Tribology International 92 (2015) 395–403

Contents lists available at ScienceDirect

Tribology International

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

Edge loading and running-in wear in dynamically loaded journal bearings

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ARTICLE INFO

Article history:
Received 24 March 2015
Received in revised form
17 June 2015
Accepted 17 July 2015
Available online 29 July 2015

Keywords:
Mixed lubrication
Non-Newtonian lubricant
Simulation
Measurement

ABSTRACT

This paper focuses on the beginning phase of hydrodynamic journal bearing life time when the first adaption of the contacting surfaces occurs. Generally, this effect is known as running-in.

Experimental data from a journal bearing test rig using a low viscosity 0W20 multi-grade automotive lubricant provide the solid basis for the simulative study of the running-in process. From these measurements and a subsequent determination of the surface roughness, parameters for the mixed lubrication contact model are derived. This analysis combined with the experimentally identified lubricant properties under high pressure and high shear rate enables the evaluation of an iterative simulation approach. In this iterative approach the bearing surface geometry is adapted stepwise until a steady state of operation is achieved.

Results show worn regions at the edge of the highly loaded bearing shell. This wear is caused by metal–metal contact due to the elastic bending of the shaft. The calculated wear depth at the edge and the expansion of the worn area in axial and circumferential direction matches the measured profile. This agreement indicates that the simple iterative approach using the Greenwood and Tripp contact model and Archard's wear equation is suitable to predict the worn surface geometry after the running-in process is completed.

Furthermore, the simulation shows that the maximum asperity contact pressure in mixed lubrication decreases with the stepwise adaption of the surface geometry, until only an insignificant metal–metal contact remains. With this adapted surface geometry, the influence of shaft speed, temperature and surface roughness is also discussed.

1. Introduction

When two contacting elements operate together for the first time, adjustments of the adjacent surfaces take place, regardless if the contact is lubricated or not. This adjustment concerns the geometrical conformity on macro and micro scale as well as changes of the mechanical and material properties [1]. Such an adjustment process occurs in the early stage of operation and is commonly called running-in.

In hydrodynamic journal bearings the softer shell generally adapts its surface to fit the harder journal during the running-in. In highly loaded regions of the bearing, abrasive wear and plastic deformation occur. These impacts affect the surface shape, the radial clearance and surface roughness. Older publications on this topic [2–5] highlighted through experiments that the shell roughness gets smoother during the running-in process especially in the very beginning. Another result implied that the final shell roughness of the worn region was similar to the harder journal roughness. The experiments were conducted on journal bearing test rigs with a static load. These test conditions lead to an extensive metal–metal contact and wear along the entire bearing width. In a more recent publication [6], a similar test rig setup was used to study the effect of lubricant additives on the friction and wear behaviour of journal bearings during several starts and stops of the journal.

However, in most technical applications, such as turbines or combustion engines, journal bearings are designed for a long life time. Thus, journal bearings mainly operate in a pure hydrodynamic lubrication regime. Usually, dynamic loads (e.g. due to eccentric masses or combustion) act on the shaft and bend it

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http://dx.doi.org/10.1016/j.triboint.2015.07.022

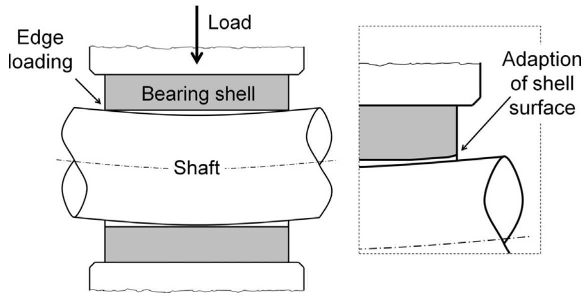

Fig. 1. Sketch of elastically deformed shaft due to external load.

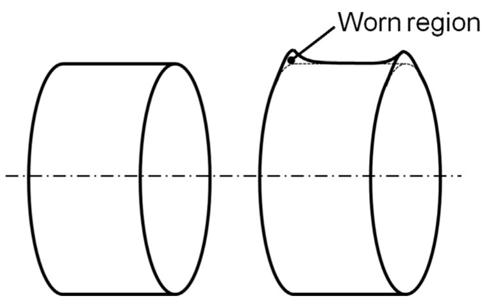

Fig. 2. Sketched surface profile of a new bearing (left) and a worn bearing after the running-in process (right).

elastically. This elastic deformation causes an unconformity of journal and shell and can lead to metal–metal contact mainly at the bearing edges (see Fig. 1). Another source for unconformity and edge loading is any minor misalignment between journal and bearing as a result of production tolerances, assembly deformation or thermal deviation.

