THEORETICAL AND EXPERIMENTAL INVESTIGATION OF EXTERNAL EXCITATION INFLUENCE ON BEHAVIOUR OF ROTOR SUPPORTED IN SLIDING JOURNAL BEARINGS


Abstract: Instability of rotors supported in sliding bearings is encountered in spite of sophisticated computational and design methods. Project dealing with possibilities of affecting rotors by external excitation of bearings was started in 2007. Test stand with rigid rotor was designed, manufactured and tested. Modification of test stand for elastic rotors was designed as a variant enabling passage through bending critical speed. Matlab model of spatial movement of the rotor in two bearings was prepared and simulation rotor run-up was carried out.

Key words: journal bearings, rotor instability, external excitation, test stand, piezoactuator control, lumped parameter model, Matlab-Simulink model, rotor run-up simulation.

1. INTRODUCTION

Instability is one of the most serious problems of high-speed rotors supported in sliding journal bearings. The project analysing possibilities of affecting the rotor through external excitation of bearing bushings was started in 2007. Test stand [1] with rigid shaft for experimental investigation of possibilities to influence rotor behaviour by external excitation was designed and manufactured. Theoretical analysis carried out indicated, that simple harmonic excitation would not be able to suppress rotor instability and that more complex course of excitation forces will be needed.

2. EXPERIMENTAL STAND

Test stand with rigid rotor supported in two sliding journal bearings was designed and manufactured in 2007 (see Fig. 1). The driving motor 3 and elastic coupling 6 enable to reach theoretically speeds well over 20.000 rpm, which is more than double the calculated stability limit of the rotor. The rotor vibrations can be measured by two pairs of relative sensors 10, located in close vicinity of bearings. Each bearing bushing can be connected to one vertically and one horizontally oriented computer controlled piezoactuator 12, by which it can be excited by arbitrary force. In 2008 the stand was assembled (see Fig. 2) and connected to hydraulic aggregate supplying oil to bearings. Driving motor was coupled to frequency converter enabling varying speed from zero to maximum of 23.000 rpm.

After repairing some minor assembly errors the rotor could be driven to 7.000 - 9.000 rpm. At this speed the motor was stopped due to exceeding maximum allowable value of electric current. As driving motor output is about five times higher than
calculated friction losses, the reason of this behaviour had to be looked for. One of the reasons can be shaft unbalance, resulting in consummation of part of motor output for shaft vibration.

Fig. 1 Test stand with rigid rotor

Fig. 2 Photograph of assembled test stand

The more serious reason of higher passive resistance is imperfect alignment of bearing and driving motor axes. Although installed elastic coupling has two joints, it is not able to compensate misalignment, which is obviously greater than prescribed value. With bearing clearance of 0.05 mm and bearing span of 200 mm the real clearance (“lift” of the shaft in bearings) is due to mutual bearing misalignment still smaller. When turning the shaft, one can feel distinct increase of resistance in some shaft positions, which substantially increase calculated friction losses. Misalignment of the motor and test shaft axes is confirmed also by very “special” journal trajectories in both bearings, which are presented in Fig. 3 (bearing 1 is nearer to the motor). Both measured signal are somewhat influenced also by electric disturbances, probably from frequency converter and oil pump.

To reach nominal maximum motor speed it would be necessary to increase bearing clearance, by which the rotor stability limit will be at the same time decreased. Essential is also dynamic balancing of the shaft and coupling. To achieve perfect decoupling of the test shaft from driving motor, another flexible coupling can be installed between original coupling and the shaft.
In order to be able to investigate experimentally also behaviour of elastic shafts, e.g. to test the passage through bending critical speeds and possibility to affect vibration amplitudes in this area, a modification of test stand was designed (see Fig. 4). Rigid shaft was replaced by elastic one, with diameter between bearings reduced to 10 mm, and bearing span was increased from 200 to 300 mm. With one disk of 0.5 kg in the middle the bending critical speed of the rotor is about 9,000 rpm.

**Fig. 4 Test stand with elastic rotor**

### 3. Muszynska Lumped Parameter Model of the Rotor System

There are many ways how to model a rotor system, but this paper prefers an approach, which is based on
- the concept developed by Muszynska [1] and supported by Bently Rotor Dynamics Research Corporation,
- the lubricant flow prediction using a FE method for Reynolds equation solution [4].

The reason for using Muszynska approach is that this concept offers an effective way to understand the rotor instability problem and to model a journal vibration active control system by manipulating the sleeve position by means of actuators [1], which are a part of the closed loop composed of proximity probes and a controller. The solution of
the Reynolds equation gives more precise rotor dynamic characteristics including rotor stability.

