International Journal of Mechanical Sciences 85 (2014) 152–159

Contents lists available at ScienceDirect

International Journal of Mechanical Sciences

journal homepage: www.elsevier.com/locate/ijmecsci

On-line algebraic identification of eccentricity parameters in active rotor-bearing systems

M. Arias-Montiel [a,*], F. Beltrán-Carbajal [b], G. Silva-Navarro [c]

[a] Universidad Tecnológica de la Mixteca, Instituto de Electrónica y Mecatrónica, C.P. 6900 Huajuapan de León, Oaxaca, Mexico
[b] Universidad Autónoma Metropolitana, Unidad Azcapotzalco, Departamento de Energía, C.P. 02200 Mexico, D.F., Mexico
[c] Centro de Investigación y de Estudios Avanzados del I.P.N., Departamento de Ing. Eléctrica, Sección de Mecatrónica, C.P. 07360 Mexico D.F., Mexico

ARTICLE INFO

Article history:
Received 27 September 2013
Received in revised form
30 April 2014
Accepted 24 May 2014
Available online 2 June 2014

Keywords:
On line identification
Disturbance estimation
Rotor bearing system

ABSTRACT

In this work the on-line unbalance parameter identification problem in a rotor-bearing system is dealt by an algebraic approach. The system has two disks asymmetrically located along the shaft which is supported by a conventional ball bearing at one end and by an active suspension at the other one. The Finite Element Method (FEM) is used in order to obtain a reduced order model for the rotor-bearing system. The identification process is carried out on-line and the proposed method requires only the lateral shaft displacements at the disks' location measurements to estimate both, disturbance forces caused by unbalance and eccentricity parameters. FEM model and identified unbalance parameters are used to synthesize an active control scheme in order to attenuate the lateral vibration amplitudes in the rotor-bearing system. Numerical results show the fast convergence of the estimated parameters and disturbances to the real ones and considerable reductions in vibration amplitudes when the system passes through its first critical speed.

1. Introduction

At present, dynamic models and numerical simulations are very important tools in the design and analysis of rotating machines, so that it is necessary to get models which represent real systems' behavior with a good precision. The accuracy of a model is determined by comparing the response of the model and the response of the real system to the same input signal [1]. In rotating machines the presence of mass unbalance is inevitable. This unbalance causes centrifugal forces of which magnitudes depend on the rotor mass, angular speed and distance between geometric center and center of mass of the rotor (eccentricity) [2,3]. Eccentricity represents one of the most difficult parameters to measure or to estimate in a rotor-bearing system and, consequently, it is an important source of less of accuracy for its mathematical model [4].

Different approaches to solve the problem parameters' identification in rotor-bearing systems have been proposed. Yuan-Pin and An-Chen [5] developed a method for estimating unbalance distributions of flexible shafts and constant eccentricities of rigid disks based on the transfer matrix method. They presented numerical results for a system with only a disk and operating to constant speed. De Queiroz [6] presented a method to identify the unknown unbalance parameters of a Jeffcott rotor based on a dynamic robust control technique, in which the disturbance forces are estimated and then, from these forces, the magnitude and phase of the unbalance are obtained. This strategy is proved by numerical simulations and the rotational speed of the machine has to satisfy the persistency of excitation condition in order to guarantee the convergence of the method. Mahfoud et al. [7] proposed a method to identify the matrices of a state variable model for rotordynamics systems using curve fitting techniques and optimization procedures based on least-squares methods, measuring the full state vector (displacement, speed and acceleration). The external forces can be found proposing an inverse problem from the model with the matrices previously determined. Sudhakar and Sekhar [8] estimated the unbalance faults in a Jeffcott-like rotor system with a fault identification approach, obtaining good results in both numerical and experimental ways, showing the need of new methods and techniques to solve the unbalance parameters estimation problem.

