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DESIGN OF THE MAGNETICALLY SENSITIVE HYDRODYNAMIC BEARING FOR THE EXPERIMENTAL ROTOR RIG

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Abstract: *In a hydrodynamic bearing, the relative motion between the rotating shaft and stationary housing is separated by a thin film of a lubricant. The magnetically sensitive lubricants with the promise of a semi-active control are able to improve the vibration response of the rotor system supported by the hydrodynamic bearings. A design of the magnetically sensitive hydrodynamic bearing is briefly introduced in this paper. The presented experimental rotor rig is proposed to analyse the large range of load conditions and rotational speeds of the bearing. The designed hydrodynamic bearing is intended to test the ferrofluids, magnetorheological oils, and nano-micro composite magnetic fluids. The apparent viscosity of the magnetically sensitive lubricant is altered by the magnetic field generated by the electric coil and thus the position of the bearing journal is shifted. The experimental measurement results show that the rheological behaviour of the lubrication layer with a magnetorheological fluid is significantly influenced by a magnetic field. The raising magnitude of the current in the electric coil leads to an increase in the bearing's load performance.*

Keywords: Magnetically sensitive fluid, Hydrodynamic lubrication, Sommerfeld number, Experimental measurement, Load bearing capacity.

1. Introduction

The hydrodynamic bearing design is dependent on the predefined operation conditions of the rotating machine, on the geometrical dimensions of the bearing, and the physical properties of a lubricant. The bearing capacity, journal circumferential rotational speed, frictional losses, lubricant heating, vibration damping, and bearing life is determined by the design factors. The use of a magnetically sensitive fluid allows to improve bearing performance in a wide range of operating speeds and bearing loads (Zapoměl and Ferfecki, 2022). The magnetorheological fluids and nano-micro composite fluids, see (Susan-Resiga and Vékás, 2018), offer the possibility to adjust the apparent viscosity of a lubricant in the hydrodynamic bearing by a magnetic field.

Lampaert (2020) introduces the theoretical and experimental research focused on magnetic fluids in bearings and sealings systems. For the bearings with magnetorheological fluids, the rheological textures are developed to improve the performance of the bearings and for the ferrofluids, two design concepts of bearings are described: the pressurized type and the pocket type. The comparison of the theoretical and experimental measurement results with the hydrodynamic bearing lubricated by a ferrofluid and magnetorheological oil is presented by Ureta et al., (2009). The enhancement of designed bearing performance is achieved only by the use of a magnetorheological fluid.

The paper describes the new concept design of the magnetically sensitive hydrodynamic bearing. The work aims to show the assembled experimental test rig with the proposed bearing and to give explanations of the measured response behaviour of the bearing lubricated by a magnetorheological oil. The results measured on the experimental test rig are qualitatively fully consistent with the computational simulations.

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2. Design of the hydrodynamic bearing for the magnetically sensitive fluids

The hydraulic parameters of the designed bearing were determined by the computational approach based on a numerical solution of the Reynolds equation presented in the book by the authors Budynas and Nisbett (2006). Rheological properties of the lubrication film of the magnetic fluid are altered with an external magnetic field. The distribution of the magnetic induction in a fluid film and the leakage of the magnetic flux out of the magnetic circuit was investigated with the magnetostatic finite element analysis.

