HYDRODYNAMIC LUBRICATION IN POROUS JOURNAL BEARINGS: COMPARISON BETWEEN EXPERIMENTAL AND SIMULATION DATA

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1 INTRODUCTION

In many industrial applications like automotive and household appliances the most commonly used types of bearings are the sintered bearings. The advantage of using such bearings lies in the fact that once impregnated with lubricant they can perform at satisfactory levels without requiring additional impregnation. Conversely, the current knowledge about the exact behaviour of sintered bearings is rather limited, mainly due to the complexity of the porous matrix. In this sense, a novel self-consistent description of the lubricant flow through sintered journal bearings has been devised. Here the spatially variable permeability and the viscosity obtained from measurements of industrially employed lubricants and bearing serve as key input data. Amongst others, the model under investigation allows for the prediction of the hydrodynamic branch of Stribeck curve for a given permeability. It is the main goal of this paper to discuss the comparison of these results with experimental ones obtained recently.

2 DESCRIPTION OF THE THEORETICAL MODEL

The model investigated consists of the classical bearing-shaft system where a thin film of lubricant separates the two parts. As a peculiarity associated with porous journal bearings, the entire system is considered as insulated, i.e there is neither oil supply nor loss of lubricant through the seat surface or the outer margin of the gap. Assumptions like laminar flow inside the separation gap and very small inertial forces compared to viscous forces, renders us to apply the well-known Reynolds equation with the ultimate goal to obtain the pressure distribution in this region.

Nevertheless, one additional phenomenon was taken into consideration when modifying the Reynolds equation, specifically cavitation. Other papers have treated this problem extensively [1], [2], yet our model takes on a different approach. It is assumed that cavitation develops as a two-phase homogeneous mixture of lubricant and vapour occurring in the divergent part of the bearing. The fluid is assumed to undergo a thermodynamically stable phase transformation where the pressure change is insignificantly small. This region will be referred to as the cavitation region. Here the density is not a constant, but, in non-dimensional form, takes values between 0 and 1. Furthermore there is another

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contribution to the gap flow, namely the one due to the mass exchange of the lubricant with the sintered bearing. This term is mathematically described by Darcy's law for flow through porous media, and further on coupled with the Reynolds equation by means of mass conservation principles. Hence, at the interface between the fluid film and the homogeneous mixture there is zero net flux. The cavitation region is restricted to the lubrication gap as it can be shown that the interior of the sinter matrix is indeed free of cavitation. Two parameters that play an important role on the final solution of the problem were identified, the eccentricity ratio ε and the factor K that measures the strength of the coupling between Reynolds' equation and Darcy's law. Numerically, the governing equations are solved by adopting a modification of the Elrod's model [3], where an advantageous pressure-density relationship is used. The choice of lubricant and geometrical characteristics as input parameters are briefly explained in the subsequent section.

3 CONFIGURATION

The need of improving the performance of lubricants has always been a main concern for engineers. Among various novel potential lubricants, a serious amount of research has recently been dedicated to ionic liquids (IL). Although their properties vary greatly with the variation of the cation and anion chains, they have proven to have a set of properties that make them appealing for industrial applications: high thermal stability, low flammability and very low vapour pressures. The latter was particularly interesting with our bearing application, where the fluid was expected to undergo sudden changes in pressure that affect the continuity of the lubricant film.

IL1 VG32	Porosity: 20% Load: 0.5N/mm²	Porosity: 20% Load: 1.5N/mm²	Porosity: 25% Load: 0.5N/mm²	Porosity: 25% Load: 1.5N/mm²
IL2 VG150	Porosity: 20% Load: 0.5N/mm²	Porosity: 20% Load: 1.5N/mm²	Porosity: 25% Load: 0.5N/mm²	Porosity: 25% Load: 1.5N/mm²
IL3 VG220	Porosity: 20% Load: 0.5N/mm²	Porosity: 20% Load: 1.5N/mm²	Porosity: 25% Load: 0.5N/mm²	Porosity: 25% Load: 1.5N/mm²

