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Numerical simulation of feedback controlled fluid-induced instabilities in rotor system supported by hydrodynamic bearings

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In certain operating conditions, the oil-film forces, by which the hydrodynamic bearings act on the rotor, destabilize and induce a self-excited vibration. There are many solution based on modifications of the bearing geometry to enlarge the operational range of the hydrodynamic bearing as a tilting pad bearing, lemon bore, etc. or usage of the dampers whose operation is based only on the dissipation of mechanical energy. In this paper several concepts based on controlled kinematic excitation of the bearing shells is investigated by computer simulation and by experimental test. The mathematical model of the uncontrolled and controlled rotor system contains the nonlinear hydrodynamic forces determined by the solution of the Reynolds equation assuming short case of bearing. To solve the equation of motion the Runge-Kutta method of the 4th order with Dormand-Prince modification and variable length of the integration step is used. The computer simulations were performed for rotor system without feedback control and with feedback controller. The main objectives of the numerical analysis were determination of the stability regions of the vibration excited by the imbalance forces. Results of the computer simulations proved that the analyzed approaches reduce amplitude of the rotor vibration caused by the imbalance forces. Experimental test and computer simulations demonstrate that the active vibration control increases considerably the rotation speed when the self-excited vibration of the oil-film occurred.

Key words: rotor system, hydrodynamic bearing instability, kinematic excitation

1. Introduction

Hydrodynamic bearings are often used due to their good damping characteristics and high load capacity. However, if the bearings operate in higher revolutions, the fluid excited instabilities, which are known as oil whirl [1] and oil whip [2] can appear. The oil whirl is a subsynchronous precessional movement with a frequency close to half of the rotational speed. The hydraulic forces, by which the oil-film acts on the shaft, destabilize

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the rotor motion and induce a self-excited vibration with typically very high amplitude of the vibration. This produces a large eccentricity of the shaft journal that could collide with the bearing liner due to roughness of both surfaces. The shaft journals and bearing liners are overloaded by the pressure peaks in the oil-film. These undesirable operating conditions reduce service lifetime or even they could cause a sudden failure of the rotor system.

In order to develop high performance rotating machines supported by the hydrodynamic bearings, growing attention has been paid to passive or active vibration control to improve their dynamic properties. It is known that the modification of the shape of the bearing gap (by a tilting pad bearing, a lemon bore, etc.), the increasing the system damping, etc. are passive techniques of the vibration attenuation. The passive damping devices operate on a principle of dissipation of mechanical energy and they do not destabilize vibration of the rotor. Therefore, in the cases the rotor systems operate in a varying of conditions (such as a wide range of speeds, a strong outer excitation, more variable loads, etc.) an active control by implementation of various kinds of devices (by an active journal bearing, a magnetic bearing, a piezoelectric bearing pusher, etc.) is more efficient.

Modeling and analysis of a rotor system with a new type of active oil bearing is presented in the paper [3]. By adjusting the chamber pressure, it can be controlled the deformation of the sleeve and subsequently the thickness of the oil-film, which changes dynamical properties of the rotating machine. The active bearing is located in the middle of the rotor. Distribution of the oil-film pressure was modeled by means of the Reynolds equation. The presented numerical computations indicate that stability of the equilibrium position of a rotor system can be significantly improved with the proposed active bearing. A complete mathematical model for a multi-bearing rotor system incorporating this type active journal bearing has been presented in [4].

In the paper [5] the active control device including active magnetic bearing is described. This system is used to suppress instability of hydrodynamic bearing and to increase the operating range. The forces generated in journal bearing are obtained using the short bearing approximation of Reynolds equation. The mathematical model of the rotor system is built assuming the elastic but massless shaft supported by two identical hydrodynamic bearings. Position-, acceleration- and damping-feedback controllers used for the active magnetic bearing to remove the journal bearing instability are investigated and compared theoretically and numerically. All three controllers show efficiency in controlling the hydrodynamic bearing instability, however, the best properties embodies the damping feedback controller.

A wide class of active damping rotor supports employs piezoelectric elements to improve stability properties of the hydrodynamic bearings. Przybylowicz [6] presented concept based on setting the position of the bearing housing by means of piezoelectric actuators. Motion of the journal is measured by the sensors and the signal activates the actuators via a velocity feedback. Analytical examinations proved that the presented method brings additional inertia- and damping-like terms to the equations of motion and both parameters strongly contribute to the dynamic response of the journal bearing system. Carried out numerical simulations proved that the method is effective as it increases the critical speed twice or even more.

The dynamics rotor system supported by the hydrodynamic bearings with passive or active elements attenuating its vibration is inherently a nonlinear problem. Therefore nonlinear mathematical models must be implemented in modeling of the dynamics rotor system because important phenomena such as self-excited vibration and limit cycles can not be observed in linear models. Nevertheless in many cases linearized approaches have been used in the modeling and solving of rotor dynamic problems.

The dynamic characteristics of synchronously controlled hydrodynamic bearing in order to suppress whirl instability and to reduce the unbalance response with linearized model of the rotor-bearing can by found in [7]. The oil-film forces are computed by means of the Reynolds equation. The stability threshold and steady-state response of the rotor system is greatly improved and influenced by the phase difference and control gain.

