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Predicting friction reliably and accurately in journal bearings—A systematic validation of simulation results with experimental measurements

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It is the aim of this work to predict friction in journal bearings reliably and accurately under realistic dynamic working conditions. To this purpose elastohydrodynamic (EHD) calculations using an extensive oil-model and including an approach to the conformal roughnesses of the bearing surfaces are carried out for transient loads typical for current utility vehicles (40 MPa) as well as for considerably higher specific loads (70 MPa) and for different lubricants (SAE10, SAE20, SAE30 and SAE40) to account for a large span of working conditions ranging from full film lubrication to mixed lubrication with metal–metal contact.

The results obtained from this simulation model are compared to measurements performed on a journal bearing test rig. We find that the results of the presented approach agree very closely with the experimental values. The presented approach allows consequently to investigate the effectiveness of changes in bearing geometry, bearing materials, bearing surface roughness, lubricant viscosity and engine operating conditions to reduce friction in journal bearings.

1. Introduction

In the automotive and industrial sector, consumer demand and the increasingly tight environmental legislation motivate the development of more efficient internal combustion engines (ICE). One possibility to achieve this goal is to reduce power losses due to mechanical friction. Despite all the efforts and developments in engine design and oil formulation so far to reduce these friction in engines stay an active topic. More specifically, friction losses in journal bearings are generally estimated to cause about 25% of the total mechanical power losses; for a specific automotive ICE estimations for bearing losses reach even about 40% of the total mechanical power losses [1]. The associated potential for energy or fuel savings [2] can only be realized if friction and the accompanied tribology is properly taken into account in early development already on the simulation level. Therefore, the reliable prediction of friction losses plays a crucial role in developing future economic engines. Further, to avoid reliability issues the working conditions (e.g. the peak oil film pressure and minimum oil film thickness) have to be predicted realistically.

While there exist numerous studies in the literature that are concerned with various physical properties of journal bearings under dynamic loads, like e.g. asperity contact pressure, minimum oil film thickness, temperature distributions amongst others using different approaches [3–11] or the influence of various properties of the lubricant [3,12–14] on journal bearings, to the knowledge of the authors this is the first work that systematically validates results obtained from a single simulation model over a large range of working conditions comprising lubrication with different oil-grades and involving a large span of different specific loads in direct comparison to experimental measurements in terms of friction prediction.

In the following a generic simulation model is presented that is able to predict friction accurately and reliably in journal bearings for a large range of working conditions. It can consequently be used as a tool to investigate the potential for measures to reduce friction in real applications.

Therefore, the presented work aims to cover current and to some extent future requirements in engines. The dimensions of the journal bearings studied in this work are typically found in contemporary utility vehicles, while the investigated loads either represent or exceed current common working conditions and target future developments. While in the following different loads and different lubricants are investigated, the influence of different rotational speeds of the journal and the necessity of an extensive oil-model are studied elsewhere [15].

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Concerning friction in bearings, the lubricant plays a crucial role. The engine oil used for lubrication is today a highly sophisticated complex that has to perform many different tasks. In terms of friction, however, there is still no other easily accessible way to reduce the inherent hydrodynamic losses due to viscosity except by reducing the viscosity itself and this is commonly done in industry. The consequently reduced oil film thickness increases, however, the likelihood of asperity contact, therefore, modern oils include a comprehensive additive package to reduce wear and friction in case asperity contact actually occurs. The efficiency of reducing friction in engines using optimal lubricants is documented in various works [16–18,2,19]. Within this work, we strive to account for friction modifying additives by using a low boundary friction coefficient that is evidenced experimentally [20,21].

While the focus in this work are monograde oils as they are used in large stationary engines, the results also apply correspondingly to multigrade oils as monograde oils can be understood as special case of multigrade oils at extreme shear rates where the viscosity improver additives are no longer effective.

Reliably investigating friction and wear in bearings need a sophisticated physical model that describes elastic structure deformation, full film lubrication, modification of the fluid flow due to surface roughness and asperity contact at the same time.

Consequently, the concept of this paper is as follows: in Section 2 a short introduction to the different lubrication regimes is given, Section 2.1 discusses the averaged Reynolds equation and the following Section 2.2 presents the asperity contact theory of Greenwood and Tripp. Section 3 gives an overview of the employed measurement test rig and Section 4 describes the concept of the simulation and the details of the employed oil-model. Thereafter in Section 5, results obtained from the calculations for the various working conditions are discussed and compared to experiment. Finally, Section 6 concludes this work with a summary.

2. Friction in journal bearings

In an ICE, journal bearings are generally exposed to different operation conditions in terms of load, speed and temperature. As depicted in Fig. 1, depending on relative speed, load and viscosity the operating conditions reflected as friction coefficient range from purely hydrodynamic lubrication with a sufficiently thick oil film to mixed or even boundary lubrication with severe amounts of asperity contact.

In this work different conditions are covered, ranging from purely hydrodynamic to mixed lubrication with significant power losses due to asperity contact.

