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Self-excited and flow-induced vibrations of a rotor supported on journal bearings

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A rotor supported on hydrodynamic journal bearings represents a complex dynamic system, whose vibrations are significantly influenced by fluid films in the journal bearings. Except for resonant frequencies, dangerous self-excited vibrations of the rotor due to oil-whirl and oil-whip instabilities can occur under certain conditions (e.g. [1], [4]). Flow-induced vibrations can occur when journal bearings are poorly lubricated. These vibrations are sub-synchronous and they do not pose a danger for the rotor (e.g. [2], [5]). Self-excited and flow-induced vibrations of the rotor can be predicted using computational models or they can be detected experimentally. Close understanding of behaviour of the journal bearing before, during and after the self-excited and the flow-induced vibrations is the main motivation for the complex research of dynamics and hydrodynamics of the rotor-bearing system. Deep knowledge of relations between the dynamics of a fluid film in journal bearings and the dynamic response of rotating systems of these types of vibration can help to improve designs of many modern rotating machines.

During last few years many aspects regarding the influence of the fluid film in journal bearings on the dynamics of rotor systems were introduced (e.g. [4], [5]). Recent development in numerical methods for nonlinear models and numerical continuation methods allowed the oil-induced instabilities and resulting bifurcations to be studied even deeper. E.g. De Castro et al. [1] implemented nonlinear hydrodynamic forces and predicted oil whirl and oil whip for a real vertical rotor train and for a horizontal test rig. Sub-synchronous fluid-induced vibrations were observed e.g. by DeCamillo in thrust bearings [2]. DeCamillo noted that such vibrations usually occur in poorly lubricated bearings but he was unable to identify all conditions that can lead to the reported sub-synchronous vibrations. The problem of the sub-synchronous fluid-induced vibrations is not commonly studied in available literature and, therefore, the investigation of this problem is a challenge.

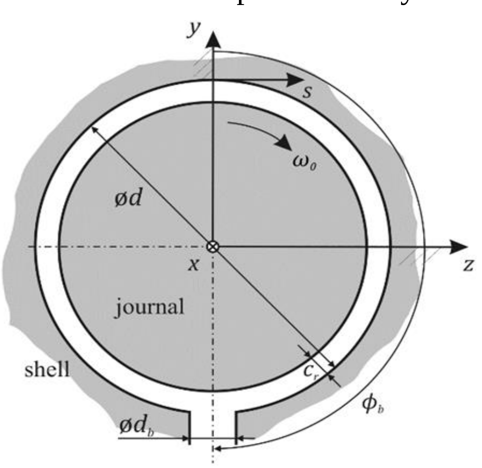

Fig. 1. Geometry of the journal bearing

There are several common methods for the modelling and dynamic analysis of rotating systems. The approaches are based on the finite element method (for the one-dimensional Euler-Bernoulli and/or Timoshenko continua) or on multibody dynamics. Journal bearings are represented by nonlinear forces acting at points corresponding to the bearing support. The oil film dynamic behaviour is described by the Reynolds equation. It has to be solved in each time integration step and the resultant pressure distribution is transformed into dynamic forces, which are subsequently included into equations of motion.

Equations of motion for the rotor supported on journal bearings that are used in this work are derived based on a multi-body formalism. Motions of the rotor are decomposed to global (gross) motions and elastic motions (vibrations) in this approach. In order to simulate both types of the motion, the system of differential algebraic equations is employed, [3], (time t is omitted for a better clarity)

$$\mathbf{M}\,\ddot{\mathbf{q}} + \mathbf{f}^{rb}(\mathbf{z},\dot{\mathbf{z}}) = \mathbf{f}^{gyr}(\mathbf{z}) + \mathbf{f}^{j}(\mathbf{z},\boldsymbol{w}) + \mathbf{f}^{e}(\mathbf{z}) - \mathbf{D}\,\dot{\mathbf{q}} - \mathbf{K}\,\mathbf{q}, \tag{1}$$

