# THE INFLUENCE OF CONTROLLED BUSHING MOVEMENT ON BEHAVIOUR OF A ROTOR IN SLIDING BEARINGS

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*Test stand for investigation of the influence of bearing bushing movement control on behaviour of a rigid rotor supported in sliding journal bearings was designed. The stand was equipped with two pairs of piezoactuators, enabling to move each bearing bushing in two directions, and with two pairs of relative sensors tracing shaft movement at both bearings. The initial tests showed quite unexpected phenomena, which should be cleared up, before experiments with controlled bearing bushing movement could be started. Finally the system began to operate according to predictions and it was possible to start intended experiments. Some results of the rotor behaviour with and without piezoactuator action are presented.*

Keywords: *test stand, hydrodynamic journal bearings, rigid rotor, controlled bushing movement, piezoactuator, relative sensor, oil whirl instability, elastic support*

## 1. Introduction

Rotor instability is one of the most serious problems of high-speed rotors supported in sliding bearings. With constantly increasing parameters of new machines problems with rotor instability are encountered more and more often. To study possibilities of affecting rotor behaviour by controlled movement of bearing bushings, test stand was designed, manufactured and assembled in the scope of Czech Science Foundation grant project, which was inspired by work [3]. Test stand located at Technical university of Ostrava showed some features, which were quite unexpected in such a simple system. The reasons of this unorthodox behaviour had to be found before experiments with bearing bushing excitation could be started.

## 2. Test stand design

As can be seen in Fig. 1, test stand [1] 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 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. Shaft movement is ob-

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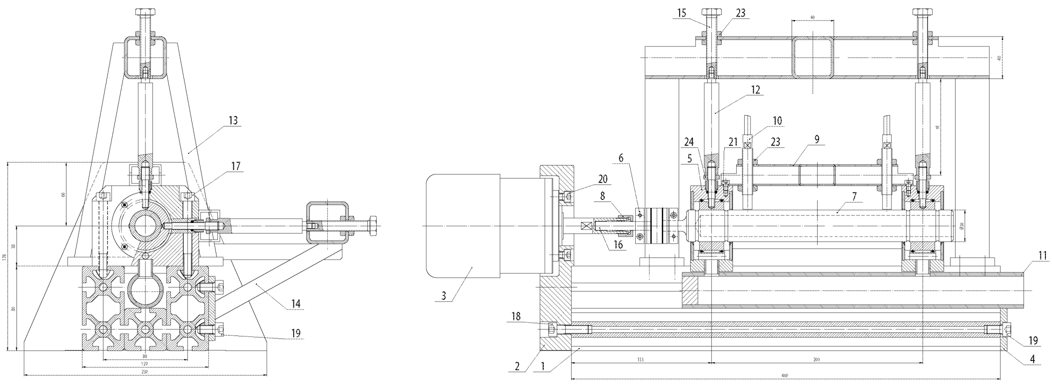


*Fig.1: Test stand cross section*

served 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 used. 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. Putting the test stand into operation

The 1st test shaft was hollow in order to achieve mass as low as possible. However, as is apparent from Fig. 2, the measured orbits had quite unexpected shape [2]. Peaks with amplitude of about 10 $\mu$m could be observed on vibration signals or journal trajectory (see Fig. 2).

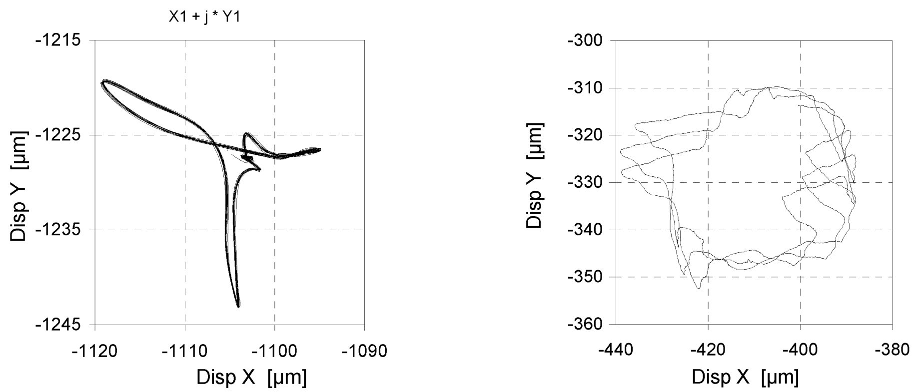


*Fig.2: Sample of journal trajectory with the first test shaft*

The signal from well-balanced shaft (left record) shows practically only disturbances, while the shaft unbalanced by disk (right record) exhibits certain orbit with disturbances. According to representative of the sensor supplier, the disturbances were caused by interference of sensors located in two perpendicular directions due to small shaft diameter (they recommend minimum diameter of 50 mm while the test shaft was only 30 mm in diameter).

