FLEXIBLE ROTOR SUPPORTED BY MAGNETIC BEARINGS: DESIGN AND EXPERIMENTAL RESULTS

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Magnetic bearings present a number of interesting characteristics which provide some advantages over conventional bearings for a variety of applications. First of all, they allow the suspension of rotors or other mechanical components without any contact, eliminating mechanical wear and lubrication systems, reducing consequently the power comsumption. Furthermore, magnetic bearings present high dynamical performance enabling their use as active damping actuators in vibration control.

Magnetic bearings have successfully been applied in turbomachinery, centrifuges, vacuum pumps, canned pumps, machine tool spindles, robotics, vibration isolating contactless actuators and spacecraft equipment.

The present paper describes the development of a vertical flexible rotor totally supported by magnetic bearings that operate in rotations beyond the first natural bending frequency. Figure 1 presents the basic constructive components of the mechanical test rig. The rotor is composed by two shafts connected by an elastical coupling which allows flexural, axial and torsional vibration modes.

The axial magnetic bearing provides the rotor suspension and two radial ones ensures the radial position control. Contactless sensors monitor the axial and radial positions. The rotor is driven by an induction electrical motor excited by a frequency converter, allowing rotor speed control.

The rotor mechanical design was performed looking for a low bending frequency system, in order to get a safe test condition and allowing the dynamic study of a general rotor-bearing system with elastical coupling.

The electromagnetic components of the bearing were designed employing a software Finite Element Method-FEM developed at Santa Catarina Federal University-UFSC. The following design parameters were taken into account: physical dimension limitations, electromagnetic and electrical saturation limits and the maximum feasible stiffness.

A suitable rotor physical model was achieved and the differential equations of motion were deduced for numerical simulation tasks. The rotor geometrical and physical characteristics were determined and the dynamical ones were identified through "Bump-Tests" using a Spectrum Analyses.

Closed-loop control of the rotor bearing system is essential for stable operation. Figure 2 shows a basic block diagram of the whole system.

The control block is composed by an axial and a radial control system. The radial control is designed employing the Poles and Zeros Allocation Method, based on the knowledge of the mechanical resonance characteristics and looking for a stable and well damped passage over the critical speeds, two rigid body modes and the first bending mode. The axial control is designed adopting a notch filter allocated at the first axial resonance because the objective was neither control nor excite the rotor on it.

Both control systems, radial and axial, were tested through numerical simulation with suitable mechanical models which was performed in System Dynamic Analysis-ADS software developed at COPESP. Basically, in this software, the mechanical system may be represented by a differential equation, as follow:

$$\mathbf{M}*\ddot{\mathbf{Y}} + \mathbf{D}*\dot{\mathbf{Y}} + \mathbf{K}*\mathbf{Y} = \mathbf{F} \qquad (1)$$

where:

- $\mathbf{M}$ → Mass and Inertia matrix
- $\mathbf{D}$ → Internal damping and gyroscope effects matrix
- $\mathbf{K}$ → Internal stiffness matrix
- $\mathbf{Y}$ → Position variable vector
- $\mathbf{F}$ → External excitation vector (magnetic bearing forces)

Equation (1) may be expressed in a State-Variable form:

$$\dot{\mathbf{X}} = \mathbf{A}_m*\mathbf{X} + \mathbf{B}_m*\mathbf{F} \qquad (2)$$

where:

$$\mathbf{X} = \begin{bmatrix} \dot{\mathbf{Y}} \\ \mathbf{Y} \end{bmatrix} \quad \rightarrow \text{Mechanical state variable vector}$$

$$\mathbf{A}_m = \begin{bmatrix} \emptyset & \mathbf{I} \\ \mathbf{M}^{-1}*\mathbf{K} & \mathbf{M}^{-1}*\mathbf{D} \end{bmatrix} \quad \rightarrow \text{Mechanical system matrix}$$

$$\mathbf{B}_m = \begin{bmatrix} \emptyset \\ \mathbf{M}^{-1} \end{bmatrix} \quad \rightarrow \text{Mechanical control matrix}$$

The linearized magnetic bearing system may be represented by the following equations:

$$\begin{cases} \dot{\mathbf{Z}} = \mathbf{A}_e * \mathbf{Z} + \mathbf{B}_e * \mathbf{X} & (3) \\ \mathbf{F} = \mathbf{C}_e * \mathbf{Z} & (4) \end{cases}$$

where:

$\mathbf{A}_e$ → Electrical system matrix
$\mathbf{B}_e$ → Electrical control matrix (Sensor constants and linearized bearing gains)
$\mathbf{C}_e$ → Electrical system matrix
$\mathbf{Z}$ → Electrical state variable vector

Equations (2), (3) and (4) may be associated, resulting:

