The paper discusses the results of testing a prototype of the active vibration control of radial sliding bearings with the use of piezoactuators. The main aim of active control is to prevent instability due to the oil film and thus to extend the operating speed range of a bearing journal. The effect of the controller gain and the amplitude of the parametric excitation on the operating speed range will be demonstrated by experiments.

1. Introduction

The speed range of rotors supported by hydrodynamic journal (sliding) bearings limits a phenomenon which is known as the instability due to the oil film. After crossing a certain speed threshold the bearing journal starts to whirl and the whole machine starts to vibrate. Passive measures against machine vibration are based on modification of the geometric profile of the bearing bushing. The bushing having a lemon or elliptical or multilobe bore is introduced. It is possible to suppress instability with the use a pressure dam or tilting pads. 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 sliding bearings was developed in the Czech Republic under the grant projects which were sponsored by the Czech Science Foundation [1]. Research work was carried out at the VSB Technical University of Ostrava and the research company TECHLAB Ltd.

The active vibration control of the journal bearings uses the bushing which can move freely and is not rotatable. The position of the bushing is governed by a pair of mutually perpendicular piezoactuators and the position of the journal is measured by a pair of the proximity probes. The sketch of the controllable journal bearing arrangement is shown in Fig. 1. Symbols of the spring elements in this Fig. 1 denote the elastic O-rings. The O-rings together with the piezoactuators determine the stiffness of the connection of the bushing with the bearing housing.

With regard to future industrial applications, we focused on rigid rotors and a standard type of sliding bearings with the only exception that is possibility to control the bushing position with respect to the bearing housing. The journal movement in two directions is measured as close as possible to the bearing bushing. Some publications deal with the vibration of flexible rotors, which have a large deformation in the middle between the bearings. Such a large displacement is easily mea
able by eddy-current sensors [2] [3] in contrast to the movements of the rigid rotors, which can be measured only with the use of capacitive sensors.

Many authors pay attention to the active control of the sliding bearings with the use of active magnetic bearings (AMB) [4] and giant magnetostrictive material (GMM) [5]. Although the authors of the paper which deals with the use of 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 could not arise at this low rotational speed. Therefore, the active vibration control was not aimed at eliminating instability of sliding bearings. Magnetorheological liquids are attracting attention to the vibration damping of the rotor as well, but the delayed response of the liquid viscosity to the magnetic field change is in the order of ten milliseconds that does not allow using this method for the real-time control of the high speed rotors [6]. Piezoactuators as a tool to control of rotating machines have been intensively investigated in the literature since the end of 1980's. The advantage of journal bearings with piezoactuators compared to AMB is that the bearing bushing mounting stiffness remains unchanged in the case of an accidental loss of the electric power supply. One of the first contributions is dated from the beginning of the 1990's [7].

This paper focuses on the way how to transform information on the position of the journal to the voltage supply of piezoactuators in real time. The input of the controller is the position of the journal and the output is the voltage which powers the piezoactuators. The controller is chosen to be of the proportional type. The effect of a constant controller gain on vibration damping is first presented, and then the effect of parametric excitation is discussed. Influence of the controller gain on the speed limit for stable operation of the rotor has been published previously [1] but with less stiffness of the piezoactuator holder than we use for testing now. A fundamental research in the field of instability of the non-linear mechanical systems was done by Tondl [8]. We focused on the problem of the active vibration damping of rotors with the use of parametric excitation of journal bearings. The research work was carried out under the project No. P101/12/2520 which was funded by the Czech Science Foundation. The results are described in the second part of the paper.

2. Test stand design

The longitudinal section of the test stand is shown in Fig. 2 while some details can be seen on photos in Fig. 3. The test stand consists of the rigid shaft of 30 mm diameter supported in two cylindrical hydro-dynamic journal bearings of the length of 30 mm. An input for the oil inlet is in the horizontal plane of symmetry of the bushing. The results of the experiments are presented only for the radial clearance of 55 μm. The bearing span is 200 mm. An inductive motor of 400 Hz drives the rotor and therefore the maximum rotational speed is 23k RPM.