However, as was ascertained by many trials, e.g. with one sensor removed, the reason was not sensor interference, but apparently inhomogeneity of the shaft material. As such a signal was unusable for piezoactuator control, new shaft was manufactured without the hollow and with somewhat smaller diameter to achieve higher clearance.

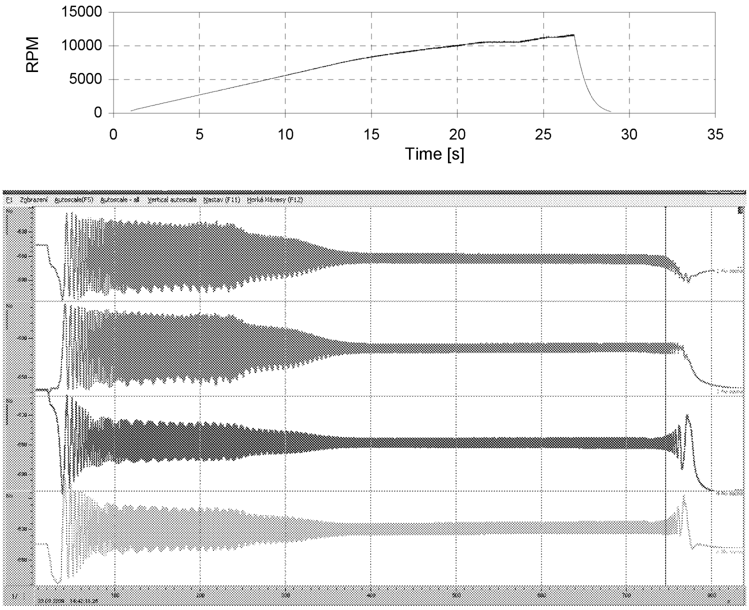


*Fig.3 Rotor run-up with clockwise direction of rotation*

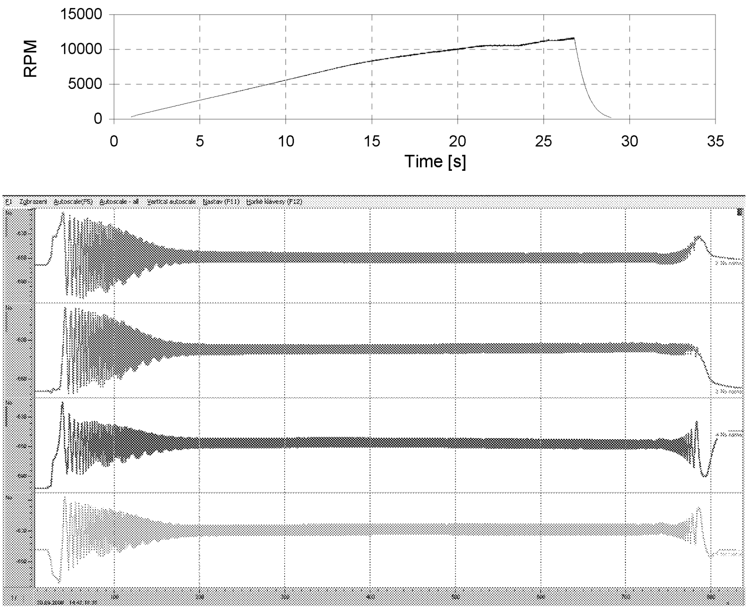


*Fig.4: Rotor run-up with anticlockwise direction of rotation*

*Top down : bearing 1 (rotor free end) – horizontal direction, vertical direction,*
*bearing 2 (coupling side) – horizontal direction, vertical direction.*

With this shaft and oil VG32 grade it was possible to reach speeds up to 12 000 rpm. However, the rotor behavior was far from what was expected. As can be seen from records in Fig. 3 and 4, the rotor was unstable from the beginning of run-up to some speed, where instability identifiable by high vibration amplitude ceased, and rotor further operated in stable regime. Moreover, the cessation of instability was dependent on direction of rotation, as is clear by comparing Fig. 3 and 4.

