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EFFECTS OF ACTIVE VIBRATION CONTROL OF JOURNAL BEARINGS

ÚČINKY AKTIVNÍHO TLUMENÍ VIBRACÍ KLUZNÝCH LOŽISEK

Abstract

The article deals with the stability of the movement of the journal inside the bushing of the hydrodynamic plain bearings, which limits the operational range of rotor rotational speed. The purpose of active vibration control using piezoactuators is to increase the limit of the rotor rotation speed without onset of a whirl. It is also required maintaining the bearing journal position without vibration and reducing friction losses at the steady-state rotation. The main topic is the presentation of the results of the test of the journal bearing with the use of a test bench.

Abstrakt

Článek se zabývá stabilitou pohybu čepu uvnitř pouzdra kluzného hydrodynamického ložiska, která omezuje provozní rozsah otáček rotoru. Účelem aktivní kontroly vibrací pomocí piezoaktuatorů je zvýšit hranici rychlosti otáčení rotoru, aniž by vznikl vír, tj. kroužení osy rotoru kolem osy pouzdra. Je také zapotřebí udržet polohu čepu ložiska bez vibrací a snížit ztráty třením při ustálených otáčkách. Hlavním tématem článku je prezentace výsledků testu kluzného ložiska na zkušebním stavu.

Keywords

Active vibration control, journal bearings, plain bearings, sleeve bearings, oil whirl, piezoactuators, proximity probes.

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 Fig. 1. To study possibilities of affecting rotor behavior by controlled movement of bearing bushings, a test rig was designed, manufactured and assembled.

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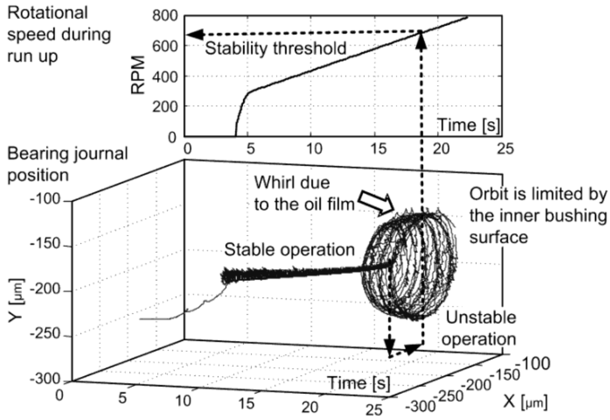

Fig. 1 Sliding hydrodynamic bearings instability due to the oil film

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, see Fig. 2. Even though there are many solutions based on mentioned passive improvements, the approach to preventing the journal bearing instability, presented in the paper, is based on the use of the active vibration control (AVC). In the introduction it should be emphasized that research of active vibration control was aimed at rigid rotors and the standard design of sliding bearings, where the journal displacement is measured at the closest position to the bearing bushing.

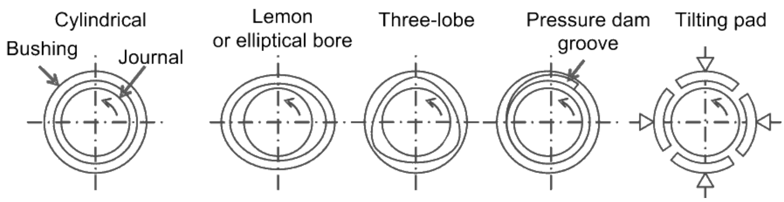

Fig. 2 Passive suppression of vibrations of the whirl type

Many authors pay attention to the active control of sliding bearings with the use of active magnetic bearings (AMB) [1] and giant magnetostrictive material (GMM) [2]. 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 sliding bearings, but only at positioning the shaft axis. Favorite means for rotor control are magnetorheological liquids, but the delayed response of the liquid viscosity to the magnetic field change does not allow using this method for the closed-loop control of high speed rotors [3]. Piezoactuators as a tool to control of rotating machines have been intensively investigated in the literature since the end of 1980's. 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 an accidental loss of electric power supply comparing AMB. One of the first original contributions dated from the beginning of the 1990's [4]. These papers did not study the effect of the oil film on the onset of instability and its suppression using the active vibration control. Worth mentioning are papers [5] and [6] dealing with the problem of the rotor instability.

Interest in active control of journal bearings has recently appeared at Graz University of Technology in Austria, where is aimed at reducing the bearing wear, and in China, where it is prepared such type of bearings for the spindle of machine tools [7].