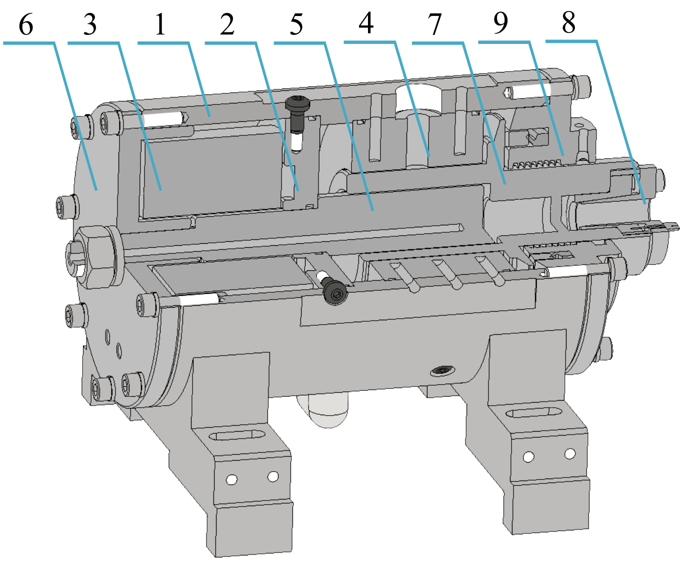

Fig. 1: Sketch of the designed hydrodynamical bearing.

The bearing shell with a fixed cylindrical geometry with the inner diameter of 60.6 mm was designed in three variants with *L/D* (the length *L* and the inner diameter *D* of the bearing) ratios of 1.0, 0.5, and 0.25. The radial clearance of the bearing is 0.3 mm. For the fluid dynamic viscosity of 0.042 Pas and the static applied load of 50 N, the performance factors of the bearing's shell were computed: the Sommerfeld number of 0.756, 0.378, and 0.189 and the relative eccentricity of 0.179, 0.551, and 0.841. For a higher preload, an additional static load can be applied on the rotor with the designed mechanism.

The housing (1) of the designed hydrodynamic bearing (see Fig. 1) consists of a cylindrical tube with two welded blocks for mounting to the base. The bearing housing is made of soft magnetic steel. In the bearing housing, there is the disc (2) of the brass material separating the inner space for electric coil (3) and the chamber with the bearing shell (4) made of steel material. The ferromagnetic pin (5) is inserted into the disc and the electric coil. The pin is bolted to the end shield (6) and the bearing housing. The hollow journal (7) of aluminium material is inserted onto the pin. The thin film of a magnetically sensitive lubricant is supplied into the clearance between the inner surface of the bearing shell and the outer surface of the hollow journal. The cone clamping element (8) is used to connect the shaft to the hollow journal. The aluminium front shield (9) is screwed to the bearing housing. To prevent a leakage flow of the oil in the clearance between the front shield and the hollow journal the labyrinth seal and the circumferential V-ring seal are utilized. The O-ring seals are used to avoid contamination of the equipment, namely the electric coil and the external environment.

The magnetic sensitive fluid is supplied into the upper part of the bearing shell by a single inlet. The two outflows are located in the bottom part of the bearing. Holes for monitoring of the magnetic field and the temperature distribution in the components during the bearing operation are drilled in the pin, bearing housing, and shell.

The electric coil generates the magnetic flux through the end shield, bearing housing and shell, lubrication film of magnetic fluid, hollow journal, and pin and closes back to the end shield. The magnetic forces between the hollow journal and either the pin or the bearing shell are suppressed by the non-ferromagnetic material of the journal.

3. Description of the experimental rotor rig for testing of the magnetically sensitive bearing

The main components of the proposed experimental test rig for the magnetically sensitive bearing are shown in Fig. 2. The foundation (1) of 25 mm-thick plate is made of the EN AW 2024 aluminium alloy. The rotor base (2) is the U 180 hot rolled profile with a length of approximately 1.5 meter. The bearing housing (3) for a single row deep groove ball bearing, the proximity probe stand (4), and the rotor protection stand (5) are made of the EN AW 7075. The positions of the bearing housings, stand for the probes, and vibration protection can be easily adjusted on the rotor base according to the measurement requirements.

The rotor is driven by a low-voltage 2-pole asynchronous electric motor (6) with a power of 2.2 kW. A frequency converter (7) of the rated power of 3.0 kW is used to control the motor speed. The motor is dynamically balanced to grade B and therefore it can be safely operated at up to 6000 rpm. Two kinds of flexible couplings were used: the jaw coupling (8) made of aluminium and bellows coupling made of steel material. Shafts (9) with a diameter of 15 mm are made of C45 material, and numerous shaft lengths of up to 800 mm can be used. The three discs (10) with a diameter of 100 mm are mounted on the shaft. The discs are made of steel grade 11600 and weight of 4.5 kg in total. On the opposite side of the motor, the cone clamping element WSR 220 (11) is installed to connect the rotor with the proposed bearing.

