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FEEDBACK CONTROL OF JOURNAL BEARING INSTABILITY

ZPĚTNOVAZEBNÍ ŘÍZENÍ NESTABILITY KLUZNÉHO LOŽISKA

Abstract

The interaction between the rotating system and the oil film in journal bearing causes an undesirable dynamic behaviour, which is called the oil whirl and the oil whip. The presented paper deals with the reduction of oil whirl effects by means of the feedback control of the bearing shell by kinematic excitation. The numerical simulations were performed for the rotor system supported in journal bearings without feedback control and with PD feedback controller as well. The influence of PD controller parameters of the equilibrium position, the stability in small surrounding of the equilibrium position, the response on the unbalance force excitation and the vibration oil whirl rise limit are determined.

Abstrakt

Vzájemné ovlivňování otáčejícího se rotoru s olejovým filmem kluzného ložiska způsobuje nežádoucí dynamické chování, které se nazývá "oil whirl" a "oil whip". Představený článek se zabývá snížením účinků "oil whirl" pomocí zpětnovazebního kinematického buzení ložiskové pánve. Numerické simulace byly provedeny pro neřízenou a PD regulátorem zpětnovazebně řízenou rotorovou soustavu uloženou v kluzných ložiskách. Je určen vliv parametrů PD regulátoru na rovnovážnou polohu, na stabilitu v blízkém okolí rovnovážné polohy, na odezvu od buzení nevývahou a na vzrůst kmitání na hranici "oil whirl".

1 INTRODUCTION

The journal bearings are often used due to their good damping characteristics and high load capacity. However if the bearings are lightly loaded and operated in higher revolutions the fluid excited instabilities known as oil whirl and oil whip can appear. The oil whirl phenomenon occurs when the shaft is whirling at a frequency close to one-half of rotor angular speed. When the rotation speed reaches twice the first natural frequency the oil whip instability can be detected. In case of strong outer excitation of the rotor it is not possible to avoid the increase of those effects by improvement of dynamic properties, for example by use of the tilting pad journal bearing. One possibility to reduce the effects of oil instabilities is to control the journal bearing properties by feedback excitation of the bearing shell.

In the paper [2] an active magnetic bearing is used for instability passage of journal bearing and for increasing of the operating range. The forces generated in journal bearing are obtained using the short bearing approximation for Reynolds equation. The rotor system mathematical model is build under assumption of elastic but massless shaft supported in two same journal bearings. Position-, acceleration- and damping-feedback controllers used for the active magnetic bearing removing the journal bearing instability are examined and compared theoretically and numerically. All three controllers show efficiency in controlling the journal bearing instability whereas the best properties embodies the damping feedback controller.

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The dynamic characteristics of synchronously controlled journal bearing in order to suppress whirl instability and to reduce the unbalance response are investigated in article [1]. The oil film forces are computed by means of Reynolds equation. It is assumed that the rotor is rigid and the oil film force in the journal bearing is approximated by linear theory. 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 journal bearings is discussed in the paper [6]. The rotor system motion equation has two degrees of freedom and is build for rigid shaft supported in two same journal bearings and their forces are expressed in the base of Muszynska's relation [4]. The numerical simulations of the rotor system with feedback controller have shown that it is possible to reduce the response on the unbalance forces excitation and also to increase the stability threshold [4] in the direction to higher revolutions.

At present time the feedback controlled journal bearings in the field of rotor dynamics are a subject of intensive research, both experimental and theoretical. The testing stand for mitigation of the fluid-instability of journal bearing with kinematic excitation was proposed in the company TECHLAB, Ltd. The aim of presented contribution is to show that the response magnitude on the unbalance force excitation and the rise of oil whirl phenomenon can be reduced by the feedback PD controller. Compare to other authors the numerical simulations are performed for the feedback controlled rotor system, where the nonlinear oil film forces determined from the solution of the oil film Reynolds equation assuming that case of cylindrical short cavitated bearing.