There are two promising ways how to control the rotor movement, either by magnetic bearings or by piezoactuators, which are added to the system. However, AMB are still very expensive and cases, where it is possible to install AMB to already designed rotor, are very rare. As already mentioned, the ABM requires retainer bearings in case of power failure, which complicates machine design. On the other hand, installation of a piezoactuator into bearing housing is relatively easy. The non-rotating loose bushing can be inserted into any bearing housing. The only problem is to find piezoactuators for reasonable prize.

2 ACTIVE VIBRATION CONTROL OF JOURNAL BEARINGS

Vibrations of the shaft can be suppressed using the system for an active vibration damping with piezoelectric actuators to move the 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 Fig. 3.

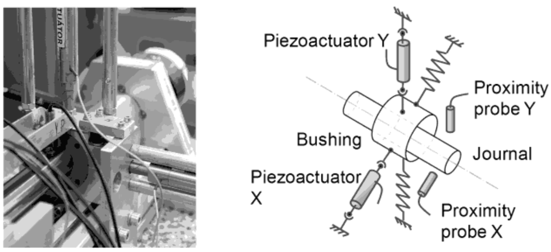

Fig. 3 Movable bushing inserted into the housing

Two stacked linear piezoactuators are used to actuate the position of the bearing journal via the position of the bearing bushing. The position of the journal or shaft is sensed by a pair of capacitive sensors. The system of the actively controlled journal bearings is the 1st functional prototype in the known up to now [8]. 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 accuracy of micrometers, which is important e.g. for grinding and milling spindles. Active vibration control is a novelty which faces to the mentioned instability of the journal bearings. The working prototype of the active vibration control system for the journal bearings was developed in the Czech Republic.

3 TEST STAND AND INSTRUMENTATION

The performance of the actively controlled bearing was tested on the test bench (Rotorkit) with the span of bearing pedestals of 200 mm. The bearing diameter is 30 mm and the length-to-diameter ratio is equal to one, see Fig. 4. An inductive motor of 400 Hz drives the rotor and therefore the maximum rotational speed is 23k rpm. An input for the oil inlet is in the horizontal plane of symmetry of the bushing. As is evident from the photo in Fig. 4 the supports of the piezoactuators were reinforced by additional bars. Bending and torsion load of the piezoactuators are excluded by using flexible tips and mounting procedure. The position of the bearing journal is measured by a pair of the proximity probes which are capacitive sensors originated from the MICRO-EPSILON 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. The magnitude of the capacitive sensor error is less than 1 μm. As was stated before the bearing bushings are actuated by means of 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 μm, the pushing force is up to 3000 N and the pulling force is only up to 700 N.

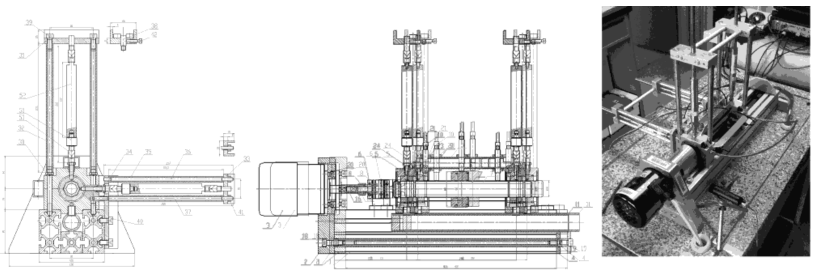

Fig. 4 Rotorkit

An electronic feedback which is shown in the diagram in Fig. 5 is created for each of the two directions of the motion of the bearing journal in the bearing bushing. The control system is thus composed of two independent control loops, each of which has its own controller. Both the controllers are of the proportional type. The controllers were created as a digital in the signal processor of the dSpace type. The sampling frequency is chosen equal to 5 kHz.

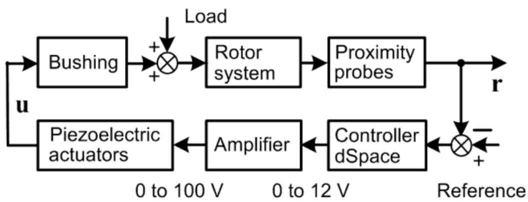

Fig. 5 Active vibration control system

The threshold of stability in angular velocity can be calculated by the Muszynska formula [9]

$$\Omega_{CRIT} = \sqrt{K/M}\,/\lambda, \qquad (1)$$

where:

K – stiffness of the oil film $\left[\frac{\mathrm{N}}{\mathrm{m}}\right]$,

M – rotor mass $[\mathrm{kg}]$,

λ – dimensionless parameter, which is slightly less than 0.5 $[-]$.