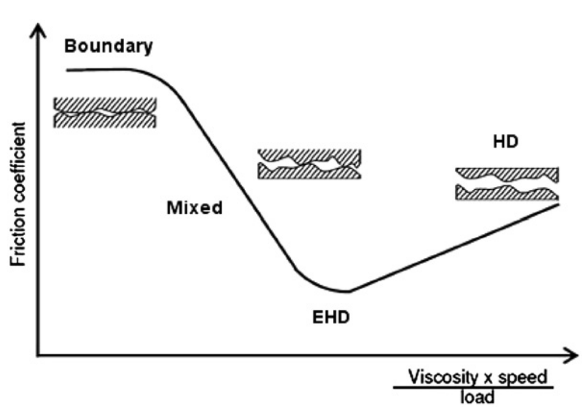

Fig. 1. The Stribeck-plot showing the different regimes of lubrication: hydrodynamic (HD), elastohydrodynamic (EHD), mixed and boundary lubrication.

To investigate the origins of friction in a journal bearing, Eq. (1) summarizes the different friction mechanisms: the first two terms describe the hydrodynamic losses within the lubricant, namely the frictional force due to pressure driven flow R_{Press} and due to shear flow R_{Shear}, and the last term denotes friction due to asperity contact, i.e. the friction due to metallic contact between the sliding surfaces, R_{Bound}:

$$R_{\mathrm{tot}} = R_{\mathrm{Press}} + R_{\mathrm{Shear}} + R_{\mathrm{Bound}}, \tag{1}$$

which can be written in the following form:

$$R_{\mathrm{tot}} = \int_{-B/2}^{B/2}\int_{\varphi_1}^{\varphi_2} \frac{h}{2}\cdot\frac{\partial p}{\partial \varphi}\cdot \mathrm{d}\varphi\cdot \mathrm{d}z + \int_{-B/2}^{B/2}\int_{\varphi_3}^{\varphi_4} \eta_p \cdot \frac{R}{h}\cdot (R\cdot\omega + L(e,\dot{e},\dot{\gamma},\varphi))\, d\varphi\cdot dz + \mu_{\mathrm{Bound}}\cdot p_{\mathrm{a}}\cdot A_{\mathrm{a}}, \tag{2}$$

where h denotes the bearing clearance, $\partial p/\partial\varphi$ is the pressure gradient along the azimuth angle φ, z denotes the axial direction, B is the width of the bearing shell, η_p is the dynamic oil viscosity that is taken into account with its pressure p dependency in this work. R is the inner shell radius and ω is the angular velocity of the shaft. φ_1,φ_2 are the begin and end of the hydrodynamic pressure distribution in the fluid film and φ_3,φ_4 denote the begin and end of the oil filled part of the bearing, respectively. μ_{Bound} represents the boundary friction coefficient, p_{a} is the asperity contact pressure and A_{a} is the area that experiences asperity contact with this asperity contact pressure. The term $L(e,\dot{e},\dot{\gamma},\varphi)$,

$$L(e,\dot{e},\dot{\gamma},\varphi) = \dot{e}\cdot\sin\varphi - e\cdot\dot{\gamma}\cdot\cos\varphi \tag{3}$$

comprises the movement of the shaft within the journal bearing, with (e,γ) denoting the polar coordinates of the shaft displacement within the bearing shell and $(\dot{e},\dot{\gamma})$ their time derivatives.

2.1. Average Reynolds equation for rough bearings

To calculate the movement of the journal, as needed for Eq. (2), and to obtain the corresponding pressure distribution within the oil film an average Reynolds equation is used, that takes into account the roughness of the adjacent surfaces.

When the typical minimum oil film thickness is of comparable magnitude to the surface roughness, the lubricating fluid flow is also affected by the surface asperities and their orientation. To account for this modification of the fluid flow we use the average Reynolds equation as developed by Patir and Cheng [22,23], which can be written in a bearing shell fixed coordinate system as

$$-\frac{\partial}{\partial x}\left(\theta\phi_x \frac{h^3}{12\eta_p}\frac{\partial p}{\partial x}\right) - \frac{\partial}{\partial z}\left(\theta\phi_z \frac{h^3}{12\eta_p}\frac{\partial p}{\partial z}\right) + \frac{\partial}{\partial x}\left(\theta(\overline{h}+\sigma_s\phi_s)\frac{U}{2}\right) + \frac{\partial}{\partial t}(\theta\overline{h}) = 0, \tag{4}$$

where x, z denote the circumferential and axial directions, θ is the oil filling factor and $h,\overline{h}$ are the nominal and average oil film thickness, respectively. Further, U denotes the journal circumferential speed, η_p is the pressure dependent oil viscosity and σ_s is the combined (root-mean-square) surface roughness. ϕ_x,ϕ_z,ϕ_s represent the flow factors that actually take into account the influence of the surface roughness.

The average Reynolds equation maps the fluid flow affected by surface roughness onto a modified fluid flow between perfectly smooth surfaces by the use of pressure flow factors ϕ_x,ϕ_z and the shear flow factor ϕ_s. The pressure flow factor depends only on the oil film thickness, the combined surface roughness σ_s and its relative orientation to flow, usually quantified with the aid of the Peklenik-factor γ,

$$\gamma = \frac{\lambda_{0.5x}}{\lambda_{0.5z}}, \tag{5}$$