Table 1: Test matrix

For these reasons, the lubricants chosen for both theoretical and experimental testing were ILs with different viscosity grades (32, 150, and 220). All experiments were carried out on spherical sintered-iron bearings with the inner radius R_i = 8 mm, length L = 11 mm and maximum wall thickness T = 3.75 mm. Variations of porosity and load according to Table 1 aim to predict the conditions under which the bearings perform best from the point of view of frictional

behaviour. Porosities of 20% and 25% correspond to measured average permeabilities of $6 \cdot 10^{-15}$ Pa·s and $1{,}5 \cdot 10^{-14}$ Pa·s respectively, and provide with an oil content of at least 90% of the open porosity. The bearings are radially loaded with either 0,5 N/mm² or 1,5 N/mm² at a time.

4 RESULTS AND DISCUSSION

Experimental measurements cover the whole range from boundary lubrication regime to fully hydrodynamic lubrication, while numerical simulations account only for the latter.

4.1 Experimental

In the field of Tribology Stribeck curves play an important role due to the possibility of identifying the transition speeds from one lubrication regime to the other. The faster the transition to hydrodynamic lubrication, the less wear of the parts, and thus, a more successful lubrication process. This is why the main focus of our investigation was to obtain most accurate friction values in relation with the rotational speeds, which later on can be compared to simulation results. All experiments carried out for each configuration comprised 3 types of measurements: running-in period for a fixed time during which the maximum reached temperatures were recorded, a series of 3 Stribeck runs that deliver the friction values versus the rotational speed, and short-time tests for low constant speeds in order to record the values for the frictional forces in the boundary lubrication regime. In the first instance it was found that the temperatures reached during the Stribeck runs are greatly comparable with the temperatures reached at the end of 5 hours of operation, when the system appears to have achieved thermo dynamical equilibrium. One critical example that supports this finding is the temperature profile obtained for IL3 on a sinter bearing with 20% porosity and loaded with 0.5 N/mm², shown in Figure 1. After 5 hours of operation (Figure 1a) the maximum temperature in the system is about 55 °C. On the other hand, during the Stribeck runs (Figure 1b), by varying the rotational speed from 0 to 600 rpm in 50 seconds, from 600 rpm to 3000 rpm in 90 seconds and further on maintaining it at 3000 rpm for 25 seconds, the system reaches a temperature of approximately 50 °C. Thus, this particular variation profile of the speeds chosen for the Stribeck runs leads to similar temperatures obtained as if the system were operating at constant high speed for longer time, the case of practical applications.

From the temperature-speed curve (Figure 1b) it is also possible to obtain the variation of the lubricant viscosity with temperature. For every value of the temperature we can calculate the dynamic viscosity of the lubricants by applying the commonly adopted Ubbelohde-Walther formula:

$$\lg\lg(\eta + a) = K - m \lg T ,$$

where a, K and m are known parameters chosen in relation with the lubricant

properties, and T and η are the usual notations for temperature and dynamic viscosity, respectively. In this way a functional relationship between viscosity and rotational speed is obtained, a specific restraint that finally enters in the simulation code.