As is evident from the photo on Fig. 3 the supports of the piezoactuator were reinforced by additional bars. Bending and torsional load of the piezoactuators are excluded by using flexible tips and mounting procedure.
3. Test stand instrumentation

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. In contrast to the capacitive sensors, the eddy-current sensors produce the error of the magnitude of around 10 microns. This error can be explained by the inhomogeneity of the surface layer of the rotating bearing journal.

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 100 V peak value at the output (LVPZT). Control voltage for the piezoactuators can only be positive. 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. The resonant frequency of the piezoactuators is of order thousands of Hertz while the frequency range for transferring a control signal with no distortion is required to be of order hundreds of Hertz.

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 noisy signal produced by the proximity probes is the reason, for which the derivative feedback was not used.

The controllers were created as a digital in the environment of the signal processor of the dSpace type which is shown in the diagram in Fig. 4. The input and output voltage of the DC amplifier is shown also in this figure. The sampling frequency is chosen equal to 5 kHz.

4. Closed control loop

A variable $u$ in Fig. 4 is a control variable and a variable $r$ is a controlled variable. The controlled variable is a two-component coordinate of the bearing journal axis while the control variable is a two-component coordinate of the bushing axis as is shown in Fig. 5. Because both the variables indicate coordinates in the plane, then they can be considered as two-component vectors. The same meaning as the vector has a complex variables. The real part of this variable has the meaning of the $X$-coordinate while the imaginary part has the meaning of the $Y$-coordinate, therefore it is possible
to denote it as $r = X + jY$. The origin $(0, 0)$ of the coordinate system in the complex plane is situated in the center of the mentioned cylindrical bearing bore.

Figure 4. Active vibration control system.

Figure 5. Coordinate system in the complex plane.

There are many ways how to model journal bearings, but this paper prefers a lumped parameter model, which is based on the concept developed by Muszynska [9]. This concept assumes that the oil film acts as a combination of the spring and damper, which rotates at an angular speed $\lambda \Omega$ (variables will be defined hereinafter). The reason for using this concept was that it offers an effective way to understand the rotor instability problem and to create a model of a journal vibration active control by manipulating the bushing position with the use of actuators, which are a part of the closed loop system composed of the proximity probes and the controller. The correctness of this approximation confirms Mendes and Cavalca [10]. The entries of the stiffness and damping matrices according to Muszyńska’s model and the calculation of these matrix entries using Reynold’s equation agree. Some entries are constants and others are linear functions of the rotational speed. The equation of motion in the complex form for the rigid rotor operating in a small, localized region in the journal bearing, is as follows [1]

$$M \ddot{r} + D (\dot{r} - \dot{u}) + (K - jD\lambda \Omega) (r - u) = F_p,$$

where $M$ is the total rotor mass, $\Omega$ is the rotor angular velocity, $K$ and, $D$ are specifying proportionality of stiffness and damping to the relative position of the journal center displacement vector, $\lambda$ is a dimensionless parameter, which is slightly less than 0.5 and $F_p$ is a disturbing force.

Active vibration control of journal bearings uses the bushing position as the control variable $u$ and the shaft position as a controlled variable $r$. The control variable is an output of a controller. The controller transforms an error signal computed as a difference of a reference and actual position of the journal. As is evident from the block diagram in Fig 6, the controller is of the proportional type with the gain $K_p$.

The Laplace transfer function relating the displacement of the bushing to the displacement of the shaft is given by

$$G_s(s) = \frac{D s + (K - jD\lambda \Omega)}{M s^2 + D s + (K - jD\lambda \Omega)}.$$
The stability margin can be calculated under assumption that the open-loop frequency transfer function $G_0(\omega) = K_p G_s(\omega)$ of the control loop in Fig. 6 is equal to -1. The frequency of the steady-state vibration at the stability margin is given by $\omega = \lambda \Omega$ and $K_p = \omega^2 M/K - 1$. If the feedback gain $K_p$ is positive then the maximal rotational speed $\Omega_{MAX}$ for the rotor stable behavior is as follows

$$\Omega_{MAX} = \Omega_{CRIT} \sqrt{K_p + 1}.$$  

This formula enables to calculate the critical rotational speed of the rotor without any control $K_p = 0$. Increasing of the stability margin for the rotational speed of the rotor is possible by introducing an additional feedback.