Effect of the position feedback on the stability limit was analyzed in the article [8]. It has been proven that if the open-loop gain K_P is positive, then the maximal rotational speed $\mathbf{\Omega}_{MAX}$ for the rotor stable behavior is as follows:

$$\Omega_{MAX} = \Omega_{CRIT}\sqrt{K_P + 1}.. \qquad (2)$$

4 INFLUENCE OF THE CONTROLLER PROPORTIONAL GAIN ON INSTABILITY THRESHOLD OF JOURNAL BEARINGS

The instability onset of the bearing journal motion inside the bushing arises when crossing the threshold value of rotational speed Eq. (2). This phenomenon means that the steady-state rotation of the journal is not stable and the journal axis starts to whirl at the frequency which is 0.42 to 0.48 multiple of the frequency of rotational speed of the rotor. The effect of active vibration control for the constant gain of the controller on the threshold of instability is presented in this chapter. Rotor speed increases according to a ramp function as it is shown in the left panel of the Fig. 6. The time history of the axis coordinates of the bearing journal is shown in the right panel of the Fig. 6. The -coordinate corresponds to horizontal direction and the -coordinate is for vertical direction. Active vibration control is switched OFF for this measurement. Instability occurs at about 2k rpm. Oscillations of the bearing journal are limited by the journal clearance within the bushing. Measurements in this article were carried out on the shaft with the radial clearance of 55 μm. The time histories of the axis coordinate of the bearing journal are also shown in Fig. 7. The active vibration control for these measurements is switched ON. The controller gain for the measurement in Fig. 7 is the maximum value we used for the tests. Instability of the bearing occurs at the rotational speed about 12k rpm. Vibrations during the instable motion of the journal are also limited by the journal clearance within the inner gap of the bearing bushing.

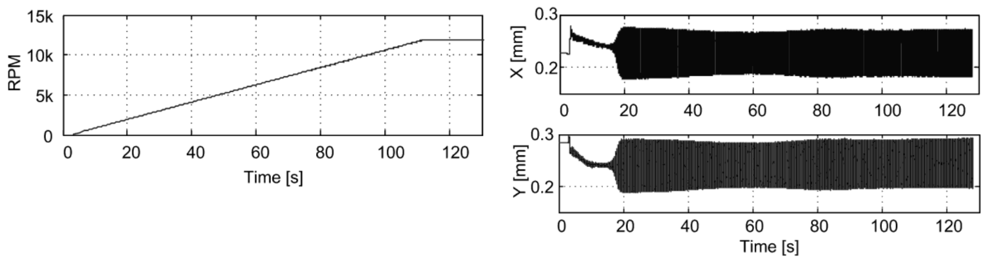

Fig. 6 RPM and the journal position as a function of time for tests without active vibration control

The active vibration control is not turned ON at zero rpm of the rotor but after finishing a transient process, which ends by lifting the journal to approximately the middle position in the vertical direction which takes approximately 15 seconds for the given rate of the increase of speed. The transient of the journal seems to be reverse for the vertical motion of the bearing journal in the right panel of the Fig. 6 and in the left panel of the Fig. 7. The scale for the vertical motion is reverse in these figures, meaning upside-down. The relationship between horizontal and vertical movement of the journal shows the orbit in the right panel of Fig. 7. The orbit is a trajectory of the journal axis. The shape of the orbit is approximately circular when instability occurs. The maximum rotational speed for stable behavior of the journal in the bearing bushing increases to six times using feedback. According to Eq (2) this value corresponds to the open-loop gain which is equal 35.