The hydrodynamic bearing (12) can be supplied with a classical oil or a magnetically sensitive fluid. In the developed fluid feed system, the bearing inlet is not pressurized. The single inlet orifice with a diameter of 10 mm is located near the upper reservoir (13). The outlets lead to the bottom stainless-steel reservoir (14) where the fluid is stored. The fluid system is closed by the gear pump (15) with a maximum lift height of 25 m, which pumps the fluid into the upper reservoir. The temperature of the fluid is controlled and can be cooled in the bottom reservoir with a submersible cooling coil.

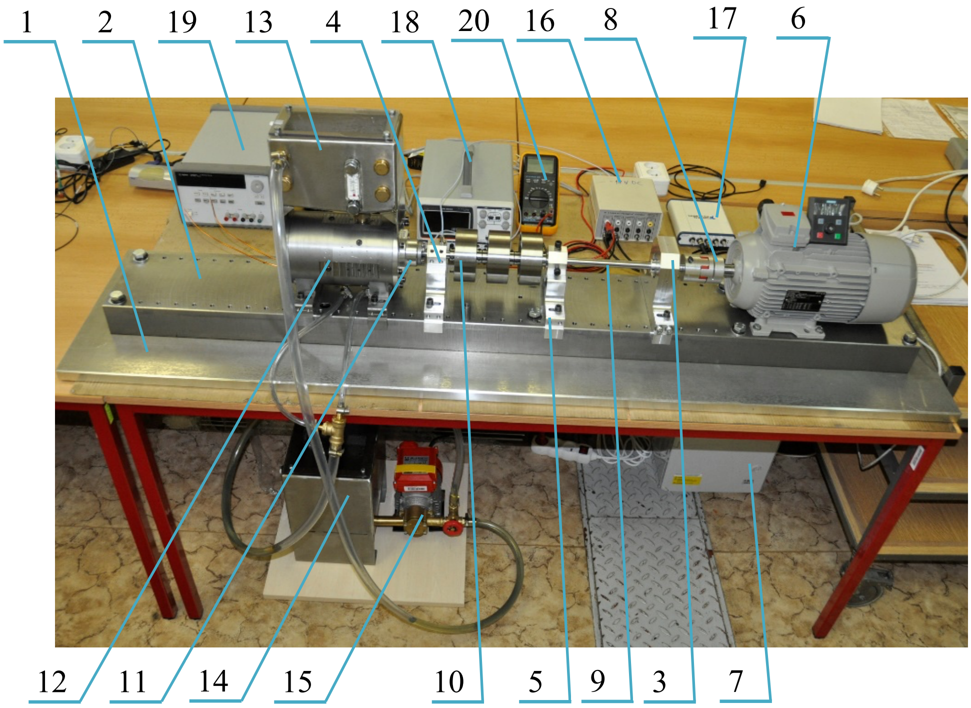

Fig. 2: Image of the designed experimental rotor rig with the hydrodynamical bearing.

The shaft displacement is monitored in two perpendicular directions by 3300 XL NSv eddy current proximity probes mounted in the probe stand with a slight offset to eliminate crosstalk. The probe produces a linear voltage signal proportional to the relative shaft displacement in the range of up to 1.5 mm. The Rotor Kit Proximitor Assembly (16) provides power to probes and amplifies the voltage signal from the probes. The time dependent voltage signals are connected to a five-channel data acquisition module (17), the NI USB-4432 with a sample rate of up to 102.4 kS/s. The time dependent displacements, shaft orbits, filtration of signals, and FFT spectrum are visualized in real-time by our custom application developed in the MATLAB software.

