

AN ACTIVELY CONTROLLED JOURNAL BEARING WITH INCREASED RESISTANCE TO INSTABILITY

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Journal hydrodynamic bearings are used for their high radial load capacity and operation at high speeds. The only disadvantage is the excitation of vibrations, called an oil whirl, after crossing a threshold of the rotational speed. The whirl can be suppressed using the controlled motion of the non-rotating loose bushing which is actuated by piezo-actuators. The displacement of the piezo-actuators is governed by a feedback which reduces the difference between the required and true position of the bearing journal. This new bearing design enables not only to increase the threshold of instability but also serves to maintain the desired bearing journal position with the accuracy of micrometers. The article deals with the design of an actively controlled bearing with special regard to the installation of piezoelectrometers. Part of the article is also a description of the influence of active vibration control on bearing performance characteristics and bearing friction losses.

KEYWORDS

journal bearings, non-rotating loose bushing, piezo-actuators, active vibration control, proximity probes

1 INTRODUCTION

Journal hydrodynamic bearings (alternatively called sleeve bearings or plain bearings for radial load) are a standard solution to support rotors. Their advantage is a possibility to carry high radial load and to operate at high rotational speeds. The disadvantage of the journal bearings is the excitation of unwanted rotor vibrations by whirling of the journal in the bearing bushing. The bearing journal becomes unstable as the journal axis begins to perform a circular motion that is bounded only by the walls of the bearing bushing. The instability of the journal position arises after exceeding the threshold of the rotational speed. The described unstable behavior of the journal bearing is called oil whirl and is explained by the properties of the oil film present. The magnitude of the mentioned threshold depends on the radial bearing clearance, bearing load and the viscosity of lubricating oil.

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, or use of tilting pads. Even though there are several solutions based on mentioned passive improvements this article deals with the use of the active vibration control (AVC) with piezo-actuators as a measure to prevent instability. The principle of an actively controlled bearing and the correspondent technical drawing is illustrated in Fig. 1.

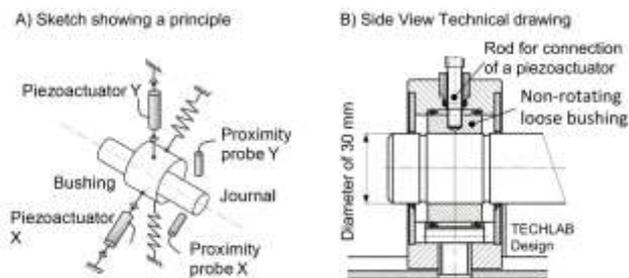


Figure 1. Actively controlled journal bearing

Active magnetic bearing was previously considered as the only actively controlled bearing [Furst 1988]. Later published a paper dealt with a giant magnetostrictive material (GMM) [Lau 2009] to actuate the position of the non-rotating loose bushing. Although the article on using GMM stated that experiments can be carried out up to 1700 rpm, they publish only measurements at 350 rpm which could not excite instability at such low rotational speed. The unstable behavior of the bearing journal is a problem of high-speed rotors, but in this case active vibration control was aimed only at maintaining a required position of the shaft axis. Piezo-actuators as a tool to control of rotating machines have been intensively studied in the literature since the end of 1980's. The linear piezo-actuators can create a large force on a very short track [Buchacz 2014]. The advantage of the journal bearings with piezo-actuators is that the bearing bushing mounting stiffness remains unchanged in the case of a loss of the electric power supply comparing the active magnetic bearing (AMB). However, AMB are still very expensive and cases, where it is possible to install AMB to already designed rotor, are very rare. On the other hand, installation of a piezo-actuator into bearing housing is relatively easy. The non-rotating loose bushing can be inserted into any bearing housing.

