7th International DAAAM Baltic Conference
"INDUSTRIAL ENGINEERING"
22-24 April 2010, Tallinn, Estonia

MEASUREMENT OF OIL FILM PRESSURE IN HYDRODYNAMIC JOURNAL BEARINGS

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Abstract: *Hydrodynamic journal bearings are critical power transmission components in various machines and, therefore, knowing the true operating conditions of hydrodynamic journal bearings is essential. Oil film pressure is one of the key parameters describing the operating conditions in hydrodynamic journal bearings. The aim of this study was to measure the oil film pressure in real hydrodynamic journal bearings under realistic operating conditions. A versatile bearing test rig with hydraulic loading system was used as the main test apparatus. The oil film pressure was measured by optical pressure sensors integrated in the bearing. Experimental oil film pressure data was compared with results from simulations. The results can be used in development of safer and more efficient machines.*

Key words: lubrication, optical pressure sensor, tribology

1. INTRODUCTION

The power density in various machines is increasing year by year due to growing demands for mechanical and economic efficiency. One of the consequences of an increase in power density is that critical power transmission components have to carry increasingly high loads.

Hydrodynamic journal bearings are typical critical power transmission components that carry high loads in different machines. Therefore, it is essential to know the true or expected operating conditions of the bearings. Numerous studies of the operating conditions of hydrodynamic journal bearings have been made during the last decades. Still, the case is far from closed.

The operating conditions of hydrodynamic journal bearings can be described by a set of tribological variables called key operating parameters. The key operating parameters most directly related to the bearing-lubricant-shaft contact are the oil film temperature, oil film thickness and oil film pressure. Until now, oil film pressure in hydrodynamic journal bearings has been studied mainly by mathematical means, because its experimental determination has been a demanding or even an unfeasible task. Under real operating conditions, there are typically many practicalities that complicate the experimental determination of true oil film pressure. The oil film may be extremely thin and therefore sensitive to different disturbing factors. In addition, the oil film pressure may be extremely high or may have a high dynamic variation.

The main aim of the study was to determine the oil film pressure in hydrodynamic journal bearings carrying realistic loads. The study also included the determination of the oil film temperature, operating range and friction loss.

The operation of journal bearings was studied both by experimental means in a laboratory environment and by theoretical means using mathematical simulations. The main test apparatus was a versatile bearing test rig with an advanced measuring system for determining major tribological variables of journal bearings. Common journal bearings running under hydrodynamic operating conditions were studied. The experiments were made in true scale and realistic bearing loads were used. The study focused on the main bearings of a high-speed diesel engine. The oil film pressure was measured by optical pressure sensors. Advanced simulation software was used for the bearing simulation.

The oil film pressure in a hydrodynamic journal bearing was measured with parameters of diesel engine operating range. By determining the oil film pressure in hydrodynamic journal bearings, it is possible to increase knowledge about the true operating conditions of bearings. The knowledge can be used in the development of safer and more efficient machines.

2. STATE OF ART

To measure the oil film pressure, thin film pressure sensors consisting of thin material layers on the sliding surface of the bearing were used by Ichikawa *et al*. [1], Mihara *et al*. [2,3], Mihara and Someya [4] and Someya and Mihara [5]. The pressure sensors typically consist of three layers of material with a total thickness of about 6 μm [5]. The pressure sensitive part of the sensor is called sensitive film. It is in the middlemost layer and is made of Manganin, an alloy of copper, manganese and nickel. This pressure-sensitive part is connected as the fourth resistor in a Wheatstone bridge. The operation of the thin film pressure sensor is based on the change of resistance in the pressure sensitive part. The lowest layer electrically insulates the sensitive film from the bearing shell. The uppermost layer protects the sensitive film from wear. Because the surface of the protection film is plain, the thin film pressure sensor resembles a common plain coating on the sliding surface of the bearing. Masuda *et al.* [6] carried out test rig experiments in which oil film pressure in a connecting rod big-end bearing was measured by a semi-conductor strain gauge-type transducer embedded in the surface of the crankpin. They used a slip ring to transmit the pressure signal from the crankpin to the data recorder. Brito *et al*. [7] used high precision Bourdon pressure gauges attached via manifold valves to a series of bores in the circumferential centreline of the bush to measure the oil film pressure in a journal bearing with two axial grooves. They used a total of fourteen pressure gauges to determine the oil film pressure distribution. Sinanoğlu *et al.* [8] determined the oil film distribution by using sixteen manometer tubes placed in the bearing shell. The first optical pressure sensor to measure oil film pressure in journal bearings was developed by Ronkainen *et al*. [9] and its different versions were used in this study.

