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A DYNAMIC ANALYSIS AND CONTROL OF AN ACTIVE MAGNETIC BEARING

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Abstract. *Magnetic bearings are devices that provide support for any ferromagnetic body (rotor) using magnetic forces. There are several advantages to the use of these devices, being one the absences of contact between the rotor and the support. For that reason, the magnetic bearings require no lubrification. In addition, there is no friction on the surface of the device, making maintenance costs low. Furthermore, they can operate at high speeds and have a controllable dynamic. Active magnetic bearings are characterized by being unstable, requiring a control system to ensure stability, i.e. to keep the rotor positioned at the centre even in the presence of a disturbance. Thus, in order to solve the problem of stability, this paper proposes the dynamic analysis of an active magnetic bearing, for which was designed a control model. With that purpose, a simulation was performed, and a testing bench was built for the validation of the results.*

Keywords: *Rotordynamic, Active Magnetic Bearing, Control*

1. INTRODUCTION

Active Magnetic Bearings (AMBs) have frequently been applied in rotating machines such as pumps, turbines, compressors, and turboexpanders (Siva Srinivas *et al.*, 2018; Aeschlimann *et al.*, 2017; Jayawant and Masala, 2016). These are used to maintain the relative position of a rotating assembly (rotor) to a stationary component (stator) without any kind of mechanical contact by means of actively controlled electromagnetic forces, therefore, a closed-loop controller is required to maintain stability. Due to non-contact, the operation takes place without friction, reducing wear on the bearings and consequently maintenance costs low (Stephan *et al.*, 2013). No lubrication is required in the AMB, therefore there are no oil contamination problems (Wan, 2020). The AMB is also applied in medical devices like artificial heart pumps where are used to reduce friction and oil contamination (Sun *et al.*, 2020). These advantages justify the use of AMB as an alternative to replace the current conventional bearings in modern machines, mainly because they are suggested in applications where the operating environment becomes more extreme, such as extreme temperatures, corrosive working fluids, and contamination-free (Mushi *et al.*, 2012).

An AMB system is a combination of different mechanical, electrical and electromagnetic components specifically amplifiers, sensors, controllers, rotors, electromagnets and data acquisition. Therefore, is a complex mechatronic system. Although a device with relatively high cost compared to conventional bearings, it offers technological advantages such as controllable dynamics to the desired application by tuning of the control loop, accurate equilibrium position of the rotor also due to the control loop, diagnostics and online operations are readily performed and used to check the operating conditions and performance of the system (Pesch and Scavelli, 2019).

One of the greatest challenges in using the AMB is to control and stabilize the rotor so that it can support loads and ensure good dynamic behaviour against high spin speeds and disturbances that may occur. For this, new control strategies that can meet these needs must be developed and tested. Indeed, several control algorithms have been applied, the traditional controllers being the most used nowadays (Borase *et al.*, 2020). Examples of control strategies in an AMB system can be seen in the work of Ritonja *et al.* (2010), where PID controller was implemented to measure and analyze the behaviour of the system, additional dynamics, parameter variations, and noise. Simple control for a PI/PD with cascade connection is given in Polajžer *et al.* (2006). Siqueira and Pinto (2013) used the sliding mode control technique within a computational model of an unbalanced rotor supported by two magnetic bearings, this type of control has the advantage of order reduction, rejection of external disturbances and insensitivity to parameter variation.

This work studies the dynamic behaviour of the magnetic bearing system, evaluating the characteristics of the response for cases where the rotor is subject to static (without rotation), transient and dynamic (with rotation) behaviour. The tests were performed on an experimental bench with a conventional PID controller implemented in an embedded electronic platform for the control of two magnetic bearings. In the experiment, advantages the AMB system has for the desired behaviour, which is to maintain the rotor position even in the presence of disturbances are presented and discussed.