The stabilization of the rotor system supported in hydrodynamic bearings is discussed in papers [8] and [9]. The motion equation of the rotor system has two degrees of freedom and is built for rigid shaft supported by two identical hydrodynamic bearings and their forces are expressed on the base of Muszynska's relation [2]. The numerical simulations of the rotor system with feedback controller have shown that it is possible to reduce the response amplitude on imbalance forces excitation and increase the stability threshold in the direction of higher speed revolutions.

Therefore, presented paper is focused on the assembly of nonlinear mathematical models and their solutions for a testing stand designed to verify the influence of the oil whirl and oil whip phenomenon with feedback control of hydrodynamic bearing dynamic properties. Compared to many others articles [7], [9] and [10] the mathematical model of the uncontrolled and controlled rotor system contains the nonlinear relations for oil-film forces. The numerical simulations were performed for the rotor system without feedback control and with feedback controller. The equilibrium position was found out. The stability in the small range around the equilibrium position and the response to the imbalance forces were examined and the limit for stability threshold was determined.

In order to verify the results of the numerical simulation the testing stand for research active control of the hydrodynamic bearing was designed by the TECHLAB, Ltd. This device were assembled and initiated into operation at the VSB - Technical University of Ostrava. The increasing of threshold speed for development of oil-film instability was verified by experiments on testing stand with bearing shells actuated by controlled piezoactuators.

2. Motion equation of rotor system supported by hydrodynamic bearings

The following properties are assigned to investigated rotor system:

(i) the shaft mass is concentrated in the shaft journal centres,

(ii) the shaft is considered to be a rigid body,

(iii) the rotor is supported by two identical hydrodynamic bearings and the oil-film forces are determined by the solution of Reynolds equation with the assumption of cylindrical short cavitated (π-film) bearing,

(iv) the gyroscopic effect of the shaft and inertia effect of bearing shells are neglected,

(v) the rotor system is loaded by imbalance force caused by the unbalanced shaft and by its own-weight and

(vi) the rotor is rotating in a constant angular velocity.

Under these assumptions the motion equation of the rotor system is set up in the stationary coordinate system, (left Fig. 1), x, y, z

$$\mathbf{M}\,\ddot{\mathbf{q}}_J = -2\,\mathbf{f}_B(\mathbf{q}_J,\,\mathbf{q}_B,\,\dot{\mathbf{q}}_J,\,\dot{\mathbf{q}}_B,\,t) + \mathbf{f}_{EQ} + \mathbf{f}_U(t)$$

$$\mathbf{M} = \begin{bmatrix} m & 0 \\ 0 & m \end{bmatrix},\ \mathbf{q}_J = \begin{Bmatrix} y_J \\ z_J \end{Bmatrix},\ \dot{\mathbf{q}}_J = \begin{Bmatrix} \dot{y}_J \\ \dot{z}_J \end{Bmatrix},$$
$$\ddot{\mathbf{q}}_J = \begin{Bmatrix} \ddot{y}_J \\ \ddot{z}_J \end{Bmatrix},\ \mathbf{q}_B = \begin{Bmatrix} y_B \\ z_B \end{Bmatrix},\ \dot{\mathbf{q}}_B = \begin{Bmatrix} \dot{y}_B \\ \dot{z}_B \end{Bmatrix}, \tag{1}$$
$$\mathbf{f}_B = \begin{Bmatrix} f_y \\ f_z \end{Bmatrix},\ \mathbf{f}_{EQ} = \begin{Bmatrix} 0 \\ -mg \end{Bmatrix},\ \mathbf{f}_U = \begin{Bmatrix} m\varepsilon\omega^2\cos(\omega t) \\ m\varepsilon\omega^2\sin(\omega t) \end{Bmatrix},$$

where y_J, z_J, $\dot{y}_J$, $\dot{z}_J$ and $\ddot{y}_J$, $\ddot{z}_J$ are displacements, speeds and accelerations of the shaft journal centre in the horizontal and vertical vibration plane (Fig. 1), y_B, z_B and $\dot{y}_B$, $\dot{z}_B$, are displacements and speeds of the bearing shell centre in the horizontal and vertical vibration plane, f_y is the horizontal and f_z vertical component of the hydrodynamic force, m is the shaft mass, ε is the shaft unbalance, ω is the angular velocity of shaft rotation, t is time and g is the gravitational acceleration.

The scheme of hydrodynamic bearing is given in the right Fig. 1. The bearing geometry is described in the stationary coordinate system, x, y, z, where the x-axis is perpendicular to the projection plane formed by the rotor-fixed coordinate system, r, t, x, where the r-axis goes through the bearing shell centre O_B and shaft journal centre O_J.

The hydrodynamic bearings are implemented into the computational model by means of force couplings. The clearance between the journal and the liner of the bearing is very narrow and therefore for determination of the force by which the layer of lubricant acts on the rotor journal the classical theory of lubrication based on application of a Reynolds equation has been used.