2 TEST RIG FOR TESTING BEARINGS

For developing a new design of the actively controlled bearing, a test rig was built, see Fig. 2. This figure provides different views of the test status. An inductive motor of 400 Hz drives the rotor, and therefore the maximum rotational speed is 23k rpm. The engine is connected to the rotor via the Huco diaphragm coupling. The bearing diameter is 30 mm, and the length-to-diameter ratio is equal to about 0.77. The span of bearing pedestals is of 200 mm. The results of the experiments presented in this article are for the radial clearance of 55 μm . Also, the journals of the other clearance are available for testing. The performance of the actively controlled bearing was tested on the test bench (Rotorkit) of the TECHLAB design [Simek 2010], [Simek 2014]. Additionally, it should be emphasized that research was focused at rigid rotors and the journal bushing of the cylindrical bore, where the journal motion is measured at the location closest to the bearing bushing. The research work resulted in putting into operation of the active vibration control system, which became the first functional bearing prototype known up to now [Tuma 2013]. The mechanical arrangement of the actively controlled bearing is shown on the right of Fig. 1. Oil leakage from the volume between the bearing body and the loose bushing and the piezo-actuator rod is sealed with rubber O-rings. The bearing oil of the OL-P03 type (VG 10 grade, kinematic viscosity 2.5 to 4 mm^2/s at 40 $^{\circ}\text{C}$) was used to all the tests at the ambient temperature about of 20 $^{\circ}\text{C}$. The selected oil type enabled to reach the maximum rotational speed, while the instability threshold was much lower. Input for the oil inlet is in the horizontal plane of symmetry of the bushing. As it was stated

before, vibrations of the rotor is suppressed using the system for active vibration control with piezoelectric actuators enabling to move the non-rotating loose bushing in two directions. The motion of the bearing bushing is controlled by the controller, which responds to the change in position of the bearing journal related to the bearing housing. Two stacked linear piezo-actuators are used to actuate the position of the bearing journal via the position of the bearing bushing. The bearing uses a cylindrical bushing which did not require unique technology of production and assembly. This new bearing enables not only to damp vibrations and to prevent instability but also enables to maintain the desired bearing journal position with an accuracy of micrometers



Figure 2. Test rig for testing bearings

3 ACTIVE VIBRATION CONTROL

3.1 Piezo-actuators

The non-rotating loose bushings are actuated with the use of the piezo-actuators oriented in two perpendicular directions and fastened to the bearing body. A part of the mechanical structure for connecting the piezo-actuator to the bearing body and bushing will be referred to as support in the article. The preloaded piezo-actuators are of the P-844.60 type and originate from the Physik Instrumente Company. The piezo-actuator are powered by the variable electrical voltage ranging up to 120 V although it is a low voltage type (LVPZT). As the output voltage of the controller does not exceed 12 V a DC amplifier is needed. The free stroke of the piezo-actuator ranges up to 90 micrometers. The blocking pushing force is up to 3 kN and the pulling force is only up to 700 N. The stiffness of the piezo-actuator is $K_{PA} = 33 \text{ MN/m}$. The asymmetry of the limit values of the force is due to the risk that the piezo-actuator can be destroyed by tensile stress. The piezo-actuator stack is therefore preloaded inside its housing and operates mainly in the pressure mode. The stiffness of the O-ring seals is indicated by $K_{OR} = 5.5 \text{ MN/m}$. The O-ring stiffness measurement was together with the bearing bushing which was connected in the series. The wall thickness of the bushing is 8 mm, therefore its stiffness is many times larger than the stiffness of the O-ring. For a considerable length of used piezo-actuators, an auxiliary frame must be used for their connection to the bearing housing. Bending and torsion load of the piezo-actuators is excluded by using flexible tips and mounting procedure. The stiffness of the piezo-actuator support K_S plays a significant role in the active vibration control. Low stiffness of the support may reduce the stroke of the bearing bushing.