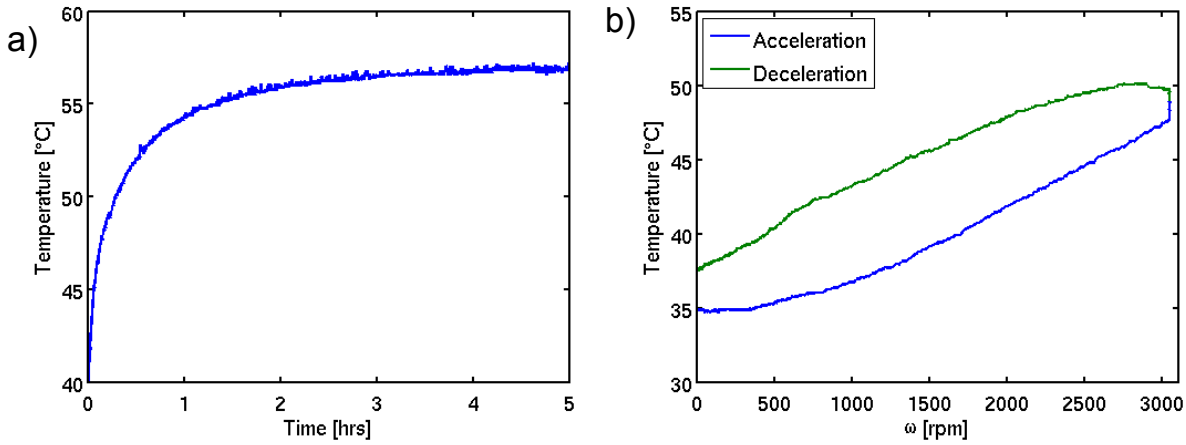

Figure 1: IL3-20%-0.5 N/mm² at 3000 rpm (a) and Stribeck run (b)

4.2 Comparison with simulation

The numerical computations deal with the same configurations described so far, yet, the controlling parameter is not the external applied load, but the eccentricity ratio ε. This parameter describes the magnitude of the displacement between the centre of the shaft and the centre of the bearing ($\varepsilon = 0$ the bearing and the shaft are concentric, $\varepsilon = 1$ the bearing and the shaft are in contact). Due to numerical reasons the maximum value of ε that leads to converged solutions in our simulations is approximately 0.7. Nevertheless, within this range of eccentricity ratios we are still able to reach load values very close to the experimental ones.

Since the theoretical model is a steady state system, it is reasonable to compare results only for the acceleration phase of the Stribeck runs. Figure 2 shows the assessment of all the numerical calculations versus experimental measurements. In all experimental curves we notice slightly increased amplitudes of oscillations around the values of 750 rpm and 1750 rpm, which can be attributed to a common behaviour of the step motor that has been used on the experimental test rigs. From the point of view of the applied load, the general trend is that bearings having to support higher loads show lower friction coefficients in hydrodynamic regime, especially for more viscous lubricants. As expected, porosity also plays a role in the lubrication system, although a generally valid tendency cannot be advanced.

The comparison reveals an apparent discrepancy between the calculated Stribeck curves and those obtained experimentally. However, for low to medium viscosities and a relatively high load the correlation appears to be the best (Figure 2 a, b, c, d), but for highly viscous lubricants at a low load it is rather

poor: a careful analysis of the numerical results for a given load and a constant rotational speed shows that the friction force and, hence, the friction number are found to be almost proportional to the eccentricity, which explains the observed offset. As a noteworthy aspect, cavitation plays a pronounced role for high loads and our model predicts the behaviour of the bearing with satisfactory accuracy.

