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3D thermohydrodynamic analysis of a textured slider

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Analysis of a 3D inlet textured slider bearing with a temperature dependent fluid is performed. Numerical simulations are carried out for a laminar and steady flow. Hot and cold lubricant mixing in the groove is modelled and examined for different operating conditions. Thermohydrodynamic performance of the bearing is analyzed for different texture lengths.

Results show that texture has a stronger and positive influence on load carrying capacity when thermal effects are considered. This beneficial effect is at a maximum for the longest dimples with a length shorter than the pad length. Texture is also beneficial for the load carrying capacity when the sliding speed and inlet flow rate are varied. The load carrying capacity of the slider can be increased by up to 16% in severe operating conditions (high sliding speed).

1. Introduction

Tribological contacts in slider and journal bearings are the most investigated geometries when hydrodynamic lubrication problems are modelled. Many aspects have been analyzed under the past few years in order to improve either the performance or the modelling of these contacts. In such fluid films, the validity of Reynolds equation is still discussed when some discontinuities in the geometry occur. Inertia effects have to be taken into account in some cases whereas they can be skipped for some other cases [1,2]. Many cavitation models have also been introduced and compared since they may predict differently the film rupture and reformation [3,4]. The energy equation and its boundary conditions have also been investigated. Texturing one of the surfaces is a recent advance that has been considered. This has shown a significant improvement in performance in certain cases e.g. increasing load carrying capacity and minimum film or reducing friction. Tonder [5,6] pointed out that introducing a series of dimples or roughness at the inlet of a sliding surface contact can generate extra pressure and thus support higher load. Brizmer et al. [7] strengthened this idea as they showed that a partial texturing creates a collective dimple effect and generates substantial load carrying capacity useful for finite and long sliders. Dimple dimensions were investigated by Ronen et al. [8] who showed that an optimum value of the "pore" depth over diameter ratio gives a minimum value for friction force. Yu and Sadeghi [9] also studied the effect of grooves on load support for a thrust washer with focus on groove depth, width, shape, and quantity. An optimum groove depth providing maximum load support as well as an optimum number of grooves was found. Siripuram and Stephens [10], analysing a single cell dimple, found that the asperity size that minimizes the friction coefficient is dependent on the shape and orientation for the case of a recessed asperity. The depth and position of a pocket in a pad bearing were also found to be of importance for reduction in the friction coefficient, Brajdic-Mitidieri et al. [11]. For another kind of texture such as an heterogeneous slip/no slip surface, the load support can also be increased in a slider bearing as it decreases the resistance to flow [12].

Experimental analysis of lubricated contacts with a micro-textured surface have also shown positive effects [13,14]. Snegovskii and Arnautova [13] reported that dimples machined on the shaft surface allowed a significant increase in load carrying capacity for a journal bearing operating at high speeds. This work was then continued with tests of the same bearing in the low speed range [14]. According to the authors, frictional losses were reduced by 10–15% with a dimpled shaft whereas the load capacity was improved by a factor of 1.5–2 at high sliding speeds.

Most of the theoretical studies were performed for an isothermal flow. Nevertheless, an energy equation has been included in some models in order to study the influence of thermal effects [2,15–18]. Fillon and Bouyer [15], analysing a wear defect in a journal bearing, showed an advantage for such a geometry. For high speed bearings submitted to a light load the wear defect is responsible for an improvement in thermohydrodynamic (THD) performance by reducing the maximum temperature. Thermal effects were studied by Yu and Sadeghi [16] regarding the performance of thrust washers. According to the

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authors, thermal effects do not reduce only load and friction, but they also increase the side leakage. Thermal effects have greater influence on the load when the groove depth and number increase. Kucinschi et al. [17] also analyzed a 3D thrust washer with a thermoelastohydrodynamic (TEHD) model. The effects of the particular shape of the stator on the pressure and temperature fields were analyzed. Dobrica and Fillon [2] used a THD model to treat discontinuous domains such as Rayleigh step bearings. The authors conclude that the Navier–Stokes based model is more accurate for all the configurations studied (film thickness increased, velocity varied), but is unusable in 3D studies since it has slow convergence rate. Later on, the same authors studied a 3D slider pocket bearing and determined the optimal size of the pocket regarding the load carrying capacity by taking into account the thermal effects [18]. The study also concludes that similar load and maximal temperature is obtained for a pocket slider and an equivalent inclined slider.

All these studies including thermal effects often assumed an imposed temperature at the inlet and ambient pressure on the side for a slider bearing. This assumption is generally made for slider bearings rather than journal bearings as the domain is generally modelled without periodicity and without inlet lubricant: the slider is taken away from its environment [19].

