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Influence of periodic excitation on self-sustained vibrations of one disk rotors in arbitrary length journals bearings

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ABSTRACT

Interaction of forced and self-sustained vibrations of one disk rotor is described by nonlinear finite-degree-of-freedom dynamical system. The shaft of the rotor is supported by two journal bearings. The combination of the shooting technique and the continuation algorithm is used to study the rotor periodic vibrations. The Floquet multipliers are calculated to analyze periodic vibrations stability. The results of periodic motions analysis are shown on the frequency response. The quasi-periodic motions are investigated. These nonlinear vibrations coexist with the periodic forced vibrations.

1. Introduction

Journal bearings are frequently used in rotating machineries. The fluid films of the journal bearings interact with the rotors, which can result in violent self-sustained system vibrations. The rotors are always unbalanced, which leads to the system forced oscillations. Therefore, the interactions of the forced and self-sustained vibrations can be observed in the rotor systems. The rotor vibrations cause their fatigue breaking. Therefore, many efforts were done to analyze the vibrations of the rotor systems. Here, only several papers devoted to this subject are considered. Modern methods of nonlinear mechanics including nonlinear modes are used to analyze rotor dynamics [1,2]. The finite element approach for the calculations of the fluid film forces in the form of the Taylor series with respect to the generalized coordinates and velocities is suggested in [3]. Muszynska [4,5] considers the rotor with one journal bearing. She obtains the rotor vibrations model, which is based on the experimental data. Ping et al. [6] use the finite element method to discretize the elastic rotor vibrations. The analytical model of the nonlinear forces of the journal bearing is derived to study the nonlinear vibrations of the rotor system. The variational approach is used to determine the journal bearing fluid film pressure by Zheng et al. [7]. The Rayleigh method is applied to minimize the obtained functional. Ding and Leung [8] analyze transients in the rotors accounting the journal bearings hydro dynamical forces. The stiff unbalanced rotor supported by two journal bearing is treated by Liu et al. [9]. They assumed that the journal bearing has infinite length. The mathematical model of the rotor plane motions accounting both hydro dynamical phenomena in fluid films and nonlinear forces of elastic supports is analyzed by Chen and Yan [10]. The dynamics of the stiff rotor, which is described by five generalized coordinates, is analyzed by Adiletta et al. [11,12]. Periodic and quasi-periodic motions of the rotor-bearing-seal coupled system are analyzed in [13]. The asymptotic approach based on the multiple scales expansions is applied to analyze the dynamics of one disk rotor on the nonlinear flexible base in [14]. The applications of nonlinear normal modes for rotor nonlinear dynamics analysis are reviewed in [15]. The rotor dynamics is discussed in the books [16–19].

In the majority of the above-mentioned papers the balanced rotor self-sustained vibrations, which originate due to the interaction of fluid films with rotor journals, are analyzed. But, it is well-known, that the rotor is always unbalanced. Therefore, the interaction of forced and self-sustained vibrations can be observed. In this paper this interaction is investigated in the rotors with arbitrary length journals bearings. The quasi-periodic vibrations, which take place due to this interaction, are observed.

The rotor with arbitrary length journals bearings is analyzed in this paper. The combination of the shooting technique and the

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continuation method is applied to calculate the disk periodic vibrations. The Floquet multipliers are calculated to analyze periodic vibrations stability. The quasi-periodic motions are observed. The coexisting of the periodic motions and the quasi-periodic vibrations is detected. The Poincare sections of quasi-periodic motions are analyzed.

2. The problem formulation

The nonlinear dynamics of the horizontal rotor in two journal bearings is considered. The sketch of this rotor is shown in Fig. 1a. The rotor consists of the weightless shaft and the disk, which is attached asymmetrically with respect to the shaft ends. The shaft is supported by two journal bearings. The disk is attached to the shaft at the distant D_ε from the disk center of mass. The generalized coordinates (x_1, y_1) and (x_2, y_2) describe the motions of the journals $\boldsymbol{A}$ and $\boldsymbol{B}$, respectively. The vibrations of the disk in plane (xoy) are described by two generalized coordinates (x, y) (Fig. 1). The angles θ_1 and θ_2 are described the disk rotation about x and y axes, respectively. These angles are shown in Fig. 1b. The forces of the fluid films are applied to the journals A and B (Fig. 1a). The projections of these forces on the axes x, y are denoted by $F_x(x_i, y_i)$, $F_y(x_i, y_i)$, $i = 1, 2$, respectively. The sketch of the cylindrical journal bearings is shown in Fig. 1c. The shaft is rotated with the angular velocity Ω about z axis.

Due to the action of the gravity force, the rotor takes up the equilibrium position, which is described by the values of the generalized coordinates $\bar{x}$, $\bar{y}$, $\bar{\theta}_1$, $\bar{\theta}_2$, $\bar{x}_1$, $\bar{y}_1$,$\bar{x}_2$, $\bar{y}_2$. The detailed derivation of the equations of the system motions are published in [2,14]. These equations can be presented in the following form:

$$\begin{aligned}
&m\ddot{x} = \tilde{F}_x(x_1, y_1) + \tilde{F}_x(x_2, y_2) + m\Omega^2 D_\varepsilon \cos \Omega t;\\
&I_e\ddot{\theta}_2 - I_p\Omega\dot{\theta}_1 + l_1\tilde{F}_x(x_1, y_1) - l_2\tilde{F}_x(x_2, y_2) = 0;\\
&m\ddot{y} = \tilde{F}_y(x_1, y_1) + \tilde{F}_y(x_2, y_2) + m\Omega^2 D_\varepsilon \sin \Omega t;\\
&I_e\ddot{\theta}_1 + I_p\Omega\dot{\theta}_2 - l_1\tilde{F}_x(x_1, y_1) + l_2\tilde{F}_x(x_2, y_2) = 0,
\end{aligned} \tag{1}$$

