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ACTIVE VIBRATION CONTROL OF JOURNAL BEARINGS TO PREVENT INSTABILITY CAUSED BY AN OIL FILM

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Summary: *The paper deals with a working prototype of a system for the active vibration control of journal bearings to prevent rotor instability due to the oil film. The controllable journal bearing consists of movable bushing, the position of which is actuated with the use of piezoactuators. The first tests show that the active vibration control extends considerably the range of the operational RPM.*

1. Introduction

Rotor instability due to the oil film is one of the most serious problems of high-speed rotors supported in journal bearings. With constantly increasing parameters of new machines, problems with rotor instability are encountered more and more often. To study possibilities of affecting rotor behavior by controlled movement of bearing bushings, a test stand was designed, manufactured and assembled. Even though there are many solution based on passive improvements of the bearing geometry to enlarge the operational RPM range of the journal bearing as a lemon bore, pressure dam, tilting pad, etc., the approach to preventing the journal bearing instability, presented in the paper, is based on the use of the active vibration control.

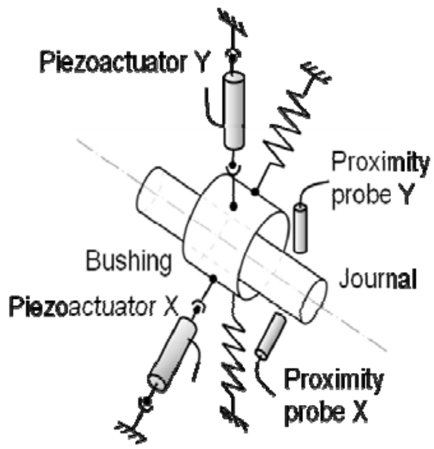

Figure 1 Arrangement of the controllable journal bearing

2. A test stand for research active vibration control

Many authors pay attention to the active control of magnetic bearings, but use of piezoactuators is rarely studied. Worth mentioning are articles [Carmignani et al, 2001] and [Rho & Kim 2002]. Because of the lack of information, it was decided to start research of methods suppressing sliding journal bearing instability with by active vibration control. The research work was granted by the Czech Grant Agency as a part of the research project No. 101/07/1345 “Active control of journal bearings aimed at suppressing the rotor instability”. The control system adds an electronic feedback to the rotor-bearing system by active control of a movable bushing. The current passive damper changes into an active component of the system with controllable properties. Objectives of the research work can be broken into the following tasks:

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1. Experimental verification of theoretical knowledge about conditions of the onset of rotor instability.
2. Building up an experimental device used for tests of active control of the journal bearings, with the aim of suppressing rotor instability.
3. Algorithm of stabilization of shaft movements inside the journal bearing, verified on the experimental device.

The project ended in 2009. This paper informs about results of the tests of the active vibration control system. A laboratory test facilities including the journal bearing equipped with the movable bushing was designed by the TECHLAB Ltd., Prague. Technical University of Ostrava developed the control system, which is based on piezoactuators, and put the system into operation [Šimek et al, 2010] [Tůma et al, 2010].

The sketch of a controllable journal bearing arrangement, which is implemented for the active vibration control, is shown in figure 1 while a technical drawing can be seen in Fig. 2. The test stand in figure 2 consists of a rigid shaft 7 supported in two cylindrical hydrodynamic journal bearings. Bearing bushings are supported in rubber “O” rings, which ensure sealing of oil inlet and at the same time enable movement of bushings within the clearance in bearing casing. Bearing bushings can be excited by means of piezoactuators 12 oriented in vertical and horizontal directions and fasted to the frames 13 and 14. The preloaded open-loop LVPZT piezoactuators are of the P-842.40 and P-844.60 type. They require a low voltage amplifier with the output voltage of 100 V. The pushing force produced by the P-842.40 type is of 800 N and the pulling force only 300 N. The displacement is up to 90 µm. The same displacement reaches the LVPZT piezoactuator of the P-844.60 type while the pushing force of the P-844.60 type is up to 3000 N and the pulling force is up to 700 N. The test shaft is driven by high-frequency motor 3 through elastic coupling 6, constituting two joints, so that the shaft is decoupled from motor and free to move.

