Piezoelectric actuators in the active vibration control system of journal bearings

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Abstract. The advantage of journal hydrodynamic bearings is high radial load capacity and operation at high speeds. The disadvantage is the excitation of vibrations, called an oil whirl, after crossing a certain threshold of the rotational speed. The mentioned vibrations can be suppressed using the system of the active vibration control with piezoeactuators which move the bearing bushing. The motion of the bearing bushing is controlled by a feedback controller, which responds to the change in position of the bearing journal which is sensed by a pair of capacitive sensors. Two stacked linear piezoeactuators are used to actuate the position of the bearing journal. This new bearing enables not only to damp vibrations but also serves to maintain the desired bearing journal position with an accuracy of micrometers. The paper will focus on the effect of active vibration control on the performance characteristics of the journal bearing.

1. Introduction
The advantage of journal hydrodynamic bearings (alternatively called sleeve bearings or plain bearings for radial load) is high radial load capacity and operation at high speeds. The disadvantage is the excitation of shaft vibrations, called an oil whirl, after crossing a certain threshold of rotational speed which is dependent on the radial bearing clearance and the viscosity of lubricating oil. Once an unstable motion of the bearing journal occurs, the machine starts to vibrate, and its operating speed cannot be increased. Demonstration of the rotor instability during run-up is shown in figure 1.

Figure 1. Journal hydrodynamic bearings instability due to the oil film.
The rotational speed of the rotor increases linearly with time at the slew rate of 7k rpm per minute. Journal equilibrium position is achieved after 20 seconds from the start-up. The bearing journal is moved...
to the side while lifting up. A passive way how to suppress vibrations consists in adjusting the shape of the bearing bushing, such as lemon or elliptical bore of the bushing, a groove or tilting pads. Even though there are many solutions based on mentioned passive improvements, the approach to preventing the journal bearing instability is based on the use of the active vibration control (AVC) in the paper. To study possibilities of affecting rotor behaviour by controlled movement of bearing bushings, a test rig was designed, manufactured and assembled. The development work resulted in the design of the active bearings control system, which became the first functional bearing prototype in the known up to now [1]. In the introduction, it should be emphasised that research of active vibration control was aimed at rigid rotors and the standard design of cylindrical journal bearings, where the journal displacement is measured at the closest position to the bearing bushing.

1.1. Active vibration control

Vibrations of the shaft can be suppressed using the system for an active vibration damping with piezoelectric actuators to move the floating bearing bushing in two directions. The motion of the bearing bushing is controlled by the electronic controller, which responds to the change in position of the bearing journal with respect to the bearing housing. The mechanical arrangement of the actively controlled bearing is shown in figure 2. Two stacked linear piezoactuators are used to actuate the position of the bearing journal via the position of the bearing bushing as is shown in figure 3. The position of the journal or shaft is sensed by a pair of capacitive sensors with accuracy better than 1 micrometer. It works with a cylindrical bushing which did not require special technology of production and assembly. This new bearing enables not only to damp vibrations, but also serves to maintain the desired bearing journal position with an accuracy of micrometers.

1.2. Past developments

Many authors pay attention to the active control of journal bearings with the use of active magnetic bearings (AMB) [2] and giant magnetostrictive material (GMM) [3]. Although the authors of the paper on using GMM state that experiments can be carried out up to 1700 rpm, they publish only measurements at 350 rpm. The instability due to the oil film is a problem of high-speed rotors. With utmost probability, instability could not arise at such low rotational speed as 350 rpm. Therefore, the active vibration control was not aimed at eliminating instability of journal bearings, but only at positioning the shaft axis. Piezoactuators as a tool to control of rotating machines have been intensively investigated in the literature since the end of 1980's [4]. The linear piezoactuators can create a large force in a very small track. The advantage of journal bearings with piezoactuators is that the bearing bushing mounting stiffness remains unchanged in the case of a loss of electric power supply comparing AMB.

1.3. Test stand
The performance of the actively controlled bearing was tested on the test bench with the span of bearing pedestals of 200 mm. The bearing diameter is 30 mm and the length-to-diameter ratio is equal to about 0.77. The radial clearance of the journal is 55 micrometers. The instability of the oil whirl type starts at 2k rpm. The active vibration control extends stable operating rotational speed range up to 12k rpm. An inductive motor of 400 Hz drives the rotor and therefore the maximum rotational speed is 23k rpm. Input for the oil inlet is in the horizontal plane of symmetry of the bushing. The position of the journal is measured by a pair of the proximity probes which are capacitive sensors originated from the Microepsilon Company. The sensors are of the capaNCDT CS05 type with a measurement range of 0.5 mm. An advantage of the capacitive sensors is that it is not necessary to ground the shaft.

