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Influence of flattening of rough surface profiles on the friction behaviour of mixed lubricated contacts

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ABSTRACT

The roughness of tribological surfaces can have a huge influence on the friction behaviour of lubricated contacts. On this account the impact of surface flattening on the friction coefficient is investigated in this work. In terms of energy saving and sustainability, a lower friction coefficient and a shorter run in phase causes mostly decreasing wear and leads to a stable running behaviour of frictional systems. Hence, different journal bearing materials with a different strength are investigated focusing on surface flattening. The whole numerical investigation is modelled with the finite elements software Abaqus (Dassault Simulia) and is taking place at the microscopic level. All investigations are based on the mixed lubrication model from Lorentz [1] (2013).

1. Introduction

Reducing wear and friction in sliding systems is an important element to develop high-efficiency drive trains. Thus, it is possible to reduce the CO_2 emissions and a better economic performance could be reached. For example, the mechanical friction loss in crankshaft bearings in a petrol engine represents about 12–18% of the power dissipation [2]. To reduce these considerable friction losses in powertrain units, it is necessary to investigate and understand complex tribological phenomena at the microscopic scale. Limitations of experimental device require new methods to support the understanding of these complex phenomena. Therefore numerical methods like computational fluid dynamics (CFD) or finite elements method (FEM) were taken into account to increase the knowledge about friction and wear in microscopic contacts.

This paper reports mixed lubrication in journal bearings, which is investigated with a numerical approach based on the FEM. Especially the effect of surface flattening in the run-in phase is taken into account, to determine the influence of elastic and plastic deformation on the friction coefficient. In the model, surface roughness of turning processes and plastic hardening due to deformation of the asperities are considered.

There are similar research works done on this topic which are outlined in the next section as well as different techniques for modelling fluid-structure interactions in rough surfaces. In the third section is given a short summary about the existing approach for modelling mixed lubrication and the alterations in consideration of detecting the ratio of bearing contact area to total area are described. The fourth section shows results about surface flattening for different materials and the influence on the friction coefficient calculated between two rough surfaces, occurred in a drilling process.

2. State of the art

Line contacts in mixed lubrication with rough surfaces were investigated numerically by Chang, [3] with sinusoidal generated profiles. These investigations concerned models with elastohydrodynamic contacts. These theories were then carried on by Mihailidis [4] who calculated the friction coefficient of mixed lubricated line contacts. As well as the model of point contact from Redlich [5], these numerical models were working in a stationary regime. The reason for that is a huge calculation time, necessary to link fluid and structure equations if time integration is chosen.

As a consequence of the huge calculation time, similar approaches were impossible to be used on models at the macroscopic scale. On this account, different solutions were used in parallel to these discretisation methods: robust and very useful analytical approaches were developed for journal bearings [6] based on the Stribeck curve. Other approaches, such as multilevel systems of equations were used by Hu [7], but did not allow the same level of details as conventional CFD or FEM models. Whole preceding investigations were not taking into account thermal exchange neither between fluids and solids nor directly inside of the fluids and solids.

Larsson [8], who modelled surface roughness at the microscopic scale by using the FEM was able to describe real rough

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surface characteristics but did also not take into account thermal heat fluxes. At the macroscopic scale only Jackson [9] took into account thermal effects but this time without modelling any adhesion or roughness characteristics.

On the other hand, parallel to last described investigations, an analysis combining asperity contacts and taking into account thermal effects was achieved by Zhai [10] but without impacting the fluid (no thermal coupling between the fluid and the asperities). This was done by Wiersch [11] who integrated thermal conditions in elastohydrodynamics, meaning coupling fluid structure and thermal behaviour. Knoll [12] and Bartel [13] also developed similar methods to investigate journal bearings, and based on the flow factor theory [14] (correction of the Reynolds equation).

Mixed lubrication contacts phenomenon is to be regarded in transient conditions with a relative displacement of two rough surfaces. In usual EHD models, relative motion between the asperities is not taken into account. If real surfaces are considered, a relative movement may lead to heavy fluid mesh distortions caused by the movement. For this reason, adaptive and remeshing methods were developed [15]. Based on such approaches, Albers et al. [16] analyzed rough surfaces under mixed lubrication conditions, allowing calculating the part of solid and hydrodynamic friction firstly with Arbitrary Lagrangian–Eulerian approaches (ALE). This is only possible when, the fluid flow is not broken, that means, in hydrodynamic conditions with an adequate film thickness in the lubrication gap.

Then, the use of Coupled Eulerian–Lagrangian methods (CEL) enabled it, to overpass fluid mesh distortion limitations [17] by evaluating in a first step the potential of the method in a two dimensional analysis. Further analyses performed in a three dimensional configuration, this time in real mixed lubrication conditions, displayed encouraging results [18]. Also in this context wear is of high interest.

The research field of lubrication of seals [19] and joint protheses [20] use similar models. Taking into account transient conditions, heat generation and real measured rough surfaces, the models need then to be calculated of different machining conditions and directions.