The GPE-2323 (18) and E3632A (19) power supplies are utilized for the powering of the probes and the electric coil in the housing of the hydrodynamic bearing. The average relative clearance between the shaft surface and the probe tips is set by the digital multimeter (20).

4. Measurement results of the dynamical properties of the designed hydrodynamic bearing

The experimental results depicted in Fig. 3 has been obtained for the hydrodynamic bearing lubricated with the MRF-122EG magnetorheological fluid. The proximity probes were positioned near the location of the bearing (see Fig. 2). The location of the shaft centre corresponds to the mean value of the time dependent displacement measured for the period of 10 s. An increase in the rotational speed of the rotor shifts the shaft centre to the bearing centre. The shift between the maximum and minimum rotor speed is approximately

160 μm in the vertical direction and 40 μm in the horizontal direction. The measurements were terminated when the upper rotation speed range approached the first bending mode shape. In Fig. 3 on the left, the '+' and '*' symbols mark the bearing centre path when 0.0 A and 0.7 A current is applied to the bearing coil, respectively. The curvature of the bearing centre path is governed by the direction of the rotor rotation. It is obvious that the raised current in the electric coil caused a change in the stiffness of the lubrication film and increased the bearing load capacity.

The measured orbits and locations of the shaft centre for a speed of 15 Hz are shown in Fig. 3 on the right. The coloured orbits are depicted for the raw measured data, and the black ones show the filtered orbits with the synchronous rotations. The rise of the current caused the change of the orbit shape and shift of the orbit centre closer to the bearing centre.

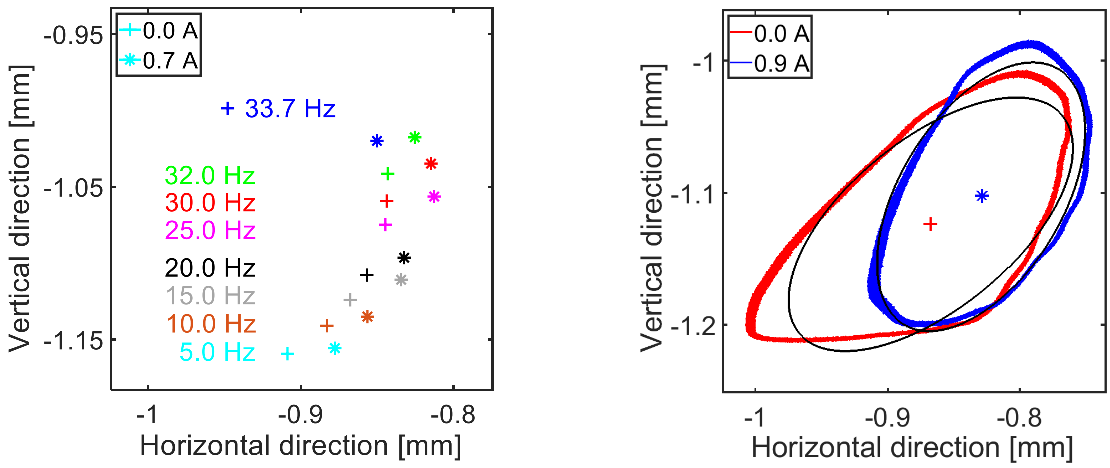

Fig. 3: Locations of the shaft centre (left) and trajectories of the shaft centre for a speed of 15 Hz (right).

5. Conclusions

This paper introduces a basic design overview of the magnetically sensitive hydrodynamic bearing. The proposed experimental rig has a modular character that allows to perform tests of various rotor-bearing configurations. The measurement results of the MRF-122EG magnetorheological fluid from LORD Corporation have been discussed. The experimental results show that the lubricating film with a magnetorheological fluid is sensitive to the variations of the current in the electric coil of the bearing. The rise of the magnitude of the applied current causes a shift of the rotor journal towards the bearing centre which means the bearing load capacity has been raised. The measured results with the developed magnetically sensitive hydrodynamic bearing are in accordance with the predicted behaviour obtained utilizing computational simulations.

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