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Enhanced performance of optimised partially textured load bearing surfaces

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ABSTRACT

Textured surfaces have been shown to provide enhanced tribological performance in a variety of contacts. Numerical analysis and optimisation methods are combined for application-oriented texture optimisation. However, an analytical approach is advantageous in providing more generic in-depth understanding of the nature of the relationships between texture parameters and objective functions, such as enhanced load carrying capacity and reduced friction. The paper outlines such an approach to obtain a set of global optimum design parameters for partially textured surfaces. The optimised results are expressed in dimensionless form, which enables their use for a variety of applications. The performance of optimised partially textured sliding surfaces is compared with the other conventional bearing geometries in their optimum state.

1. Introduction

Surface textures introduced on contacting surfaces act as micro-reservoirs of lubricant under operating conditions which do not otherwise yield a coherent film of lubricant. Such conditions include, inter alia, poor contact kinematics as in stop-start and reciprocating motions, high contact loads and poor conjunctional geometrical profiles. Surface textures create micro-wedges, acting as micro-bearings on their own accord. There has been an increasing volume of research on tribological performance of textured surfaces since the late 1990's [1,2] due to the advances made in their relatively cost-effective manufacture. These include laser surface texturing (LST) [3–7], which has attracted much interest. Owing to the diversity of load bearing conjunctions and operating conditions, there has been a plethora of reported research focused on particular applications such as journal bearings [8–11], thrust bearings [12,13], piston ring and cylinder liners [4–6,14,15], mechanical seals [1,2,16] and rolling element bearings [17,18] among other cases. There have also been studies on optimisation of the textured surfaces for specific applications [19]. However, due to a relatively large number of texture design parameters and operational conditions, a unified set of guidelines for the optimisation process for any given application has not hitherto emerged despite several in-depth contributions [20–23].

Rahmani et al. [24,25] proposed an analytical approach for investigation of tribological performance of partially textured sliding surfaces under hydrodynamic regime of lubrication. The objectives were to maximise hydrodynamic load carrying capacity and minimise viscous friction or the required flow rate. Their results provided a set of local optima, where for a given texture length ratio an optimum set of textured length and depth ratios could be specified for any given number of texture features. Alternatively, optimum values of texture length and depth ratios could be determined for a given number of textures at various texture length ratios. These characteristic ratios are defined in Section 2.

The advantage of the analytical approach is in providing an in-depth understanding of the relationships between the texture design parameters and stated measures of performance. Therefore, the underlying physical relationships can be readily obtained through an analytical approach. In addition, analytical solutions provide time-efficient solutions when compared with the numerical computations, using iterative techniques. The inherent error in the numerical method is also a factor that originates from its approximate nature. The advantage of numerical approach is dealing with complex geometries or those caused by any thermo-elastic effects. Furthermore, more complex texture shapes or patterns may be studied using numerical approaches.

The current study builds upon the analytical approach in Ref. [25] to provide a global set of optimum parameters, which would maximise the attainable load carrying capacity. It also provides a comparison between the performance of textured surfaces with other commonly used slider geometries, all with their optimum design; wherever analytically feasible.

2. Geometrical features of a textured contact

Fig. 1 shows a two-dimensional schematic representation of a textured sliding surface. The parameters used to characterise a textured

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surface are: (i)- the number of textures, (ii)- the location of the textured area within the overall contact, (iii)- geometrical properties of texture features including their cross-sectional profile, maximum depth and width, and (iv)- the axial interspatial spacing between successive textures (pitch).

Using the notations in Fig. 1, the following dimensionless parameters are defined [26,27]:

- Texture depth ratio: This is the ratio of the maximum to the minimum gap between the counter face surfaces.

$$\xi = \frac{h_m + h_d}{h_m} = 1 + \frac{h_d}{h_m} \tag{1}$$

- Leading and trailing edge ratios: These are the ratios of the leading and the trailing edges of the textured region to the overall bearing axial length:

$$\alpha_0 = \frac{l_0}{L},\ \alpha_n = \frac{l_n}{L} \tag{2}$$

- Texture length ratio: This is the ratio of the texture length at the base to the interspatial space between successive texture features:

$$\varepsilon = \frac{d_x}{l_x} \tag{3}$$

- Textured length ratio: This is the ratio of the length of textured area to the overall bearing axial length:

