Proceedings of the International Conference on Applied Mechanical Engineering Research (IC-AMER2019)
NIT Warangal, India – May 02-05, 2019
Paper No. XXX (The number assigned when the papaer is accepted)

Quantitative Estimation of Unbalance and AMB's parameters in a rigid rotor levitated by Active Magnetic Bearings

Prabhat Kumar*, Vikas Kumar, Kundan Kumar, Lakhan Singh Meena
Department of Mechanical Engineering, NIT Manipur, Imphal West, 795004, Manipur, India

*Corresponding Author Email: ysprabhat.pamho@gmail.com

Abstract

Online identification of most severe unbalance fault in a high speed rotating machinery system such as pumps, gas turbines, etc. is very essential for their smooth functioning. In the present paper, a model based identification algorithm has been developed to estimate quantitatively the unbalance parameters such as magnitude and phase of unbalance as well as force-displacement stiffness and force-current stiffness constants of AMBs in a mathematical model of a rotor system with active magnetic bearing. A mathematical model consisting of a rigid rotor with a disc at middle position levitated by two active magnetic bearings are developed for this estimation purpose. Equations of motion of the rotor system are derived and numerically simulated to generate displacement and current response of the rotor system in time domain. A fast Fourier Transform technique is utilized to convert the time domain response into frequency domain signal, which further used in developed identification algorithm to estimate the unbalance fault parameters as well as AMB's constants. It has been found that the estimated results are highly stable and have good accuracy.

Keywords: Active Magnetic Bearing, Fast Fourier Transform, Identification, Unbalance

1. Introduction

Rotating elements are quite common to our use now a days. Usually the rotating machines such as centrifugal pumps, gas turbines, aircraft engines, etc. have bearings to provide supports to the rotor through physical contact but, the recent trend is to use Active Magnetic Bearings (AMBs) that levitate the rotor system in the air using electromagnetic force. Thus, the rotor does not feel any frictional resistance or wear while in rotation. This system is quite advantageous as there is absence of lubrication, high precision and long-life time system monitoring and there is cost-down in maintenance also, but above all it also facilitates smooth functioning at ultra-high speed. These AMBs find application where lubrication may be prohibitive, as in vacuum pumps as well as medical appliances [1]. In the course of gaining this high speed, often the rotating machines comes through a phase when the rotor starts vibrating severely with large amplitude and sometime it even fails. The speed at which this phenomena occurs is called critical speed. The culprit here is most of the time, the unbalance. A rotating system with unbalance is often noisy, unsafe and short-lasting and requires more maintenance. Rotating unbalance is mainly caused by uneven distribution of mass around an axis of rotation. Mathematically, it is the product of the rotor mass and its eccentricity (the distance of the centre of gravity of the rotor from its centre of rotation) [2]. This unbalance is monstrous during high speed rotation. So the slightest unbalance can be potentially of high impact. Nordmann and Aenis [3] have used AMBs as sensor elements to measure both outputs and inputs and as actuator elements to excite the rotor system for detection and diagnosis of faults in centrifugal pumps. Sinha [4] and Edwards et al [5] have given an overview of vibration-based condition monitoring of rotating machine. De Queiroz [6] developed a new method for identification of the unknown unbalance parameters of a simple Jeffcott like rotor by exploiting a dynamic robust control mechanism. He used unbalance disturbance forces by active feedback control mechanism to identify the unbalance-related parameters. Markert et al. [7] and Platz et al.[8] presented a model to generate a dynamic behaviour of an unbalance rotor system utilizing least-squares fitting approach in the time domain. Sudhakar and Sekhar [9] presented three different approaches based on least squares fitting technique to identify the unbalance fault in a rotor system supported by two conventional ball bearings. More significant information's such as locality and severity of fault can be provided by model based diagnostic techniques. Zhou and Shi [10] have given a very comprehensive idea of active unbalance control systems.