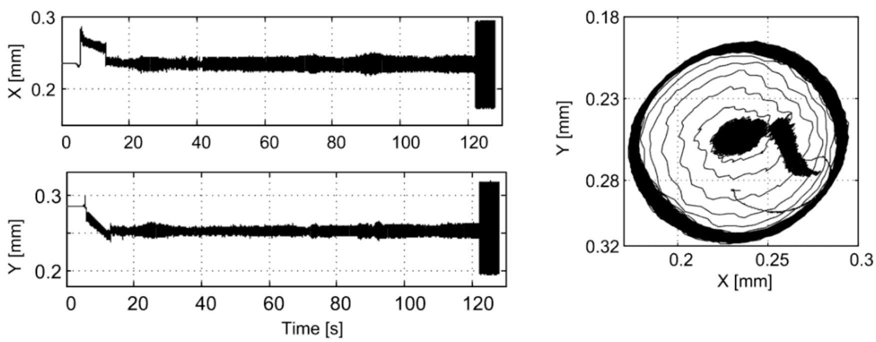

Fig. 7 RPM and the journal position as a function of time for tests with active vibration control ON

5 INFLUENCE OF THE PARAMETRIC EXCITATION ON INSTABILITY THRESHOLD OF JOURNAL BEARINGS

The linear proportional controller was used for active vibration control in the previous chapter. Parametric excitation means that at least one parameter of the system varies periodically in time according to a sinusoidal function. The gain of the proportional controller was selected as this varying parameter. The system becomes non-linear and non-stationary. The gain of the proportional controller is given as follows:

$$K_P(t) = K_{P0}\left(1 + \alpha \sin(\omega_0 t)\right), \tag{2}$$

where:

α – dimensionless amplitude of excitation $[-]$,

K_{P0} – static gain factor $[-]$,

ω_0 – angular frequency of excitation $\left[\frac{\text{rad}}{\text{s}}\right]$.

Dohnal [10] has solved a similar problem for magnetic bearings. Our experiments on the test bench were conducted for the following amplitudes of excitation α = 0, 0.1, 0.15, and 0.2. The static gain was the same as the gain of the previous experiments with the linear controller. The excitation frequency was selected 30 Hz, which is approximately equal to the frequency of vibration at the low rpm. Rotor speed increases according to a ramp function as is shown in the left panel of the Fig. 8. Results of measurement are shown in the two panels as in the previous figures. The time histories in the right panel of the Fig. 8 correspond to an operation when the active vibration control is OFF. The effect of the amplitude of the parametric excitation on the journal movement during rotational velocity run up is shown in Fig. 9. The best choice of the excitation amplitude is α = 0.15, for which is the position of the journal almost without oscillations. The amplitude of the residual oscillation of the journal does not exceed 8 μm. Precision ball bearings (so called Deep groove ball bearings) which are offered by SKF have a radial clearance (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.

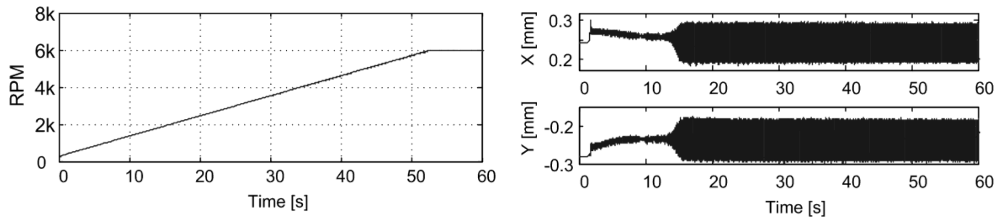

Fig. 8 RPM and the journal position as a function of time for tests with active vibration control OFF

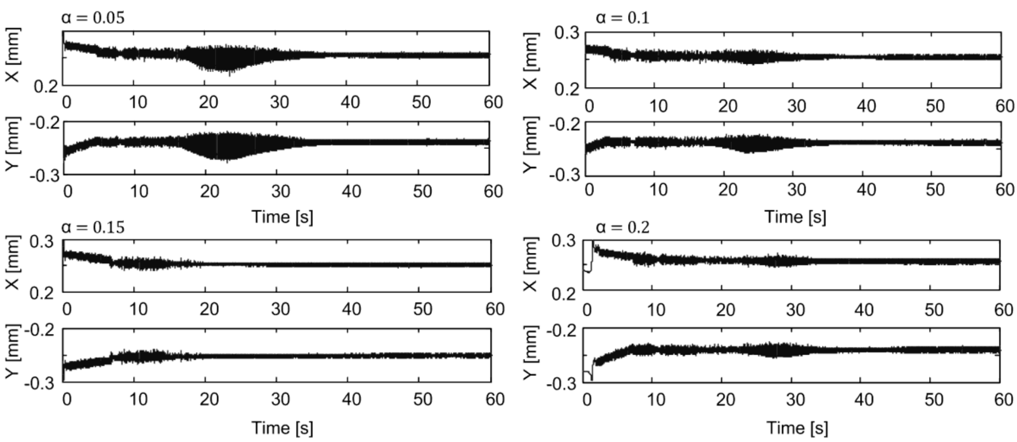

Fig. 9 The journal position as a function of time for tests with active vibration control ON