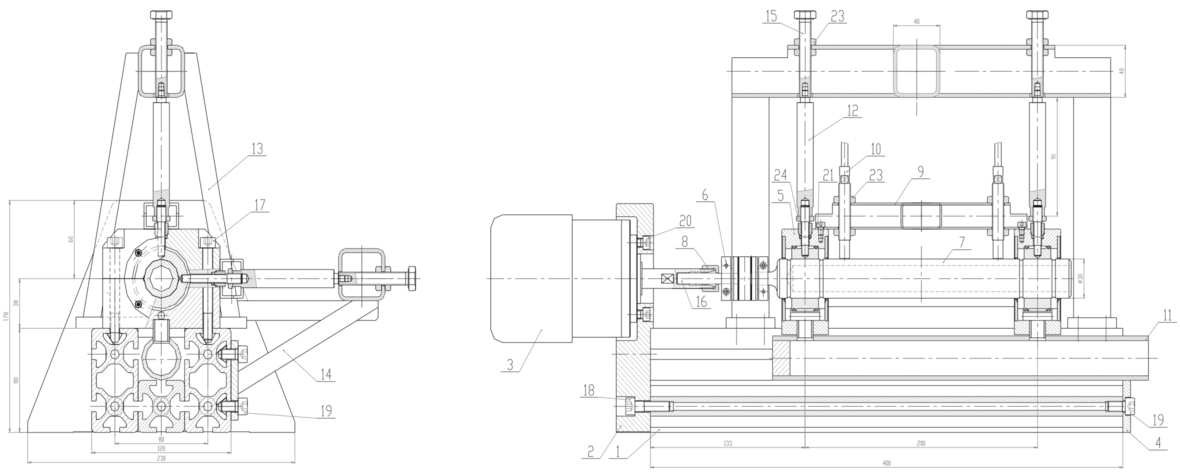

Figure 2 Test stand cross section

Shaft movement is observed by two pairs of relative sensors 10, working on eddy current principle. Sensors IN-085, supplied by the firm Brüel & Kjær, alternatively with sensors from Bently Nevada Rotor Kit were tested. Both these sensors work on the principle of eddy current. It is also possible to put one or two discs on the shaft, thus increasing bearing load and rotor mass. However, lowest stability limit should be achieved with the minimum bearing load, i.e. with hollow shaft without discs. Test stand was designed for speeds up to 23.000 rpm; maximum achieved speed was about 20.000 rpm, although calculated bearing losses are much lower than proposed drive motor output.

3. Operation without active control

When putting into operation of the test stand, we met these problems

- choice of lubricating oil determining the journal looses due to the friction
- measurement accuracy of shaft position
- mounting of piezoactuators to avoid torsional loading and enable adjusting position at the accuracy of micrometers.

To reach the maximum motor speed higher than 6000 RPM it was necessary to increase bearing clearance to 90 μm with simultaneous decrease of calculated rotor stability limit. As a lubricant was initially used the hydraulic oil of the VG 32 class (kinematic viscosity of up to 32 mm^2/s at 40 ^0C) and then bearing special oil for high-speed grinder spindle bearing of the OL-P03 type (kinematic viscosity 2.5 to 4 mm^2/s at 40 ^0C). The hydraulic oil enabled to reach the maximum rotational speed of 16000 RPM and the instability onset at the same value of RPM. The bearing special oil enabled reach the motor maximum nominal rotational speed 23600 RPM while the instability onset starts at 4300 RPM. All tests were undertaken at ambient temperature about 20 ^0C. Lubricating oil was not preheated during tests.

The first measurement showed the shape of the rotor centerline trajectory (orbit) considerably differing from a circle or an ellipse. To explain this phenomenon, the reasons were looked for in the uniformity of motor rotation, the misalignment of the motor and rotor axes, the oil pump and the interference of the proximity probe output signals. To achieve perfect decoupling of the test shaft from driving motor, another flexible coupling was installed between original coupling and the shaft. The rotation uniformity is not as smooth as when a DC motor is used. The interference between signals causes an error below 1 μm in measurements. Inspection of shaft geometrical deviations as it is a non-circularity error showed that this deviation is less than 1 μm as well. Finally it was proved that the non-homogeneity of the rotor material magnetic properties is the main source of the proximity probe periodical error. The error signal is repeated synchronously with rotor rotation. The same material is passing-by below the tip of the proximity probe after one quarter of the rotor revolution; therefore a phase shift of the period quarter may be discovered. It proves that the proximity probe output depends not only on distance from the shaft surface but on magnetic end electric property of the rotor ferrous material. Peak-to-peak value of the regular periodic error is equal to 11 μm. The spectrum of the measurement periodic error is composed from harmonics of the rotor rotational frequency. To avoid this measurement error of the proximity probe the moving average corresponding to one complete revolution was evaluated.

