A REYNOLDS-TYPE EQUATION FOR BEARINGS WITH STOCHASTIC ROUGHNESS UNDER ELASTOHYDRODYNAMIC LUBRICATION REGIME

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ABSTRACT
The concept of elastohydrodynamic lubrication (EHL) refers to lubrication scenarios where the deformation of surfaces under pressure significantly influences the thickness of the lubricating film. At nominal pressures ranging from 2 to 7 MPa, the elastic deformation of lubricated surfaces holds substantial importance, directly affecting the performance of bearings [1]. The presence of such pressures cannot be disregarded since the corresponding elastic deformations directly impact the performance of the bearings.

Surface roughness in lubrication phenomena of sliding surfaces has long been a subject of intensive studies since it significantly affects the performance and operational parameters of the lubricated contacts. In order to account for surface roughness, a modification of the film thickness is performed; the nominal film thickness h is augmented by the geometry of the stator's and rotor's surface deviations from smooth surface, r_1 and r_2 respectively. The effect of surface roughness on EHL, and lubrication phenomena in general, has been treated by two approaches, namely the deterministic and the stochastic, can be followed. The deterministic approach is highly accurate, but case-specific; representing the roughness profile geometry calculates the parameter values for the specific roughness. A different profile geometry would result in different pressure generation, thus different parameters. On the other hand, the stochastic approach formulates the roughness via its statistical properties. The stochastic representation of the surface roughness relies on its average m_r and standard deviation σ_r. Based on these moments, averaged Reynolds-type equations are derived, which describe the mean values of the operational parameters of the bearing

Recently, a Reynolds-type Equation for hydrodynamically lubricated bearings has been derived by the authors [2] and utilized to calculate the average values of a bearing's operational parameters given by Eq. (1):

$$\nabla\left(\begin{pmatrix} h^3 - 3\sigma^2 g^2(t)h - 3\sigma^2 g^2(t)h^2F_0 \\ +3\sigma^4 g^4(t)F_0 \end{pmatrix}\nabla m_p\right) = 6U\eta\frac{\partial h}{\partial x} \qquad \textbf{(1)}$$

In the present work we aim to extend Eq. (1), in order to account for elastic deformations of the bearing surfaces. Following this, equations for the calculation of the principal

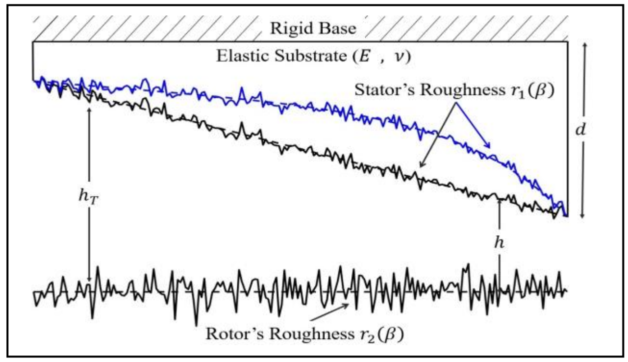

Fig.1: Film geometry in rough surfaces (black and blue lines represent the non-deformable and deformable bearing bush)

operational parameters, i.e. the load carrying capacity W, the friction force F, and the friction coefficient μ, are also derived.

The elastic deformations of the bearing surfaces can be generally calculated by an equation of the form of Eq. (2):

$$h_E = f(p, d, E, k) \qquad \textbf{(2)}$$

where f is a function of the generated pressure p, the depth (thickness) of the deformable bearing bush d, as well as of the elastic modulus E and parameter k of the bush material, with parameter k being dependent on the Poisson ratio.

The mathematical formulation of the new equation yields the original Reynolds Equation for smooth surfaces under elastohydrodynamic lubrication, by setting the roughness's standard deviation equal to zero ($\sigma \to 0$), and the Reynolds-type Equation (1) by setting the elastic deformations equal to zero (assuming $E \to +\infty$).

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