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Calculation of Stribeck curves for (water) lubricated journal bearings

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Abstract

This paper describes a mixed elastohydrodynamic lubrication (EHL) model for finite length elastic journal bearings. The finite element method was employed to discretise the coupled system of 2D–3D Reynolds-structure equations and to compute Stribeck curves at constant load. As underrelaxation strategies have been found to be insufficient for an iterative solution of this problem, artificial dynamics have been added to the numerical structure equations in order to solve for stationary solutions of the fluid–structure problem. An ideal plastic asperity contact model together with an effective film thickness formulation according to Chengwei and Linqing was employed in order to compute the contact pressure in mixed lubrication. The method presented in this paper is applied to a typical water lubricated journal bearing problem. Computed Stribeck curves are presented and the numerical performance of the method is evaluated.

Keywords: Mixed lubrication; Soft-EHL; EHL; Journal bearing; Stribeck curve calculation; Fluid–structure interaction

1. Introduction

Many elastohydrodynamic (EHL) journal bearing models and algorithms solving the coupled system of equations arising from the interaction between the lubricating fluid and the deformation of the bearing have been published [1–5]. Early EHL journal bearing models, such as presented by Higginson [1], apply a 1D *thin layer* elastic deformation model in order to calculate the bearing deflection and show how the load capacity for a given eccentricity ratio is affected by the bearing flexibility. Oh and Huebner [2] developed a 2D–3D finite element approach to the EHL journal bearing problem and employed a staggered iterative algorithm to solve for the fluid–structure equilibrium. It was concluded in their work that an elastic bearing is certainly not inferior compared to the rigid bearing, but has the ability to distribute the load over a larger bearing surface area and that for the same minimum film thickness and peak pressure a higher load capacity can be obtained. With respect to their numerical solution method, they remarked that it did not converge when the bearing deformation was of the same order of magnitude as the film height. Even with various underrelaxation strategies, convergence of their solution was not ensured.

Potential asperity contact at high-eccentricity ratios is left out of consideration in these early publications. More recently, the paper of Wang et al. [5] incorporates asperity contact by the elastic–plastic asperity contact model of Lee and Ren in order to study mixed lubrication (ML) phenomena. Three major factors affecting lubrication performance were studied: elastic deformation of the bearing, surface roughness effect on lubrication and asperity contact pressure. Bearing deflection was evaluated by reduction of the full 3D FEM stiffness matrix into a 2D flexibility matrix. The average flow model of Patir and Cheng (P&C) was employed to account for roughness effects on lubrication.

A most useful tool for the design of journal bearings is the Stribeck curve as it clearly points out the critical journal velocity at which the transition from EHL to ML takes place or at which velocity a certain acceptable coefficient of friction is exceeded. For soft-EHL problems, such as arising from polymer bearings, it became clear that

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Nomenclature

c	radial clearance (m)
C	damping matrix (N s/m)
D	bearing diameter (m)
e	eccentricity (m)
E	Young's modulus (Pa)
f	bearing coefficient of friction (–)
f_c	coefficient of friction in dry contact (–)
F_c	nodal contact force (N)
F_f	nodal fluid force (N)
h	nominal film thickness (m)
h_t	effective film thickness (m)
$\bar{h}_T$	mean film thickness (m)
H	hardness (Pa)
K	stiffness matrix (N/m)
L	bearing length (m)
W	bearing load (N)
W_t	target load (N)
p	total pressure (Pa)
p_f	hydrodynamic pressure (Pa)
p_c	contact pressure (Pa)
R	journal radius (m)
S	scaling factor (–)
S_q	surface roughness parameter (m)
t	bearing thickness (m)
$\mathbf{u}$	vector with nodal displacements (m)
$\dot{\mathbf{u}}$	discrete time derivative of $\mathbf{u}$ (m/s)
U, U_1, U_2	surface velocity (m/s)
x, y, z	local cartesian coordinate system (m)
X, Y, Z	global cartesian coordinate system (m)
α	numerical damping coefficient (s)
Δt	time step (s)
η	dynamic viscosity (Pa s)
v	Poisson ratio (–)
ω	journal frequency (rpm)
ϕ	circumferential coordinate (rad)
$\phi_{px}, \phi'_{px}, \phi_{py}, \phi'_{py}$	pressure flow factor (–)
ϕ_s, ϕ'_s	shear flow factor (–)
ρ	fluid density (kg/m^3)
σ_0	projected bearing pressure (Pa)
θ	load angle (°)