The time history of the rotor centerline displacement during run up with extremely low viscous oil VG10 without preheating is shown in figure 3. For clockwise rotation the journal centre trajectory in bearing supporting the free end of the rotor is shown in figure 4. It is technically impossible to increase the rotor rotational speed smoothly from 0 RPM. Rotor speed is changed by jumping on the value of 230 RPM and then continuously increases up to the rotational speed meet the rotor instability condition. The journal starts at the bottom of the bearing sleeve and with increasing speed it moves up in direction of rotation. At the level of the sleeve centre the journal starts to move towards the centre of the bearing. With infinite speed or zero load the journal centre coincidences with bearing centre, which is generally unstable position in circular bearing. The instability onset occurs at about 4300 RPM.

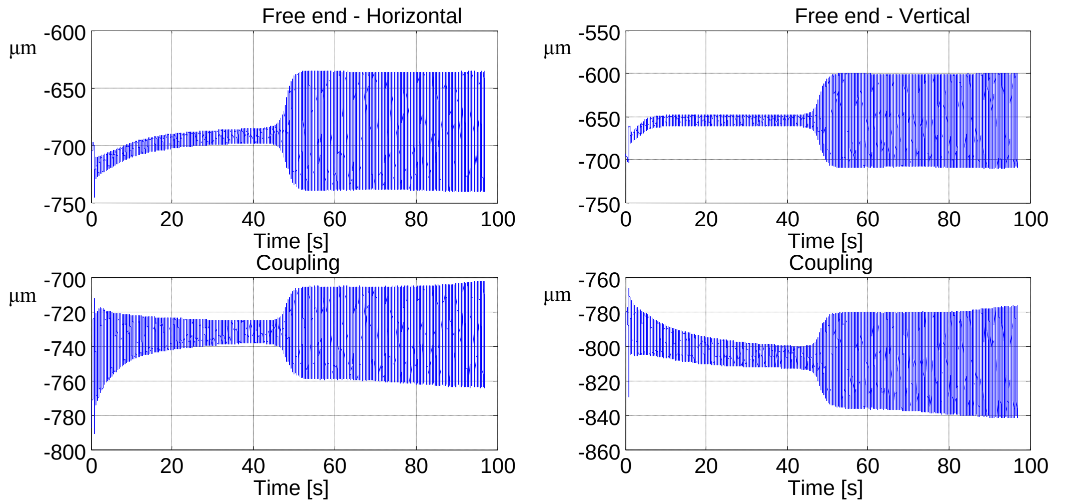

Figure 3 Time history of the rotor centerline displacement during run up with extremely low viscous oil VG10

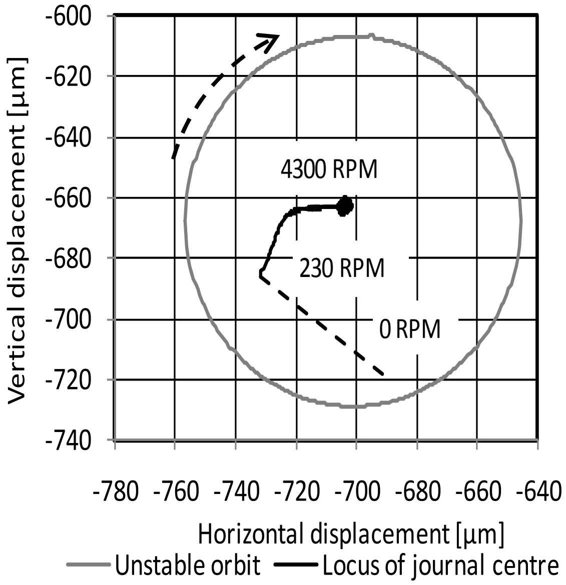

Figure 4 The Journal center trajectory during rotor run up (free end of rotor)