2 MOTION EQUATION OF ROTOR SYSTEM SUPPORTED IN JOURNAL BEARINGS

The following properties are assigned to investigated rotor system: (i) the shaft mass is concentrated to the of shaft journal centre, (ii) the shaft is considered as a rigid body, (iii) the rotor is supported in two identical journal bearings and the oil film forces are determined from the solution of Reynolds equation with the assumption of cylindrical short cavitated bearing, (iv) the gyroscopic effect of shaft and inertia effect of bearing shell is neglected, (v) the rotor system is loaded by unbalance force caused by the unbalanced shaft and by its self-weight and (vi) the rotor is rotating in constant angular velocity.

Under those assumptions the rotor system motion equation is built up in the stationary coordinate system, x, y, z

$$\mathbf{M}\,\ddot{\mathbf{q}}_\mathrm{J} = -2\,\mathbf{f}_\mathrm{B}\left(\mathbf{q}_\mathrm{J}, \mathbf{q}_\mathrm{B}, \dot{\mathbf{q}}_\mathrm{J}, \dot{\mathbf{q}}_\mathrm{B}, t\right) + \mathbf{f}_\mathrm{EQ} + \mathbf{f}_\mathrm{U}(t),$$

$$\mathbf{M} = \begin{bmatrix} m & 0 \\ 0 & m \end{bmatrix},\quad \mathbf{q}_\mathrm{J} = \begin{Bmatrix} y_\mathrm{J} \\ z_\mathrm{J} \end{Bmatrix},\quad \mathbf{q}_\mathrm{B} = \begin{Bmatrix} y_\mathrm{B} \\ z_\mathrm{B} \end{Bmatrix},\quad \dot{\mathbf{q}}_\mathrm{J} = \begin{Bmatrix} \dot{y}_\mathrm{J} \\ \dot{z}_\mathrm{J} \end{Bmatrix},\quad \dot{\mathbf{q}}_\mathrm{B} = \begin{Bmatrix} \dot{y}_\mathrm{B} \\ \dot{z}_\mathrm{B} \end{Bmatrix},\quad \ddot{\mathbf{q}}_\mathrm{J} = \begin{Bmatrix} \ddot{y}_\mathrm{J} \\ \ddot{z}_\mathrm{J} \end{Bmatrix}, \tag{1}$$

$$\mathbf{f}_\mathrm{B} = \begin{Bmatrix} f_\mathrm{y} \\ f_\mathrm{z} \end{Bmatrix},\quad \mathbf{f}_\mathrm{EQ} = \begin{Bmatrix} 0 \\ -m\,g \end{Bmatrix},\quad \mathbf{f}_\mathrm{U} = \begin{Bmatrix} m\,\varepsilon\,\omega^2\cos(\omega\,t) \\ m\,\varepsilon\,\omega^2\sin(\omega\,t) \end{Bmatrix},$$

where y_J, z_J, $\dot{y}_\mathrm{J}$, $\dot{z}_\mathrm{J}$ and $\ddot{y}_\mathrm{J}$, $\ddot{z}_\mathrm{J}$ are displacements, speeds and accelerations of shaft journal centre in horizontal and vertical vibration plane, y_B, z_B and $\dot{y}_\mathrm{B}$, $\dot{z}_\mathrm{B}$ are displacements and speeds of bearing shell centre in horizontal and vertical vibration plane, f_y is the horizontal and f_z vertical component of the oil film 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. In case of uncontrolled rotor system all members located in the vectors $\mathbf{q}_\mathrm{B}$ and $\dot{\mathbf{q}}_\mathrm{B}$ are equal to zero.