6 REDUCING MECHANICAL POWER LOSSES IN ACTIVELY CONTROLLED BEARINGS

Power losses in the journal bearings were estimated from the electric power which is consumed by frequency convertor and motor. Dependence of electrical power upon rotational speed of the motor was measured with and without active control as it is shown in Fig. 10. 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 7000 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 sliding bearings by 27 %. The bearing clearance is of 90 μm for the bearing journal of the diameter 30 mm. As a lubricant the hydraulic oil of the OL-P03 type (VG 10 grade, kinematic viscosity 2.5 to 4 mm^2/s at 40 °C) was used. All tests were undertaken at ambient temperature about 20 °C. For small power loss by friction in the bearings the actively controlled bearings can be used in systems for storing the kinetic energy as they are flywheels that spin at high speed. Longer life compared with roller bearings, it is another advantage of this type of bearings [7].

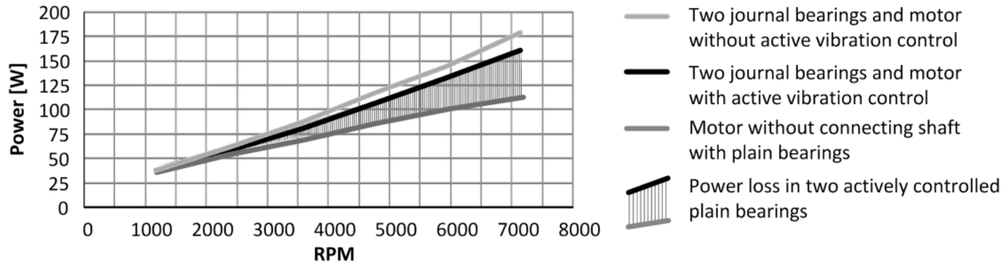

Fig. 10 The electric power consumed by the frequency convertor, motor and bearings

7 STIFFNESS OF ACTIVELY CONTROLLED BEARINGS

The rotor of the test bench has a mass of 1.5 kg, and therefore is under influence of the gravity and centrifugal forces due to the rotor unbalance. During active vibration control a dynamic force induced by the piezoactuators acts at the bearing bushing. The magnitude of this force is in the range from 100 to 200 N and has relatively stable amplitude. Users of the actively controlled journal bearing will be interested to know the maximum value of the radial force which is allowed.

The bearing bushing is suspended on a pair of piezoactuators and the bearing journal is supported by an oil wedge. According to catalogue data we used a linear piezoactuator which is able to generate force of 3000 N in pressure or 700 N in tension on the track in the range of 90 μm. These parameters correspond to the piezoactuator stiffness of 33 MN/m. Stiffness of the sealing O-rings 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 stiffness of the oil film in neighborhood of the central position within the bushing bore is of 4 kN/m. This stiffness increases by many orders of magnitude if the journal is approaching the bearing bushing wall. Stiffness of the journal support is defined first of all by stiffness of the oil film. A steady-state error in a non-controlled bearing originates due to a radial load which can be considered as a disturbance. A proportional controller governing a system of journal bearings in the closed-loop with an open-loop gain K_P reduces the steady-state error (K_P + 1) times what results in increase of the oil wedge stiffness (K_P + 1) times compared to the design without a feedback. An integration controller reduces steady-state error to zero, which corresponds to a theoretically infinite stiffness. Allowable forces, however, are limited by the load capacity of the piezoactuators. The pressure force is thus less than 3 kN. Notice that on the market there are piezoactuators enabling to generate forces up to 20 kN.

8 CONCLUSIONS

Experiments prove the correctness of the theoretical prediction which refers to the extending of the operating range of plain bearings when active vibration control is used. The experiments with the time-periodic changes of the controller gain confirm the positive effect on the vibration response. The control system maintains the bearing journal at the chosen position with the same precision as the precision ball bearings. The actively controlled journal bearings reduce the mechanical power losses by friction.

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