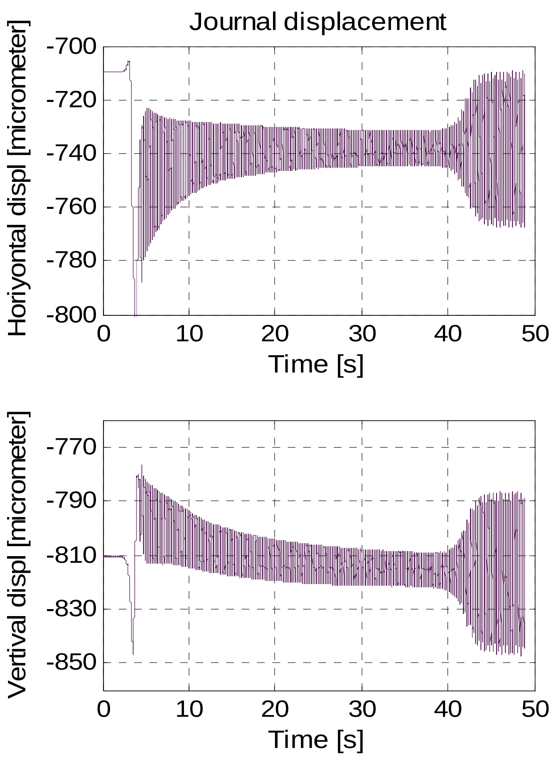

Figure 5 Time history of the journal center trajectory during rotor run up (end at the driving motor)

As shown in figure 5, the rotor end connected to the driving motor via a coupling, becomes unstable immediately after starting run-up and vibrates with frequency slightly less than half of the rotor rotational frequency. The journal centerline is above the bearing axis and in asymmetric position with respect to the vertical bearing symmetry plane due to the force resulting from bending stiffness of the membrane elastic coupling. Vibration of the whirl type is slowly damped without a tendency to magnify. The rotor is pushed up because of its low weight.

4. Piezoactuators mounting

Choice of the piezoactuator type was verified by measurement of the dependence of acting force on the open-loop piezoactuator travel. The result of the piezoactuator loading by a rod, acting at the bushing in the horizontal direction, is shown in figure 6. As the measurement shows, the piezoactuator load capacity is satisfying. A flexible tip was used to attach the piezoactuator to the bushing rod and the frame structure for compensation of misalignment and possible bending load.

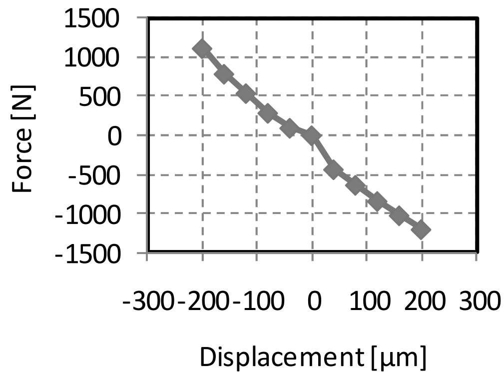

Figure 6 Dependence of piezoactuator force on displacement

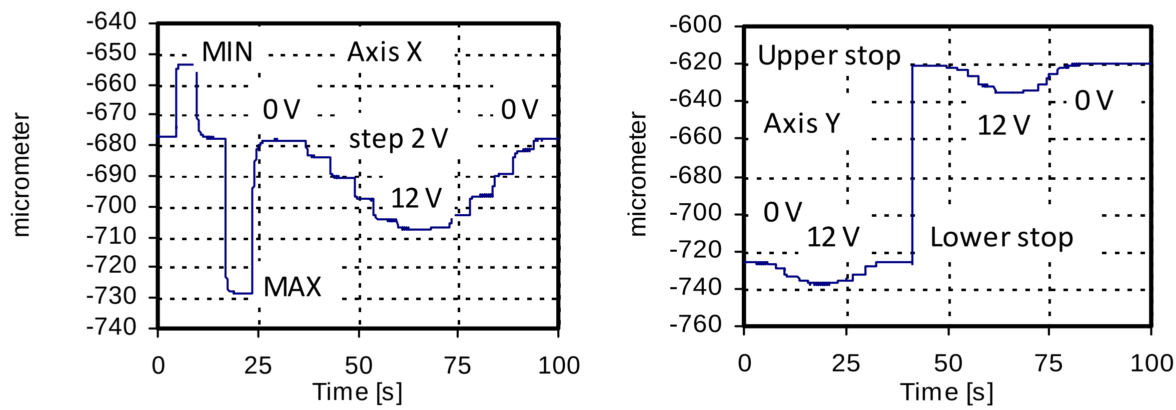

Figure 7 Effect of the amplifier input voltage on the shaft position change in the horizontal and vertical direction

The test stand instrumentation allows active vibration control only in the journal bearing at the opposite side to the driving motor. Before the operational tests begin, the initial position of the piezoactuators has to be adjusted in the middle position of the operating travel range of the piezoactuators. This position corresponds to half the output voltage of the controller, the full range of which is equal to 12V. A screw at a holder is tightened in this position. The range of the shaft displacement for the full scale of the controller output voltage is shown in figure 7 for the horizontal directions and in figure 4 for the vertical direction of the displacement.

