Tribology International 98 (2016) 212–219

Contents lists available at ScienceDirect

Tribology International

journal homepage: www.elsevier.com/locate/triboint

Short Communication

# Surface roughness effects in hydrodynamic bearings

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ARTICLE INFO

*Article history:*
Received 18 June 2015
Received in revised form
17 February 2016
Accepted 18 February 2016
Available online 26 February 2016

*Keywords:*
Slider bearing
Hydrodynamic lubrication
Roughness
Fourier transform

ABSTRACT

An analytical solution method for the effects of general surface roughness in hydrodynamic bearings is presented and illustrated with the analysis of wide exponential land slider bearings. The method is based on the solution of Reynolds equation using perturbation techniques under the assumption of small amplitude waviness coupled with linear superposition using Fourier transforms. To verify the analytical solution, full numerical results were obtained using finite differences discretization techniques and multigrid methods for convergence acceleration. Comparisons were made for cases involving surface features given either as a sinusoidal wave or as a single Gaussian dent. A good agreement was found, therefore verifying the accuracy of the analytical scheme. The applicability, limitations and possible extensions to the methodology are also discussed.



## 1. Introduction

One of the first methods proposed to quantify the effect of surface roughness in hydrodynamic lubrication is the Stochastic approach of Tzeng and Saibel [1,2], where roughness is treated as a random variable characterized by an experimentally determined probability density function. This concept proved to be very popular and was adopted and extended by several authors to investigate for instance the effects of transverse and longitudinal roughness [3], directional or isotropic roughness with average flow factors [4–6] and general lubricant flow interactions caused by two-dimensional roughness [7].

Despite their popularity, the main drawback of stochastic methods is that they do not account for specific features of the roughness. Therefore, deterministic studies have long been pursued. These, however, have been mostly confined to the analysis of waviness due to the inherent mathematical complexity to simulate a general surface roughness. Burton [8] performed a steady-state asymptotic analysis of the effects of two-dimensional waviness on parallel plates, highlighting possible conditions for film breakdown. Dowson and Whomes [9] considered an axial waveform on a rigid cylinder to represent turned and ground surfaces, and found friction coefficient reductions when compared to smooth surfaces. Hargreaves [10] investigated the effect of a stationary sine wave on a rectangular slider bearing and found that the load-carrying capacity can be enhanced depending on the amplitude and frequency of the waviness pattern. Lin [11] included three-dimensional waviness in a journal bearing under steady-state conditions. An increase in side-leakage flow and a decrease of friction coefficient were found by increasing asperity height. Tønder [12] analyzed a sinusoidal wave sliding over a smooth inclined pad and found periodical variations in load-carrying capacity only for wavelengths longer than the bearing length. With very short wavelengths, the load carrying capacity appeared to be unaffected compared to smooth surfaces.

The above described studies have consistently found that the effects of surface roughness can in some situations be beneficial to the lubrication conditions. This has led researchers to investigate the effects of adding carefully controlled patterns of micro-features onto the bearing surfaces [13–16]. The patterns can be added to the entire load bearing area looking to exploit the so-called dimple effect of local cavitation [17,18]. Alternatively, partial texturing of the surfaces can be used [19,20]. Here the improvement in lubrication conditions is attributed to a collective effect of individual dimples, which can be translated as an equivalent average converging shape between the mating surfaces as in a step bearing.

Despite the progress made in this field, deterministic studies involving roughness have been mostly limited to well-defined geometrical features such as dimples, squares, and waviness. There is no general methodology available for deterministic analysis of real roughness, which still remains a challenging topic even if choosing to employ advanced numerical solution methods. For instance, the discrete representation of the roughness requires a large number of numerical grid points, thus leading to long solution times already when the roughness is considered stationary.

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http://dx.doi.org/10.1016/j.triboint.2016.02.027
0301-679X/

**Notation**

| | |
|---|---|
| $f$ | load carrying capacity, N |
| $H$ | dimensionless film thickness |
| $h$ | lubricant film thickness, m |
| $H^*$ | dimensionless integration constant |
| $h_0$ | lubricant film thickness with smooth surfaces, m |
| $h_e$ | inlet film thickness, m |
| $H_m$ | dimensionless smooth minimum film thickness |
| $h_m$ | minimum film thickness, m |
| $i$ | complex number |
| $K$ | kurtosis of the surface roughness |
| $l$ | bearing length, m |
| $P$ | dimensionless lubricant pressure |
| $p$ | lubricant pressure, Pa |
| $p_0$ | lubricant pressure with smooth surfaces, Pa |
| $R$ | general roughness profile |
| $RMS$ | root mean square of the surface roughness |
| $Sk$ | skewness of the surface roughness |
| $T$ | dimensionless time |
| $t$ | time, s |
| $u_1$ | lower surface velocity, m s$^{-1}$ |
| $u_2$ | upper surface velocity, m s$^{-1}$ |
| $u_e$ | lubricant entrainment velocity, m s$^{-1}$ |
| $X$ | dimensionless coordinate |
| $x$ | coordinate, m |
| $X_0$ | dimensionless start position of surface features ($T=0$) |
| $x_e$ | length of exponential shape, m |
| $X_p$ | dimensionless position of surface features |
| $Z$ | dimensionless coordinate |
| $z$ | coordinate, m |
| $Z_1$ | dimensionless surface waviness |
| $z_1$ | surface waviness, m |