The thickness of the oil-film h at any location of the bearing circumference is defined by the following relation derived e.g. in [11].

$$h(\varphi) = \delta - e_J\cos(\varphi-\gamma),\ \ \delta = R - r_J,\ \ e_J = \sqrt{(y_J-y_B)^2+(z_J-z_B)^2}. \tag{2}$$

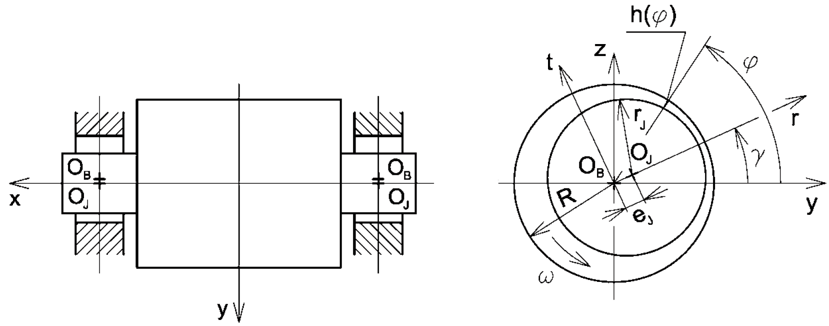

Figure 1. Scheme of a rigid rotor system with hydrodynamic bearings (left) and coordinate frames of the hydrodynamic bearing (right).

δ is the radial gap in case of centric shaft journal position, R is the radius of bearing shell, r_J is the radius shaft journal, e_J is the eccentricity between the bearing shell centre and shaft journal centre, γ is the position angle of the line of centres and φ is the circumferential coordinate.

The hydrodynamic force is determined by the solution of the Reynolds equation in the case of short bearing theory. The hydrodynamic force acting in the bearing centre can be expressed by radial f_r and tangential f_t component in the rotor-fixed coordinate system. According to [11], their relations for the short bearing are:

$$f_r = \eta R L \left(\frac{L}{\delta}\right)^2 \left[(\omega - 2\dot{\gamma})\frac{\varepsilon_J^2}{(1-\varepsilon_J^2)^2} + \frac{\pi(1+2\varepsilon_J^2)\dot{\varepsilon}_J}{2(1-\varepsilon_J^2)^{5/2}}\right],$$
$$f_t = -\eta R L \left(\frac{L}{\delta}\right)^2 \left[(\omega - 2\dot{\gamma})\frac{\pi\varepsilon_J}{4(1-\varepsilon_J^2)^{3/2}} + \frac{2\varepsilon_J\dot{\varepsilon}_J}{(1-\varepsilon_J^2)^2}\right], \tag{3}$$

$$\varepsilon_J = \frac{e_J}{\delta}, \quad \dot{\varepsilon}_J = \frac{(\dot{y}_J - \dot{y}_B)\cos(\gamma) + (\dot{z}_J - \dot{z}_B)\sin(\gamma)}{\delta},$$
$$\dot{\gamma} = \frac{-(\dot{y}_J - \dot{y}_B)\sin(\gamma) + (\dot{z}_J - \dot{z}_B)\cos(\gamma)}{\varepsilon_J}. \tag{4}$$

ε_J is the relative shaft journal centre eccentricity, L is the bearing length and η is the oil dynamical viscosity. Dot $(\cdot)$ denotes the first differentiation of the relative shaft journal centre eccentricity ε_J, the position angle γ and others with respect to time.

The goniometric functions of position angle γ according to the right Fig. 1 can be written as follows:

$$\cos(\gamma) = \frac{y_J - y_B}{e_J}, \quad \sin(\gamma) = \frac{z_J - z_B}{e_J}. \tag{5}$$

The hydrodynamic force is included in the motion equation of the rotor system (1) by its horizontal f_y and vertical f_z components expressed in the stationary coordinate system. Forces components can be written as:

$$f_y = f_r \cos(\gamma) - f_t \sin(\gamma), \quad f_z = f_r \sin(\gamma) + f_t \cos(\gamma). \tag{6}$$

3. Computational procedures for determination equilibrium position and stability evaluation

If a rotor system is loaded only by static forces (e.g. own-weight, constant forces, etc.) and does not vibrate, the motion equation (1) is transformed into the equation of equilibrium position:

$$\mathbf{0} = -2\,\mathbf{f}_B(\mathbf{q}_{EQ}, \mathbf{0}, \mathbf{0}, \mathbf{0}, 0) + \mathbf{f}_{EQ}, \tag{7}$$

where $\mathbf{q}_{EQ}$ is the vector of static displacements of the shaft journal centre, $\mathbf{f}_B$ is the vector of hydrodynamic forces and $\mathbf{f}_{EQ}$ is the vector of static forces exerting on the rotor system and its elements are shown in the motion equation (1).

The equilibrium equation of the rotor system is solved by the Newton-Raphson's method on the base of nonlinear algebraic equation (7). The load caused by the rotor system own-weight $\mathbf{f}_{EQ}$ acts in the vertical plane, however the equilibrium position of the shaft centre is shifted in the horizontal and vertical directions (see the left Fig. 5). This mutual dependence of the displacements in horizontal and vertical directions is caused by the hydrodynamic forces.

