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Identification of dynamic parameters of active magnetic bearings in a flexible rotor system considering residual unbalances☆

Yuanping Xu[a,b], Jin Zhou[a,*], Zongli Lin[c], Chaowu Jin[a]

[a] College of Mechanical and Electrical Engineering, Nanjing University of Aeronautics and Astronautics, Nanjing 210016, China
[b] Laboratory of Robotic Systems, École Polytechnique Fédérale de Lausanne (EPFL), Lausanne 1015, Switzerland
[c] Rotating Machinery and Control Laboratory (ROMAC), University of Virginia, Charlottesville, VA 22904-4743, USA

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ABSTRACT

Active magnetic bearings (AMBs) support rotors using electromagnetic forces rather than mechanical forces. It is necessary to identify the AMB force coefficients (equivalent stiffness and damping) since they play key roles in the rotordynamic analysis. The identification is usually performed by analyzing the unbalance response. However, the presence of unknown residual unbalances reduces the identification accuracy and a rigid rotor model is only valid when the rotating speed is far below the first bending critical speed. Therefore, this paper proposes an identification algorithm to estimate the stiffness and damping parameters for a flexible rotor AMB system in the presence of unknown residual unbalances by using two independent unbalance response data sets. The proposed algorithm is first evaluated by numerical calculation for accuracy and then applied experimentally in the identification of a flexible rotor AMB system operating at speeds ranging from 3,000 rpm (50 Hz) to 30,000 rpm (500 Hz). The identification results are verified in the end.

1. Introduction

AMBs could provide non-contact electromagnetic forces rather than mechanical bearings. The contactless feature between rotor and bearings permits no mechanical wear, elimination of lubrication, long life expectation, low costs and high attainable rotating speeds [1–2]. Consequently, AMBs have been increasingly used in a wide range of rotating machinery applications [3–4]. When controlled by a PID controller, the rotor-AMB system has an additional attractive feature. The electromagnetic force parameters, equivalent stiffness and damping, could be tuned easily [5] since they are closely related to the control coefficients.

For a rotor-AMBs system, the identification usually could be classified into the following two aspects: (1) identify the system models for the controller design; (2) identify the closed-loop equivalent stiffness and damping coefficient for the rotordynamics analysis. For the first aspect, since AMB is open loop unstable and the high performance multiple input and multiple output (MIMO) control algorithms rely on system models, it is necessary to identify the system models before the controller design [6–12]. For the equivalent stiffness and damping identification, these coefficients are the foundation for the rotordynamics analysis since the controller effects are considered in these coefficients. [5,13–17]. In this paper, we focus on equivalent stiffness and damping identification.

For the equivalent stiffness and damping identification, only a few works have been reported and most of these works have been done for a rigid rotor model at zero rotating speed. The investigation of Humphris et al. [13] shows that the varying amounts of position and velocity feedback affect the stiffness and damping characteristics of the bearing, respectively. Williams et al. [14] acquired the theoretical stiffness and damping properties of the AMB from the controller transfer function. However, the time delay in the digital control system was not considered. Under the shaker excitation, Lim et al. [15] identified the parameters of hybrid magnetic bearings for blood pump applications with a PID controller. Tsai et al. [16] applied the wavelet transform algorithm to identify the stiffness and damping coefficients. Zhou et al. [17] experimentally obtained these coefficients on the basis of a rigid rotor model and minimized the errors using the error response surface method.

Our previous study [18] proposed an identification method for a rotating flexible rotor AMBs system using unbalance excitation. Although unbalance excitation is simple and permits identification under the rotating condition, the disadvantage is that the existence of residual unbalance on the rotor could incur errors in the unbalance excitation

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* Corresponding author.
E-mail addresses: ypxu@nuaa.edu.cn (Y. Xu), zhj@nuaa.edu.cn (J. Zhou), zl5y@virginia.edu (Z. Lin), jinchaowu@nuaa.edu.cn (C. Jin).

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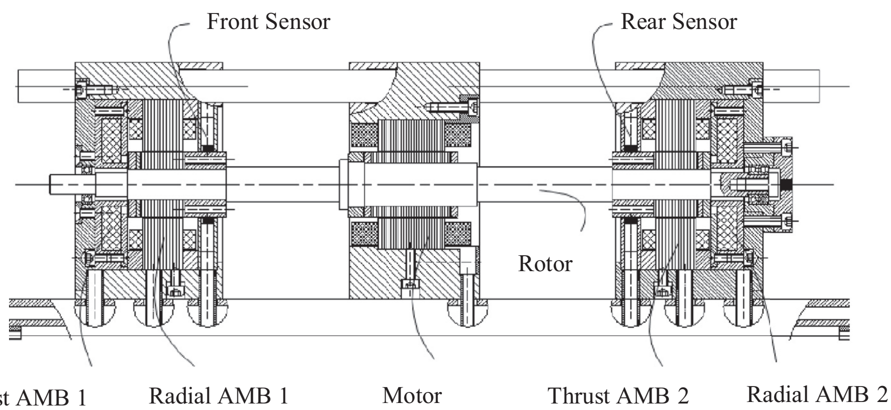

Fig. 1. The schematic view of the rotor AMBs test rig.

force calculation. Because the residual unbalance distribution is unknown and may change during operation for a flexible rotor. In this paper, we extended this identification algorithm considering the adverse effects of the residual unbalance in a flexible AMB rotor system. The extended algorithm is not only evaluated very well in simulation, but also employed to perform the experimental identification ranging from 3000 rpm (50 Hz) to 30,000 rpm (500 Hz), which is beyond first bending critical speed. The experimental identification results are also verified in the end.

