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The Effect of Anisotropic Bearings on Dynamics and Stability of the Ball-Spring Auto-balancer

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ABSTRACT: In recent years, there has been much interest in the use of automatic dynamic ball balancers to balance the rotors with varying imbalances. An automatic dynamic ball balancer is a device that can automatically compensate the variable imbalances of a rotor in certain working conditions without having to stop the rotating equipment. Ball-spring auto-balancer is a new type of automatic ball balancers which has two main advantages over the traditional ones, i.e. it causes the rotor vibration amplitude at the transient state to be small and it has a wider balanced stable region. Bearings are one of the most important mechanical components due to their considerable influence on the dynamic behavior and stability of the rotary systems. In previous studies, the dynamic behavior and stability of the rotor with isotropic bearings equipped with a ball- spring auto-balancer has been analyzed. However, in practice, the bearings have generally anisotropic behavior due to the manufacturing process. In this study, the dynamics and stability of a rotor with anisotropic bearings equipped with a ball-spring autobalancer are investigated via the multiple scales method for the first time. The results show that the anisotropic bearings do not impair the main advantages of the ball-spring auto-balancers, and as the anisotropic parameter increases the balanced stable region decreases.

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1. INTRODUCTION

The imbalance is one of the main causes of undesired vibration in rotating systems. This phenomenon leads to adverse effects in rotating equipment such as reducing the bearing life, increased maintenance costs, and etc. For a rotor with a fixed amount of imbalance, the conventional balancing procedure is applicable. However, for a rotor with varying imbalance during the operating conditions, the conventional balancing method is not sufficient. This observation persuades the implement of automatic ball balancers which can passively reduce the vibration amplitude of the imbalance rotor. The capability of ball balancers which can automatically compensate the variable imbalances makes this device to be used currently in industrial applications such as CD- ROMs, DVD drives [1] and etc. The first investigation on Automatic Ball Balancer (ABB) was carried out by Thearle [2]. Jinnouchi et al. [3] showed that, although the ABB can eliminate the undesired vibrations of the imbalance rotor above the first critical speed, it leads to large vibration at low speeds. Jung and DeSmidt [4] investigated the nonsynchronous vibration behavior for a rotor with ABB by implementing the harmonic solution.

Reviewing the previous studies show that, although the traditional ABB has numerous advantages, it has two main deficiencies: First, the ABB amplifies the vibration amplitude of the rotor in speeds below the first critical speed. Second, it limits the balanced stable region of the rotor [5]. In this regard, recently the modified ABB model entitled ball- spring autobalancer has been proposed by Rezaee and Fathi [6], for solving the mentioned disadvantages. In previous studies, the dynamics and stability of rotor mounted on isotropic bearings equipped with a ball-spring Auto-Balancer (AB) has been investigated via the numeral method. However, the bearings have generally anisotropic behavior due to the manufacturing process. Therefore, in this study, the dynamics and stability of a rotor with anisotropic bearings equipped with a ball-spring autobalancer are investigated by implementing the multiple scales method for the first time. For this purpose, first the mathematical model of a rotor with anisotropic bearings equipped with a ball-spring AB is constructed and the nonlinear governing equations of motion are derived. Then due to the advantage of the analytical methods over the numerical ones, the vibration response of the system is acquired via the multiple scale method. Finally, the system stability is analyzed and the balanced stable regions are obtained.

2. MATHEMATICAL SIMULATION

Fig. 1 illustrates the model of a rotor supported by two anisotropic bearings and equipped with a ball-spring AB. The position of each ball is defined by a radial distance, δ_i , and an angle, ϕ_i , $i = 1, 2, 3, \ldots, n$, where n is the number of the balls.

Here, to extract the equation of motion, the kinetic energy, potential energy, and Rayleigh's dissipation function are derived. The kinetic energy of the rotor with a ball- spring

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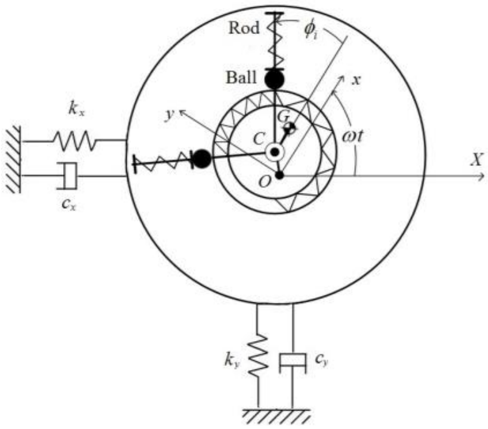

Fig. 1. Schematic of a rotor with anisotropic bearings equipped with an automatic ball-spring balancer.