3. Model

The investigations of this work are based on the mixed-lubrication model of Lorentz. The boundary conditions for this model are detailed in Ref. [1]. The tangential adhesion effects derived from the Bowden and Tabor adhesion criterion [21] are assumed valid in this work. Output variables apart from the contact stresses are the distribution of film thickness in the lubrication gap, the global and local friction coefficient and the temperature in the frictional contact, present in mixed lubrication regimes. By means of these output data, highly loaded areas can be detected and conclusions of area with a high wear rate can be assessed. The generation of output variables is based on a Matlab code and is fully automatic. Both contacting solids have the same turned surface and their roughness is 0.704 μm. The length and width of the body is 225 μm and the high is 90 μm. The solid part with the relevant surface is generated with a Matlab script. The surface data was measured with a white-light-interferometer and the waviness of the profile was filtered out. So only the roughness of the surface was taken into account. The solid part with the real roughness is generated by using B-Splines according to the measured surface points. The file format of the solid part is IGES and therefore easy to handle in Abaqus. The generated solid part with the real roughness is displayed in Fig. 1.

The existing model is extended in this work to a force-controlled load application of the upper body against the lower one. In most applications, the load should be applied due to a force or a pressure control. Until now, for better convergence behaviour, most models used a way controlled load application. With a given value of displacement of the upper body, a lubrication gap could be adjusted. As a consequence, if there are many asperities in contact, the normal force increased, because the upper body could not move back. The same if there are fewer asperities in contact, the normal force decreased. Hence, it was impossible to get the model with a constant normal load during the sliding phase.

The setting of the developed approach for the solid and fluid parts of the model is displayed in Figs. 2 and 3. To control leakage of the fluid, the flow rate at the edges of the fluid volume in x- and z-direction is zero (Fig. 3). These boundary conditions are only valid for the fluid. If the upper rough body is pressed against the lower one, in times of contact of the asperities, there a vibrations due to the impact of the body. In this case, an absorbability is necessary, to reduce these vibration. For comparison, in Fig. 4 the normal force profiles of a force of 10 N of a damped and undamped loading are shown.

Clearly, the vibrations, represented by the oscillating trend of the undamped model are visible. The damping was realized by four dashpots, connecting the upper body with a damping plate. The damping plate is spatially fixed and is defined as a rigid body with an unlimited stiffness.

The existing model was also extended to the constitutive model by Johnson–Cook [22], with the material responses strain hardening, strain-rate effects and thermal softening, to describe the hardening effects in the boundary layer, caused by the plastic deformations in the frictional contact. The model is expressed as

$$\sigma = \left[A + B\varepsilon^n\right]\left[1 + C\ln\dot{\varepsilon}^*\right]\left[1 - T^{*m}\right] \quad (1)$$

where ε is the equivalent plastic strain, $\dot{\varepsilon}^* = \dot{\varepsilon}/\dot{\varepsilon}_0$ is the dimensionless plastic strain rate for $\dot{\varepsilon}_0 = 1.0\ \mathrm{s}^{-1}$, and T^* is the homologous temperature. A, B, n, C, and m are the five material constants. The values of the constants are not widely available. They are only known

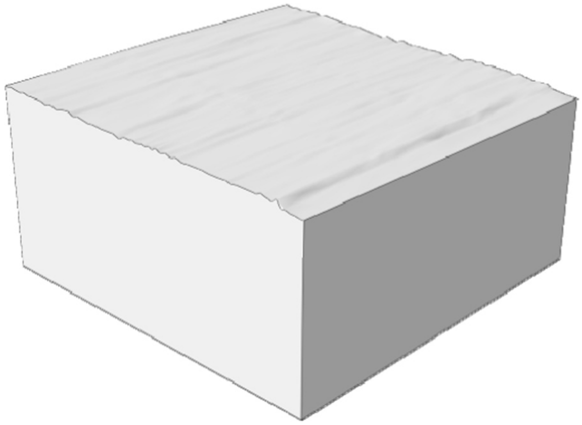

Fig. 1. Solid with a real turned surface.

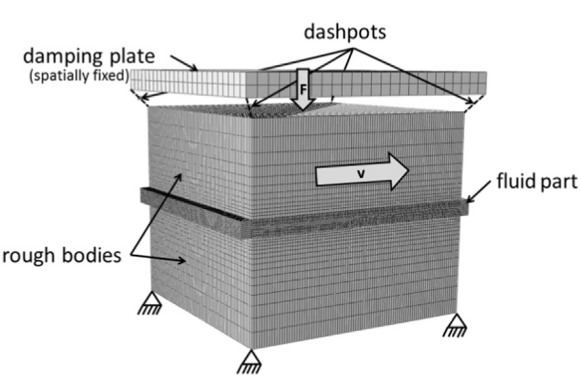

Fig. 2. Updated mixed lubrication model with applied boundary conditions.