The remainder of this article is organized as follows. In Section 2, the experimental rotor bearing system adopted in this paper is described. The following section Section 3 describes the proposed identification algorithm. The evaluation of the algorithm is presented in Section 4 and Section 5 describes experimental results and the verification. Conclusions are drawn in Section 6.

2. Test rig description and flexible rotor modeling

2.1. AMBs system description

Fig. 1 shows the experimental AMBs rotor system employed in this paper. The rotor is supported by two radial and two thrust AMBs and is designed to have a maximum speed of 60,000 rpm. Two rolling element ball bearings are installed as backup bearings to protect the AMBs when the rotor is out of control. The rotor is 0.468 m long and weighs around 2.4 kg. The rotor displacement vibration is detected by eddy current displacement sensor (sensitivity: 20,000 V/m; cut-off frequency: 5 kHz) and a 1.5 kW AC asynchronous induction motor is used for spinning the rotor. Table 1 summarizes the physical properties of the two radial AMBs.

2.2. Flexible rotor modeling

The finite element method (FEM) is employed for the rotordynamics analysis and we compiled this finite element (FE) rotordynamics analysis software based on Nelson–Timoshenko beam theory [19] using Matlab software. The rotor is split into fifty three elements and the equation of motion (EOM) for the rotor in this paper is written as,

$$\mathbf{M}_R\ddot{q} + (\mathbf{C}_R + \Omega\mathbf{G}_R)\dot{q} + \mathbf{K}_R q = f(t), \tag{1}$$

Table 1
Radial AMBs specifications.

Parameter	Value
Coil number of signal pole	75
Magnetic pole area	2×10^{-4} m^2
Bias current	1.5 A
Air gap	0.3 mm

where $\mathbf{M}_R$, $\mathbf{C}_R$, $\mathbf{K}_R$ are square symmetric mass, damping and stiffness matrices, respectively; $\mathbf{G}_R$ is the skew symmetric gyroscopic matrix. Ω represents the rotation speed; q and $f(t)$ are the displacement and the external force vectors, which are written as

$$\begin{aligned} q &= [q_1 \ \cdots \ q_{B1} \ \cdots \ q_{B2} \ \cdots \ q_N]^{\mathrm{T}}; \\ q_i &= [x_i \ \ y_i \ \ \beta_{xi} \ \ \beta_{yi}], \\ f &= [f_1 \ \cdots \ f_N]^{\mathrm{T}} \end{aligned} \tag{2}$$

where q_{B1} and q_{B2} represent the displacements at the front, rear AMBs; x_i and y_i represent translations in the x and y directions; β_{xi} and β_{yi} are the angular displacements about the y and x axes, respectively. The finite element rotor model has been updated and verified by previous work [20] since an accurate rotor model is the premise of identification.

Fig. 2 shows the theoretical free-free undamped mode shapes of the rotor used in the AMBs system. The first bending critical speed of this rotor is around 480 Hz (28,800 rpm). In order to visualize the gyroscopic effects, a Campbell diagram is drawn at 10^6 N/m support stiffness in Fig. 3. With the gyroscopic effects, the resonance frequencies are the functions of the rotor speed. In Fig. 3, the red dashed line indicates the operating speed of the rotor and large synchronous vibration will occur when it intersects with the rotor natural frequencies.

2.3. Equations of motion

The coordinate system of the rotor is pictured in Fig. 4 and the identification in this paper focuses on two orthogonal directions of each radial AMBs.

The electromagnetic force provided by the AMBs are modeled as stiffness $\mathbf{K}_B$ and damping $\mathbf{C}_B$. Considering the residual unbalances effects on the rotor, we can obtain the following EOM for the rotor AMBs system,

$$\mathbf{M}_R\ddot{q} + (\mathbf{C}_R + \mathbf{C}_B + \Omega\mathbf{G}_R)\dot{q} + (\mathbf{K}_R + \mathbf{K}_B)q = f_{unb} + f_{res}, \tag{3}$$

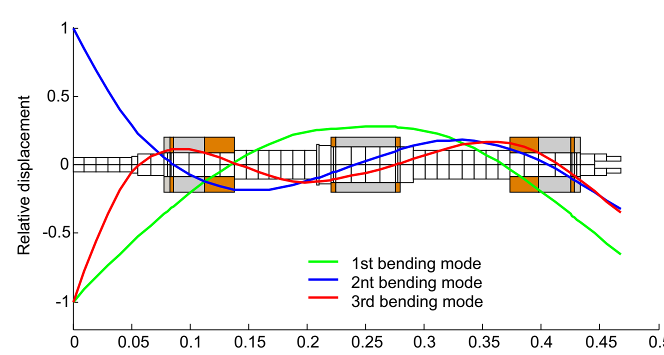

Fig. 2. Theoretical free-free mode shapes of the rotor.