$$\mathbf{S}(\boldsymbol{\theta}_B)\,\dot{\boldsymbol{\theta}}_B = \boldsymbol{\Omega}_B, \tag{2}$$

$$\boldsymbol{\theta}_B^{\top}\,\boldsymbol{\theta}_B = 1, \tag{3}$$

$$\mathbf{r}(\mathbf{q}) = \mathbf{0}. \tag{4}$$

The position, the orientation and the deformation of the rotor are included in state vector $\mathbf{z} = [\mathbf{x}_B^{\top}, \boldsymbol{\theta}_B^{\top}, \dot{\mathbf{x}}_B^{\top}, \boldsymbol{\Omega}_B^{\top}, \mathbf{q}^{\top}, \dot{\mathbf{q}}^{\top}]^{\top}$. Vector $\mathbf{x}_B$ and quaternion $\boldsymbol{\theta}_B$ characterize the position and the orientation of the rotor, respectively, and vector $\mathbf{q}$ contains elastic coordinates relative to a coordinate system whose position is defined by vector $\mathbf{x}_B$ and whose orientation is given by four Euler parameters in quaternion $\boldsymbol{\theta}_B$. Vector $\mathbf{w}$ is composed of state vectors of all bodies coupled to the rotor. $\mathbf{M}, \mathbf{D}, \mathbf{K}$ are constant matrices, which characterize mass, damping and stiffness of the rotor, respectively. $\mathbf{f}^{rb}, \mathbf{f}^{gyr}$ are vectors of forces, which result from the rigid body accelerations and gyroscopic effects, respectively. Vector $\mathbf{f}^{j}$ accommodates forces in couplings and $\mathbf{f}^{e}$ contains prescribed external forces and moments.

Equation (2) describes the relation between angular velocity $\boldsymbol{\Omega}_B$ of the global motion and quaternion $\boldsymbol{\theta}_B$, (3) is the normalization condition and (4) is introduced in order to obtain a unique separation of the global and the elastic coordinates. These additional equations are described in detail e.g. by Offner et al. [3].

The forces acting on the rotor in its journal bearings are obtained using the solution of the Reynolds equation. These so-called hydrodynamic forces can be evaluated by integrating a pressure in an oil film over the surface of the bearing. Here it is assumed, that the oil is an incompressible Newtonian fluid, the film is thin and the flow in the film is laminar. Furthermore, cavitation may occur and mass conservation in cavitated areas is considered. Pressure $p = p(s,x,t)$ is then governed by the Reynolds equation in the form (see e.g. [5]):

$$\frac{\partial}{\partial s}\left(\frac{\theta\,h^3}{12\,\mu}\,\frac{\partial p}{\partial s}\right) + \frac{\partial}{\partial x}\left(\frac{\theta\,h^3}{12\,\mu}\,\frac{\partial p}{\partial x}\right) = \frac{u_j+u_s}{2}\,\frac{\partial(\theta\,h)}{\partial s} + \frac{\partial(\theta\,h)}{\partial t}, \tag{5}$$

where s, x are the circumferential and axial coordinates, respectively. These coordinates are depicted in Fig. 1. Function $h = h(s,x,t)$ determines the gap between a journal and a shell, $\theta = \theta(s,x,t)$ is the percentage of the bearing gap that is filled with oil, $\mu = \mu(s,x,t)$ is the dynamic viscosity of the oil and u_j, u_s are the surface velocities of the journal and the shell, respectively. Note that there are two unknown variables in (5): p and θ. For $t = 0$, ratio θ is prescribed by initial conditions (usually $\theta = 1$) and (5) is solved for p. If p drops below the value of saturation pressure p_c at any node then the Gümbel condition (i.e. $p = p_c$) is applied and (5) is solved for θ.