3. MATERIALS AND METHODS

A versatile bearing test rig (Fig. 1) was the main test apparatus in this study. The main design of the test rig was simple, spacious and based on the use of replaceable components. The test rig consisted of a frame and a bearing unit as well as loading, drive, lubrication, control and measuring systems.

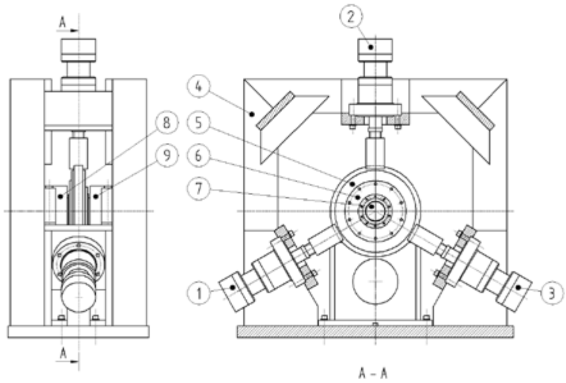

Fig. 1. The test rig with three hydraulic cylinders (1, 2 and 3) generating the bearing load. The cylinders were placed in the frame (4) and they pushed the force ring (5), housing (6) and bearing against the shaft (7) with the supporting bearings (8 and 9).

The frame of the bearing test rig consisted of a coarse base plate and two firm angular beams with housings for the load cylinders. The width, height and length of the frame were circa 1.2, 1.1 and 0.6 m, respectively.

The bearing unit included the bearing, housing with seal ring covers, force ring with heating resistors, alignment arm, shaft and supporting bearings with negligible bearing play.

The main components of the loading system were three servo valve controlled hydraulic cylinders and a hydraulic plant power unit. The cylinders pushed the force ring, housing and bearing against the shaft, thus producing the bearing load. The nominal load range of the bearing test rig was from 0 to 100 kN.

The drive system drove the shaft and its main components were a frequency modulator, three-phase motor, drive shaft, pulse sensor and coupling with an integrated torque transmitter. The nominal output of the motor was 11 kW and the rotational speed range of the shaft was from 0 to 3000 1/min.

The lubrication system supplied the lubricating oil for the bearing and its main components were an oil pump, oil tank, filters and heating devices with thermostats. The oil inlet pressure range was from 0 to 10 bar and the oil inlet temperature range was from room temperature to 100 ^{o}C.

The control system controlled the bearing load generated by the hydraulic cylinders and its main components were a controlling computer with a signal processor card and an analogue output board, along with analogue proportional-integral-derivative amplifiers, anti-aliasing filters, three two-stage servo directional valves with analogue amplifier modules, six cylinder pressure transmitters, a pulse sensor and pulse counter electronics. The pulse sensor and pulse counter outputs were used by the signal processor to produce the cylinder forces (Fig. 2), which were synchronised with the rotation of the shaft. The synchronised operation of the cylinders made it possible to generate various bearing load patterns, such as static and rotating loads.

The measuring system measured and recorded the data. Its main components were a measuring computer, signal processing modules and various measuring devices such as optical pressure sensors based on optical fibre technology.

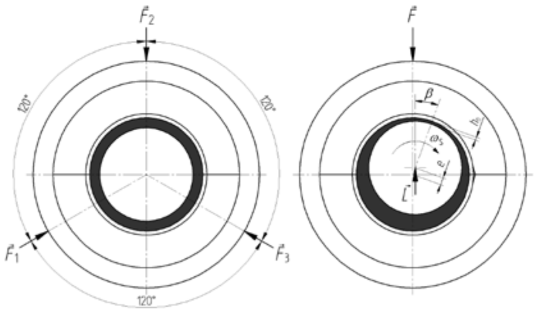

Fig 2. Cylinder forces F_1, F_2 and F_3 pushed the bearing against the shaft. Their resultant, load F, was the counterforce of the bearing load L. With rotating loads, the shaft, resultant and bearing load rotated with the angular velocity ω_S.