2. ACTIVE MAGNETIC BEARING MODELING

AMBs use electromagnetic force to levitate a shaft and keep it in its reference position, which is possible by actively controlling the current that flows through the electromagnets, avoiding contact between the mechanical parts. Thus, a magnetic field strength can act in a controlled manner is necessary . To understand how these forces are generated in a magnetic bearing, the generation of forces between an electromagnet and a ferromagnetic material is considered. Using the principle of virtual work, the determination of magnetic forces f can be calculated from Eq. (1) (Schweitzer and Maslen, 2009).

$$f = -\frac{\partial U_a}{\partial x} = \frac{B_a^2}{\mu_0} A_a = \frac{\mu_0 A_a n^2}{4} \left(\frac{i}{x}\right)^2 = k \left(\frac{i}{x}\right)^2. \tag{1}$$

where U_a, x, B_a, μ_0, A_a, n, k and i are the energy stored in the volume of the air gap, air gap, magnetic flux density, magnetic permeability of a vacuum, cross-section area of the air gap, the number of turns, constant and current, respectively.

In Equation (1), the force produced by the electromagnet is proportional to the square of the current in the coil i and inversely proportional to the square of the air gap. We can also verify they are non-linear. Thus, this relationship can be linearized around an operating point for the air gap (x_0) and the bias current (i_0).

Electromagnetic forces are only of attraction. In order to produce forces along with two opposite directions, actuators are normally arranged in pairs (Fig. 1). This allows full control of the rotor in one direction. For a pair of magnets, the magnetic force f_x represents the difference between the forces in the positive and negative direction, that is, the resultant force in the x direction. For this case, the actuator currents are defined as the sum of the bias current (i_0) and a control current (i_x) (current variation) for the positive actuator and the difference $i_0 - i_x$ for the negative actuator. The air spaces are defined by the position deviation x and the nominal stationary air space (x_0), so the terms $x_0 - x$ and $x_0 + x$ are inserted. Thus, Eq. (2) is obtained:

$$f_x = f_+ - f_- = k \left(\frac{(i_0 + i_x)^2}{(x_0 - x)^2} - \frac{(i_0 - i_x)^2}{(x_0 + x)^2} \right). \tag{2}$$

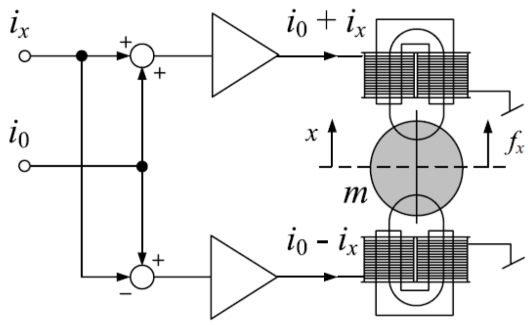


Figure 1: Differential operation of an active magnetic bearing (Schweitzer and Maslen, 2009).

Considering the system works in ranges close to an operating point, where the position variation is small, Eq. (2) can be linearized and the expression for the force of the AMB can be written according to Eq. (3):

$$f_x = k \frac{i_0}{x_0^2} i_x + k \frac{i_0^2}{x_0^3} x. \tag{3}$$

or in a compact form:

$$f_x = k_i i_x + k_s x. \tag{4}$$

k_i coefficient is the current stiffness, being the actuator gain, that is, variation due to the change in the winding current. k_s coefficient is the position stiffness or stiffness of the open-loop system, corresponding to the variation in the rotor position, it has a negative value for most configurations, which leads to destabilizing the magnetic bearing. When the rotor moves, the force increases by pulling it in the same direction as its movement. Choice of the bias current i_0 and the value of the air gap at the equilibrium position x_0, determine the parameters k_i and k_s of the resulting magnetic force. Similarly, it can be performed for a new electromagnetic pair in the other direction to generate positioning forces.

3. CONTROL SYSTEM

AMBs are devices require a closed-loop control system to ensure rotor stability and levitation (Stephan *et al.*, 2013). Its application in magnetic bearings is wide, we can mention some works related to the use of the PID controller in Sabirin *et al.* (2007), Ritonja *et al.* (2010), Polajžer *et al.* (2006), Yoon *et al.* (2013) and Shata *et al.* (2018). The magnetic bearing closed-loop control system consists of a system in which the sensor signal is measured and compared to a reference value, thus generating an error signal. This signal will be sent to the controller that will process and transfer the output signal to the plant of the system to be controlled so that the desired behaviour is achieved, that is, the stabilization of the rotor position.