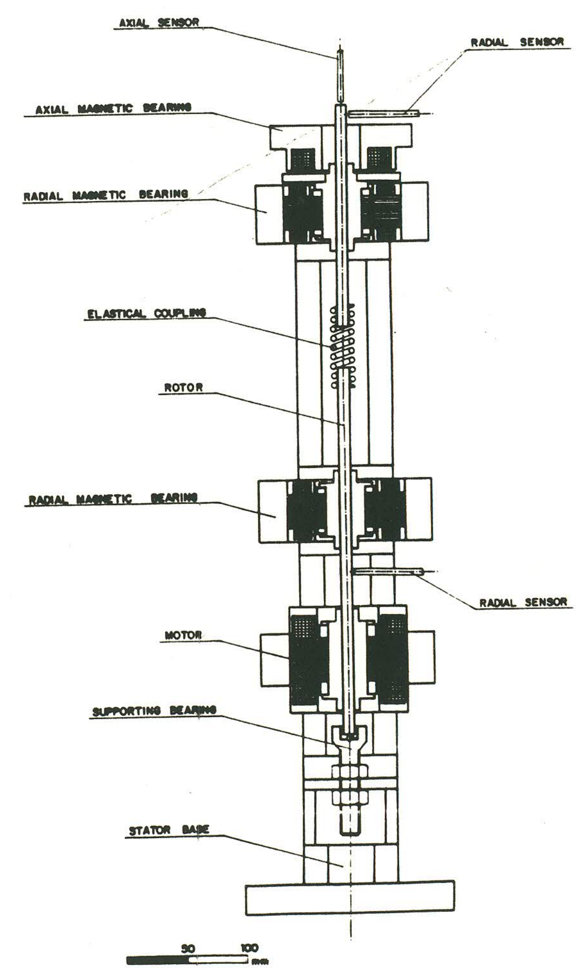
$$\begin{Bmatrix} \dot{\mathbf{X}} \\ \dot{\mathbf{Z}} \end{Bmatrix} = \begin{bmatrix} \mathbf{A}_m & \mathbf{B}_m * \mathbf{C}_e \\ \mathbf{B}_e & \mathbf{A}_e \end{bmatrix} * \begin{Bmatrix} \mathbf{X} \\ \mathbf{Z} \end{Bmatrix} \qquad (5)$$

Equation (5) corresponds to the complete linearized rotor-bearing closed-loop system and the eigenvalues and eigenvectors calculation and analysis, employing ADS software, constitute an effective design tool for stability analysis.

After the implementation of the magnetic bearings control systems, the complete test rig is experiment for actual stability behavior verification and dynamical parameter measurements.

Magnetic bearings provide remarkable advantages over other kinds of bearings: they may be used as contacless exciters for dynamic identification tests during the machine operation. A filtered noise signal may be introduced and the spectrum analysis carried out. This test so called "bump-test on line" and was applied in the present work.

The reasonable agreement between experimental results simulation shows that the presented design procedure of flexible rotors supported by magnetic bearings presented here can be successfully applied



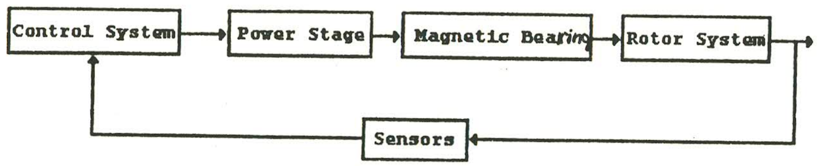


Figure2-Rotor-Bearings System BlockDiagram

Parameter Estimation of a Rotor System Excited by unmeasured Stochastic Forces

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Normaly, on the parameter estimation of systems, the measurements of the excitation and the response are made. Then, using different methods, its is possible to estimate the parameters from the mathematical model associated to the physical system.
In the specific case of mechanical systems, a particular internal structure for the dynamic matrix are found when the vector space notation is used. This structure can be used for example, to decrease the number of state variables to be measured for the complete identification of the system. These particularities were explored by some authors. In the class of mechanical systems excited by stochastic forces, others characteristics are found. These characteristics can also be fully used in an identification procedure. The stochastic excitation naturaly present in the process can be of great value, if it is reach enough in the frequency domain, to excite the frequencies of interest. On the other hand, the measurement of such excitation forces is very dificult to be realized, when not impossible. The possibility to deal with the information contained only in the response of such systems opens another research area in the identification field. Some authors have already worked on this topic with success. In the present work the used identification procedure is based on the Ljapunov Matricial Equation. In this method the estimation equations are obtained from the matricial relation

$$A \; Rxx(\tau) + Rxx(\tau)A^T = - BQB^T e^{A^T \tau},$$

With A(nxn): dynamic matrix of the system
B(nxu): input matrix
Rxx (nxn): autocorrelation matrix of the state vector x (t) with time lag τ
Q(uxu): excitation intensity matrix

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