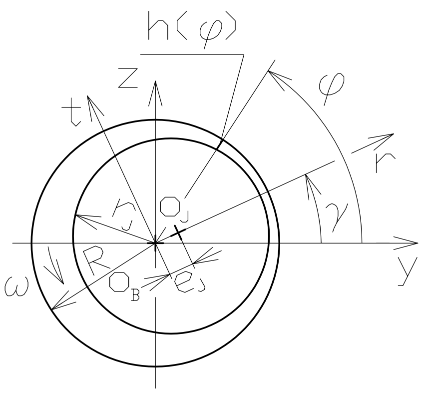

Fig. 1 Coordinate frames of the journal bearing

The bearing geometry (Fig. 1) is described in the stationary coordinate system, x, y, z, where the x-axis is perpendicular to the projection plane and in the rotor-fixed coordinate system, r, t, x, where the r-axis goes through the bearing shell centre and shaft journal centre. The oil film thickness of cylindrical bearing in general position is

$$h(\varphi) = \delta - e_{\mathrm{J}} \cos(\varphi - \gamma), \; \delta = R - r_{\mathrm{J}}, \quad e_{\mathrm{J}} = \sqrt{(z_{\mathrm{J}} - z_{\mathrm{B}})^2 + (y_{\mathrm{J}} - y_{\mathrm{B}})^2}, \tag{2}$$

where 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 centre shaft journal and φ is the absolute circumferential angle.

The oil film force is determined from the solution of the oil film Reynolds equation for the case of cylindrical short cavitated (π-film) bearing. The oil film 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 [3] their relations for cylindrical short cavitated bearing are

$$f_{\mathrm{r}} = \eta \, R \, L \left(\frac{L}{\delta}\right)^2 \left[(\omega - 2\,\dot{\gamma}) \frac{\varepsilon_{\mathrm{J}}^2}{(1-\varepsilon_{\mathrm{J}}^2)^2} + \frac{\pi \left(1 + 2\,\varepsilon_{\mathrm{J}}^2\right)\dot{\varepsilon}_{\mathrm{J}}}{2\left(1-\varepsilon_{\mathrm{J}}^2\right)^{5/2}}\right],$$
$$f_{\mathrm{t}} = -\eta \, R \, L \left(\frac{L}{\delta}\right)^2 \left[(\omega - 2\,\dot{\gamma}) \frac{\pi\,\varepsilon_{\mathrm{J}}}{4\,(1-\varepsilon_{\mathrm{J}}^2)^{3/2}} + \frac{2\,\varepsilon_{\mathrm{J}}\,\dot{\varepsilon}_{\mathrm{J}}}{\left(1-\varepsilon_{\mathrm{J}}^2\right)^2}\right] + 2\,R\,L\,p_{\mathrm{a}}, \tag{3}$$

$$\varepsilon_{\mathrm{J}} = e_{\mathrm{J}}/\delta, \; \dot{\varepsilon}_{\mathrm{J}} = \frac{(\dot{y}_{\mathrm{J}} - \dot{y}_{\mathrm{B}})\cos(\gamma) + (\dot{z}_{\mathrm{J}} - \dot{z}_{\mathrm{B}})\sin(\gamma)}{\delta}, \; \dot{\gamma} = \frac{-(\dot{y}_{\mathrm{J}} - \dot{y}_{\mathrm{B}})\sin(\gamma) + (\dot{z}_{\mathrm{J}} - \dot{z}_{\mathrm{B}})\cos(\gamma)}{\varepsilon_{\mathrm{J}}}, \tag{4}$$

where ε_{J} is the relative shaft journal centre eccentricity, L is the bearing length, η is the oil dynamic viscosity and p_{a} is the ambient pressure at the bearing faces. The goniometric functions of position angle γ according to the Fig. 1 can be written as follows

$$\cos(\gamma) = \frac{y_{\mathrm{J}} - y_{\mathrm{B}}}{e_{\mathrm{J}}}, \; \sin(\gamma) = \frac{z_{\mathrm{J}} - z_{\mathrm{B}}}{e_{\mathrm{J}}}. \tag{5}$$

The oil film force is included in the rotor system motion equation (1) by horizontal f_{y} and vertical f_{z} component expressed in the stationary coordinate system. They can be written as

$$f_{\mathrm{y}} = f_{\mathrm{r}} \cos\gamma - f_{\mathrm{t}} \sin\gamma, \; f_{\mathrm{z}} = f_{\mathrm{r}} \sin\gamma + f_{\mathrm{t}} \cos\gamma. \tag{6}$$

If a rotor system is loaded only by static forces (e.g. self-weight, etc.) and does not vibrate, the motion equation (1) is transformed into the set of algebraic equations. The equilibrium equation of the rotor system is solved by the Newton-Raphsons method on the base of nonlinear algebraic equations.