As it was mentioned the signal from the proximity probes is connected to the dSpace signal processor. The output of the signal processor is connected to the input of the amplifier that powers the piezoactuator. The electronic feedback (see figure 8) in the presented experiments was of the proportional controller type. Although improved dynamic properties of the control loop require adding a derivative component, the noisy signal produced by the proximity probes is the reason why the derivative feedback was not used [Víteček et al, 2008].

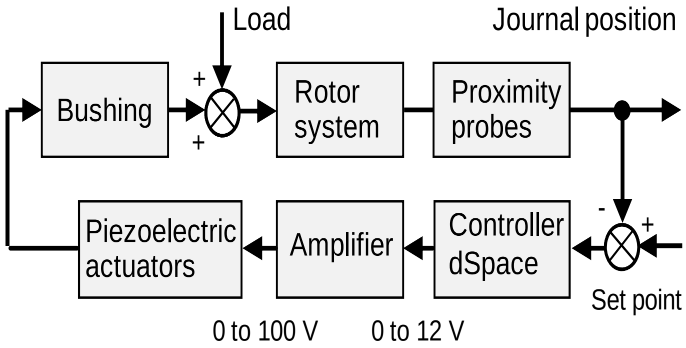

Figure 8 Active vibration control system

5. Experiments with the active vibration control of journal bearings

The rate of increasing the rotor rotational speed was set to 7000 RPM in a minute in experiments with active vibration control, as is shown in figure 9. Because the piezoactuators cannot cover the whole range of possible changes in the position of the shaft centerline from 0 RPM, the active vibration control is switched ON when the shaft lifts up into the safety stabilized position without oscillations, say approximately at 3000 RPM.

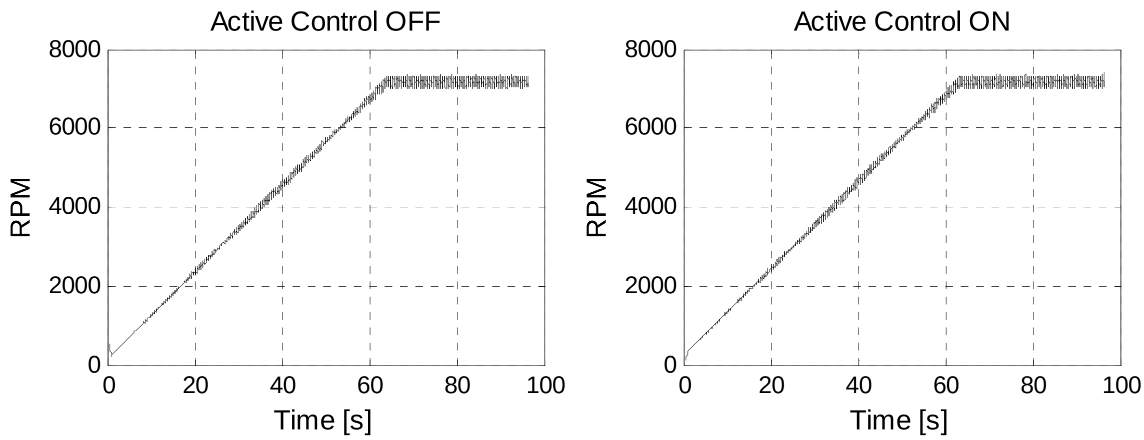

Figure 9 Time history of the rotor rotation speed when the active vibration control is ON and OFF

For the oil of the OL-P03 type the onset of instability starts at 4300 RPM. As it is clear from figure 10, if the active control is switched ON the onset of instability is shifted to 7300 RPM. The result of measurements at half the open-loop gain (50%) is shown in the middle part of figure 10. The onset of instability occurs at about 6200 RPM. Of course, the active vibration control is immediately switched OFF after starting the unstable oscillation at the subharmonic frequency of the shaft rotation frequency. The output of the signal processor is saturated on the full voltage range. As it is demonstrated in figure 10 the active vibration control significantly extends the range of operating rotational speed of about 3000 RPM in comparison with the operating range without the active vibration control. The active vibration control is clearly seen as a complementary way to the traditional journal bearing design modifications and other means, which prevent instability or shift the rotor instability onset to higher rotational speed.