AB can be expressed as:

$$
\begin{aligned}
T &= \frac{1}{2}J\omega^2 + \frac{1}{2}M\left[\dot{x}^2 + \dot{y}^2 + 2(x\dot{y} - \dot{x}y)\omega + (x^2+y^2)\omega^2\right] \\
&+ \frac{1}{2}m_d\left(2x\varepsilon\omega^2 + \varepsilon^2\omega^2 + 2\varepsilon\dot{y}\omega\right) + \frac{1}{2}m_b\sum_{i=1}^{n}\Big[2\delta_i\left(\dot{\phi}_i+\omega\right) \\
&\times\left(\dot{y}+\omega x\right)\cos\phi_i + \dot{\delta}_i^2 - 2\delta_i\left(\dot{\phi}_i+\omega\right)\left(\dot{x}-\omega y\right)\sin\phi_i \\
&+ \delta_i^2\left(\dot{\phi}_i+\omega\right)^2 + 2\dot{\delta}_i\left(\dot{x}-\omega y\right)\cos\phi_i + 2\dot{\delta}_i\left(\dot{y}+\omega x\right)\sin\phi_i\Big]
\end{aligned}
\tag{1}
$$

where $M = m_d + n\times m_b$ and m_d is the mass of the rotor and m_b is the mass of a ball and ω is the angular velocity of the rotor. J is the mass moment of inertia of the disk with respect to G. Neglecting gravity, the potential energy can be obtained as below:

$$
\begin{aligned}
V &= \frac{1}{2}k_x x^2 + \frac{1}{2}k_y y^2 + \frac{1}{2}k_r\sum_{i=1}^{n}\left(\delta_i - a\right)^2 \\
&+ \frac{1}{2}k_p d^2\left[\left(\phi_2 - \phi_1 - \frac{2\pi}{n}\right)^2 + \cdots + \left(\phi_n - \phi_{n-1} - \frac{2\pi}{n}\right)^2\right]
\end{aligned}
\tag{2}
$$

where k_x and k_y is the equivalent stiffness of the bearings in x and y directions respectively, k_p and k_r are the stiffness of the peripheral and radial springs of the balancer, respectively. The Rayleigh's dissipation function can be expressed as:

$$
\begin{aligned}
F &= \frac{1}{2}c_x\left(\dot{x}^2 - 2\dot{x}y\omega + \omega^2 y^2\right) + \frac{1}{2}c_y\left(\dot{y}^2 + 2x\dot{y}\omega + \omega^2 x^2\right) \\
&+ \frac{1}{2}c_r\sum_{i=1}^{n}\left(\dot{\delta}_i^2 + \delta_i^2\dot{\phi}_i^2\right)
\end{aligned}
\tag{3}
$$

where c_x , c_y is the equivalent damping constant of the bearings in x and y directions respectively, and c_r is the damping coefficient of the balls. The nonlinear equations of motion are derived by putting Eqs. (1) to (3) into the Lagrange equations. Also to generalize the results, the following nondimensional parameters are defined:

$$
\begin{aligned}
&\bar{x} = \frac{x}{\gamma},\ \bar{y} = \frac{y}{\gamma},\ \bar{\delta}_i = \frac{\delta_i}{\gamma},\ \tau = \omega_n t,\ r = \sqrt{\bar{x}^2 + \bar{y}^2},\ \gamma = \frac{k_r a}{k_r - m_b\omega^2} \\
&\zeta_x = \frac{c_x}{2M\omega_n},\ \zeta_y = \frac{c_y}{2M\omega_n},\ \bar{m} = \frac{m_b}{M},\ e = \frac{\varepsilon}{\gamma},\ \bar{\omega} = \frac{\omega}{\omega_n},\ \beta = \frac{c_r}{2m_b\omega_n} \\
&\omega_b = \sqrt{\frac{k_r}{m_b}},\ \omega_n = \sqrt{\frac{k_x}{m_d}},\ f = \frac{\omega_b}{\omega_n},\ \eta = \frac{k_p d^2}{m_b\gamma^2\omega_n^2},\ \sigma = \frac{k_y}{k_x}
\end{aligned}
\tag{4}
$$

where ω_n and ω_b are the natural frequencies of the rotor and the ball- radial spring assembly, respectively. σ is the ratio of the equivalent stiffness of the bearings in y direction to x direction. One can obtain the dimensionless form of the equations of motion by implementing the nondimensional parameters. In order to acquire the vibration response of the system, equations of motion must be solved. For doing this task by the perturbation method, the multiple scale method is used.