The rotor system eigenvalues are calculated under the assumption that the nonlinear coupling vector of the oil film forces $\mathbf{f}_{\mathrm{B}}$, in the motion equation (1), is expanded to the Taylor series in the surroundings of equilibrium position. The stability of the rotor system equilibrium position can be judged based on the size of eigenvalues real parts of linearized motion equation.

The rotor system response on the unbalance force excitation is determined by integration of nonlinear motion equation.

3 MOTION EQUATION OF ROTOR SYSTEM WITH FEEDBACK CONTROLLER

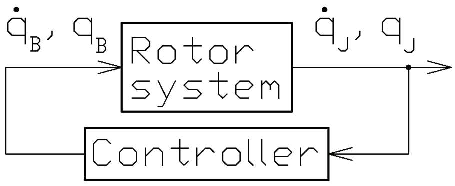

Fig. 2 Scheme of a feedback controller

The oil film force instability is controlled by kinematic excitation (Fig. 2), which is actuating on bearing shell in opposite direction to displacements of shaft journal, thus removing effects of these forces. The bearing shell is kinematically excited in horizontal and vertical vibration direction with one feedback controller.

The time history of the kinematic excitation is determined in the dependence of used controller type (P and PD). The aim of the control is to reach the zero shaft journal centre shift (the requested shift value is equal to zero) and so the kinematic excitation time history (centre-displacements and speeds of the bearing shell) for feedback PD controller can be determined from the relations

$$\mathbf{q}_{\mathrm{B}} = -k_{\mathrm{p}}\,\mathbf{q}_{\mathrm{J}} - k_{\mathrm{d}}\,\dot{\mathbf{q}}_{\mathrm{J}}\,,\ \dot{\mathbf{q}}_{\mathrm{B}} = -k_{\mathrm{p}}\,\dot{\mathbf{q}}_{\mathrm{J}} - k_{\mathrm{d}}\,\ddot{\mathbf{q}}_{\mathrm{J}}\,, \tag{7}$$

where k_p and k_d is the proportional and derivation constant. The scheme of this controlled is at the Fig. 2. The motion equation of the rotor system supported in journal bearings with feedback controller can be obtained by replacing bearing shell displacements $\mathbf{q}_{\mathrm{B}}$ and speeds $\dot{\mathbf{q}}_{\mathrm{B}}$ into the motion equation (1).

4 DEVELOPING TESTING STAND

In the company TECHLAB, Ltd. the testing stand (Fig. 3) making it possible to investigate the instability suppression of journal bearing with bearing shell outer kinematic excitation was proposed. In this stand there is the hollow shaft with annular section driven by the electromotor with maximum revolutions of 24000rpm. This shaft is supported in two same journal bearings. The bearing shell can be kinematic excited by practically arbitrary time history by piezoelectric actuators located in the stand. The rotor dynamic properties can be influenced by feedback kinematic excitation during unstable run whose parameters are determined on the base of measured displacements, speeds and accelerations respectively.

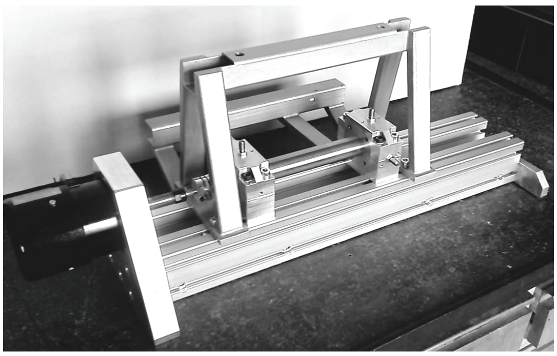

Fig. 3 Photography of mounting some parts of the testing stand

The debugging and checking of the testing stand properties is in progress at the Department of Automation and Control of Technical University of Ostrava in the present time.