Four bearings with diameter of 85 mm and width of 32 mm were used in the experiments and their main design was based on a main bearing of a high-speed diesel engine. In the bearing used in oil film pressure measurements, the shell was made of steel, the bearing alloy was lead-free bronze, the sliding surface was non-plated and the lubricating oil was supplied through the shaft (Fig. 3). Small cavities with measurement membranes were made on the outer surface of the upper bearing sleeve for the optical pressure sensors.

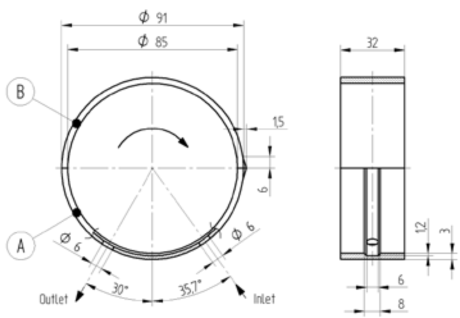

Fig. 3. Bearing had an oil groove in its lower sleeve. The oil inlet and outlet holes were blocked. The lubricating oil ran from the bearing into two outlet channels in the housing.

Three shafts made of steel were used and their nominal diameter and length were 85 mm and 600 mm, respectively. The shaft used in oil film pressure measurements had lapped sliding surface and one radial oil-feeding hole (Fig. 4).

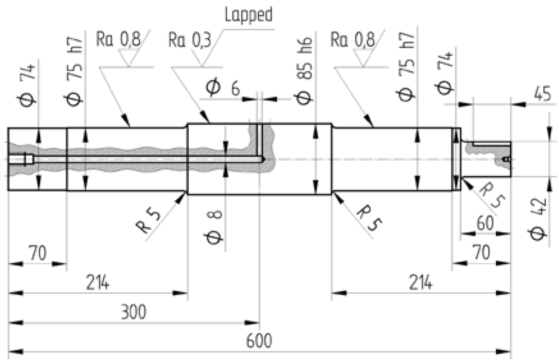

Fig. 4. Shaft with a radial oil-feeding hole.

Two housings were used and their nominal outer diameter was 270 mm and width was 80 mm. They consisted of half-round conical sleeves with lubrication channels. In the housing used in oil film pressure measurements, there were holes for seven optical pressure sensors and one thermocouple (Fig. 5). The design of the housings made it possible to use realistic pre-tensioning caused by oversized bearing shells.

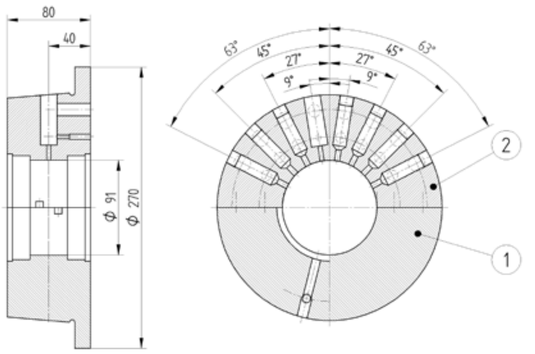

Fig. 5. Housing with holes for optical pressure sensors and a thermocouple.

Seven lubricating oils were used. Two of them were real engine lubricants. The lubricating oil used in the oil film pressure measurements was a combined cylinder and crankcase lubricant of medium speed diesel engines. It was manufactured from mineral base oils and blended as SAE 40. Its density was 916 kg/m^3 (at 15 ^{o}C) and kinematic viscosity was 139 mm^2/s (at 40 ^{o}C).

To determine operating range of the bearings, an experimental method called isotherm mapping [10] was used in the test rig experiments. The friction loss of the bearings was analysed by performing an experimental heat flow analysis. The oil film temperature was estimated to be equal to the operating temperature of the bearing. The operating temperature was measured by thermocouples. The oil film pressure was measured by optical pressure sensors. They were integrated in the bearing in such a way that the sliding surface of the bearing remained unchanged and acted as a measurement membrane.

The bearing simulation method was based on numerical solution of the Reynolds differential equation. The simulations were made by the AVL EXCITE simulation software. The simulation model consisted of an elastohydrodynamic bearing with 19 axial and 121 circumferential nodes, a shaft with five mass points, and an anchored housing. The lubricating oil temperature was assumed to be equal to the measured operating temperature of the bearing.