The uncontrolled rotor system eigenvalues are calculated under the assumption that the nonlinear coupling vector of the hydrodynamic forces $\mathbf{f}_B$, in the motion equation (1), is expanded in the Taylor's series in the surroundings of equilibrium position:

$$\mathbf{f}_B(\mathbf{q}_J, \mathbf{q}_B, \dot{\mathbf{q}}_J, \dot{\mathbf{q}}_B, t) = \mathbf{f}_B(\mathbf{q}_{EQ}, \mathbf{0}, \mathbf{0}, \mathbf{0}, 0) + \mathbf{D}_B\,(\dot{\mathbf{q}}_J - \mathbf{0}) + \mathbf{D}_K\,(\mathbf{q}_J - \mathbf{q}_{EQ}) + \ldots, \tag{8}$$

$$\mathbf{D}_B = \left[\frac{\partial \mathbf{f}_B}{\partial \dot{\mathbf{q}}}\right]_{\substack{\mathbf{q}_J=\mathbf{q}_{EQ},\, \mathbf{q}_B=\mathbf{0} \\ \dot{\mathbf{q}}_J=\mathbf{0},\, \dot{\mathbf{q}}_B=\mathbf{0}}}, \quad \mathbf{D}_K = \left[\frac{\partial \mathbf{f}_B}{\partial \mathbf{q}}\right]_{\substack{\mathbf{q}_J=\mathbf{q}_{EQ},\, \mathbf{q}_B=\mathbf{0} \\ \dot{\mathbf{q}}_J=\mathbf{0},\, \dot{\mathbf{q}}_B=\mathbf{0}}} \tag{9}$$

and then the motion equation can be rewritten as:

$$\mathbf{M}\,\ddot{\mathbf{q}}_J + 2\,\mathbf{D}_B\,\dot{\mathbf{q}}_J + 2\,\mathbf{D}_K\,\mathbf{q}_J = -2\,\mathbf{f}_B(\mathbf{q}_{EQ}, \mathbf{0}, \mathbf{0}, \mathbf{0}, 0) + 2\,\mathbf{D}_K\,\mathbf{q}_{EQ} + \mathbf{f}_{EQ} + \mathbf{f}_U(t), \tag{10}$$

where $\mathbf{D}_K$ and $\mathbf{D}_B$ are Jacobi matrixes of partial derivatives with respect to the displacements and velocities components, $\mathbf{M}$ is the mass matrix of the rotor system, $\mathbf{f}_U$ is vector forces exerting on the rotor system and $\mathbf{q}$, $\dot{\mathbf{q}}$ are vectors displacements and velocities of the rotor system.

The stability of the rotor system equilibrium position can be judged based on the size of eigenvalues real parts of linearized motion equation homogenous part (10). The equilibrium position is stable if the all real parts of eigenvalues are negative.

4. Motion equation of rotor system with feedback controller

The hydrodynamic force instability is controlled by kinematic excitation (Fig. 2), which actuates the bearing shell in opposite direction to displacements of the shaft journal centre, thus removing effects of these forces away. The bearing shell is kinematically excited in the horizontal and vertical directions with single feedback controller.

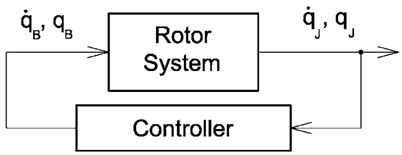

Figure 2. Scheme of a feedback controller.

The history of the kinematic excitation is determined depending on used controller type (P, PD and PID). The aim of the control is to reach the zero shaft journal centre displacement (the requested displacement value is equal to zero) and so the kinematic excitation time history (centre-displacements and speeds of the bearing shell) for feedback PID controller it can be determined out of the relations:

$$\mathbf{q}_B = -k_p\,\mathbf{q}_J - k_d\,\dot{\mathbf{q}}_J - k_I \int \mathbf{q}_J\,dt, \quad \dot{\mathbf{q}}_B = -k_p\,\dot{\mathbf{q}}_J - k_d\,\ddot{\mathbf{q}}_J - k_I\,\frac{d}{dt}\int \mathbf{q}_J\,dt \qquad (11)$$

where k_p, k_d, and k_I are the proportional, derivation and integration constants. The scheme of this control is displayed in the Fig. 2.

The motion equation of the rotor system supported by the hydrodynamic bearings with feedback controller can be obtained by replacing bearing shell displacements $\mathbf{q}_B$ and speeds $\dot{\mathbf{q}}_B$ (11) in the motion equation (1). If the rotor system is not controlled, the bearing shells do not move and therefore elements located in the vectors $\mathbf{q}_B$ and $\dot{\mathbf{q}}_B$ are equal to zero. The calculation of the equilibrium position and the stability judgment in its small surrounding is performed in similar way as in the case of uncontrolled rotor system.

5. Testing stand description and first result of experimental test

The TECHLAB, Ltd. assembled the testing stand (Fig. 3) are making it possible to investigate the instability suppression of hydrodynamic bearing with bearing shell outer kinematic excitation. In this stand [12] there is the hollow shaft with annular section driven by the electromotor with maximum revolutions of 24000 rpm. This shaft is supported by two identical hydrodynamic bearings. The bearing shells can be kinematically excited by practically arbitrary time history by piezoelectric actuators located in the testing stand. The rotor dynamic properties can be influenced by feedback kinematic excitation during operational rotation speed. Parameters of excitation are determined on the base of measured kinematic parameters.