3. RESULTS AND DISCUSSIONS

The dynamic response of the system for three different values of anisotropic parameters is presented in Fig. 2. As it can be seen from this figure, the ball- spring AB can suppress the undesired vibration of imbalance rotor with anisotropic bearings. Moreover, the results presented in Fig. 2 show that by increasing the stiffness of the bearings in y direction, the vibration amplitude of the system decreases.

In this part, the effect of bearings anisotropic parameter on the balanced stable region of the system has been investigated. In Fig. 3, the balanced stable region of the rotor

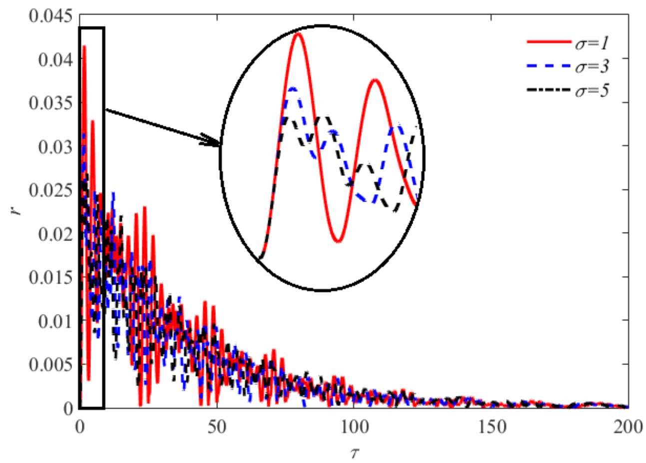

Fig. 2. Vibration response of the rotor with ball-spring AB for three different values of anisotropic parameters for $\bar{m} = 0.02$**,** $\zeta_x = 0.01$**,** $\zeta_y = 0.05$**,** $e = 0.01$**,** $\beta = 0.1$**,** $\bar{\omega} = 3$**,** $\eta = 0.004$ **and,** $f = 4$**.**

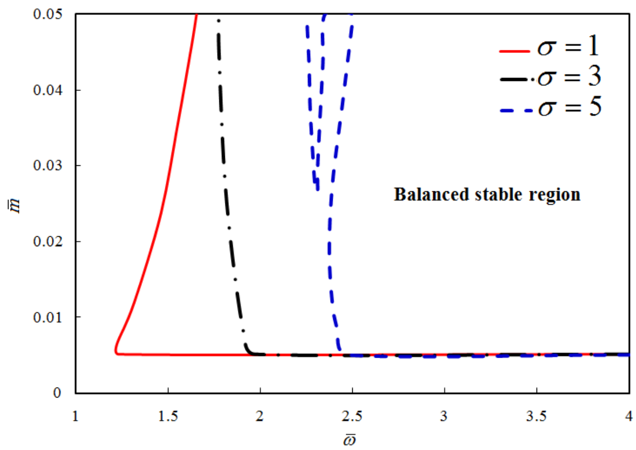

Fig. 3. Balanced stable region of the rotor with ball-spring AB for three different values of the anisotropic parameter for $\zeta_x = 0.01$**,** $\zeta_y = 0.05$**,** $e = 0.01$**,** $\beta = 0.1$**,** $\eta = 0.004$ **and,** $f = 4$**.**

supporting with two anisotropic bearings equipped with a ball-spring AB for three different values of the anisotropic parameter, σ, are depicted. As shown in this figure, by increasing the anisotropic parameter, the system balanced stable region reduces. This fact reveals the importance of considering anisotropic behavior of the bearings in the accurate stability analysis of the rotor equipped with a ball-spring AB.

4. CONCLUSIONS

In this investigation, dynamics and stability of a rotor with anisotropic bearings equipped with ball-spring AB have been studied for the first time. By implementing the Lagrange equation, the nonlinear equations of motion are derived. Vibration response and balanced stable region of the system are obtained via the multiple scales method. The results show that the bearings anisotropy affects the stability of the system, and by increasing the anisotropic parameter, the balanced stable region of the system decreases.

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