where I_e, I_p are diametrical and polar moments of the disk inertia, respectively; $I_p\Omega\dot{\theta}_1, I_p\Omega\dot{\theta}_2$ are gyroscopic moments of the disk; m is mass of the disk; the parameters l_1, l_2 are shown in Fig. 1. The fluid films forces take the forms:

$$\tilde{F}_x(x_i, y_i) = F_x(x_i + \bar{x}_i, y_i + \bar{y}_i) - F_x(\bar{x}_i, \bar{y}_i);$$
$$\tilde{F}_y(x_i, y_i) = F_y(x_i + \bar{x}_i, y_i + \bar{y}_i) - F_y(\bar{x}_i, \bar{y}_i);\ i = 1, 2.$$

The equations of two journals equilibrium are taken into account besides the equations of the disk motions [14,2]. These equations connect the generalized coordinates of the disk motions $\mathbf{q} = (x, \theta_1, y, \theta_2)^T$ to the generalized coordinates of the journals $\mathbf{q_1} = (x_1, y_1, x_2, y_2)^T$. These equations take the following matrix form:

$$\left[\tilde{\mathbf{R}}\right]\ \mathbf{q} = \left[\tilde{\mathbf{D}}\right]\ \mathbf{q_1}. \tag{2}$$

The matrix Eq. (2) is substituted into Eq. (1). As a result, the equations of the motions with respect to the disk generalized coordinates are derived. These equations have the following vector form:

$$[\mathbf{M}]\ddot{\mathbf{q}}(t) + [\mathbf{G}]\dot{\mathbf{q}}(t) = \mathbf{W}(\mathbf{q}, \dot{\mathbf{q}}) + \mathbf{D_F}(t), \tag{3}$$

where

$$\mathbf{D_F} = m\Omega^2 D_\varepsilon(\cos \Omega t,\ \ 0,\ \ \sin \Omega t,\ \ 0).$$

The matrixes $[\mathbf{M}]$, $[\mathbf{G}]$ and the nonlinear vector-function $\mathbf{W}(\mathbf{q}, \dot{\mathbf{q}})$ are discussed in the papers [2,3].

Now the forces of the fluid film are considered. It is assumed, that the fluid film occupies the region $\theta \in [0\ ;\ \pi]$ [18]. The projections of the fluid film forces on the axes x and y take the following form [3]:

$$\begin{bmatrix} F_x \\ F_y \end{bmatrix} = -\int_0^{L_B}\int_0^{\pi} \begin{bmatrix} \cos(\theta + \phi) \\ \sin(\theta + \phi) \end{bmatrix} p(z, \theta)\ R\ d\theta\ dz, \tag{4}$$

where z, θ is longitudinal and angular coordinates of the bearing; $z \in [0, L_b]$; L_b is the bearing length; R is the journal radius; ϕ is angle defined by the line of centers. The pressure of the fluid film $p(z, \theta)$ is described by Reynolds' equation [18].

The finite element approach, which is suggested in the paper [3], is used to obtain the fluid film forces. The projections of these forces F_x and F_y are obtained numerically in the form of the power series with respect to the general coordinates and the velocities

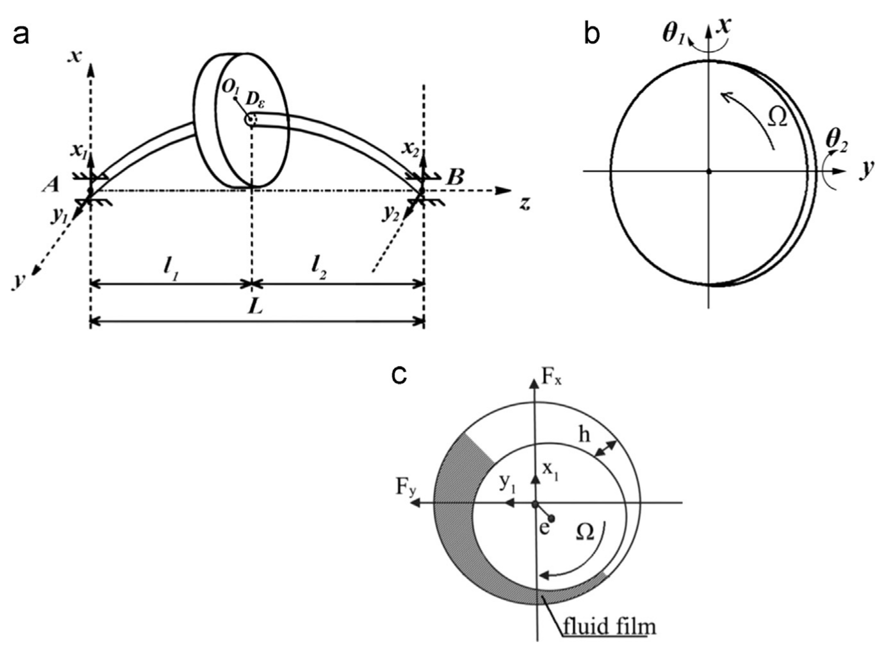

Fig. 1. Outline of one disk rotor.