The majority of geometrics and physical quantities in mathematical model are determined by means of research report [5]. The pressure field of journal bearing is approximated by the cylindrical short cavitated bearing with following parameters: L=0.015m is the bearing length, R=15.0105mm is the radius of bearing shell, r_J=14.97mm is the radius shaft journal, η=0.004Pa·s is the oil dynamic viscosity, p_a=0Pa is the ambient pressure at the bearing faces. The testing stand is supposed to be rigid, its mass is m=0.780kg and the shaft unbalance is ε=1.65·10^{-5}m.

5 NUMERICAL SIMULATION RESULTS

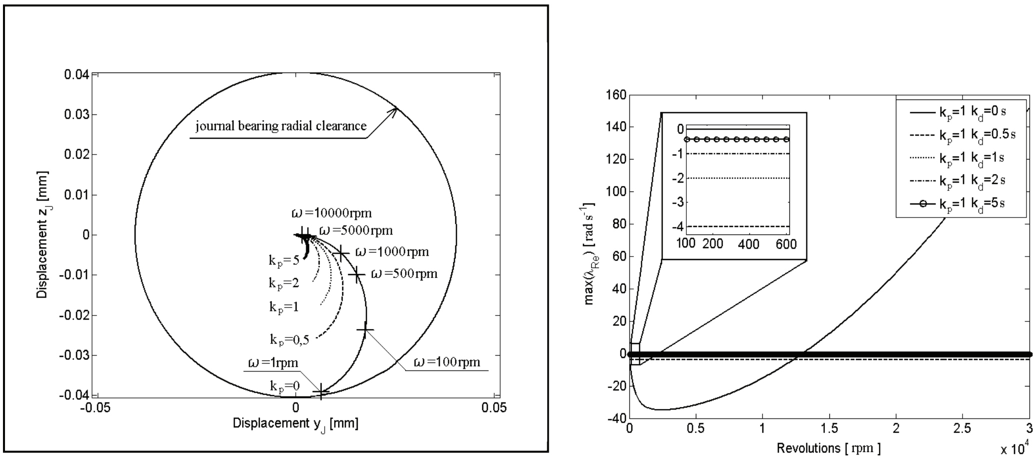

Fig. 4 The equilibrium position of the uncontrolled rotor system, controlled with feedback PD controller (left) and the maximum real part of the eigenvalue of the rotor system ($\max\{\lambda_{Re}\}$) with feedback PD controller (right)

The equilibrium position of uncontrolled rotor system and controlled one with feedback PD controller is shown by left Fig. 4. The equilibrium positions of rotor system controlled by feedback P or PD controller are identical because for mentioned controllers can be written $\dot{\mathbf{q}}_J = \mathbf{0}$ and $\ddot{\mathbf{q}}_J = \mathbf{0}$. For the higher proportional constant shaft journal centre is getting closer to the bearing shell centre (left Fig. 4). The zero control deviation is not achieved by the P and PD feedback controller because the shaft journal centre position is not situated in the bearing shell centre.

The stability of the rotor system equilibrium position can be increased by the feedback P controller and by higher proportional constant causes a wider range of stable revolutions. The stable equilibrium position of the rotor system in the whole investigated revolutions range can be ensured by PD controller (right Fig. 4). Due to the increase of derivation constant of PD controller the maximal real part of the eigenvalue decreases. Its value is practically independent of the rotor system revolutions.

The response on the unbalance force excitation of the controlled and uncontrolled rotor system is calculated by the direct integration of motion equation. The presented cascade diagrams built up on the base of Fourier spectra calculated from the steady-state component of the rotor system response for the revolution range of (1000÷30000)rpm are graphically given by Fig. 5, left Fig. 6 and Fig. 7. The synchronous excitation of the basic speed frequency ($\omega = n \cdot 60/2/\pi$, where n are revolutions), the straight-line corresponding to the ½ ($\omega = n \cdot 60/\pi$) and to double ($\omega = n \cdot 60/4/\pi$) of basic speed frequency are displayed in the bottom plane of mentioned cascade diagram.