The mechanical part of the active vibration control system for the one of two directions is shown in Fig. 3. The meaning of the individual variables of the mechanical system arrangement is explained on the left top corner of this figure. In this diagram, the force is equivalent to the electrical current and the displacement to the electrical voltage. The piezo-actuator force balances the force effect of the oil film. The free stroke of the piezo-actuator is proportional to its control supply voltage V is approximated by the formula.

$$u_x^* = kV \quad (1)$$

where k is a strain coefficient which determines strain after applying a voltage. The resulting motion u_x of the bearing bushing also depends on the load force. Interdependence of force, displacement and supply voltage including all limitations of operating conditions of the P-844.60 piezo-actuator are shown in the working graph in Fig. 4. This piezo-actuator is quite expensive, but in the last 10 years, piezo-actuator prices have fallen more than 10 times.

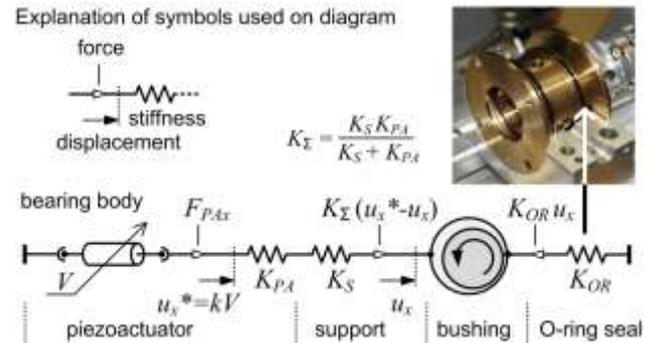


Figure 3. Mechanical part of the control system for x-axis

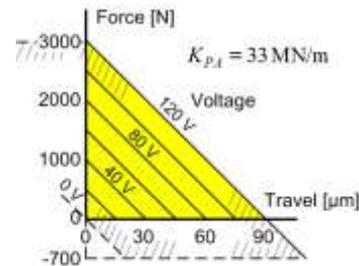


Figure 4. Working graph of the piezo-actuators of the P-844.60 type

The catalogue value of the free stroke can only be achieved for ideally rigid support, i.e., the stiffness K_S approaching infinity. The effect of the stiffness of the O-ring seal is shown in the left panel of Fig. 5. Displacement of the bearing bushing for the maximum voltage is reduced from 90 to 77 μm at the maximum voltage 120V that are not recommended to cross. The finite stiffness of the piezo-actuator support affects the virtual stiffness of the piezo-actuator as follows

$$K_{\Sigma} = \frac{K_S K_{PA}}{K_S + K_{PA}} = \frac{K_{PA}}{1 + K_{PA}/K_S} = \frac{K_{PA}}{1 + a} \quad (2)$$

where a is a multiple determining the fraction of the piezo-actuator stiffness K_S related to the support stiffness. The serial connection of the piezo-actuator and its support has a stiffness designated by K_{Σ} . Maximum displacement of the bearing bushing respecting all influences for $V = V_{MAX}$ is given by the following formula

$$[u_x]_{MAX} = \frac{[u_x^*]_{MAX}}{1 + (1 + a)K_{OR}/K_{PA}} \quad (3)$$

The real stiffness of the piezo-actuator can be reduced due to reducing the support stiffness. The effect of reduced stiffness of the piezo-actuator on the range of the control variable is shown on the right panel of Fig. 5. It is possible to compare the degree of decreasing the real piezo-actuator stiffness to the stiffness which corresponds to the ideally stiffen support.

Limiting the range of the control variable will reduce the efficiency of the active vibration control. It should be noted that the maximum displacement $[u_x]_{MAX}$ of the piezo-actuator is related to the strain coefficient, see Eq. (1).