4. RESULTS

Results from the test rig experiments by the isotherm mapping method and heat flow analysis showed that, with a sufficient bearing clearance and oil supply, safe and hydrodynamic operating range of the bearings covered sliding speeds from 2 to 12 m/s and specific loads up to 32 MPa. Usually, the friction power was below 1 kW. The friction coefficients were typically below 0.01 and the lowest values were about 0.002. Results from the oil film pressure measurements showed that the optical sensors functioned well under realistic operating conditions. The signal from the optical sensor had a good signal to noise ratio and its repeatability was good in the range studied. The measured (Fig. 6) and simulated (Fig. 7) oil film pressures increased logically as the bearing load was rotating and increased from 5 to 15 kN (or 1.8 to 5.5 MPa) with constant sliding speed of 4 m/s, oil inlet temperature of 70 ^{o}C and oil inlet pressure of 3 bar in the above mentioned range.

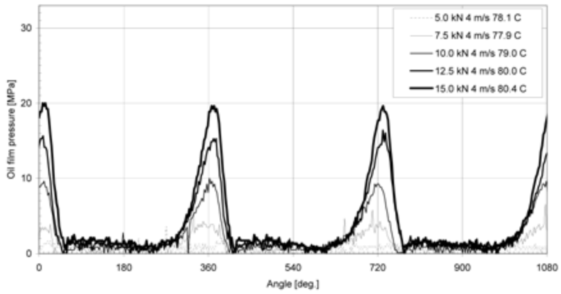

Fig 6. Measured oil film pressure with rotating loads of 5 to 15 kN and sliding speed of 4 m/s. The oil film temperature was 77.9 to 80.4 ^{o}C. The angle was 0 + N x 360 degrees (N = 0, 1, 2…) when the load F crossed the measurement point with the angle of position of +63 degrees.

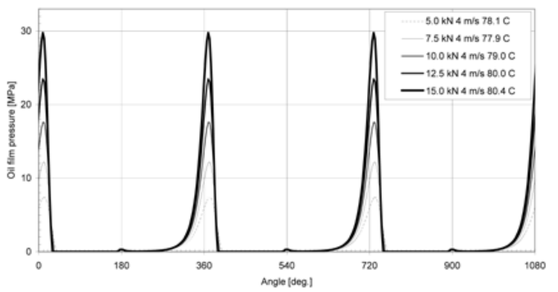

Fig. 7. Simulated oil film pressure with rotating loads of 5 to 15 kN and sliding speed of 4 m/s. The oil film temperature was 77.9 to 80.4 ^{o}C. The angle was 0 + N x 360 degrees (N = 0, 1, 2…) when the load F crossed the measurement point with the angle of position of +63 degrees.

5. DISCUSSION AND CONCLUSIONS

In this study, realistic test materials were extensively used and the experiments were made in true scale. The optical pressure sensors for measuring the oil film pressure were novel, unique and usable devices. They were based on optical fibre technology, providing benefits such as insensitivity to electromagnetic disturbances and the capability to transmit information over long distances. The optical sensors were integrated in the bearing in such a way that the sliding surface of the bearing remained unchanged, which was a highly beneficial feature. The oil film pressure was measured under hydrodynamic lubrication and realistic operation conditions. The measured and simulated oil film pressures varied logically as functions of the bearing load. The measured area of high pressure in the oil film was wider than the simulated one. The measured peak pressure was lower than the simulated peak pressure. The highest measured pressures were fairly high when compared with reported experimental results from other studies with engine bearings. Mihara *et al.* [2] measured oil film pressures up to about 33 MPa in the 62 mm diameter main bearing of a diesel engine, running at full load. Someya and Mihara [5] measured oil film pressures up to about 50 MPa in the 81 mm diameter main bearing of a diesel engine, running at full load.

By determining the oil film pressure, it is possible to increase our knowledge of the true operating conditions of hydrodynamic journal bearings. This knowledge can be used in the development of safer and more efficient machines and engines with hydrodynamic journal bearings that are carrying high loads.

6. ACKNOWLEDGEMENTS

The valuable support of the AVL List GmbH, Lubrizol Limited, Miba Gleitlager GmbH, Sisu Diesel Inc., Suomen Petrooli Oy, VTT Technical Research Centre of Finland and Wärtsilä Finland is gratefully acknowledged.

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