To verify the active journal bearing control by simulation, the simplified mathematical model is required. Till now developed model describes the displacement of the journal centerline inside a bearing housing equipped with inner bushing. It is assumed, that the bushing position inside bearing is controlled by piezoactuators as it is shown in Fig. 5. The simulation model contains equations predicting the response in the journal position to the bushing displacement. The basic model is suitable for a long flexible shaft with a flywheel, which is positioned close to the journal bearing. However, the test stand is equipped with the relatively short rigid shaft supported by two journal bearings. There is mutual influence of the control systems acting to the bushing position inside of two bearing housings. The response of the bushing displacement at the opposite journal bearing is transmitted by the gyroscopic forces of the shaft, which is rotating at the high speed. The topic of this section is focused on extending the actual model with gyroscopic effects influencing the shaft behavior during rotation with the simultaneous bushing displacements. The control system does not contain two independent two-parameter control loops (for X and Y directions at each journal bearing) but one four-parameter control system.

Both the internal forces, such as the spring, damping and tangential forces, and the external forces, such as the gravity, gyroscopic and unbalance forces, are acting on the rotor. As Muszynska has stated, these bearing forces can be modeled as a rotating spring and damper system at the angular velocity $\Omega$, where $\lambda$ is a parameter, which is slightly less than 0.5. The parameter $\lambda$ is denominated by Muszynska as the fluid averaged circumferential velocity ratio. The fluid pressure wedge is the actual source of the fluid film stiffness in a journal bearing, which maintains the rotor in equilibrium. To simplify modeling in Matlab-Simulink, the quantities like force, velocity and displacement, are position vectors of complex numbers. The letter $r$ designates the position of the journal center line in the bearing housing while the letter $u$ designates the position of the inner movable bushing as it is shown in Fig. 6. Fluid film forces acting on the rotor in coordinates rotating at the same angular frequency as the spring and damper system are given by the formula

$$F_{rot} = K(r_{rot} - u_{rot}) + D(\dot{r}_{rot} - \dot{u}_{rot}),$$

where the parameters, $K$ and $D$, specify proportionality of stiffness and damping to the relative position of the journal center displacement vector $r_{rot} - u_{rot}$ and velocity vector $\dot{r}_{rot} - \dot{u}_{rot}$, respectively. To model the rotor system, the fluid forces have to be expressed in the stationary coordinate system, in which the rotor centre-line displacement and velocity vectors are designated by $r - u$ and $\dot{r} - \dot{u}$, respectively.
Substitution into the fluid force equation results in the following formula

\[ F = K (r - u) + D (r - u) - jD\lambda\Omega (r - u), \]

where the complex term \( jD\lambda\Omega (r - u) \) has the meaning of the force acting in the perpendicular direction to the vector \( r - u \) and this force is called tangential. As the rotor angular velocity increases, this force can substantially increase and cause instability of the rotor.

The parameters \( K \) and \( D \), specifying oil film stiffness and damping, are a function of the journal centerline position vector, namely the oil film thickness. It was proved, that the closer position of the journal to the bearing wall and simultaneously the thinner the oil film, the greater is value of both these parameters. Some authors, such as Muszynska [3], assume that it is possible to approximate these functions by formulas

\[ K = K_0 \left( 1 - \frac{|r|^2}{e^2} \right)^\gamma, \quad D = D_0 \left( 1 - \frac{|r|^2}{e^2} \right)^\gamma, \quad \lambda = \lambda_0 \left( 1 - \frac{|r|^2}{e^2} \right)^\gamma \]

where \( e \) is a journal bearing clearance. The authors of this paper analyzed the other formula structure as well [5].

Due to the fact, that the shaft is considered as a rigid body, the ends of the position vectors lie on a straight line, which is identical with the shaft axis (see Fig 6). Angles \( \phi_{\text{Re}} \) and \( \phi_{\text{Im}} \) designate the inclinations of the shaft axis from the bearing housing axis, which is forming an intersection of two perpendicular planes serving for projection of the shaft axis. The plane, which is coinciding with the real axis, is horizontal while the other plane, which is coinciding with the imaginary axis, is vertical. The angle \( \phi_{\text{Re}} \) specifies the inclination of the shaft axis projection into the horizontal plane from the bearing housing axis while the angle \( \phi_{\text{Im}} \) specifies the inclination of the shaft axis projection into the vertical plane from the bearing housing axis.