To explain the influence of direction of rotation on cessation of instability one must look into equilibrium of forces acting on shaft. Centre of journal moves along so called Gümbel curve (Fig. 5), which starts at point of contact with the bearing and with increasing speed it moves up and in direction of rotation.

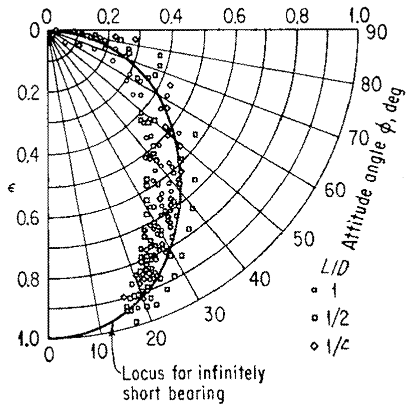


*Fig.5: Gümbel curve of journal centre in circular bearing*

With infinite speed or zero load the journal centre coincidences with bearing centre, which is generally unstable position in circular bearing. The force caused by misalignment and stiffness of the coupling acts in one direction disregarding direction of rotation, while orientation of Gümbel curve changes with direction of rotation. For one direction of rotation the effect of coupling therefore supports shift of the journal centre to bearing centre and rotor instability thus perseveres longer.

Test rotor instability was characterized by subharmonic vibration with frequency equal to roughly one half of rotational frequency. It is apparent from Fig. 6, which shows vibration in time domain and frequency spectrum (0–50 Hz) at 750 rpm (12.5 Hz).

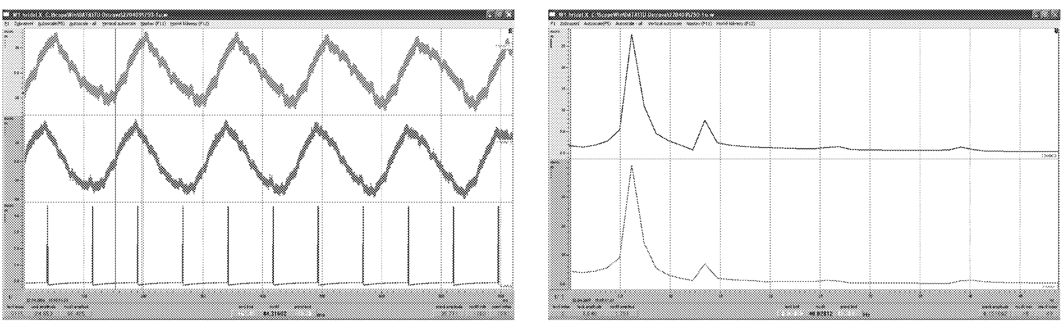

*Fig.6: Rotor relative vibration in time domain and frequency spectrum at 750 rpm*

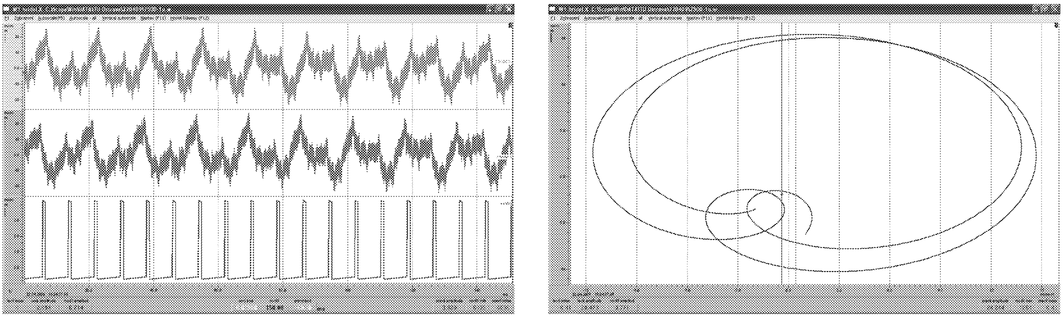

*Fig.7: Rotor relative vibration in time domain and journal centre trajectory at 7 500 rpm*