Although many research works have been previously presented on system diagnosis and unbalance fault identification in a rotor system supported by conventional bearings, the investigation upon AMB-rotor systems is not much addressed adequately, leave aside the quantitative assessment. Conventional bearings have several disadvantages such as low reliability and low efficiency and can be operated in low speeds only. An AMB system offers various advantages over conventional bearings such as rotor can be operated at high spin speeds, can reduce vibration through active and adjustable damping and stiffness coefficients employing a controller. This paper focuses on employing a mathematical model consisting of a rigid rotor with a disc at middle position levitated by two active magnetic bearings for the quantitative estimation of unbalance parameters such as eccentricity and phase of unbalance and AMB constants such as force-displacement stiffness and force-current stiffness.

2. System model configuration

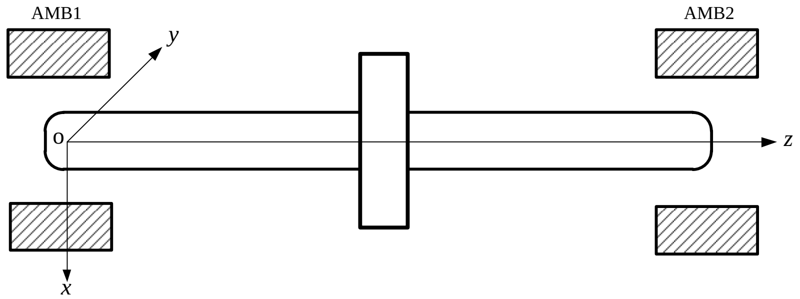

Fig. 1. A rigid rotor system with two active magnetic bearings and a heavy rotor at the middle

The rotor system used for the purpose is depicted in Fig.1. It is an unbalanced rigid rotor with a heavy disc at the middle position levitated by two AMBs. The shaft has been assumed to be rigid and massless based on lumped mass parameter model. AMBs with proportional-derivative (PD) control strategy are placed close to the rotor at both the ends. The translational degrees of freedom are considered to be in vertical (x-axis) and horizontal (y-axis) directions.

3. Unbalance force model

Unbalance fault in the rotor gives sinusoidal force. Force due to unbalance can be written in x and y directions as

$$f_{unbx} = me\omega^2 \cos(\omega t + \beta), \qquad f_{unby} = me\omega^2 \sin(\omega t + \beta) \tag{1}$$

where m is disc mass, e is disc eccentricity, ω is spin speed of rotor, β is phase of unbalance.

4. Force model due to Active Magnetic Bearings

The force due to AMBs is also bidirectional, the x- and y- components can be written as

$$f_{AMBx} = -k_s x + k_i i_x, \qquad f_{AMBy} = -k_s y + k_i i_y \tag{2}$$

where, k_s and k_i are force-displacement and force-current stiffness parameter of both AMBs. i_x and i_y are controlling current output from PD controller these can also be expressed as

$$i_{cx} = k_P x + k_D \dot{x}, \qquad i_{cy} = k_P y + k_D \dot{y} \tag{3}$$

where, k_P and k_D are proportional and derivative gains of PD controllers.

5. Equations of motion of the rotor system

Equation of motion of the rigid rotor system with consideration of inertia forces, forces due to unbalance and both AMBs in x and y direction can be expressed as

$$m\ddot{x} = f_{unbx} - 2 f_{AMBx}, \qquad m\ddot{y} = f_{unby} - 2 f_{AMBy} \tag{4}$$

Equation (4) can be written in complex form as

$$m\ddot{r} = me\omega^2 e^{j(\omega t + \beta)} - 2 f^c_{AMB} \tag{5}$$

where, complex displacement and complex controlling current are

$$r = x + jy, \qquad i_c^c = i_x + ji_y \tag{6}$$

Complex displacement response and AMB current response is utilised for the purpose of estimating unbalance parameters such as eccentricity of unbalance (e) and phase of unbalance (β) as well as AMB's constants such as force-displacement stiffness parameter (k_s) and force-current stiffness parameter (k_i). Displacement response of rotor at disc position $R(t)$ and current response at AMB position $I(t)$ can be assumed as $R(\omega)e^{j\omega t}$ and $I(\omega)e^{j\omega t}$ respectively in a harmonic form. So, the second derivative of displacement response in harmonics form can be expressed as

$$R(t) = R(\omega)e^{j\omega t}, \qquad \ddot{R}(t) = -\omega^2 R(\omega)e^{j\omega t} \tag{7}$$

FFT function of MATLAB™ is applied to convert the complex system response in time domain signal into frequency domain signal.