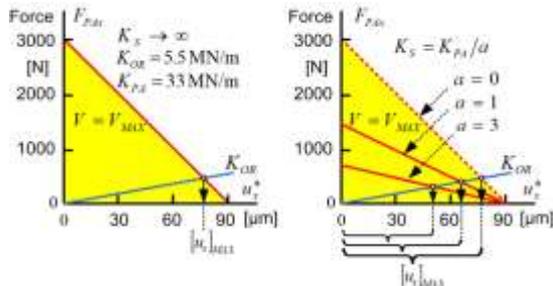


Figure 5. Working graph and point of the force balance

3.2 Control system arrangement

The control circuit of the actively controlled bearing is shown in Fig. 6. The controlled system has two inputs and two outputs. The inputs are the piezoelectric displacements that move the bearing housing in two perpendicular directions. The output of the controlled system is the position of the journal relative to the bearing body. The position of the journal is sensed by a pair of capacitive sensors of the capaNCDT CS05 type with accuracy better than 1 micrometer, supplied by the firm MICRO-EPSILON. The photo on the right of Fig. 2 shows the location of the proximity probe. The outputs of the system are connected to its inputs by an electronic feedback. Two inputs of the controlled system are the control variables as the output of two controllers. The control system is thus composed of two independent control loops, each of which has its controller 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 controller is selected as a proportional type. We consider a more complex controller types [Viteckova 2011]. Before the experiments, a simulation model was developed, on which the control algorithm was verified [Wagnerova 2016].

The controllers were created as a digital in the signal processor of the compactRIO type supplied by NI, see Fig. 7. The dSpace signal processor was used for tests, but the prototype of the actively controlled bearing is suitable to be equipped with the CompactRIO system or other development boards [Babiuch 2014]. The output voltage of the CompactRIO has to be amplified by the DC amplifier to the magnitude ranging up to 120 V to supply the piezo-actuators. The sampling frequency of the control-loops is chosen equal to 5 kHz.

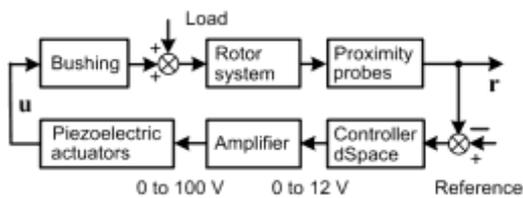


Figure 6. Active vibration control system

Calculation of instantaneous rotational speed of the inductive motor, rotational speed control according to a given program with a constant rate of the increase and decrease for governing the frequency convector and supervision of the lubricating aggregate equipment is a task for a PC.

Input and output signals are connected to the CompactRIO system, where the main program is processed in the Field Programmable Gate Array (FPGA). This programming technique

guarantees the determinism of the program which, given a particular input, will always produce the same output. Postprocessing and data storage is done in a real-time processor with a Linux-based real-time operating system. Data can be stored on an external USB disc or directly on the CompactRIO HDD and then accessed via the Web server. LabVIEW software is used for programming and visualization. Application besides communication with CompactRIO system communicates with frequency converter using OPC communication and a switching unit that controls individual switching relays via serial communication. The performance of the supervising PC allows deploying routine rotor diagnostics [Takacs 2015].

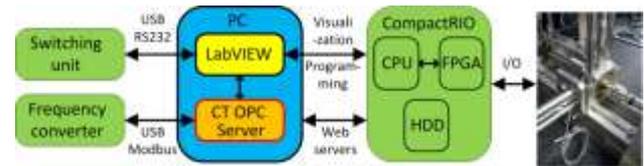


Figure 7. Compact RIO controller

3.3 Stiffness of the piezo-actuator support

The stiffness of piezo-actuator support can be measured theoretically directly, but it is more useful to assess the support stiffness by the objective of the active vibration control such as the threshold of instability. It is known that a conventional journal bearing has a limit angular rotation speed, which can be designated as ω_{CRIT} . Analysis of the effect of the proportional feedback with K_P gain on stability margin shows that the limit of the rotational speed is increased as follows [Tuma 2013]

$$\omega_{MAX} = \omega_{CRIT} \sqrt{1 + K_P} \quad (4)$$

The gain K_P determines how many times the feedback amplifies the displacement of the bearing journal on the motion of the loose bearing bushing. This gain factor is the product of the controller gain, auxiliary DC amplifier, and actual strain coefficient.