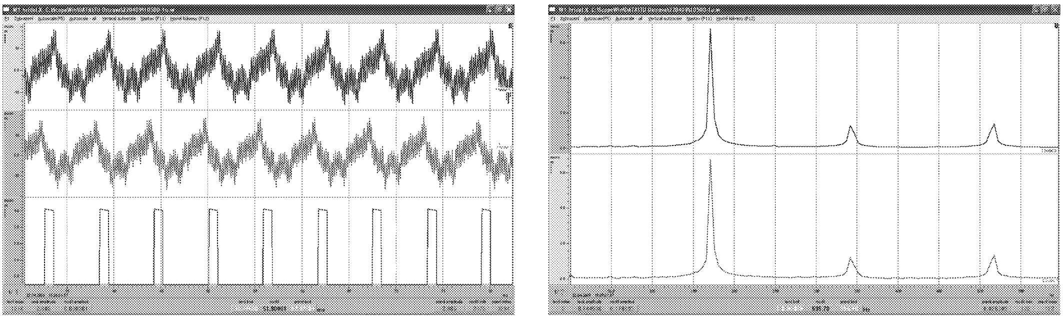

*Fig.8: Rotor relative vibration in time domain and frequency spectrum at 10 500 rpm*

Top down are rotor excursions in horizontal and vertical directions at the rotor free end and rotational speed signal. It is apparent, that dominating vibration frequency is 6.1 Hz, which is 0.49 of rotation frequency. Instability persists to about 7 500 rpm, where rotational and subharmonic frequencies become comparable, as is shown in Fig. 7.

Both subharmonic frequency, close to one half of rotation frequency, and corresponding journal centre trajectory (Fig. 7 right) are quite typical symptoms of 'half-speed whirl' or 'oil whirl' type of instability. To get clearer picture of journal trajectory vibration signals were filtered for frequencies above 200 Hz. At 10 500 rpm (175 Hz) the rotor run was quite stable, as can be seen from vibration signals and frequency spectrum (0–550 Hz) in Fig. 8.

## 4. Stability analysis

To support assumptions made in preceding chapter, an analysis was carried out to show influence of bearing load and clearance on rotor stability. Table 1 shows, that the stability margin is very strongly influenced by bearing load. For a shaft with diametral clearance of 90 $\mu$m (used for tests presented in the paper) the rotor stability limit is over 14 000 rpm for the load of 3.7 N (shaft own mass), while with load reduced to 0.1 N it is decreased to only 1 700 rpm. The influence of bearing clearance is weaker – increasing the clearance 4 times (from 50 $\mu$m to 200 $\mu$m) would reduce stability limit to roughly one half. If the shaft centre is shifted (due to coupling angular stiffness) to vicinity of bearing centre, which corresponds to situation with very low load, practically immediate onset of rotor instability can be expected, as was detected during measurement (Figs. 3 and 4).

| bearing load (N) | diametral clearance ($\mu$m) | | | |
|---|---|---|---|---|
| | 50 | 90 | 140 | 200 |
| 0.1 | 2 000 | 1 700 | 1 400 | 1 300 |
| 1.0 | 7 500 | 5 000 | 4 700 | 3 800 |
| 3.7 | 14 200 | 11 100 | 9 200 | 7 900 |
| 9.7 | 14 800 | 11 400 | 9 500 | 8 100 |

*Tab.1: Dependence of stability limit on bearing load and clearance*

Table 1 indicates stability limit (speed at which rotor instability sets in) for bearings lubricated by VG32 oil. However, the influence of oil grade is relatively weak. Warming-up of more viscous oil in bearing clearance is higher due to greater friction losses, so that the difference between oil film stiffness and damping in a bearing lubricated by oil grade VG32 or VG10 is relatively small.

The situation on test stand without piezoactuators is even more complicated due to flexible support of bearing bushings, which is necessary to enable their movement within bearing casing. With piezoactuators disconnected from bearing bushings flexible support provides additional damping suppressing instability. It can be proved by the test run, in which bearing bushing excursions were monitored simultaneously with shaft vibration during rotor run-up. As is demonstrated in Figs. 9 to 11, cessation of rotor instability interrelates with the bushing movement.