This paper investigates a 3D thermal flow in a textured slider bearing. Assumptions imposed at the boundary conditions are avoided since the geometry of the slider bearing is extended with some side channels and a fore-region which is supposed to give more realistic conditions for the temperature and pressure fields. Incoming lubricant is supplied perpendicular to the sliding direction in the fore-region (or inlet groove), therefore the Navier–Stokes equations are used since the thin film hypothesis is not fulfilled. The energy equation is used in order to model the mixing of the flow in this region and to estimate heat generation in the contact. The lubricant mixing is first described under different operating conditions then smooth and textured sliders are analyzed and compared.

2. Numerical model

2.1. Equations

The flow is modelled with the commercial code ANSYS CFX 11.0. The Navier–Stokes equations, momentum Eq. (1) coupled with the continuity Eq. (2), are solved over the domain together with the energy Eq. (3), using the finite volume method. The flow is considered laminar, steady and 3D.

$$\frac{\partial}{\partial x_j}(\rho u_i u_j) = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j}\left(\mu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right)\right) \tag{1}$$

$$\frac{\partial}{\partial x_i}(\rho u_i) = 0 \tag{2}$$

$$\frac{\partial}{\partial x_i}(\rho u_i h_{tot}) = \frac{\partial}{\partial x_i}\left(\lambda\frac{\partial T}{\partial x_i}\right) + \frac{\partial u_i}{\partial x_j}\left[\mu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right) - \delta_{ij}\left(\frac{2}{3}\mu\frac{\partial u_k}{\partial x_k}\right)\right] \tag{3}$$

where $\delta_{ij} = 1$ when $i = j$, and 0 when $i \neq j$. The total enthalpy is calculated by the following expression:

$$h_{tot} = h_{stat} + \tfrac{1}{2}V^2 \tag{4}$$

with $\frac{1}{2}V^2$ representing the kinetic energy. The static enthalpy is calculated by integrating the following expression:

$$dh = C_p\,dT + \frac{1}{\rho}\left[1 + \frac{T}{\rho}\left(\frac{\partial \rho}{\partial T}\right)_P\right]dP \tag{5}$$

2.2. Boundary conditions

The geometry used is that of a 3D slider bearing with fore-region and extended channels at the outlet and on the sides of the pad. The smooth geometry is considered together with a textured one having three rectangular dimples located in the inlet region, Fig. 1.

The following dimensions are used for the smooth bearing: $L_x = 6$ mm (length of the pad in the sliding direction), $L_z = 6$ mm (length of the pad in the direction perpendicular to the sliding direction), $h_0 = 0.03$ mm (height of the film at the outlet of the pad) and h_1 (height of the film at the inlet of the pad). The dimensions of the fore-region, as a percentage of the pad length in the sliding direction, lie in the range proposed by Zhang and Rodkiewicz [19]: $L_G = 0.28$, $L_x = 1.68$ mm (length of the fore-region) and $d_G = 0.16$, $L_x = 0.96$ mm (depth of the fore-region). As some backflow may occur at the outlet, an extended channel is created in order to avoid a recirculating flow at the boundary. Its length is $L_{out} = 4$ mm. Some side channels of width $L_c = 1$ mm are used in order to create more realistic conditions. The assumption of zero pressure on the side of the pad is discarded. The

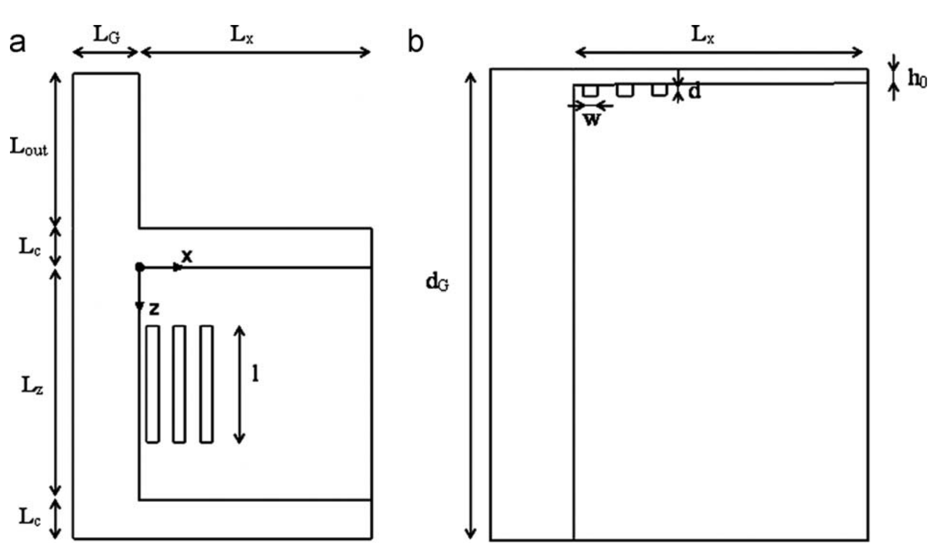

Fig. 1. Computational domain. (a) *XZ* view and (b) *XY* view (scaled by 10 in the *y*-direction).