3.1 PID Control

This work implements the independent PID controller for each degree of freedom (DOF). In this case, the system has four, two for the rotor position in a bearings and two in the other. The DOF related to the shaft rotation is not considered because is imposed by the motor. For simplicity, only the horizontal and vertical position is modelled, disregarding the axial movement.

The PID controller is a widely used control algorithm due to its simplicity, easy implementation and good performance. Many control solutions have been used over time, but the PID controller is the most used in the industry today (Borase *et al.*, 2020). The controller compares the y process value measured by proximity sensors with a y_0 reference value. The difference or error is then processed to calculate the control signal u. This signal will try bringing the measured value to the desired reference value. In this way, the controller is used to minimize the error between the measurement and the desired reference value to obtain a stable system behaviour and adequate dynamics for the working conditions. The control signal generated by the controller is described by the following Eq. (5) in the time domain.

$$u(t) = K_p \left(e(t) + \tfrac{1}{T_i} \int_0^t e(t')dt' + T_d \tfrac{de(t)}{dt} \right) \tag{5}$$

where $u(t)$ is the output signal, $e(t)$ is the difference or error between the reference signal y_0 and the system output $y(t)$, K_p the proportional gain, T_i the integral time, and T_d is the derivative time.

Equation (6) presents the PID equation in the Laplace transform domain, where the output signal is $U(s)$ and $Y(s)$ the input signal, where it is simpler to design controllers.

$$U(s) = K_p \left(1 + \frac{1}{T_i s} + T_d s \right) Y(s) \tag{6}$$

Good knowledge of the system characteristics and how the different control actions in the magnetic bearing work are essential for the controller to achieve the desired performance and for analyzing the dynamic behavior of the rotor. Figure 2 shows the dynamic behaviour of the motion control. In this simulation, with parameter values, $P = 3200$, $D = 800$ and $I = 10$, the proper behavior of the rotor is observed. The trajectories performed by the rotor start at the point of greatest displacement until reaching the origin of the coordinate system used in the simulation.

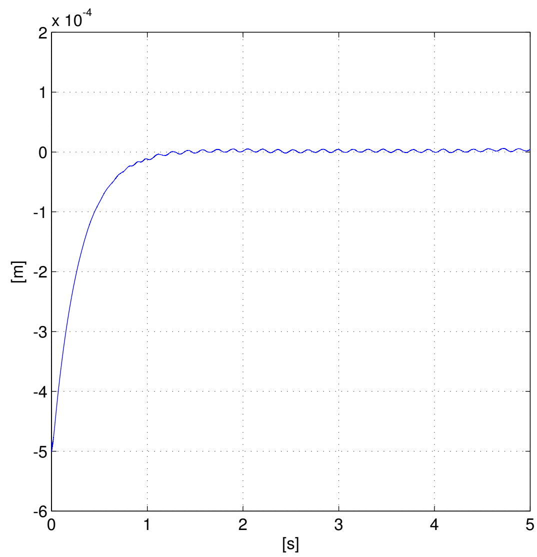


Figure 2: System dynamic behaviour for position variation in the vertical direction.

The transient response depends on the initial conditions of the system, in this case, the initial position is $-0,5mm$ on the vertical axis. Some data can be identified to analyze the step response, including the rise time and maximum over-signal. Can be observed the time it takes the system to reach a permanent response, the so-called rise time, is about $1,5s$.

3.2 Digital Implementation of Controllers

Digital implementation of the PID algorithm used to control the rotor position was performed using the function block developed by National Instrument based on FPGA. This block implements a single-precision floating-point PID controller (32 *bits*) and an anti-windup integrator calculation to limit the effect of the integral action during transients, also PI or PD controllers can be used. The control system is implemented with four cycles acting independently. Each cycle controls a rotor DOF, all operating independently in parallel, that is, each cycle runs on dedicated hardware. A discrete PID controller will read the position, calculate the error and control signal at a given sampling period T. This period should be less than the system's response time. For the PID controller in discrete form, the differential equation (Eq. 5) is transformed into a difference equation using 1st order approximation of the integral and derivative terms (Astrom and Wittenmark, 1996).