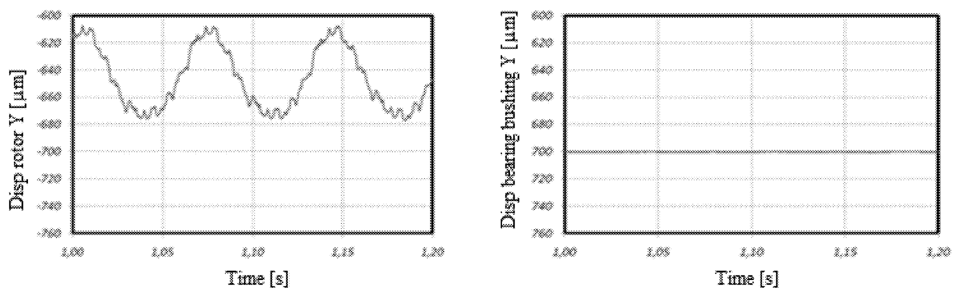


*Fig.9: Vibration of the rotor (left) and bearing bushing (right) at 1 750 rpm*

Fig. 9 illustrates the situation at 1 750 rpm; while the rotor is evidently unstable with vibration peak-to peak amplitude exceeding 70 $\mu$m, bearing bushing does not move at all. On the other hand, at 7 500 rpm (Fig. 10) vibration signals indicate stable operation of the rotor with peak-to-peak amplitude of about 15 $\mu$m, while the bearing bushing vibrates with peak-to-peak amplitude of about 10 $\mu$m.

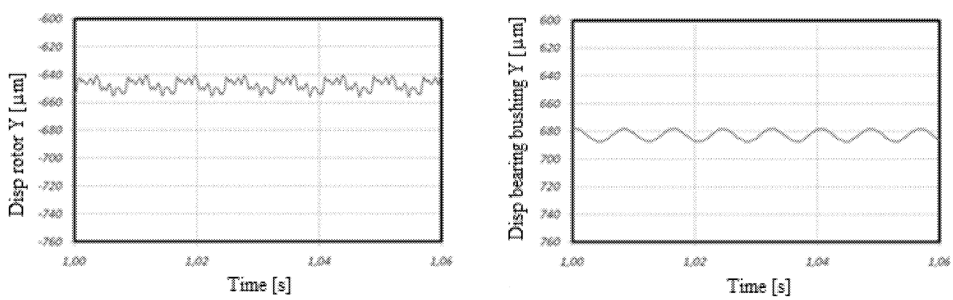


*Fig.10: Vibration of the rotor (left) and bearing bushing (right) at 7 500 rpm*

With further increasing speed the shaft vibration amplitude practically did not change, while bushing vibration amplitude grew further (Fig. 11).

It seems, that suppression of instability at 7 500 rpm was due to combined effects of shaft reaching more stable position in the bearing and of additional damping caused by bearing bushing movement.

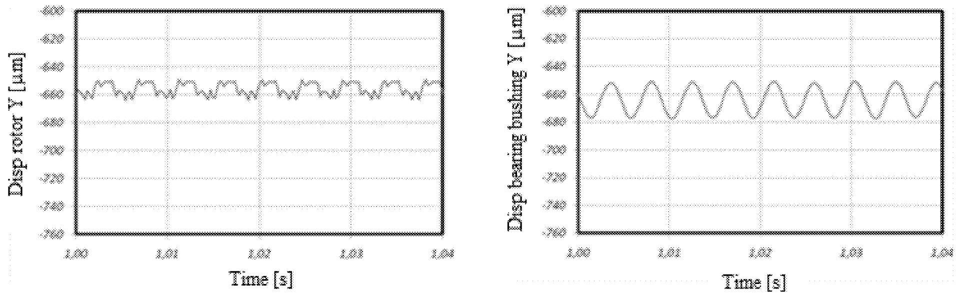


*Fig.11: Vibration of the rotor (left) and bearing bushing (right) at 10 500 rpm*