*Greek letters*

| | |
|---|---|
| $\alpha$ | dimensionless length of exponential shape |
| $\delta F$ | net variation in load due to surface roughness |
| $\Delta$ | dimensionless inlet film thickness |
| $\eta$ | lubricant viscosity at ambient pressure, Pa s |
| $\Lambda$ | dimensionless wavelength |
| $\lambda_x$ | wavelength on the $x$-axis, m |
| $\lambda_y$ | wavelength on the $y$-axis, m |
| $\omega$ | wavenumber on the $x$-axis, m |
| $\Psi$ | dimensionless wavenumber |
| $\xi$ | wavenumber on the $y$-axis, m |

Time dependent solutions can thus become prohibitive. Looking to help overcome these challenges and taking inspiration on recent work by Hooke et al. to analyze roughness effects in elastohydrodynamic lubrication (EHL) contacts [21,22], this paper proposes for the first time, to the authors' knowledge, a fast solution method based on perturbation theory and Fourier transforms to study the effects of general surface roughness in hydrodynamic bearings.

### 1.1. Objective of the present work

The aim of this paper is to introduce a general deterministic solution method for the effects of low-amplitude surface roughness in hydrodynamic bearings. Perturbation theory is adopted for the method derivation, therefore focusing on the local or micro-scale effects associated to the roughness. Phenomena such as interasperity cavitation and changes in the macro-geometry of the surfaces are assumed to either be negligible or not to be present by limiting the analysis to low-amplitude roughness. Furthermore, due to the low contact pressures typical of hydrodynamic bearings, the lubricant is considered as Newtonian, incompressible and isoviscous. Though it must be noted that these simplifications can be relaxed if desired.

The assumptions and considerations in the method derivation, such as the inclusion of smooth gap and pressure gradients that are neglected in EHL counterparts are discussed in detail together with possible accuracy limitations. The applicability of the method is then illustrated with the analysis of wide exponential land slider bearings, whereupon a practical example is given by using the method to rank the effects of different surface roughnesses.

## 2. Single wave analysis

In order to solve cases involving general surface roughness, it is necessary to first understand the effects introduced by surface roughness in the form of waviness. For that effect, consider for instance Fig. 1 for a general hydrodynamic contact geometry with known smooth surfaces film thickness ($h_0$) and pressure ($p_0$), to which a two-dimensional waviness is superposed, without loss of generality, on the top surface. Hydrodynamic bearings typically involve relatively low contact pressures such that deformations of the contact geometry and changes in lubricant density and viscosity are negligible. Therefore, assuming a Newtonian, isoviscous and incompressible fluid with negligible inertia and thin film conditions, Reynolds equation becomes:

$$\frac{\partial}{\partial x}\left(\frac{h^3}{12\eta}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial y}\left(\frac{h^3}{12\eta}\frac{\partial p}{\partial y}\right)=u_e\frac{\partial h}{\partial x}+\frac{\partial h}{\partial t} \quad (1)$$

with the lubricant film thickness given by the following expression:

$$h(x,y,t)=h_o(x,y)+z_1(x,y,t) \quad (2)$$

and the waviness by:

$$z_1(x,y,t)=h_1\exp\left[i(\omega x+\xi y-\omega u_2 t)\right] \quad (3)$$

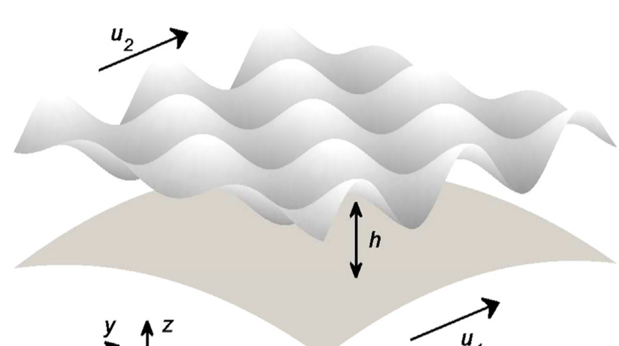


**Fig. 1.** General contact configuration for a hydrodynamic bearing with superposed waviness.