The main source of undesired vibration in rotating machinery is the mass unbalance in rotating parts [2,3]. This phenomenon can cause an unacceptable level of vibration with failures in the bearings, high levels of noise, wearing in the mechanical components and, eventually, dangerous failures in machines and, hence, control systems are needed to reduce the vibration amplitudes to acceptable values for a safe machine operation. For these purposes, many passive, semi-active and active devices have been proposed

* Corresponding author.
E-mail addresses: mam@mixteco.utm.mx (M. Arias-Montiel), fbeltran@correo.azc.uam.mx (F. Beltrán-Carbajal), gsilva@cinvestav.mx (G. Silva-Navarro).

http://dx.doi.org/10.1016/j.ijmecsci.2014.05.027
0020-7403/

(see, e.g., [9–13]). In general, a control scheme for vibration attenuation in rotating machinery is synthesized using a system model, so that, it is very important that the model represents the real system behavior with good accuracy, for this, it is necessary to have certainty in the model parameters. However, in contrast to many engineering systems, the rotor system models based on FEM are complex to be used in control engineering because of their large number of degrees of freedom. In addition, in rotor-bearing systems few measurements are available and unknown system parameters like eccentricities and exogenous forces are commonly present. In this context, asymptotic observers or state estimators can be designed, from measurements of the input and the response of the rotor-bearing system, to provide an approximation of the system states or disturbances that cannot be directly measured (see, e.g., [2,14,15]).

In this work, an on-line estimation scheme for the unbalance forces using asymptotic state observers is proposed for an asymmetrical rotor-bearing system with two unbalanced disks. For this estimation method, only measurements of the radial shaft displacements at disk locations are needed. The eccentricity in each disk can algebraically be estimated through the unbalance forces exciting the overall system. This estimation approach is used to design an active control scheme for attenuation of the vibration amplitude in the rotor-bearing system, particularly when it passes through first critical speed. Some numerical results are presented in order to show the dynamic and robust performance of the proposed estimation and control schemes.

2. Dynamical model of the rotor system with active suspension

The rotor-bearing system considered in this paper is shown in Fig. 1. The rotor-bearing system has two disks asymmetrically located along the shaft, which is supported by a conventional ball bearing at left end and by an active suspension at the right one.

The rotor-bearing system model is obtained by finite element methods, using elements' type Euler beam and the consistent matrices' approach described by Genta [16]. The system has three elements, four nodes and two degrees of freedom per node, with one radial displacement and one angle denoting the shaft deflection. One node is located at each support and one node at each disk location. Applying the boundary conditions to the model, that is, considering as rigid the left support, the generalized coordinates describing the rotor are then given by

$$\mathbf{u} = [\beta_{y1}\ R_{x2}\ \beta_{y2}\ R_{x3}\ \beta_{y3}\ R_{x4}\ \beta_{y4}]$$

where β_{yi} and R_{xi} are the shaft deflection angle and radial displacement in each node, respectively. The above system coordinates can describe the horizontal plane of motion; the vertical plane can be similarly modeled.

The active suspension mass and stiffness, disks mass and moment of inertia, are considered as lumped parameters at their corresponding degrees-of-freedom. Therefore, the system dynamics is described by the equations of motion

$$\mathbf{M}\ddot{\mathbf{u}} + \mathbf{D}\dot{\mathbf{u}} + \mathbf{K}\mathbf{u} = \mathbf{b}f(t) + \mathbf{e}_1\varpi_1(t) + \mathbf{e}_2\varpi_2(t), \quad \mathbf{u} \in R^7,\ f(t) \in R \tag{1}$$

where $\mathbf{M} \in R^7$ and $\mathbf{K} \in R^7$ are the global mass and stiffness matrices respectively and $\mathbf{D} \in R^7$ is a proportional damping matrix. These matrices are symmetric and positive definite and, therefore, the unperturbed rotor-bearing system is asymptotically stable (see, e.g., [14]). In addition, $\varpi_1(t)$ and $\varpi_2(t)$ are disturbance harmonic and synchronous forces caused by the unbalance at both disks and $f(t)$ is the control force provided by the active suspension, and the vectors

$$\mathbf{b} = [0\ 0\ 0\ 0\ 0\ 1\ 0]^T$$
$$\mathbf{e_1} = [0\ 1\ 0\ 0\ 0\ 0\ 0]^T$$
$$\mathbf{e_2} = [0\ 0\ 0\ 1\ 0\ 0\ 0]^T$$

denote the corresponding degree-of-freedom or channel where the control force and unbalance forces are entering into the system, respectively.