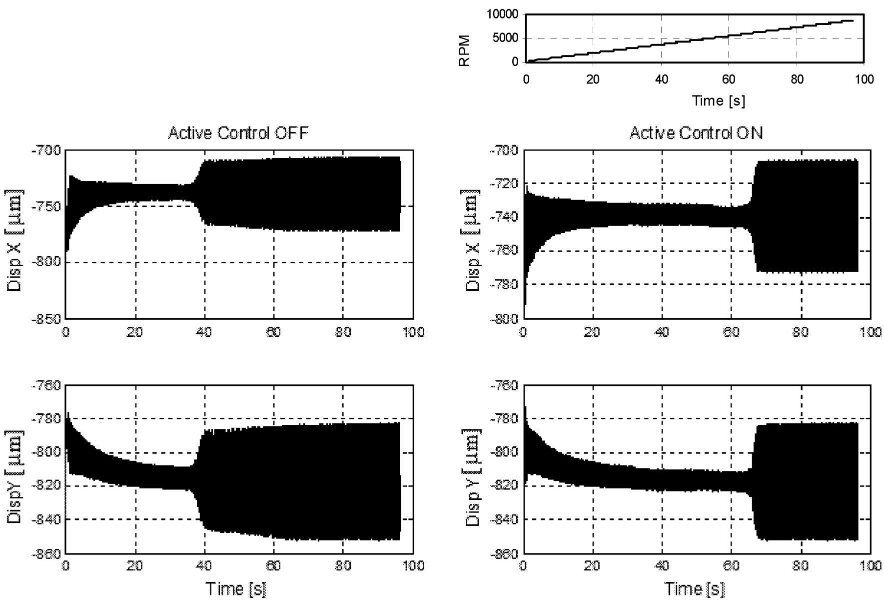


*Fig.12: Run up of the rotor with extremely low viscous oil VG10*

Replacing VG32 oil grade with extremely low viscosity oil VG10 returned the rotor behaviour to 'normal', i.e. the rotor run up was stable to certain speed, when instability set in. As can be seen from Fig. 12, rotor was stable to about 4 500 rpm. Instability, indicated by sudden increase of vibration amplitude, persevered up to almost 10 000 rpm. Vibration amplitude in region of instability reached practically the whole of bearing clearance, which was about 90 $\mu$m. Upper pair of recorded signals is from bearing 1, at the rotor free end. At the start of run-up the rotor in bearing 1 shifted somewhat up and right (looking from shaft end). Signals in bearing 2 (lower pair of records) show, that at run-up the shaft centre is moving along certain trajectory due to misalignment of shaft and motor axes. With increasing speed and growing oil film load capacity and stiffness the shaft centre is moved towards bearing centre. Trajectory of shaft centre is much smaller than at beginning of run-up, but bigger than in bearing 2, where the influence of coupling is weaker.

## 5. Correction of vibration signals

Inhomogeneity of the rotor material magnetic properties as the main source of the proximity probe periodical error was proved by means of measurement of run-out by digital indicator. Fig. 13 shows the difference in signals on original (hollow) and new (without orifice) shaft. The dependence of the proximity probe regular error $\Delta$ on the rotation angle $\alpha$ can be approximated by a sum of trigonometric function differing in the number of waves

$$\Delta(\alpha) = \sum_{n=1}^{3} a_n \cos(n\,\alpha + \varphi_n) ,$$

where $a_n$ is an amplitude and $\varphi_n$ is a phase shift. The reduction of the proximity probe error requires measurement of reference pulses and prediction of the rotational angle.

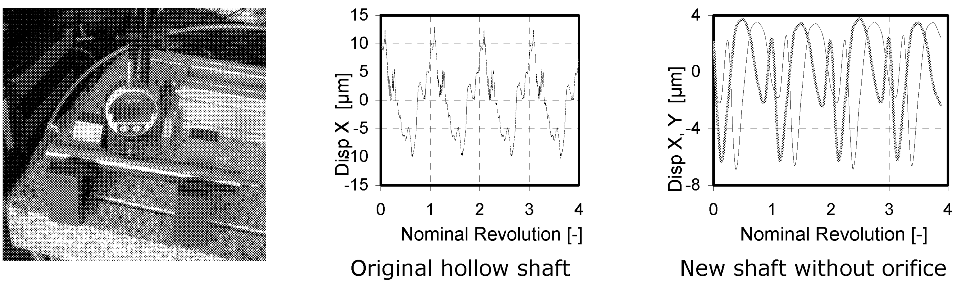


*Fig.13: Regular error of proximity probe as a function of rotation angle*

## 6. Piezoactuator choice and testing

Proper selection 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 acting on the bushing in horizontal direction is shown in Fig. 14. As the measurement shows, the piezoactuator load capacity is satisfactory.