There are two bearings supports of the rotating shaft. Let \( r \) be a position vector of the shaft center of gravity and let \( l_1 \) and \( l_2 \) be the distances of the center of gravity from the journal bearings. The force \( F \) has to be indexed according to the journal bearings

\[ F_1 = K (r_1 - u_1) + D (r_1 - u_1) - jD\lambda\Omega (r_1 - u_1), \]
\[ F_2 = K (r_2 - u_2) + D (r_2 - u_2) - jD\lambda\Omega (r_2 - u_2). \]

Let the angels \( \phi_{\text{Re}} \) and \( \phi_{\text{Im}} \) be combined into the complex variable \( \Phi = \phi_{\text{Re}} + j\phi_{\text{Im}} \).

The position vectors of the shaft ends in both the journal bearings are as follows

\[ r_1 = r + l_1 (\sin(\phi_{\text{Re}}) + j \sin(\phi_{\text{Im}})) = r + l_1 \Phi = X_1 + jY_1, \]
\[ r_2 = r - l_2 (\sin(\phi_{\text{Re}}) + j \sin(\phi_{\text{Im}})) = r - l_2 \Phi = X_2 + jY_2. \]
The first derivation of the variables $r_1$ and $r_2$ with respect to time results in

$$
\dot{r}_1 = \dot{r} + l_1 (\phi_{Re} + j \phi_{Im}) = \dot{r} + l_1 \dot{\Phi},
$$

$$
\dot{r}_2 = \dot{r} - l_2 (\phi_{Re} + j \phi_{Im}) = \dot{r} - l_2 \dot{\Phi},
$$

(6)

The equation of motion in stationary coordinates is obtained

$$
M \ddot{r} = M g + m r \omega^2 \exp(j(\omega t + \delta)) - F_1 - F_2,
$$

(7)

where $M$ is the total rotor mass and $g$ is acceleration of gravity. The unbalance force, which is produced by unbalance mass $m$ mounted at a radius $r_u$, acts in the radial direction and has a phase $\delta$ at time $t = 0$.

The shaft rotating at the high rotation speed can be considered as a gyroscope [6]

$$
A \ddot{\Phi} = l_2 F_2 - l_1 F_1 + j C \Omega \dot{\Phi},
$$

(8)

where $A$ is a moment of inertia of the shaft about its axis and $C$ is a moment of inertia of the same shaft about the axis, which is perpendicular to the shaft rotational axis.

4. Simulation study of the model behavior during run-up

As was stated before, the numeric solution of the journal equation of motion is obtained by using Matlab-Simulink. The block diagram of the rotor system is shown in Fig. 8.

To test the model response, the following values of the parameters were employed. The bushing position $u_2 = 0$, while the variable position $u_1$ corresponds to the rotation at the speed of 40 rad/s with the amplitude of 20 µm. The moments of inertia correspond to a short rigid hollow shaft of the outer diameter of 30 mm and the inner diameter of 20 mm, which was made of steel.

- $M = 2.38$ kg; rotor mass
- $\text{lam}_0 = 0.475$; fluid averaged circumferential velocity ratio (lambda)
- $K_0 = 4000$ N/m; oil film stiffness
- $D_0 = 2000$ Ns/m; oil film damping coefficient
- $e = 0.0001$ m; clearance in the journal bearing (100 µm)
- $\text{MMR} = 0.00001$ kg m product of the unbalance mass $m$ mounted at a radius $r_u$.
- $L_1 = 0.1$ m; distance of the rotor gravity center from the journal bearing 1
- $L_2 = 0.1$ m; distance of the rotor gravity center from the journal bearing 2
- $A = 0.0006$ kg m$^2$; a moment of inertia of the shaft about its axis
- $C = 0.008$ kg m$^2$; moment of inertia of the same shaft about the axis, which is perpendicular to the shaft axis
Fig. 8 Block diagram of the rotor system supported on two journal bearings

The rotational speed was increased from 0 to 400 rad/s during 10 s. The simulation results are shown in Fig. 9. The rotation of the bushing in the journal bearing 1 does not influence the position of the shaft in the journal bearing 2.

Fig. 9 Orbit plots

5. Conclusions

Test stand for investigation of possibilities to influence rotor behaviour through external excitation of sliding journal bearing bushings was designed and partly tested. Due to imperfect alignment of the motor and test shaft axis maximum design speed was not achieved, but the problem will be solved by means of increasing bearing clearance and perfect balancing of the shaft and coupling. Modification of test stand for elastic rotors, enabling e.g. passing through bending critical speed, was designed.

Lumped parameter model of the rotor system, according to concept of Muszynska and with gyroscopic effects included, was elaborated and simulation study of rotor behaviour during run-up was carried out.
6. ACKNOWLEDGEMENT

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7. REFERENCES