First experiments with an imperfect support showed displacement of the bearing bushing of less than about 20 microns at the maximum electrical voltage to supply the piezo-actuators. Maximum speeds increased by only 70%. This increase of the operational speed range was equivalent to the gain of approximately $K_P \approx 2$. These conditions were encountered at the beginning of the development when the support arrangement was provisionally extended due to the use of longer piezo-actuators as can be seen in Fig. 8. Much more rigid support is evident from the photos in Fig. 2. The expansion of the operating speed range corresponds to the gain of $K_P \approx 35$. However, the ideal solution is to install piezo-actuators into the bearing body.



Figure 8. Improved support for piezo-actuators

4 EFFECT OF THE FEEDBACK ON BEARING OPERATION

The experiments were aimed to determine the limit of the operating speed, i.e. the margin of the stable operation. The journal bearing has been tested by a run-up with constant increase of speed until the rotor becomes unstable and the whirl occurs. The summary results are shown in Fig. 9. The time history of the speed is shown in the top right panel of the above-mentioned figure. The figure is arranged in three pairs of the strip graphs of the time histories of the journal position for horizontal (coordinate X) and vertical (coordinate Y) directions. The first strip shows the run-up without the active vibration control (AVC OFF), the second strip is for AVC ON, and the third one for parametric damping ON. For AVC ON the operating range of the rotational speed was considerably increased. The deviations of the parametric damped motion were considerably reduced as well. The static gain of the proportional controller was limited only by the stability of the control loop. The gain was set as high as possible with respect to stability.

The effect of the linear proportional controller on the operational range is positive, but there are some residual deviations of the journal position. Parametric damping means that at least one parameter of the system varies periodically in time according to a sinusoidal function as was suggested by Tondl [Tondl 1991]. 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:

$$K_p(t) = K_{p0}(1 + \alpha \sin(\omega_0 t)), \quad (5)$$

where α is dimensionless amplitude of excitation, K_{p0} is static gain factor and ω_0 is angular frequency of excitation.

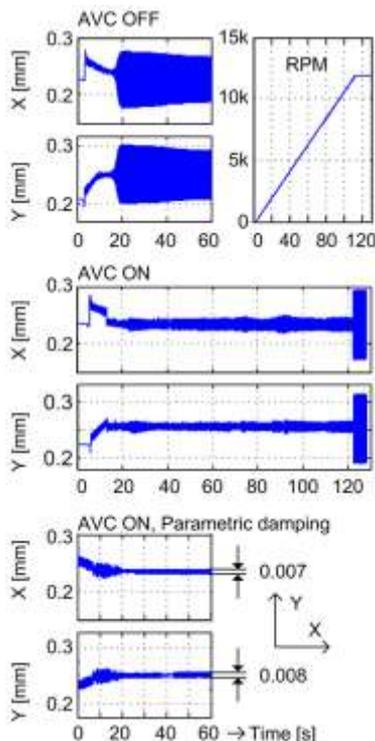


Figure 9. Active vibration control system

Dohnal [Dohnal 2011] has solved a similar problem for magnetic bearings. Our experiments on the test bench were conducted for the following amplitudes of excitation $\alpha = 0, 0.1, 0.15,$ 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. According to the theory proposed by Tondl [Tondl 1991], the frequency of 30 Hz is considered to be the basic Parametric Resonance of the controlled system. Rotor speed increases according to a ramp function as is shown in the first left panel of Fig. 9. 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 8 micrometers. The journal position variations are affected by the residual unbalance of the rotor and possibly by piezo-actuators themselves because the position measurement error is less than 1 micrometer. The bearing position without active control is unstable at speeds above 3000 RPM. Precision ball bearings (Deep groove ball bearings) which are offered by SKF have a radial internal clearance of the C2 class for 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.