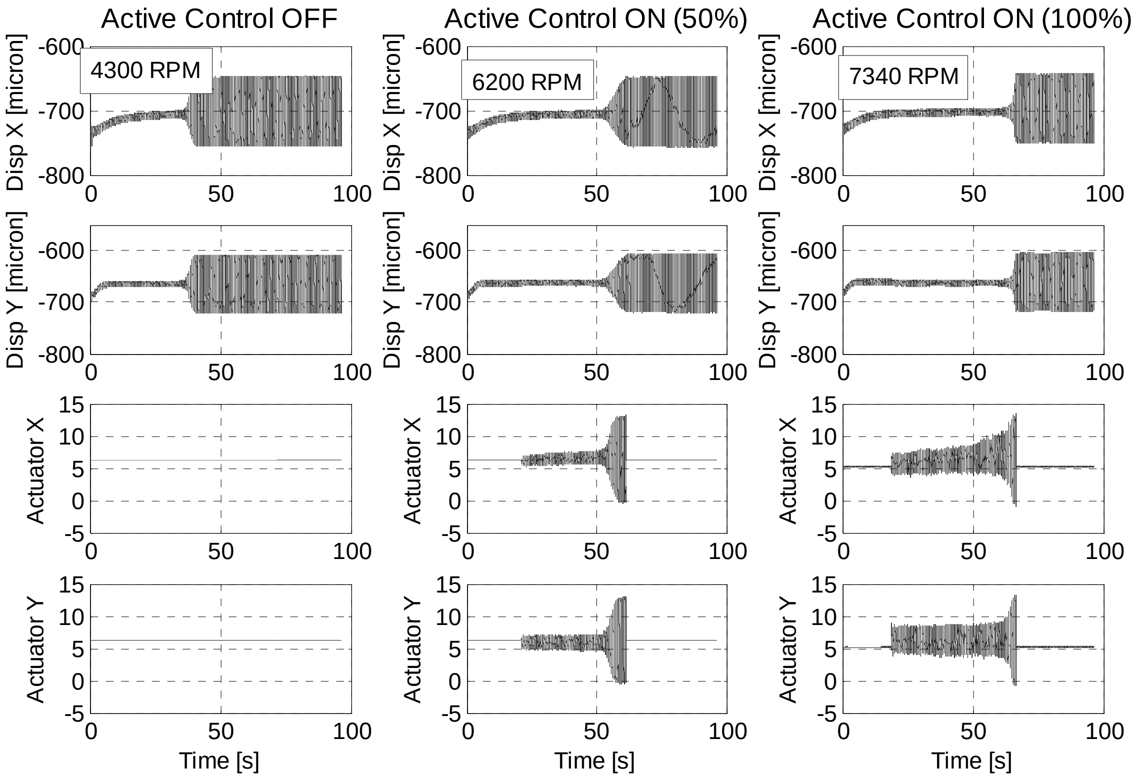

Figure 10 Time history of the rotor displacements in horizontal and vertical direction and control signals for piezoactuators when the active vibration control is ON and OFF

All the records of the rotor movement measurements originate from the PULSE signal analyzer. As already mentioned signals are affected by error of measurement. Except of the rotor instability state, the spectrum of this signal contains only harmonics of rotor speed. The error of such signals can be removed by averaging the sections of the records corresponding to one complete revolution of the rotor. The filtered curves are shown in figure 11. The same averaging method was also used for processing control signals at the output of the signal processor (see figure 12). The task of the active vibration control is to cancel only the subharmonic spectrum component of the rotor rotational frequency. This component arises in an unstable state of the rotor system and it does not almost affect the result of averaging. Amplitude of the AC component of the control signal before the onset of instability is approximately 1V (less than 10% of the full voltage range) and it is caused by the mentioned error signal. After averaging the error signal is approaching to zero. By comparison of both the control signals for piezoactuators in figure 10 and 12 it is possible to claim, that the voltage at the input of the piezoactuator amplifier will be achieved at the higher rotational speed and the range of the operational rotational speed of the rotor will be increased.