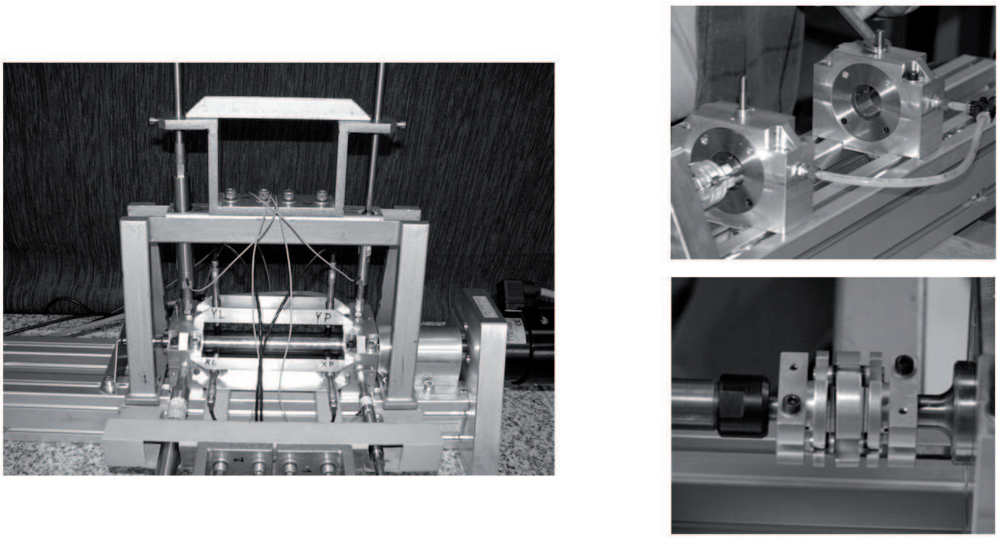

Figure 3. Photography of the testing stand designed by TECHLAB, Ltd. (left), the hydrodynamic bearings (upper right) and the flexible clutch (bottom right).

The experiments with bearing shells excited by active controlled piezoactuators connected decentrally were carried out on testing stand displayed in Fig. 3. The angular acceleration was constant and thus the increase of speed was linear during these experiments. The time history of rotor displacements near the hydrodynamic bearing is displayed in Fig. 4. The threshold of the oil whirl rise was 4300 min^{-1} without any active control of bearing shells (left Fig. 4). The speed of oil instability threshold were increased to 7300 min^{-1} (right Fig. 4) using the active P control of piezoactuators [13]. The actuators were activated at the speed 3000 min^{-1} because of limited piezoactuators working range. When the oil instability occurred, the active control and ramp up speed were switched off.

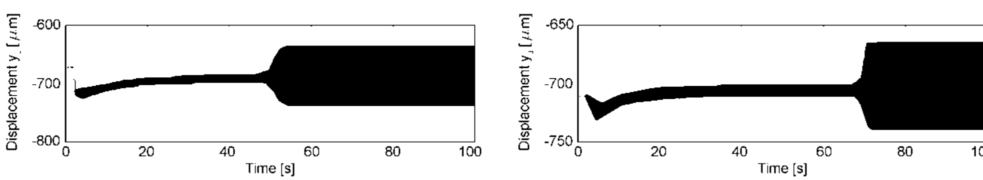

Figure 4. The displacements in horizontal direction without active control (left) and with active controlled piezoactuators (right).

6. Numerical simulation results

The computer simulations are carried out for geometrical and physical quantities according to the research reports [14] and [15] of the company TECHLAB, Ltd.

The pressure field of hydrodynamic bearing is approximated by the short bearing theory with following parameters: $L = 0.015$ m is the bearing length, $R = 15.0105$ mm is the radius of bearing shell, $r_J = 14.97$ mm is the radius shaft journal and $\eta = 0.004$ Pa·s is the oil dynamical viscosity. The testing stand is supposed to be rigid, its mass is $m = 0.780$ kg and the distance between the centre of gravity and geometric centre is $\varepsilon = 1.65 \cdot 10^{-5}$ m (shaft unbalance).

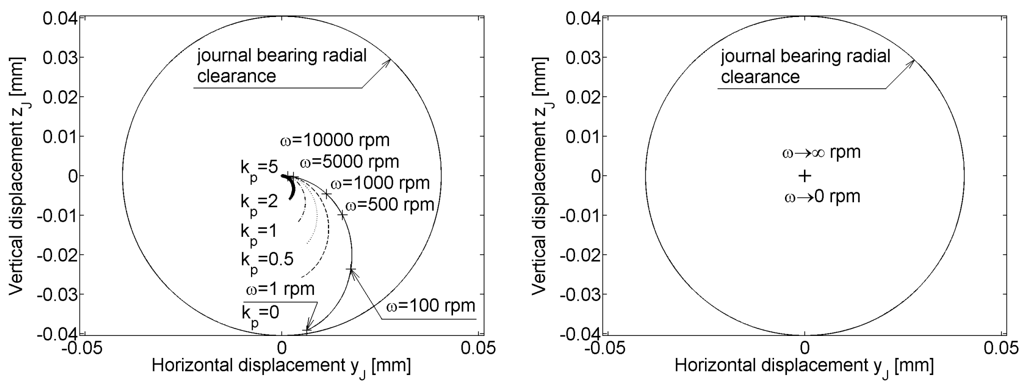

Figure 5. The equilibrium position of the uncontrolled rotor system, controlled with feedback P controller (left) and with feedback PID controller (right).