6. Procedure for quantitative estimation of unbalance and AMB's parameters

Now, complex form of unbalance can be written as

$$me\omega^2 e^{j(\omega t+\beta)} = m\omega^2 e^{j\omega t}\left(e\cos\beta + je\sin\beta\right) = m\omega^2 e^{j\omega t}\left(e_{\mathrm{Re}} + je_{\mathrm{Im}}\right) \tag{8}$$

On substituting equations (7) and (8) into equation (5), and then writing the real and imaginary part of the equations in matrix form such that unknown parameters (e_{Re}, e_{Im}, k_s and k_i) are on left side of equations and known parameter (m) as

$$\begin{bmatrix} -m\omega^2 & 0 & -R_{\mathrm{Re}}(\omega) & I_{\mathrm{Re}}(\omega) \\ 0 & -m\omega^2 & -R_{\mathrm{Im}}(\omega) & I_{\mathrm{Im}}(\omega) \end{bmatrix} \begin{Bmatrix} e_{\mathrm{Re}} \\ e_{\mathrm{Im}} \\ k_s \\ k_i \end{Bmatrix} = \begin{Bmatrix} m\omega^2 R_{\mathrm{Re}}(\omega) \\ m\omega^2 R_{\mathrm{Im}}(\omega) \end{Bmatrix} \text{ or } \mathbf{A}_1(\omega)\mathbf{x} = \mathbf{b}_1(\omega) \tag{9}$$

Equation (9) is an underdetermined system with two equations and four unknowns, so it is extremely difficult to get solution. Thus, rotor is operated with two spin speeds (ω_1 and ω_2). Then, the modified matrix equation for two speeds will be written as

$$\mathbf{A}_2(\omega)\mathbf{x} = \mathbf{b}_2(\omega) \quad \text{with } \mathbf{A}_2(\omega) = \begin{bmatrix} \mathbf{A}_1(\omega_1) \\ \mathbf{A}_1(\omega_2) \end{bmatrix}; \ \mathbf{b}_2(\omega) = \begin{bmatrix} \mathbf{b}_1(\omega_1) \\ \mathbf{b}_1(\omega_2) \end{bmatrix} \tag{10}$$

Now, there are four equations and four unknowns, so it is easy to solve this equation (10) to estimate the identifiable parameters vector as

$$\mathbf{x} = \left\{ e_{\mathrm{Re}} \quad e_{\mathrm{Im}} \quad k_s \quad k_i \right\}^T \tag{11}$$

7. Response generation and estimation of unbalance and AMB parameters

SIMULINK™ model as shown in Fig. 2 has been used to generate the response of the rotor system such as displacement and current response in x and y direction in time domain using equation(5).

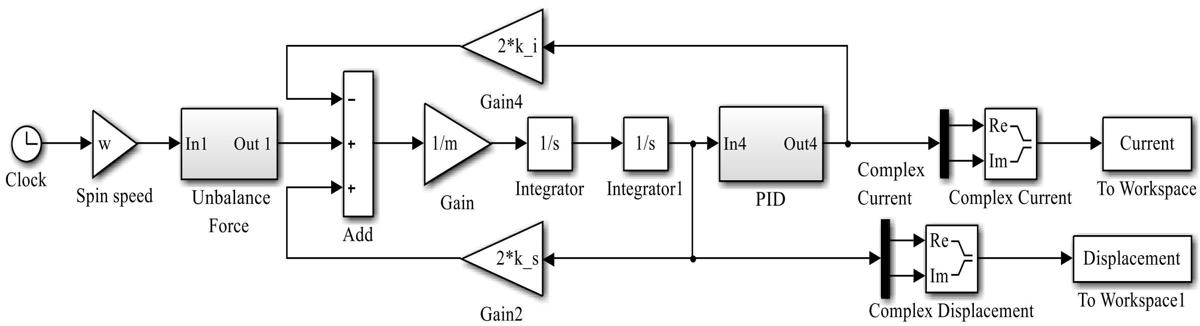

Fig. 2. SIMULINK model of the problem

The simulation for the rigid rotor system was run for 5 s using Table 1 parameters. Responses in the time domain from 4 s to 5 s was considered for further analysis. A fourth-order Runga-Kutta solver with a fixed step size of 0.0001s is used for generating system responses. Responses in time domain for the unbalanced rigid-AMB rotor system at a spin speed of 106.8 rad/s is represented in Fig. 3. FFT analysed signal for displacement and current responses with magnitude and phase at the same spin speed is presented in Fig. 4.