5. Effect of the proportional gain on instability threshold

The instability onset of the bearing journal movement inside the bushing arises when crossing the threshold value of rotational speed (3). This phenomenon means that the steady-state rotation of the journal after increasing rotational speed 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 tests of the active vibration control are performed in order to determine the threshold of the instability of the journal bearings. The best way to determine this threshold is the steady increase rate in the speed of rotation according to a linear function, called a ramp function.

![Figure 7. RPM of rotor as a function of time for tests with a constant controller gain.](image)

![Figure 8. Time history of the journal axis coordinates when the active vibration control is OFF.](image)

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 in Fig. 7. The first time history of the axis coordinate of the bearing journal is shown in Fig. 8. The X-coordinate corresponds to horizontal direction and the Y-coordinate is for vertical direction. Active vibration control is turned off for this measurement, $K_p = 0$. Instability occurs at about 2k RPM. Oscillations of the 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. 9. Active vibration control for these measurements is turned on. The controller gain for the measurement in Fig. 9 is the maximum value we used for the tests. Instability occurs at the rotational speed about 12k RPM. Oscillations at the instability movement of the journal are also limited by the journal clearance within the bushing.

As can be seen the active vibration control is switched on after finishing a transition time interval, which ends by lifting the journal to approximately the middle position in the vertical direction which takes approximately 15 seconds. The transient of the journal is reverse for the vertical movement of the bearing journal in Fig. 8 and 9. The scale for the vertical motion is reverse in this
figures, meaning upside-down. The rotor is supported by two journal bearings, active vibration control, however, was used only for the free end of the shaft which is the one that is more distant from the motor. Extending the operating speed range is possible also for active vibration control only in the horizontal direction.

The relationship between horizontal and vertical movement of the journal shows the orbit in Fig. 10. The orbit is a trajectory of the journal axis. The shape of the orbit is approximately circular when instability occurs.

![Figure 9. Time history of the journal axis coordinates when the active vibration control is ON.](image1)

![Figure 10. Orbit plot of the journal axis when the active vibration control is ON.](image2)

6. **Effect of the parametric excitation on instability threshold**

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

\[
K_p(t) = K_{p0}(1 + \alpha \cos(\omega_b t))
\]

where \( \alpha \) is a dimensionless amplitude of excitation, \( K_{p0} \) is a static gain factor and \( \omega_b \) is angular frequency of excitation. Dohnal has solved a similar problem for magnetic bearings [11]. The experiments on the test stand were conducted for the following amplitudes of excitation

\[
\alpha = 0, \ 0.05, \ 0.1, \ 0.15 \text{ and } 0.2.
\]

The static gain was the same as the gains of the previous experiment with the constant gain. 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 shown in Fig. 11. Results of measurement are shown in the two panels as in the previous figures. The time histories in Fig. 12 correspond to a mode when the active vibration control is off.

The effect of the amplitude of the parametric excitation on the journal movement during RPM run up is shown in Figs. 13 to 16. The best choice of the excitation amplitude is \( \alpha = 0.15 \), for which is the position of the journal almost without oscillations. The amplitude of the residual oscillation of the journal does not exceed 10 \( \mu \)m. This amplitude is comparable with the radial clearance of the ball bearings of the deep groove type.
7. Conclusions

The paper describes the results of two grant projects granted by the Czech Science Foundation. The last project No. P101/12/2520 was directed to the use of parametric vibration damping for active vibration control of the journal bearings with the use of the piezoactuators. The first project 101/07/1345 dealt with active vibration control as a problem of linear control theory, while the second project was focused on the controller which gain varies periodically in time according to a sinusoidal function. To analyze the efficiency of the rotor vibration damping it is necessary to use either a technique of mathematical simulations or experiments which were preferred. In the paper the results of experiments with time-periodic varying the controller gain which amplifies the error of the journal position to be transformed to the voltage at the output of the controller.
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