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Research paper

Nonlinear dynamic analysis and experimental verification of a magnetically supported flexible rotor system with auxiliary bearings

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ABSTRACT

In this paper, nonlinear dynamic analysis and experimental verification of a flexible rotor supported on the active magnetic bearings (AMB) are studied. The model of the system is formulated by eight degrees of freedom. This model takes in to account the gyroscopic moments of the disk, geometric coupling of the magnetic actuators and contact forces of the auxiliary bearings. The equations of motion are solved using the Rung–Kutta method. The effects of auxiliary bearings stiffness and rotational speed on the dynamic behavior of the system are investigated by the bifurcation diagrams, dynamic trajectories, power spectra analysis, Poincare´maps and maximum Lyapunov exponent. In the experimental test rig, two special flexible supports are constructed that can adjust the required stiffness of the auxiliary bearings. The results indicate that the auxiliary bearings stiffness and rotational speed have significant effects on the dynamic responses and can be used as effective control parameters for the system. Good correlation between the experimental and theoretical results is found. Very rich forms of periodic, quasi-periodic, period −4 and chaotic vibrations are observed. The present study can be useful in designing and selection of suitable operating parameters. As a result, the system can avoid the undesirable behavior

1. Introduction

Active magnetic bearings (AMB) possess several advantages when compared to conventional bearings. Some advantages of implementing an AMB system include the contactless operation, complete elimination of oil-based lubrication systems, low parasitic power loss, high control precision, wide operational temperature ranges, lower maintenance costs and longer system life [1,2]. Unfortunately the AMB systems are intrinsically nonlinear. These nonlinear characteristics cause nonlinear motion of the rotor, including jump phenomenon, generation of combination resonances, evidence of period-doubling bifurcations and quasi-periodic and chaotic motions [3,4]. Thus, a fundamental scientific investigation of rotor – AMB system nonlinearities is required.

Zhang et al. [5] and Amer et al. [6] proposed the use of time varying stiffness to study the transient and steady-state nonlinear responses of a rigid rotor – AMB system. They showed the potential of using time varying stiffness in the con-

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Nomenclature

A_p	area of one magnetic pole
C	external damping coefficient
$D.E.$	dissipated energy
$\bar{D}$	derivative feedback gain
E	Young's modulus of the shaft
e_A	shaft penetration depth into the auxiliary bearing A
e_D	shaft penetration depth into the auxiliary bearing D
$F_{C_{x_1}}$, $F_{C_{y_1}}$	horizontal and vertical components of the contact force in auxiliary bearing A
$F_{C_{x_3}}$, $F_{C_{y_3}}$	horizontal and vertical components of the contact force in auxiliary bearing D
$F_{M_{x_1}}$, $F_{M_{y_1}}$	horizontal and vertical components of the electromagnetic force in active magnetic bearing A
$F_{M_{x_3}}$, $F_{M_{y_3}}$	horizontal and vertical components of the electromagnetic force in active magnetic bearing D
f_{n_A}, f_{t_A}	radial and tangential components of the contact force in auxiliary bearing A
f_{n_D}, f_{t_D}	radial and tangential components of the contact force in auxiliary bearing D
g_0	radial clearance of AMBs
g_1	radial clearance of auxiliary bearings
$H(\bullet)$	Heaviside step function
I	area moment of inertia of the shaft cross-section about the neutral axis
I_0	bias magnetic coil current
I_P, I_T	polar and diametral mass moments of inertia of disk d
i_{x_1}, i_{y_1}	horizontal and vertical control currents in active magnetic bearing A
i_{x_3}, i_{y_3}	horizontal and vertical control currents in active magnetic bearing D
k_a	stiffness of the auxiliary bearings
L	length of the shaft
m_1, m_3	Mass of each shaft journal
m_2	masses of the disk d
N	number of coil turns
$\bar{P}$	proportional feedback gain
r_1	radius of disk d
S	speed parameter
T_{m_1}, T_{m_3}, T_d	kinetic energy of the lumped masses and disk
u	mass eccentricity
V	strain energy stored in the shaft
$x(z)$, $y(z)$	shape functions of the shaft
x_1, x_2, x_3	generalized horizontal coordinates
y_1, y_2, y_3	generalized vertical coordinates
α	geometric coupling parameter
α_1	ratio of position of disk d to length of the shaft
θ_x, θ_y, φ	angular displacements of disk d
μ	friction coefficient of the auxiliary bearings
μ_0	permeability of free space
ψ_A, ψ_D	inclination angles
Ω	Rotational speed of the shaft

troller to enlarge the stability region. Again, their work is largely limited to a particular choice of system parameters such as unbalanced mass.

Inayat–Hussain [7,8] studied the effects of misalignment and geometric coupling on the response of flexible rotor – AMB system. Numerical results showed that without the geometric coupling, the response of the rotor was always synchronous regardless of the values of the misalignment.

A rotor – AMB system must be equipped by auxiliary bearings which retain the amplitudes of vibrations in safe limits after their undesirable increase. More recently, Inayat–Hussain [9] also gives an insight about the response of a two dimensional rigid rotor – AMB system, which shows chaotic behavior when impacting on the wall of the auxiliary bearings. Also, the occurrence of nonsynchronous response reduces with decreasing the stiffness of auxiliary bearings for rotors with small level of imbalance.

The dynamical model of rotor - AMB system that consists of flexible shaft was examined with the consideration of the contact forces [10]. The effects of various design and operating parameters, namely the rotor imbalance, journal to disk mass ratio, shaft stiffness and magnetic bearing force, on the bifurcations in the rotor's response were investigated. Numerical