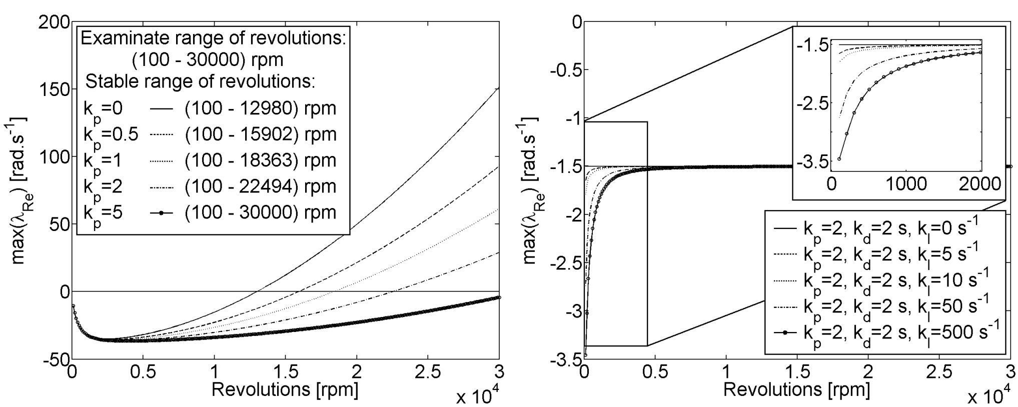

Figure 6. The maximum real part of the eigenvalue of the rotor system with feedback P (left) and PID (right) controller.

The equilibrium position of uncontrolled rotor system and controlled one with feedback P controller is shown in left Fig. 5. For the higher proportional constant of the shaft journal centre is getting closer to the bearing shell centre (left Fig. 5). The zero control deviation is not achieved by the P feedback controller because the shaft journal centre position is not situated in the bearing shell centre. The zero control deviation in the equilibrium position can be reached by a PID controller (right Fig. 5) with the arbitrary revolutions.

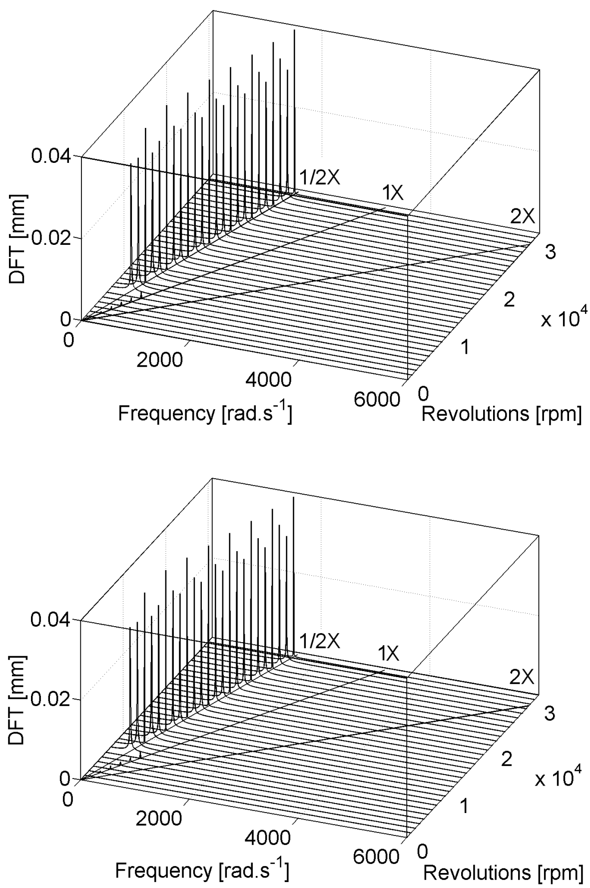

Figure 7. The cascade diagram of the Fourier's spectra determined from time history in the horizontal (upper) and vertical (lower) vibration plane of the uncontrolled rotor system.

The stability properties of the rotor system equilibrium position can be improved by the feedback P controller if greater value proportional constant is used which causes a wider range of stable revolutions (left Fig. 6). The stable equilibrium position of the rotor system in the whole investigated revolutions range can be ensured by PD controller ($k_p = 2$, $k_d = 2$ s). The maximum real part of the eigenvalue is strongly influenced by the integration constant of the PID controller (right Fig. 6). For all assumed values of the integration constants of PID controller, the maximum real parts of the eigenvalues are close to the values obtained with PD controller.