Previous publications discuss minimum oil film thickness, pressure distribution and thermal behaviour of misaligned journal bearings either experimentally [7] or by simulation [8,9]. Sun et al. [10] experimentally investigated the behaviour of misaligned journal bearings due to elastic shaft deformation. At a specific amount of rotational load he identified an increase of bearing temperature and a reduction of the maximum hydrodynamic pressure caused by metal–metal contact at the bearing edge. Priestner et al. [11] studied the frictional losses in journal bearings using an elasto-hydrodynamic simulation tool and identified high metal–metal contact pressures at the bearing edges due to shaft bending. The high metal–metal contact pressure led to overestimated friction losses. Consequently, the author adapted the geometry of the bearing shell by superposing a worn surface profile to its nominal cylindrical shape. As a result of this measure, the metal–metal contact pressure was reduced to a minimum and the predicted friction torque correlates with the measurement for a wide range of operating conditions [12].

The present study focuses on the adaption of the bearing shell profile due to severe metal–metal contact at the bearing edge (see Fig. 1). A long-term test on a journal bearing test rig was conducted with high dynamic loads. Initially, the test was performed to validate elasto-hydrodynamic simulation models with a detailed description of the oil viscosity. The impact of temperature, pressure and shear rate on journal bearing friction was presented in [13] for a wide range of operation conditions. Although, the bearing operated mainly in pure hydrodynamic conditions, the edges of the bearing shell were worn at the end of the test. The subsequent measurements of the surface geometry and surface roughness provide a solid basis for the verification of a numerical investigation of the running-in process.

The primary aim of this study is to verify an iterative approach to calculate the worn surface profile of journal bearings using the approach published by Offner [14]. Starting with the initial cylindrical

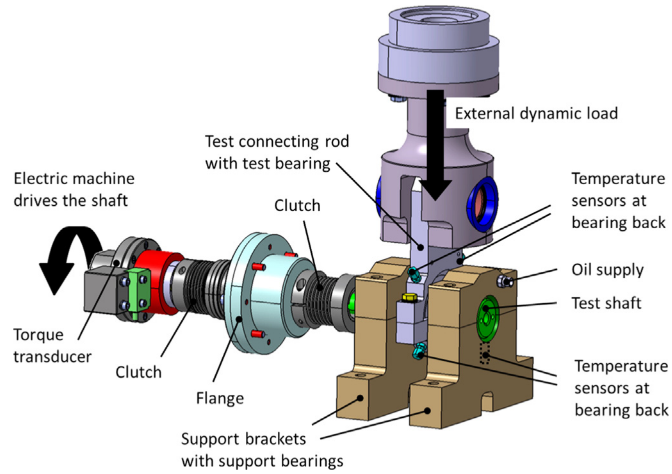

Fig. 3. Journal bearing test-rig at KS Gleitlager.

shape of the bearing shell (see Fig. 2), an elasto-hydrodynamic (EHD) simulation is performed and metal–metal contact on the bearing surface is evaluated. Using Archard's equation, a worn surface profile is generated by calculating the local amount of wear. In a following step, the worn surface profile of the first step is included into the EHD simulation and the wear calculation is performed once again. This process is repeated until metal–metal contact becomes insignificant and the surface profile allows a steady state of operation. In order to successfully verify the method, both, the input data for the contact model and the complex rheological properties of the lubricant are derived from measurements.

For a secondary aim, the study seeks to analyse under which operating condition more wear occurs due to edge loading.

Furthermore, the study seeks to support the use of simplified shell profiles in elasto-hydrodynamic journal bearing simulation to minimize metal–metal contact. Previous publications by the authors of this study have already shown that the inclusion of a simplified shell profile allows for a realistic estimation of metal–metal contact [12,15,16].

2. Test method

The test was conducted on the journal bearing test-rig at KS Gleitlager. The test-rig is shown in Fig. 3.

The test-rig consists of a rotating straight shaft which rests on two supporting journal bearings in support brackets. Via a test connecting rod an external load is applied onto the central test journal bearing. The dimensions and oil supply design of the journal bearings correspond to automotive main bearings with a 180° oil supply groove for the two supporting journal bearings. The test journal bearing corresponds to big-end bearings having an oil supply hole in the load-free (lower) shell. The bearing dimensions and materials are listed in Table 1.

An elastically clutched electric motor drives the shaft. The test connecting rod is dynamically excited by an electromechanical high-frequency pulsator, which is operated at 80 Hz. One load case is investigated with 80 kN maximum load which correspond to 100 MPa specific load in the test bearing; the dynamic load curve is shown in Fig. 4.

The test procedure started at a shaft speed of 1000 rpm and a stepwise run-up was performed up to 7000 rpm followed by a run-down to again 1000 rpm. Each step is operated for 20 min. A torque transducer, a Manner Sensortelemetrie 50 Nm with an accuracy of ± 0.15 Nm, was placed between the electric motor and the shaft to measure the friction torque generated by all three bearings.