Table 1. Nominal parameters of journal bearings (valid for both the rotors [4], [5])

Parameter	Symbol	Value	Parameter	Symbol	Value
bearing diameter	d	38.0 mm	ambient pressure	p_a	1.00 bar
bearing length	l	20.0 mm	saturation pressure	p_c	0.98 bar
radial clearance	c_r	40.5 µm	supply pressure	p_s	1.25 bar
lubricant viscosity	μ	28.3 mPa·s	supply bore diameter	d_b	5.00 mm

Different rotor-bearing systems were considered in order to investigate self-excited and flow-induced vibrations of the rotor. Geometry of the journal bearings was the same for the investigation of both vibration types, see Fig. 1. Parameters of the bearings are summarized in Table 1. The RENOLIN VG 46 lubricant is supplied to the bearings through a circular supply bore, which is located in the lower half of the bearing shell.

The proposed configuration of the test rig for the investigation of oil-whirl and oil-whip is shown in Fig. 2. The shaft in this configuration is rather slender and a massive disc is attached to the shaft.

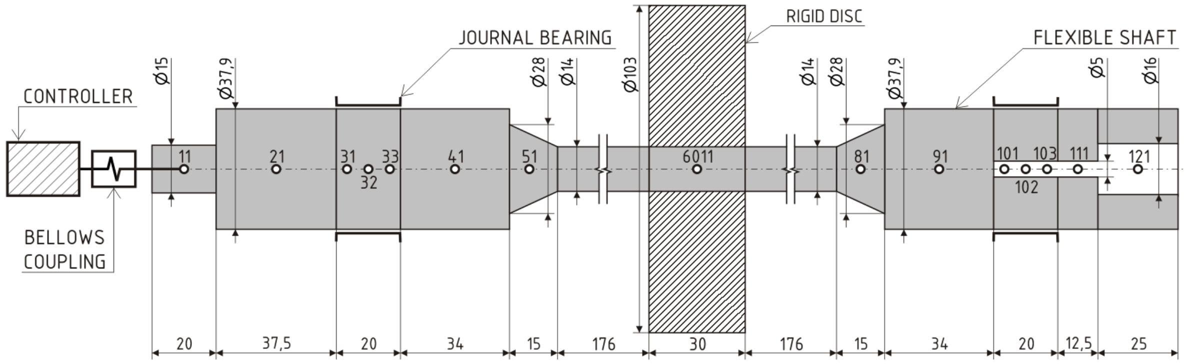

Fig. 2. Geometry and discretization of the rotor for the investigation of oil-whirl and oil-whip instabilities, [4]

The simulations suggest (Fig. 3b) that the oil-whip develops at frequency of rotation f_r in the range of 100–106 Hz (6,000–6,350 RPM) with a dominant response at 50–52 Hz, which corresponds with the first bending mode. Further simulations suggest that the threshold speed for the oil-whip is only little sensitive to the radial clearance or the lubricant dynamic viscosity. Furthermore, there is a short speed interval in which the oil-whirl takes place (92–98 Hz). Although the proposed test rig geometry is suitable for the investigation of both oil-whirl and oil-whip, a prolonged operation under oil-whip conditions is impossible because of a high level of vibrations of the disc, which exceed 1 mm peak-to-peak (Fig. 3a).