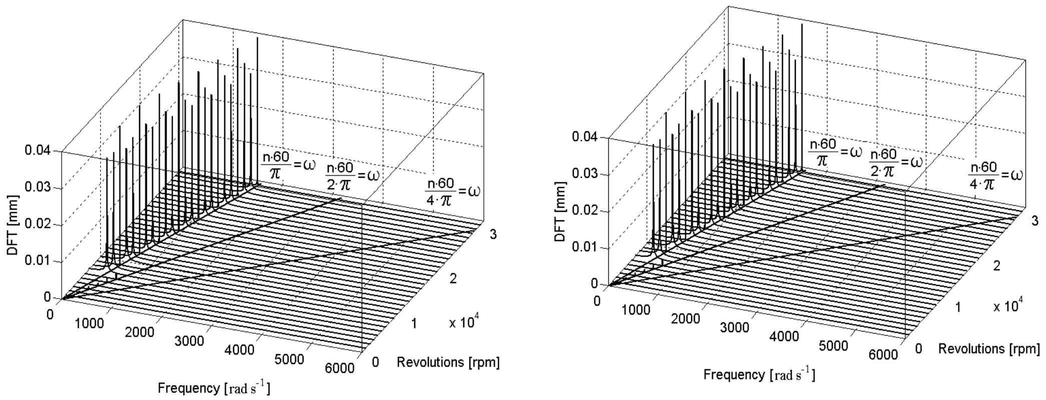

Fig. 5 The cascade diagram of the Fourier spectra from the time history in horizontal (left) and vertical (right) direction of the uncontrolled rotor system

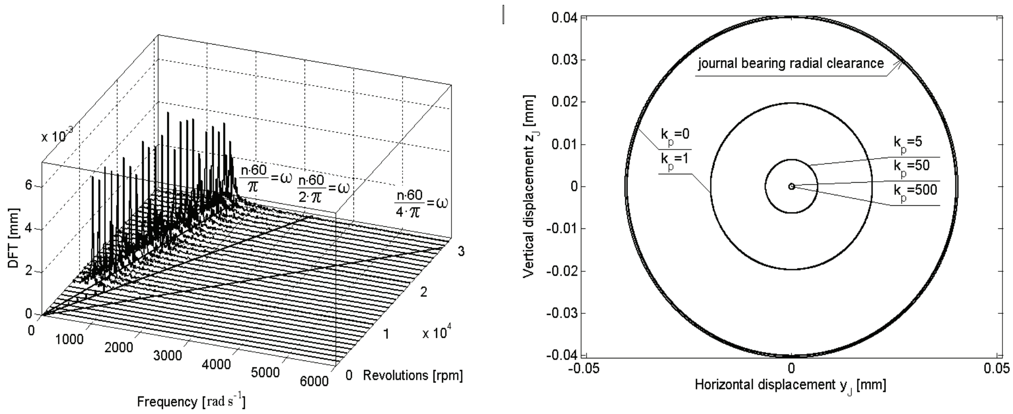

Fig. 6 The cascade diagram of the Fourier spectra from the time history in horizontal direction of the rotor system controlled by P feedback controller k_p=5 (left) and orbit of the shaft journal centre in steady-state for 8000rpm (right)

The rotor system response is in horizontal and vertical direction upon low revolutions determined by the first multiple of the speed frequency - Fig. 5, because the influence of unbalance forces dominates in the response. The half-multiple of the speed frequency starts to dominate in the revolution range (6551-6559)rpm, (Fig. 5). This interval is located slightly over the half revolutions corresponding to the stable revolutions of equilibrium position (12980rpm). So in the revolution range of the rotor system of (6560-30000)rpm is coming on a working state called oil whirl.

The cascade diagram displayed at left Fig. 6 is built from the steady-state response component calculated from the motion equation of the rotor system controlled by feedback P controller. From left Fig. 6 it is obvious, that for proportional constant of the controller k_p=5 the steady-state vibration magnitude of the uncontrolled system was reduced approximately eight times regarding to the controlled rotor system.