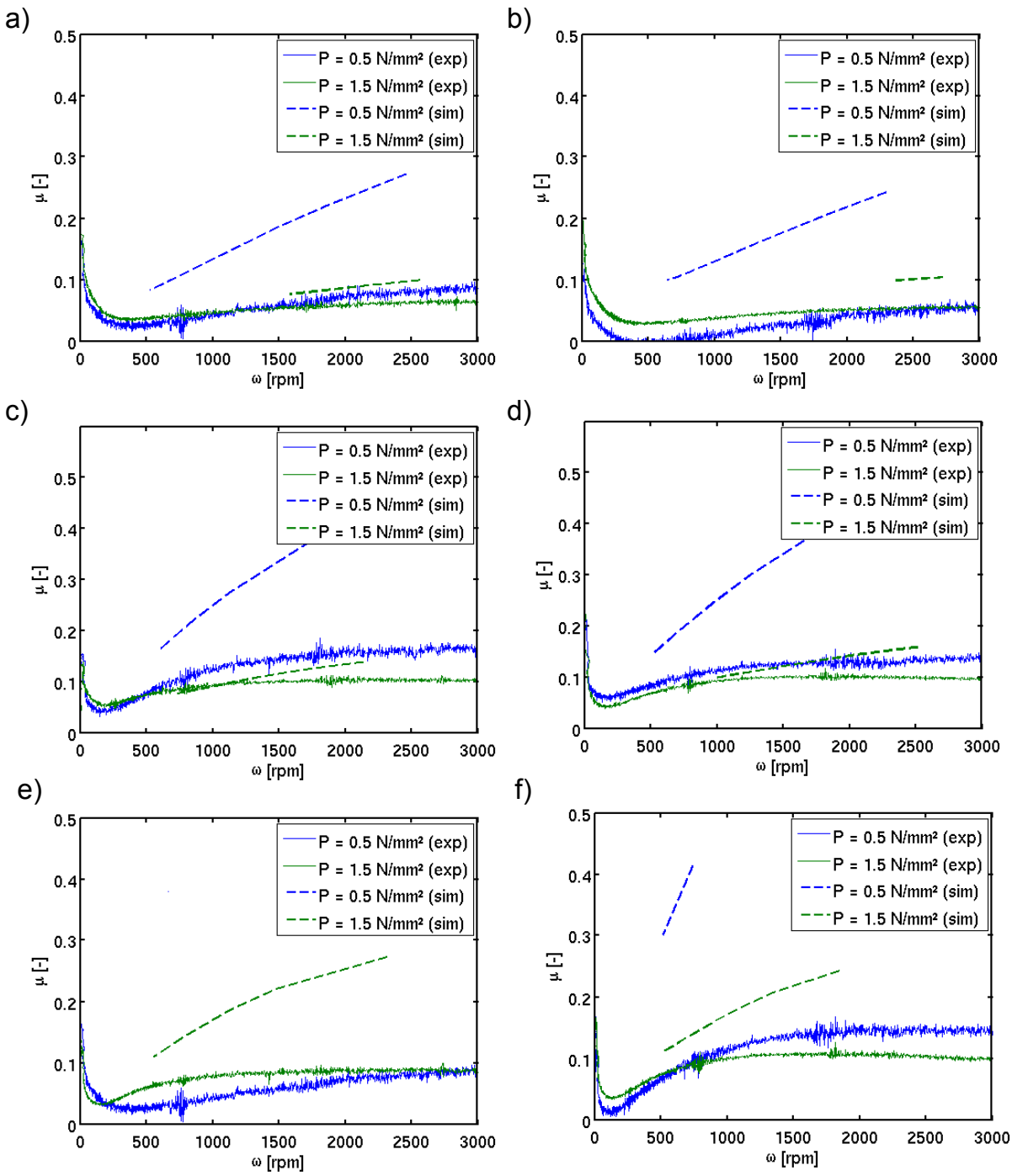

Figure 2: Comparison between friction numbers obtained experimentally (exp) and calculated (sim) for IL1 at (a) 20%, (b) 25% porosity, IL2 at (c) 20%, (d) 25% porosity, and IL3 at (e) 20%, (f) 25% porosity.

Moreover, the weak correlation in the case of highly viscous lubricants can be recognized as an effect of the non-Newtonian behaviour that has not been resolved in our numerical formulation (Figure 2 e, f). The absence of data for P = 0.5 N/mm² in Figure 2e can be explained by failure of the numerical interpolation scheme, which for the given parameters yields values occupying the mixed lubrication domain, whereas our interest was in the fully hydrodynamic one.

5 CONCLUSIONS

Apart from experimental measurements, numerical calculations on the basis of a simple but self-consistent theoretical model can provide with great insight into successful tribological design of mechanical parts. Therefore, obtaining a realistic correlation between the two could minimize the future efforts in predicting the lubrication behaviour. Data obtained so far shows encouraging perspectives, but nevertheless there is still research to do, in both experimental and simulation areas. Emphasis must be set on improving numerical resolution and minimizing errors, and in adequate calibration of the test rigs used. In particular, the deviation of the calculated from the measured data in the Stribeck curves still requires further elaborate examination of the data obtained so far. Nevertheless, at least for relatively high loads a promising agreement is detected, and the slopes of the computed curves matches those of the measured ones satisfactorily well throughout. One has to concede that the current flow description lacks, amongst other physical aspects of rather minor interest, the inclusion of non-Newtonian lubricant behaviour and a sufficiently reliable prediction of the permeability in dependence of the measured porosity. These issues and an improved cavitation model are presently under investigation.

6 LITERATURE

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