Table 1. Rotor-AMB system parameters for simulation purpose

Rotor parameters	**Assumed Values**	**AMB parameters**	**Assumed Values**
Disc mass (m)	2 kg	AMB force displacement constant (k_s)	175,440 N/m
Disc eccentricity (e)	100 μm	AMB force displacement constant (k_i)	35.09 N/A
Unbalance phase (β)	10 deg	Proportional constant of PD controller (k_P)	6000 N/m
		Derivative constant of PD controller (k_D)	3 Ns/m

It is observed from Fig. 4 that the maximum amplitude of the displacement and controlling current response is 4.347×10^{-5} m and 0.2613 A at an angular frequency of 17 Hz (or 106.8 rad/s). Real and imaginary part of responses is obtained using magnitude and phase of FFT analysed response signal at two different speeds, i.e. (106.8 and 188.5 rad/s) which is further used in equation (10) to estimate the required parameters.

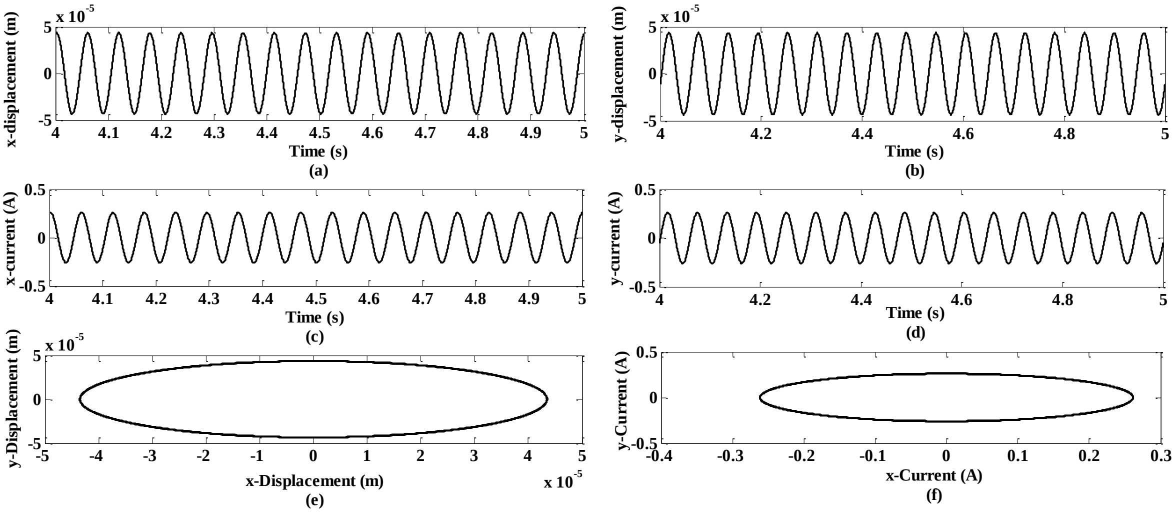

Fig. 3. Responses of rigid rotor system: (a) x-direction displacement (b) y-direction displacement (c) x-direction current (d) y-direction current (e) Displacement orbit (f) current orbit