$$\kappa = 1 - (\alpha_0 + \alpha_n) \tag{4}$$

3. Governing relationships

Reynolds equation is commonly used to model lubricated conjunctions comprising textured surface(s). To model textured features of finite lateral width (e.g. pores or chevrons), a two-dimensional form of Reynolds equation should be used, along with numerical solution techniques such as finite difference methods in order to obtain the generated pressure distribution [26,28]. For cases where the lateral texture widths may be considered large in comparison with their lengths, use of one-dimensional Reynolds equation is justified as in the case of infinite-width micro-bearings. Consequently, explicit analytical relationships for load carrying capacity and viscous friction can be derived and used to calculate the tribological performance of the given textured contact. As a result, a thorough parametric study can be conducted, through which the optimum geometrical parameters for the texture features may be determined. In such an approach, the extent to which any performance criterion may be influenced by any chosen design parameter can be ascertained. This enables the designers to identify design features with a greater degree of confidence, as well as identify the underlining physical principles.

The analytical methodology and associated boundary conditions are specified in Rahmani et al. [25–27] in some detail. The current analysis assumes an incompressible lubricant under iso-viscous condition. In addition, it is assumed that there is negligible thermal gradient in the contact domain. This is generally the case in many applications including for piston rings and cylinder liner conjunction [29], big-end bearings [30] and ball bearings [31]. Therefore, an average effective contact temperature may be assumed. The dimensionless relationships provided in Refs. [25–27] are independent of the lubricant viscosity. Therefore, when converting the results into dimensional parameters such as load carrying capacity and friction an adjusted lubricant viscosity for the contact operating temperature can be used. Furthermore, the current study is a comparative study of texture parameters for contact conjunctions which are assumed to be subjected to the same operating conditions in each studied case, with the differences accounting for the various texture parameters.

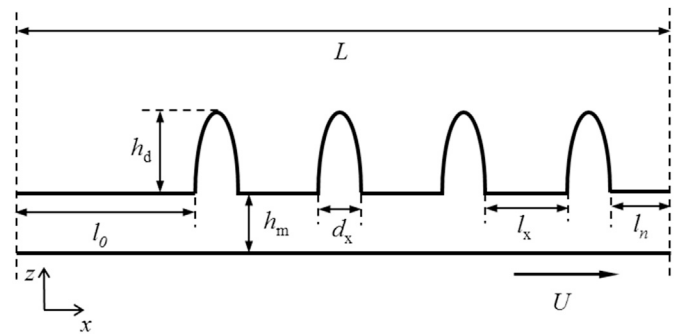

Fig. 1. Schematic of a textured contact subject to sliding motion and corresponding geometrical parameters.

The analytical relationships provided in Ref. [25] for textures of rectangular and isosceles triangular profiles are used in the current study. For clarity, these relationships are restated here.

The dimensionless performance parameters in terms of load carrying capacity and friction are defined as:

$$\Lambda_w = W\frac{1}{\eta U}\left(\frac{h_m}{L}\right)^2 \tag{5}$$

and

$$\Lambda_f = f\frac{1}{\eta U}\left(\frac{h_m}{L}\right) \tag{6}$$

where, W and f are load carrying capacity and friction per unit lateral width of the contact. In addition, η and U represent lubricant dynamic viscosity and sliding speed respectively, while h_m and L denote the minimum film thickness and the overall contact length (in the axial direction).

The ratio of dimensionless friction to dimensionless load carrying capacity is indicative of the coefficient of friction when it is purely based on texture geometrical parameters:

$$\Omega = \frac{\Lambda_f}{\Lambda_w} = \frac{f}{W}\frac{L}{h_m} = \mu\left(\frac{L}{h_m}\right) \tag{7}$$

As it can be seen Ω is a scaled-up representation of the coefficient of friction, μ, by the ratio of contact length over the minimum film thickness. Hereinafter, Ω is referred to as scaled coefficient of friction and is used in the comparison of numerical results of various bearing configurations later in Section 4.

In the case of rectangular shape texture features the dimensionless load carrying capacity and friction become:

$$\Lambda_w = \frac{3m\varepsilon\kappa(1-\kappa)(\xi-1)}{(m+\xi-1)\xi^3 - m\varepsilon\kappa\left(\xi^3-1\right)} \tag{8}$$

$$\Lambda_f = \frac{m^2\varepsilon^2\kappa^2(\xi-1)^4 + \xi(m+\varepsilon-1)\left[(m+\xi-1)\xi^3 - 2m\varepsilon\kappa(\xi-1)\left(\xi^2-\xi+2\right)\right]}{\xi(m+\varepsilon-1)\left[(m+\varepsilon-1)\xi^3 - m\varepsilon\kappa\left(\xi^3-1\right)\right]} \tag{9}$$