By defining the state vector $\mathbf{z} = [\mathbf{u}\ \dot{\mathbf{u}}]^T \in R^{14}$, the rotor-bearing system dynamics (1) can be described in state space form as follows:

$$\dot{\mathbf{z}} = \mathbf{A}\mathbf{z} + \mathbf{B}f(t) + \mathbf{E}_1\varpi_1(t) + \mathbf{E}_2\varpi_2(t), \quad \mathbf{z} \in R^{14},\ f(t) \in R \tag{2}$$

with matrices

$$\mathbf{A} = \begin{bmatrix} \mathbf{0} & \mathbf{I} \\ -\mathbf{M}^{-1}\mathbf{K} & -\mathbf{M}^{-1}\mathbf{D} \end{bmatrix}$$

$$\mathbf{B} = \begin{bmatrix} \mathbf{0} \\ \mathbf{M}^{-1}\mathbf{b} \end{bmatrix}$$

$$\mathbf{E_1} = \begin{bmatrix} \mathbf{0} \\ \mathbf{M}^{-1}\mathbf{e_1} \end{bmatrix}$$

$$\mathbf{E_2} = \begin{bmatrix} \mathbf{0} \\ \mathbf{M}^{-1}\mathbf{e_2} \end{bmatrix}$$

The rotor-bearing system parameters are shown in Table 1. For these parameters the perturbed rotor-bearing system (2) is completely controllable from the control force $f(t)$.

The dynamic behavior of the undamped system in free vibrations is given by

$$\mathbf{M}\ddot{\mathbf{u}} + \mathbf{K}\mathbf{u} = \mathbf{0}, \quad \mathbf{u} \in R^7 \tag{3}$$

and solving the so-called characteristic equation $\det(-\omega^2\mathbf{M} + \mathbf{K}) = 0$ we obtain the system natural frequencies

$$\omega_1 = 39.99\ \text{Hz} = 2399.7\ \text{rpm}$$
$$\omega_2 = 274.11\ \text{Hz} = 16,447\ \text{rpm}$$
$$\omega_3 = 836.34\ \text{Hz} = 50,181\ \text{rpm}$$

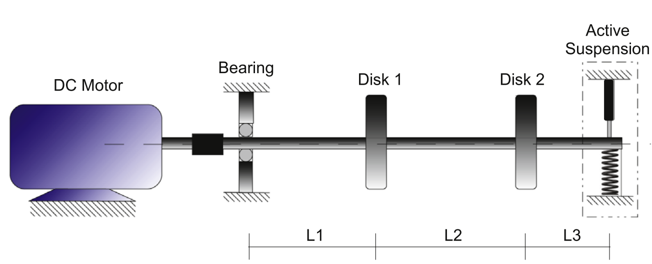

Fig. 1. Rotor system with active suspension.

Table 1
Rotor-bearing system parameters.

Parameter	Value
Shaft density, ρ	7850 kg m^{-3}
Shaft diameter, d	0.02 m
Mass per length unit, m	2.466 kg m^{-1}
Disks masses, $m_{d\ 1}$, $m_{d\ 2}$	3.9 kg
Disk diameters, d_1, d_2	0.1508 m
Element length, L_1	0.25 m
Element length, L_2	0.35 m
Element length, L_3	0.15 m
Elasticity module, E	211 GPa
Area moment of inertia of shaft, I	7.854×10^{-9} m^4
Mass moment of inertia per length unit of shaft, j	6.1654×10^{-5} kg m
Mass moment of inertia of disks, $J_{d\ 1}$, $J_{d\ 2}$	0.0057 kg m
Eccentricity of disk 1, a_1	9×10^{-5} m
Eccentricity of disk 2, a_2	11×10^{-5} m