nonlinear frictionless contact problem, typically using the finite-element method. Subsequently, a (linear) influence coefficient matrix is obtained from off-line finite-element computations, e.g., employing a nodal perturbation technique, and this matrix is used in the EHL solver [17,18]. In other words, in this approach, the nonlinear behaviour of the deforming solid is linearized at the deformed state determined by solving the contact problem. As a drawback, the friction stresses are neglected in this approach, as they are not known a priori and thus cannot be included in the contact analysis.

A general, fully-coupled nonlinear framework for modelling of soft-EHL problems in the finite deformation regime has been developed in [13,19]. In that approach, deformation of the solid is modelled using the finite element method which allows to consistently treat material nonlinearities and finite configuration changes. The fluid part is also solved using the finite element method. The Reynolds equation is formulated on the deforming contact surface of the solid. As a result, the corresponding domain and its discretization are not known a priori. In particular, the finite element mesh is defined by the deforming mesh of the contact surface of the solid. All the elastohydrodynamic couplings mentioned above are fully accounted for, and the problem is solved simultaneously for all unknowns, i.e., for displacements of the solid, lubricant pressures, and possibly other quantities involved in the model, using the Newton method (monolithic approach). Recently, the model has been combined with the mass-conserving cavitation model, and the formulation has been extended to three-dimensional problems [20].

In the present paper, the general framework [13,19,20] is further developed and is applied to study finite-deformation effects in soft-EHL. Specifically, in Section 3, the formulation is extended to the case of non-planar contact, and the related issue of the choice of the domain on which the Reynolds equation is solved is discussed in detail. Subsequently, in Section 4, the finite-deformation effects are studied for a two-dimensional problem of a rigid cylinder sliding against a coated layer and for a three-dimensional problem of an elastic ball sliding against a rigid plane.

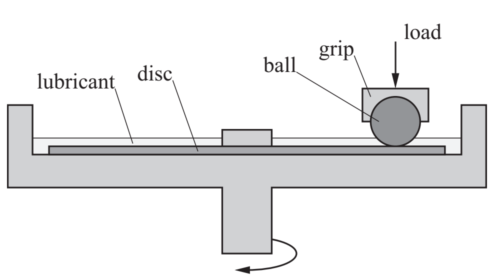

Fig. 1. Schematic of the ball-on-disc tribometer.

The main focus of this work is on the development of computational tools for modelling of soft-EHL problems in the finite deformation regime. However, in parallel, we have developed an experimental test rig suitable for examining pure-sliding lubricated contact of compliant elastomeric balls under high loads. The test rig and sample results are described in Section 2, and the measured friction coefficients are compared to the theoretical predictions.

2. Friction measurements at high contact loads

2.1. Experimental method

Friction measurements were made using a home-made ball-on-disc tribometer shown schematically in Fig. 1. In this tribometer, an elastomeric ball is placed in a grip and is loaded by a normal force against a rotating flat disc. The tribometer has been designed such that testing at relatively high normal loads is possible. The normal load is controlled by attaching a mass (dead load) to an otherwise balanced arm supporting the ball grip.

The disc is clamped to a supporting disc and both are placed in a container. A thin lubricant layer is continuously maintained on the disc surface to ensure proper lubrication conditions. The setup allows testing of steady-state lubrication in pure sliding only, and the sliding speed is adjusted by changing the angular speed of the supporting disc and the radial position of the ball. Friction force is measured by a load cell attached to the ball grip.

Nitrile butadiene rubber (NBR) balls of radius $R=10.7$ mm were used in the present study. Young's modulus was estimated as $E=3.5$ MPa by performing instrumented indentation and by fitting the resulting force–displacement response using the Hertzian contact theory. However, the material exhibits hysteretic effects even at low loading rates, hence the estimated Young's modulus is regarded approximate. A polished low-carbon steel disc was used as a counter surface. Due to a high difference in elastic stiffness, the steel disc can be assumed rigid.

The root-mean-square roughness R_q was measured using the Hommel-Etamic T8000 Nanoscan scanning profilometer: R_q of the steel disc was 0.17 μm and R_q of the rubber balls was 1.30 μm. In the latter case, the roughness was measured on the mould used for producing the balls, as the rubber is too compliant for stylus profilometry.

Distilled water and four silicone oils (Polsil OM 10, OM 50, OM 300 and OM 3000 produced by Silikony Polskie, Poland) were used as the lubricants. Polsil OM fluids are linear, non-reactive and unmodified polydimethylsiloxanes. They differ in their degree of polymerization and consequently in viscosity. The dynamic viscosity η at the test temperature of 25 °C was measured using the Brookfield HADV-III Ultra viscometer with cone/plate configuration and is provided in Table 1.

The disc was driven with a constant angular velocity which resulted in the sliding speed V between 62 and 690 mm/s for the fixed position of the ball with respect to the axis of rotation. The corresponding radius of the sliding path was 42 mm.

Table 1
Viscosity of the lubricants at the test temperature.

Fluid	Viscosity at 25 °C (Pa s)
Distilled water	0.000891
OM 10	0.00942
OM 50	0.0493
OM 300	0.3395
OM 3000	2.735

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