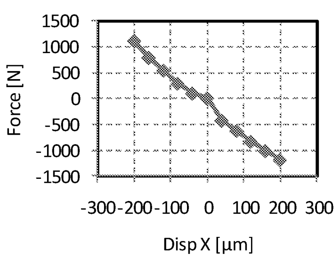


*Fig.14: Piezoactuator characteristic (force vers. displacement)*

Flexible tips were used to attach the piezoactuators to the bushing rod and the frame structure for compensation of misalignment, as is shown in Fig. 15.

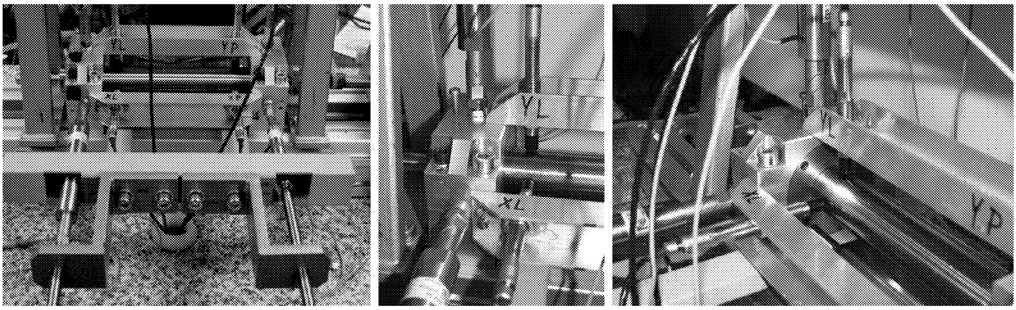

*Fig.15: Piezoactuator mounting*

## 7. Active control of bearing bushing movement

The test stand instrumentation allows active vibration control only in the journal bearing at the shaft free end. Previous to operational tests, the initial position of the piezoactuators has to be adjusted in the middle of the operating travel range. This position corresponds to half the output voltage of the controller, with full range equal to 12 V. The range of the shaft displacement for the full scale of the controller output voltage is shown for both directions in Fig. 16.

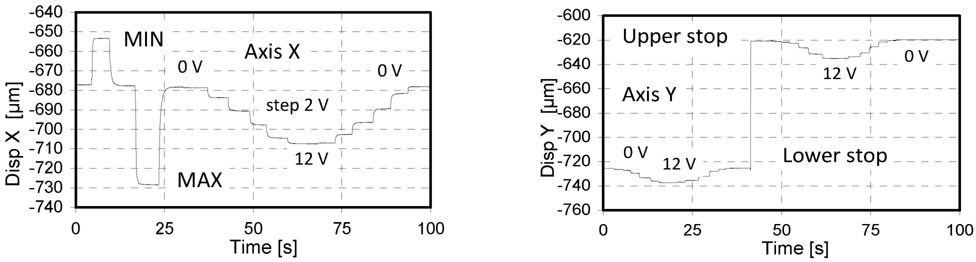


*Fig.16: Effect of the amplifier input voltage on the shaft position*

The signal from the proximity probes is led to the dSpace signal processor. The output of the signal processor is connected to the input of the amplifier, which supplies piezoactuator. Because the piezoactuators can not cover the change of the shaft centerline position from the very beginning of run-up, the active control is switched ON when the shaft lifts up into the stabilized position, which corresponds to approximately 3 000 rpm. The active vibration control is switched OFF immediately after onset of rotor instability.

As is demonstrated in Fig. 17, the active vibration control extends considerably stable operational speed range. Even if only one bearing bushing is controlled, the onset of instability is increased by about 3 000 rpm in comparison with the case without control, with instability occurring at about 4 300 rpm. Vibration control was switched of, because with further increase of speed it was not possible to restore stability; operational quantity (control variable) was out of range.