The response to the imbalance forces excitation of the controlled and uncontrolled rotor system is calculated by the direct integration of the motion equation. The Runge-Kutta method with an adaptive time step is applied for its solution. The cascade diagrams displayed in Fig. 7 and Fig. 8 were built on the base of Fourier's spectra calculated from the steady-state component of the rotor system response for the revolution range of (1000-30000) rpm. Frequency component of the synchronous vibration is commonly referred to as "1X" vibration. The straight-lines displayed in the cascade diagrams (Fig. 7 and Fig. 8) correspond to the operating speed frequency 1X, to the 1/2X and 2X (half and double of operating speed frequency). The 1X frequency dominates in the rotor system response in the horizontal and vertical vibration directions for lower revolutions - Fig. 7, because the influence of imbalance forces dominates in the response. The half-multiple 1/2X of the speed frequency starts to dominate in the revolution range (6551-6559) rpm. This interval is located slightly over the half revolutions corresponding to the stable revolutions of the equilibrium position (12980 rpm). In the revolution range of the rotor system of (6560-30000) rpm, a working state called oil whirl arises.

The frequency spectra determined from the steady-state responses calculated by the motion equation of the rotor system controlled by feedback P controller are displayed in the cascade diagrams Fig. 8. As it could be seen in the upper part of Fig. 7 and lower part of Fig. 8, for proportional constant of the controller $k_p = 50$ the steady-state vibration amplitude of the uncontrolled system was reduced approximately eighty times using the controlled rotor system.

It was found out that in the case of rotor system with feedback P controller, the threshold of the oil whirl rise was shifted to higher revolutions. The proportional constant in range $k_p = 50$ to $k_p = 500$ does not increase the stability limit significantly. The stability limit with the P controller and the proportional constant of $k_p = 500$ compared to limit with the uncontrolled rotor system was increased by about 810 rpm.

The rotor system controlled by P feedback controller for the revolutions lying over the limit for oil whirl rise (8000 rpm) in the bearing is displayed in Fig. 9. The steady-state orbit reduction of controlled rotor system with increasing of proportional constant is evident as could be seen in Fig. 9.

In the next step, the response to the imbalance forces excitation of the rotor system controlled by feedback PID controller was investigated. It was found out by the numerical simulations that the revolution threshold for oil whirl rise of the rotor system controlled by feedback PID controller is practically independent on its proportional and integration constants, at least for chosen value of the derivation constant (Table 1).

The revolution threshold of oil whirl rise for rotor system controlled by PID controller is close to the revolution threshold found out for the rotor system controlled by feedback P controller (for proportional constant equal at least to $k_p = 500$).

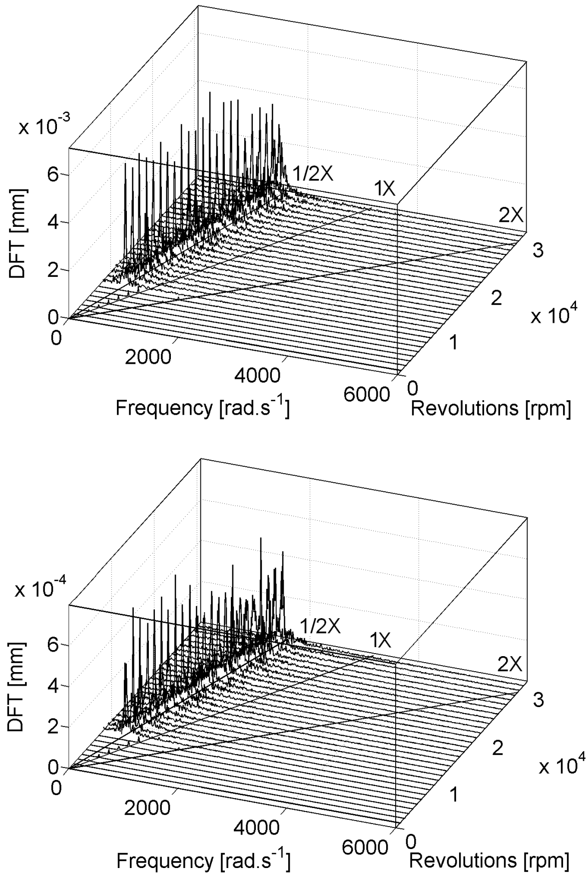

Figure 8. The cascade diagram of the Fourier's spectra determined from time histories in the horizontal vibration plane of the rotor system controlled by P controller for $k_p = 5$ (upper) and for $k_p = 50$ (lower).

7. Conclusions

In the presented contribution it is shown that by the feedback kinematic excitation of the bearing shells it is possible to increase the revolutions when the phenomenon called oil whirl rises. Simultaneously, throughout the whole investigated rotor revolution range the shaft journal centre vibration magnitude was reduced. This can be reached by P and PID feedback controller type, as it was numerically tested.

Subsequently was found out that the dynamic characteristics (such as the shaft journal centre vibration magnitude and the oil whirl phenomenon rise in the bearing) of the rotor system can be influenced by the feedback control when the nonlinear mathematic model of the oil-film forces based on the Reynolds equation solution is assumed.