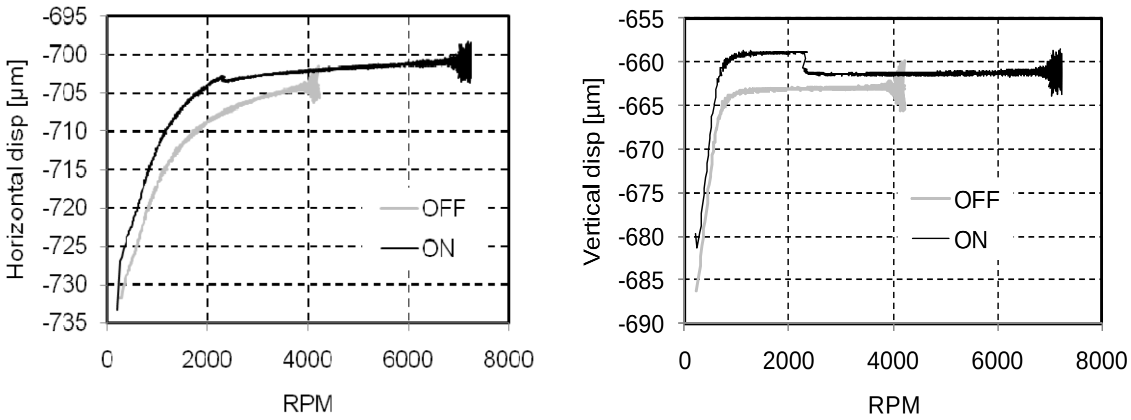

Figure 11 Averaged signals of proximity probes per a rotor revolution versus the rotor rotational speed

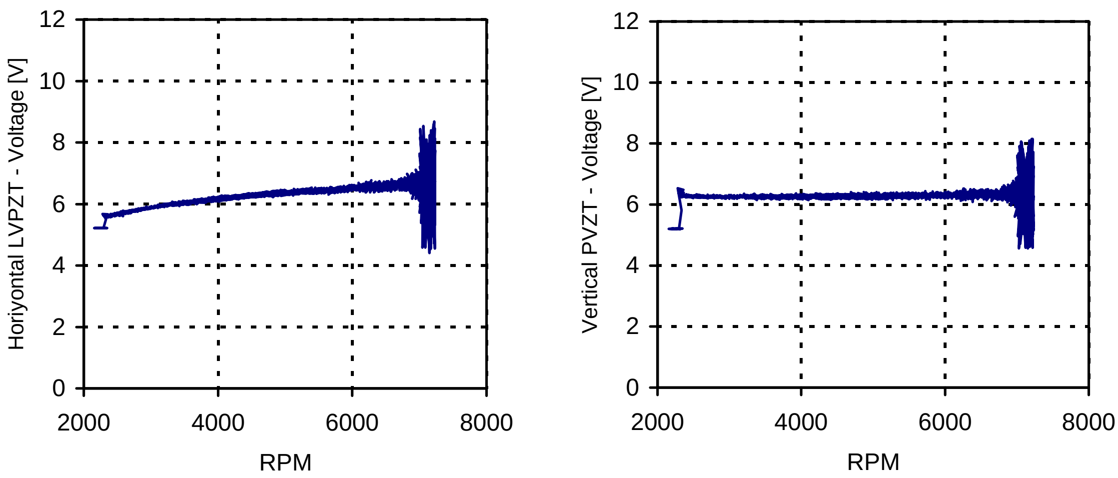

Figure 12 Averaged signals of the signal processor output voltage per a rotor revolution versus the rotor rotational speed

6. Conclusions

Test stand for experimental investigation of possibilities to affect behavior of the rotor supported in sliding bearings by external excitation was designed and manufactured. The tests carried out after test stand assembly showed some features, which had to be cleared before experiments with bearing bushing control could be started. It was proved, that unexpected rotor behavior was caused by misalignment of the shaft and drive motor axes and by non-homogeneity of shaft material. Standard behavior of the rotor was achieved with extremely low viscosity oil, with which the oil film had insufficient load capacity to shift journal centre into unstable position at the bearing centre. The proposed goal of the project was achieved by substantially increasing the onset of instability through controlled movement of only one bearing bushing. It seems, that there is a large potential for further experiments, which could lead to active control of rotor behavior of high-speed machines in real operating conditions. It is possible that the voltage at the input of the piezoactuator amplifier will be achieved at higher speeds and the range of the operational rotational speed of the rotor will be increased

7. Acknowledgement

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