solving the coupled fluid–structure equations is not straightforward [2]. The papers cited above provide useful insight into ML phenomena but cannot be used as a tool for journal bearing design optimisation. In this paper, we report about a computational method solving the mixed soft-EHL problem which can be used in optimisation of the journal bearing design.

2. Problem formulation and equations

In ideal smooth gaps that have small heights and height variations and a low Reynolds number, the flow is dominated by viscous forces and inertia effects can be neglected. The flow in such a gap is a combination of Couette and Poisseuille flow and can be described by the Reynolds equation

$$\frac{\partial}{\partial x}\left(-\frac{\rho h^3}{12\eta}\frac{\partial p_f}{\partial x}+\frac{(U_1+U_2)\rho h}{2}\right)+\frac{\partial}{\partial y}\left(-\frac{\rho h^3}{12\eta}\frac{\partial p_f}{\partial y}\right)=0, \tag{1}$$

with h the film thickness and ρ the fluid density. Since the magnitude of the pressure in a conformal contact as encountered in journal bearings remains limited, both the fluid density and viscosity η are assumed constant throughout this paper. The surface velocities are denoted by U_1 and U_2, respectively. Furthermore, p_f represents the hydrodynamic or fluid pressure. Real surfaces are not ideal smooth and the film thickness is generally given by

$$h_T = h + \delta_1 + \delta_2, \tag{2}$$

where h is the *compliance* or nominal film thickness and δ_1 and δ_2 denote the roughness amplitudes of the surfaces. With σ_1 and σ_2 the standard deviations of δ_1 and δ_2, respectively, the composite rms surface roughness—or S_q value in engineering terms—for a pair of rough surfaces is

$$\sigma = S_q = \sqrt{\sigma_1^2+\sigma_2^2}. \tag{3}$$

Embracing all surface details in a deterministic manner was—and still is—not feasible from a numerical point of view in a coupled 2D–3D EHL problem. Hence, an average rough Reynolds equation was derived by Patir and Cheng [6] correcting the Reynolds equation for the film height variations resulting from a randomly distributed surface profile:

$$\frac{\partial}{\partial x}\left(-\phi'_{px}\frac{h^3}{12\eta}\frac{\partial p_f}{\partial x}+\frac{(U_1+U_2)\bar{h}_T}{2}+\frac{\phi'_s(U_1-U_2)S_q}{2}\right)+\frac{\partial}{\partial y}\left(-\phi'_{py}\frac{h^3}{12\eta}\frac{\partial p_f}{\partial y}\right)=0, \tag{4}$$

with h the nominal film thickness and $\bar{h}_T$ the mean film thickness. For a Gaussian distributed surface roughness, leaving roughness deformation out of consideration, $\bar{h}_T$ is equal to h. The correction factors ϕ_{px} and ϕ_{py} are pressure flow factors and ϕ_s is a shear flow factor, correcting for the fluid transport by the roughness valleys. The shear flow factor, however, is equal to zero if the surfaces have the same S_q roughness [6], as is assumed throughout this paper.

Roughness deformation can be the result of local hydrodynamic pressure build up or it can result from contact between the surfaces. Due to piezoviscous effects, roughness deformation due to local hydrodynamic pressure build up plays an important role in concentrated EHL contacts—such as that occur in ball bearings—, but is of practically no importance in a journal bearing system. In