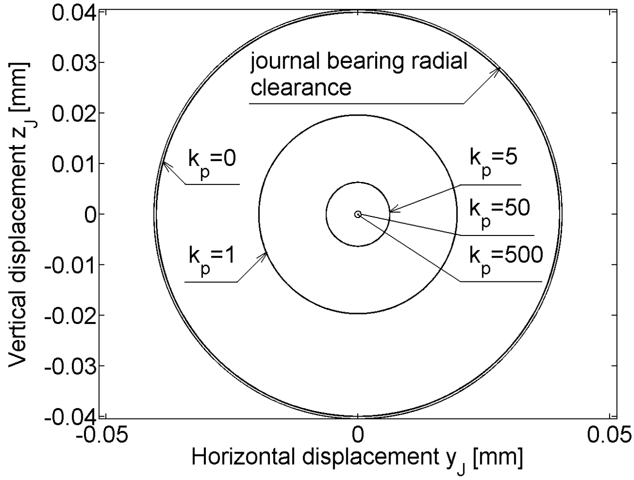

Figure 9. Orbit of the shaft journal centre in the steady-state for 8000 rpm.

Table 1. Oil whirl rise for rotor system controlled by the feedback PID controller

The threshold revolution of oil whirl rise				
Proportional constant k_p [-]	Derivation constant k_d [s]	Integration constant k_I [s^{-1}]	Oil whirl rise interval n_o [rpm]	
1	1	5	7351	7359
5	1	5	7351	7359
50	1	5	7361	7369

In the present time, the experimental results for uncontrolled and controlled rotor system by the proportional controller type are obtained. Experiment on the testing stand and numerical simulations proved that active control of hydrodynamic bearings increase the threshold speed for developing of oil-film instabilities due to self-exciting forces. The problems with misalignment, selection of oil, the way of bearing lubrication, proximity probes accuracy and speed course during experiments had to be solved during the testing stand development. Some of these problems were analyzed using numerical simulations.

The experiments enabling run up, coast down and steady-state rotation for rotor controlled by more sophisticated controllers would be performed when sufficient practice would be obtained with testing stand controlled by classical controllers. Moreover, the testing stand would be used in future. It would make possible to verify mathematical models of hydrodynamic bearings, damping of rotor vibrations, active vibration control and speed control during getting over critical speed using the testing stand.

References

[1] T. YAMAMOTO and Y. ISHIDA: Linear and nonlinear rotordynamics. A modern treatment with applications. John Wiley & Sons, Inc. 2001. 325p., ISBN 0-471-18175-7.

[2] A. MUSZYNSKA: Whirl and whip – rotor/bearing stability problems. *In Proc. of the Symp. on Instability in Rotating Machinery*. Carson City, Nevada. NASA C. P. No. 2409, (1985), 155-178.

[3] J.M. KRODKIEWSKI, Y. CEN and L. SUN: Improvement of stability of rotor system by introducing a hydraulic damper into an active journal bearing. *Int. J. of Rotating Machinery*, **3**(1), (1997), 45-52.

[4] J.M. KRODKIEWSKI and L. SUN: Modelling of multi-bearing rotor systems incorporating an active journal bearing. *J. of Sound and Vibration*, **210**(3), (1998), 215-229.

[5] A. EL-SHAFEI and A.S. DIMITRI: Controlling journal bearing instability using active magnetic bearings. *In Proc. of ASME Turbo Expo 2007*, Montreal, Canada. Paper GT2007-28059, (2007), 11p.

[6] P.M. PRZYBYŁOWICZ: Active stabilization of a rigid rotor by a piezoelectrically controlled mobile journal bearing system. *Australian J. of Mechanical Engineering*, **1**(2), (2004), 123-127.

[7] R. BYOUNG-HOO and K. KYUNG-WOONG: A study of the dynamic characteristics of synchronously controlled hydrodynamic journal bearings. *Tribology International*, **35**(5), (2002), 339-345.

[8] A. VÌTEČEK, J. TŮMA, M. VÌTEČKOVÁ: Stabilization of rotor in journal bearing. *Acta Mechanica Slovaca*, **12**(3), (2008), 899-906, (in Czech).

[9] A. VÌTEČEK, J. TŮMA, M. VÌTEČKOVÁ: Active reduction of rotor instability. *In Proc. of the Int. Conf. Cybernetics and Informatics*, Ždiar, Slovak Republic, (2008), 5p., (in Czech).

[10] J. ONDROUCH, P. FERFECKI and Z. PORUBA: Active vibration reduction of rigid rotor by kinematic excitation of bushes of journal bearings. *Metallurgy*, **49**(2), (2010), 107-110.

[11] E. Krämer: Dynamics of rotors and foundations. Springer-Verlag, 1963, 383p., ISBN 3-540-55725-3.

[12] L. Smutný, J. Škuta, R. Klečka, J. Tůma and J. Šimek: Piezoactuators in active control of journal bearing. *In 16th Int. Congress on Sound and Vibration*, The International Institute of Acoustics and Vibration (IIAV). Krakow, Poland, (2009), 16p.

[13] J. Tůma, J. Škuta, R. Klečka, J. Los, J. Šimek: A laboratory test stand for active control of journal bearing. *In Proc. of Dynamics of Machines 2010*, Institute of Thermomechanics AS Czech Republic, Prague, Czech Republic, (2010), 6p.

[14] R. Svoboda: Theoretical analysis of the influence of bearing shell external excitation on the rotor dynamic properties. Technical report Nr. 07-301, TECHLAB, Ltd., Praha, 2007, (in Czech).

[15] J. Šimek: Test stand for active control of the bearings with the aim to suppress the rotor instability. Research report Nr. 07-405, TECHLAB, Ltd., Praha, 2007, (in Czech).