5 REDUCING FRICTION LOSSES OF THE JOURNAL BEARING

Journal bearing instability increases friction losses. Quantification of the braking torque can be done by the direct measurement of torque at the motor output shaft or by the indirect measurement of the driving unit consumption. The basic power consumption of the inductive motor including the frequency converter was measured with the disconnected clutch between the motor and rotor; it means that the bearings were inoperative and the motor runs without any external loads. Power losses of the rotating system are only due to the friction of the motor bearings, fan losses, the stator copper and the iron losses in the motor. Dependence of the electrical power consumption upon the motor rotational speed was measured with and without the active control as it is shown in Fig. 10.

The difference between the electric consumption of the rotating system and the basic power consumption of the motor and the frequency converter shows that the friction losses of a pair of bearings at 7k rpm are 66 W during the unstable operation. If the active vibration control is ON, then the friction losses are of only 48 W. The active vibration control reduces the friction losses of the journal bearings by 27 %.

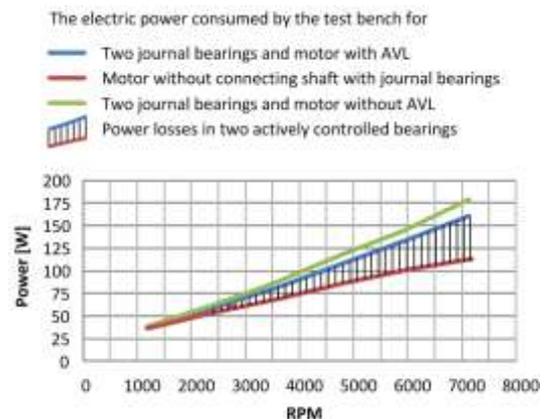


Figure 10. The electric power consumed by the frequency converter, motor and bearings

6 UPGRADED BEARING DESIGN

The piezo-actuators which are integrated directly into the bearing housing represent a more cost-effective and compact design for industrial applications. The mounting dimensions of

the upgraded bearings with the integral piezo-actuators are much smaller than for bearings in Fig. 2. The piezo-actuator can only develop compressive forces so that forces in the opposite direction should be exerted by the elastic elements.

The radial journal bearing is composed of the lower (1) and upper (2) part of the bearing body. The bearing further includes the bearing bushing (3), piezo-actuators (4), holders (5), flexible elements (6), adjusting screws (7) and fastening elements (8, 9, 10). The movement of the bearing bushing (3) is actively controlled by piezo-actuators (4) insert in-between the bearing bushing (3) and the holders (5). The preload of the piezo-actuators being achieved by the deformation elements (6) using the adjusting screws (7).

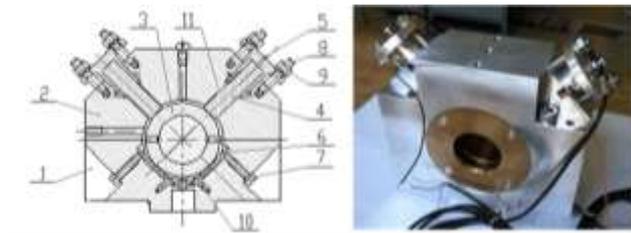


Figure 11. Upgraded design of the actively controlled bearing

The bearing body has been designed for a stack piezo-actuator made by the STEMiNC Company. The maximum input voltage of the selected piezo-actuator is 150 V. The free stroke ranges up to 50 micrometers at 150 V. The blocking pushing force is up to 1.8 kN. The stiffness of the piezo-actuator is 51 MN/m. This stiffness will be reduced due to the use of the deformation element, but the stiffness of the support will be greater in comparison with the frame supporting the PI stack piezo-actuators. Now, this piezo-actuator can be purchased at 15 times lower price than the P-844.60 piezo-actuator. The actively controlled bearing is designed for high-speed bearings as a measure against whirl instability and the use of high speeds to increase radial bearing stiffness. Interest in these deposits will increase in the context of the decline in the price of electronic components [Tuma 2018].

7 CONCLUSIONS

As was stated in the introduction, the advantage of journal hydrodynamic bearings is high radial load capacity and possibility of operation at high speeds. Predisposition to instability at high-speed rotation can be suppressed by active vibration control providing following benefits

- increasing the stable operating range of the bearing;
- reducing the friction losses of the journal bearing;
- increasing the radial stiffness of the journal bearing.
- simple manufacturing technology of cylindrical bearing bushing.