1.4. Piezoactuators

As was stated before the bearing bushings are actuated using the piezoactuators oriented in vertical and horizontal directions and fastened to the rig frame. The preloaded open-loop piezoactuators are of the P-844.60 type, the product of the Physik Instrumente Company. The piezoactuator require a low voltage amplifier with the 120 V peak value (LVPZT). The piezoactuator travel range is up to 90 micrometers, the pushing force is up to 3 kN and the pulling force is only up to 7 kN. The piezoactuator force balances the force effect of the oil film. Virtual motion of the unloaded piezoactuator is proportional to its control supply voltage \( u^* = kV \). The resulting motion \( u \) of the bearing bushing also depends on the load force as shown in the working graph of figure 4.

![Graph](image)

**Figure 4.** Mechanical part of the control loop

2. Control system and its effect on bearing operation

An electronic feedback is created for each of the two directions of the movement of the bearing journal in the bearing bushing. The control system is thus composed of two independent loops, each of which has its own controller. Both the controllers are of the proportional type. Although adding a derivative component improves the dynamic properties of the control loop, the corruption by noise of the proximity probes signal is the reason, for which the derivative feedback was not used. The controllers were created as a digital in the signal processor of the dSpace type. The input and output voltage of the DC amplifier is also shown in figure 5. The sampling frequency is chosen equal to 5 kHz.
2.1. Increasing the stable operating range of the bearing

The journal bearing has been tested at an increase in rotational speed until it becomes unstable and the journal motion is limited only by the bushing walls. The summary results are shown in figure 6. The dependence of the speed on time is shown in the left panel of figure 6, and the other three panels of this figure on the right show the dependence of the position of the bearing journal on time. An unstable motion of the journal is shown in the first of the upper mentioned three panels; the motion during active vibration control is on the second and third panel. The motion which is parametrically damped is the rightmost. The gain of the open loop of the control circuit was 35.

![Figure 6. Summary results of the active vibration control operation.](image)

The amplitude of the residual oscillation of the journal does not exceed 8 micrometers. Precision ball bearings (Deep groove ball bearings) which are offered by SKF have a radial internal clearance C2 to a diameter of 30 mm in the range from 1 to 11 micrometers. The maximum rotational speed of the 206-SFFC bearing type is only 7.5k to 13k rpm.

2.2. Reducing friction losses of the journal bearing

Power losses in the journal bearings were estimated from the electric power which is consumed by frequency converter and motor. Dependence of electrical power upon the rotational speed of the motor was measured with and without active control as it is shown in figure 7. The basic power consumption of the motor and frequency convertor was measured with the disconnected clutch between the motor and rotor; it means that the bearings were inoperative. The friction loss of a pair of bearings at 7k rpm is 66 W in an unstable operation, and if the active vibration control is ON, then the friction loss is of only 48 W. The active vibration control reduces the friction losses of journal bearings by 27 %. As a lubricant, the hydraulic oil of the OL-P03 type (VG 10 grade, kinematic viscosity 2.5 to 4 mm2/s at 40 °C) was used. All tests were undertaken at ambient temperature about 20 °C.
2.3. Increasing the radial stiffness of the journal bearing

The bearing bushing is suspended on a pair of piezoactuators and the bearing journal is supported by an oil wedge. According to the parameters in figure 4, the piezoactuator stiffness is of 33 MN/m. The stiffness of the O-ring seal is 5.5 MN/m. Force is transmitted to the bearing journal through the oil film. Based on the simulations it can be estimated that the direct stiffness of the oil film in the neighbourhood of the central position within the bushing bore is of 185 kN/m and the quadrature stiffness is still an order of magnitude larger and increases proportionally to the rotational speed. This stiffness increases by many orders of magnitude if the journal is approaching the bearing bushing wall. The stiffness of the journal support is defined first of all by the stiffness of the oil film. A steady-state error in a non-controlled bearing originates in a radial load which can be considered as a disturbance. A proportional controller which governs the system of journal bearings in the closed-loop with the open-loop gain of 35 reduces the steady-state error 36 times what results in the increase of the oil wedge stiffness 35 times compared to the design without feedback. Allowable forces, however, are limited by the load capacity of the piezoactuators. Notice that on the market there are piezoactuators enabling to generate forces up to 20 kN. The stiffness of the precision rolling bearings ranges from 100 to 200 kN/m, regardless of the load, while the stiffness of hydrodynamic bearings in neighbourhood of the central position (low load) is of the order of several kN/m. However with the active control, the stiffness can increase as much as 36-times, i.e. it can achieve values around of 100 kN/m, which is comparable to that of the precision ball bearing.

3. Conclusions

The advantage of journal hydrodynamic bearings is high radial load capacity and operation at high speeds. The disadvantage is the excitation of shaft vibrations, called an oil whirl, after crossing a certain threshold rotational speed which is dependent on the radial bearing clearance and the viscosity of lubricating oil. That provides these benefits as follow

- increasing the stable operating range of the bearing;
- reducing the friction losses of the journal bearing;
- increasing the radial stiffness of the journal bearing.

The cylindrical bushing does not require an exclusive technology of production and assembly. Electronic feedback improves the properties of the plain bearing and reduces the cost of production.

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