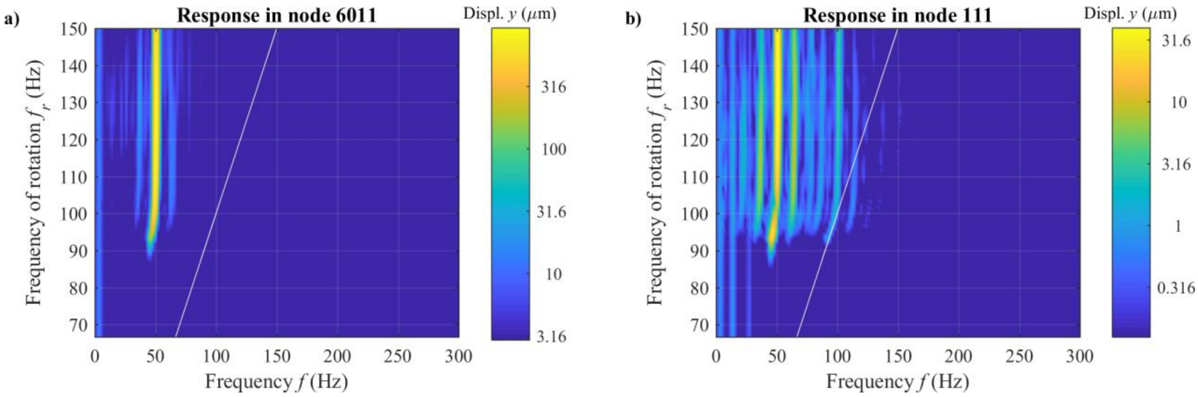

Fig. 3. Simulated steady state response of the perfectly balanced rotor from Fig. 2 (taken from [4]). Response of the rigid disc (a) and of the journal in the bearing at the non-drive end (b).

An arrangement and dimensions of the analysed system for the investigation of flow-induced vibrations are depicted in Fig. 4. The system consists of the rotor supported on two journal bearings and a controller, which controls the speed of the rotor.

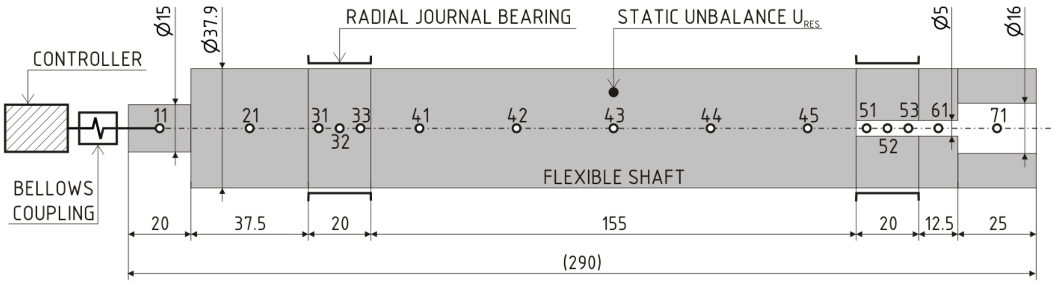

Fig. 4. Geometry and discretization of the rotor for the investigation of the sub-synchronous flow-induced vibrations, [5]

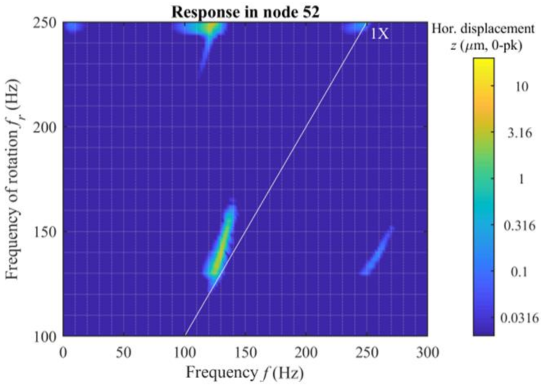

Fig. 5. Steady-state response of the perfectly balanced rotor for the investigation of the sub-synchronous flow-induced vibrations, [5]

The steady-state response of the perfectly balanced rotor was simulated for frequency of rotation f_r in the range of 100–250 Hz. The response was analysed in time interval 1–2 seconds and is depicted in Fig. 5. Flow-induced vibrations manifest themselves as a sub-synchronous component, which appears at $f_r \approx 130$ Hz and disappears at $f_r \approx 160$ Hz. Moreover, self-excited vibrations develop at $f_r \approx 245$ Hz at the frequency of $0.48 \times f_r$.

The reported sub-synchronous flow-induced vibrations can occur only if the rotor is well balanced (balance quality grade G1 or lower in accordance with ISO 21940-1), and if it is radially supported on poorly lubricated journal bearings, whose supply bores are located in the lower half of the bearings. These vibrations are stable and cover roughly 10 % of the bearing clearance and, therefore, they are not dangerous. However, they might be undesirable in precise machinery.

Acknowledgements

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