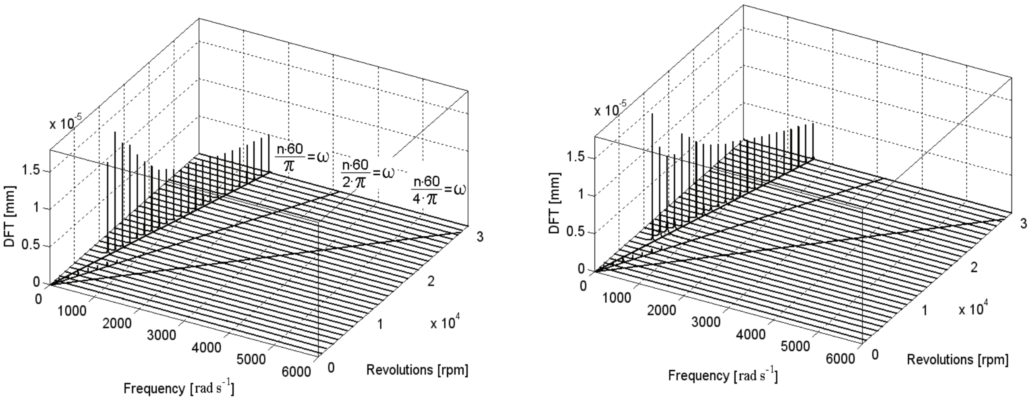

Fig. 7 The cascade diagram of the Fourier spectra from the time history in horizontal direction of the rotor system controlled by PD feedback controller k_p=50, k_d=5s (left) and k_p=0, k_d=5s (right)

The orbit of rotor system controlled by P feedback controller for the revolutions lying over the limit for oil whirl rise (8000rpm) in the bearing is displayed at right Fig. 6. In case of uncontrolled rotor system the orbit size grows to the size similar to the journal bearing radial clearance and in the frequency response dominates the half-multiple of basic speed frequency. From right Fig. 6 the steady-state orbit reduction of controlled rotor system with increasing proportional constant is obvious.

Tab 1. Oil whirl rise for rotor system controlled by feedback PD controller

Oil whirl rise limit			
Proportional constant k_p [-]	Derivation constant k_d [s]	Oil whirl rise interval [rpm]	
0	0	6551	6559
1	0	6921	6929
5	0	7201	7209
50	0	7341	7349
500	0	7361	7369
1000	0	7361	7369
5000	0	7361	7369
0	1	7361	7369
0	5	7361	7369
0	10	7361	7369
50	1	7361	7369
50	5	7361	7369

It was shown that the steady-state vibration magnitude can be even reduced by feedback D controller. Compare to the response calculated by P controller is the steady-state vibration magnitude approximately 200 times reduced (Fig. 7).

The stability limit of the P controller with the proportional constant of k_p=500 compared to the stability limit of the uncontrolled rotor system was increased by about 810rpm. The growth of proportional constant from the value of k_p=500 to k_p=1000 and derivation constant from the value of k_d=1s to k_d=10s did not cause the stability shift in the direction of higher revolutions (Tab. 1).

6 CONCLUSIONS

In the presented contribution it is shown that by the feedback kinematic excitation of the bearing shell it is possible to increase the revolutions until the phenomenon oil whirl rises. Simultaneously it was proven that the dynamic characteristics of the rotor system can be influenced by PD feedback control when the nonlinear mathematic model of the oil film forces based on the Reynolds equation solution is assumed. Further, in the whole investigated rotor revolution range the shaft journal centre vibration magnitude was reduced. It was found out by the numerical simulations that the revolution limit for oil whirl rise of rotor system controlled by feedback PD controller is practically independent of its proportional constant, at least for chosen values of the derivation constant.

In the present time the results reached for uncontrolled rotor system help with the debugging and testing of the testing stand. As soon as the testing stand will come into the service the results for controlled rotor system will be experimentally tested as well.

ACKNOWLEDGEMENTS

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