Active vibration control is evidently another possibility to prevent instability or shift the rotor stability margin to higher speed, which up to now could be carried out only by modifications of journal bearing design or by change of operational parameters.

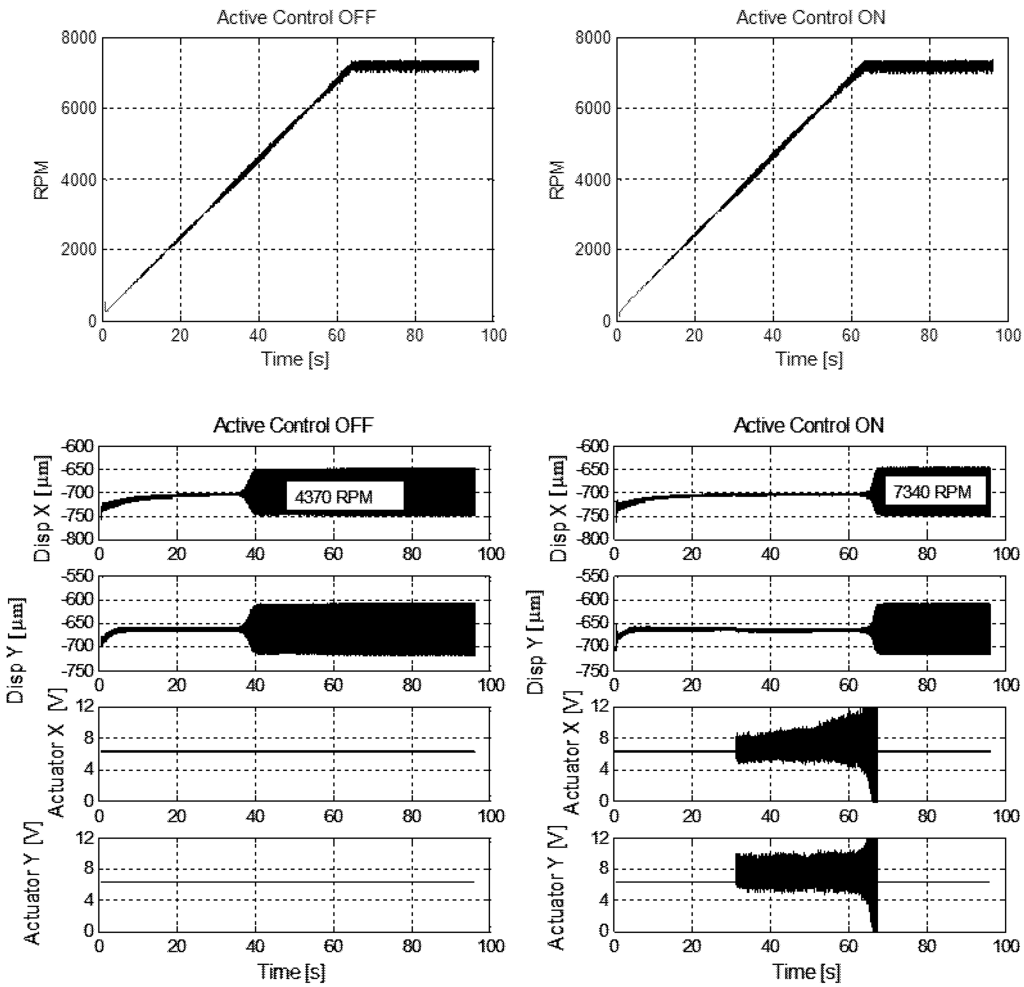


*Fig.17: Effect of active vibration control on the rotor instability onset*

## 8. Conclusions

Test stand for experimental investigation of possibilities to affect behaviour of the rotor supported in sliding bearings by external excitation was designed and manufactured. This idea was inspired by the work [3], which showed new possibilities of suppression of self-excited vibration. The tests, carried out after test stand assembly, showed some features, which had to be cleared up before experiments with bearing bushing control could be started. It was proved, that unexpected results were caused by misalignment of the shaft and drive motor axes and by inhomogeneity of shaft material. Standard behaviour of the rotor was achieved with extremely low viscosity oil, with which the oil film had insufficient load capacity to shift shaft 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 dynamics of high-speed machines in real operating conditions.

### Acknowledgement

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