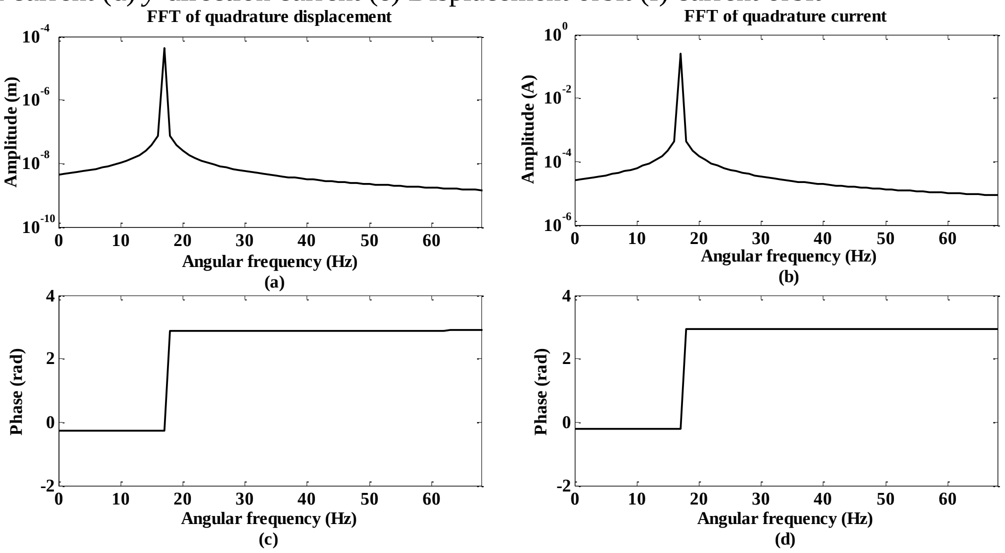

Fig. 4. FFT analysed responses: (a) amplitude of displacement (b) amplitude of current (c) phase of displacement (d) phase of current response

Table 2 summarizes the estimated values of the unbalance and AMB's parameters and their error percentage with respect to assumed ones. It can be observed the error percentage is very less for all parameters. Unbalance parameters such as eccentricity (e) and phase (β) are found to be very much accurate and stable using the developed identification algorithm.

Table 2: Assumed and estimated values of parameters related to unbalance fault and AMBs.

Parameters	**Symbol**	**Assumed values**	**Estimated values**	**Error %**
Unbalance magnitude	e (μm)	100.00	99.83	-0.17
Unbalance phase	β (deg)	10.00	10.04	0.40
Force-displacement constant of AMBs	k_s (N/m)	175,440	172,265	-1.81
Force-current constant of AMBs	k_i (N/A)	35.09	34.24	-2.42

8. Conclusions

This paper concludes with development of identification algorithm for quantitative estimation of unbalance and AMBs parameters in rigid rotor system levitated by two active magnetic bearings. For accomplish this purpose, a mathematical model is presented. Equations of motion of the considered rotor system is derived and numerically simulated to obtain displacement and current response of the system. FFT analysed response in frequency domain was used in the algorithm for online estimation of the parameters. Estimation results show that the algorithm is robust and effective providing good accuracy and stable estimated values.

Acknowledgements

The first author acknowledges students and faculty members of National Institute of Technology, Manipur for their admirable help and support.

References

[1] M. Kasarda, "An overview of active magnetic bearing technology and applications," *Shock and Vibration Digest, 2000,* vol. 32, pp. 91-99, 2000.
[2] R. Tiwari, *Rotor Systems: Analysis and Identification*. Boca Raton: CRC Press, 2017.
[3] R. Nordmann and M. Aenis, "Fault diagnosis in a centrifugal pump using active magnetic bearings," *International Journal of Rotating Machinery,* vol. 10, pp. 183-191, 2004.
[4] J. K. Sinha, "Health monitoring techniques for rotating machinery," University of Wales, Swansea, 2002.
[5] S. Edwards, A. Lees, and M. Friswell, "Estimating rotor unbalance from a single run-down," in *IMechE conference transactions*, 2000, pp. 323-334.
[6] M. De Queiroz, "An active identification method of rotor unbalance parameters," *Journal of Vibration and Control,* vol. 15, pp. 1365-1374, 2009.
[7] R. Markert, R. Platz, and M. Seidler, "Model based fault identification in rotor systems by least squares fitting," *International Journal of Rotating Machinery,* vol. 7, pp. 311-321, 2001.
[8] R. Platz, R. Markert, and M. Seidler, "Validation of online diagnostics of malfunctions in rotor systems," in *IMechE conference transactions*, 2000, pp. 581-590.
[9] G. Sudhakar and A. Sekhar, "Identification of unbalance in a rotor bearing system," *Journal of Sound and Vibration,* vol. 330, pp. 2299-2313, 2011.
[10] S. Zhou and J. Shi, "Active balancing and vibration control of rotating machinery: a survey," *Shock and Vibration Digest,* vol. 33, pp. 361-371, 2001.