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. It has been demonstrated that the CompactRIO control unit can handle bearing control in real-time.

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REFERENCES

- [Babiuch 2014] Babiuch, M. Net Micro Framework Gadgeteer Measurement Applications Development. In: Proceedings of 15th International Carpathian Control Conference ICC'2014. Velké Karlovice, Czech Republic, May 28-30, 2014, pp. 10-13. ISBN: 978-1-47-993528-4, DOI: 10.1109/CarpathianCC.2014.6843560.
- [Buchacz 2014] Buchacz, A., Płaczek, M. and Wrobel, A. Modelling and analysis of systems with cylindrical piezoelectric transducers. *Mechanika*, 2014, Vol. 20, Issue 1, (87-91)
- [Dohnal 2011] Dohnal F., Markert R. and Hilsdorf T. Enhancement of external damping of a flexible rotor in active magnetic bearings by time-periodic stiffness. In: Proceedings of the SIRM, Internationale Tagung Schwingungen in rotierenden Maschinen, Darmstadt, Deutschland, 2011
- [Furst 1988] Furst, S. and Ulbrich, H. An Active Support System for Rotors with Oil-Film Bearings, In: Proceedings of IMechE, Serie C, 1988, (61-68), paper 261/88
- [Lau 2009] [5] Lau, H.Y., Liu, K.P., Wang, W., and Wong, P.L. Feasibility of Using Giant magnetostrictive material (GMM) Based Actuators in Active Control of Journal Bearing System, Proceedings of the World Congress on Engineering 2009 Vol II, WCE 2009, July 1-3, 2009, London, U.K.
- [Simek 2010] Simek, J., Tuma, J., Skuta, J. and R. Klecka, R. Unorthodox Behavior of a Rigid Rotor Supported in Sliding Bearings. In: Proceedings of the Colloquium Dynamics of Machines 2010, Inst. of Thermomechanics, Prague, February 2-3, 2010, (85-90)
- [Simek 2014] Simek, J., Tuma, J., Skuta, J. and Mahdal, M. Test stand for affecting rotor behavior by active control of sliding bearings. In: Proceedings of the Colloquium Dynamics of Machines 2014, Inst. of Thermomechanics, Prague, February 4-5, 2014, (151-156)
- [Takacs 2015] Takacs, G., Vachalek, J., Rohal-Ilkiv, B. Online structural health monitoring and parameter estimation for vibrating active cantilever beams using low-priced microcontrollers. In: Shock and vibration. Vol. 2015, 14 p., online. ISSN 1070-9622.
- [Tondl 1991] Tondl, A. Quenching of self-excited vibrations. Academia, Prague 1991
- [Tuma 2013] Tuma J., Simek J., Skuta J. and Los J. Active vibrations control of journal bearings with the use of piezoactuators. *Mechanical Systems and Signal Processing*, 2013, Vol. 36, (618-629)
- [Tuma 2018] Tuma J., Kozubkova M., Pawlenka M., Mahdal M., and Šimek J. Theoretical and experimental analysis of the bearing journal motion due to fluid force caused by the oil film. *MM (Modern Machinery) Science Journal*, October_2018, (2466-2472)
- [Viteckova 2011] Viteckova, M., Vitecek, A., and Babiuch, M. Unified Approach to Analog and Digital Two-Degree-of-Freedom PI Controller Tuning for Integrating Plants with Time Delay. *Acta*

Montanistica Slovaca, ročník 1(2011), No. 1, pp. 89-94. ISSN 1335-1788

[Wagnerova 2016] Wagnerova, R. and Tuma, J. Use of complex signals in modelling of journal bearings. In: 8th Vienna International Conference on Mathematical Modelling, MATHMOD 2015; Vienna; Austria; February 18-20 2015, (520-525).

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