The proportional term, which does not need to be approximated, is given by

$$P(k) = k_p e(k) \tag{7}$$

The integral term is discretized using Forward Euler:

$$I(k+1) = I(k) + \frac{k_p T}{T_i} e(k) \tag{8}$$

The derivative term is discretized using Backward Euler:

$$D(k) = k_p \frac{T_d}{T} \big(e(k) - e(k-1)\big) \tag{9}$$

where k is discrete-time, e is error, T is the sampling period, Kp is the proportional gain, Ti is the integration time, and Td the derivation time.

Defining $K_i = \frac{k_p T}{T_i}$ and $K_d = k_p \frac{T_d}{T}$, and making the necessary substitutions, we have

$$u(k) = P(k) + I(k) + D(k) \tag{10}$$

The controller was implemented in FPGA with a sampling frequency of 10 kHz $(100\mu s)$ which can be changed through the user interface. This value was chosen based on the guidance of Schweitzer and Maslen (2009), who recommends a range between 5 to 10 kHz for standard magnetic bearing systems with a digital control implementation and also by ISO-14839-4 (2012), where it shows this frequency is sufficient to control most rotors.

4. DESCRIPTION OF THE AMB SYSTEM

The bench consists of the shaft/rotor supported by two radial AMBs, a polypropylene plate to limit the radial movement of the rotor, also as protection of the stator or rotor surface due to eventual failures, and axial support to prevent movement in the longitudinal direction. The AMB system is composed of position sensors, power amplifiers, an embedded controller, responsible for signal processing, and a drive system through an AC motor, whose speed control is done through a PWM signal and measured with the aid of a digital tachometer. These elements can be identified in Fig. 3.

The bearing has a nominal clearance of 1.0 mm radial air gap with the rotor at the operating point. In this way, the rotor can move radially in a range of 0 to 2.00 mm. However, the maximum movement clearance is given by the support bearing (also called auxiliary or backup or retainer or touchdown bearings), whose function is to avoid contact between the rotor and the bearing in case of malfunction. In this case, the clearance between the rotor and support is 0.5 mm, therefore, the radial displacement region of the rotor starting from the operating point is 0 to 1.0 mm. It is verified that the sensor has sufficient reading amplitude to register all the movement according to its reading direction. During the installation of the sensors, a displacement of 1.1 mm between the sensor and the rotor from the point of the operation was considered. Thus, the operating range of the sensors varies from 0.5 to 1.6 mm. Table 1 presents the main constructive parameters of the AMB system.

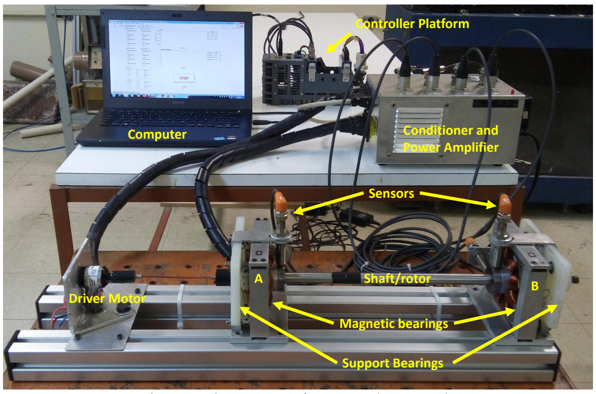


Figure 3: Experimental Bench.

Table 1: AMB System Parameters.

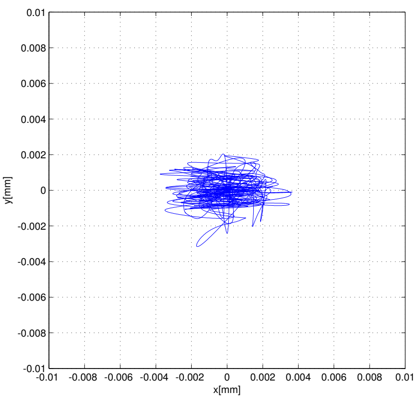
Parameter	Value	Unit
Number of coil turns per pair	260	–
Resistance of each coil pair	$1,2$	Ω
Base current in the coils	$1,0$	A
Rotor/Bearing Clearance (gap)	$1,0$	mm
Rotor Clearance/Support	$0,5$	mm
Shaft length	400	mm
Shaft/rotor assembly mass	$1,0$	kg
Distance between bearings	315	mm

5. EXPERIMENTAL RESULTS

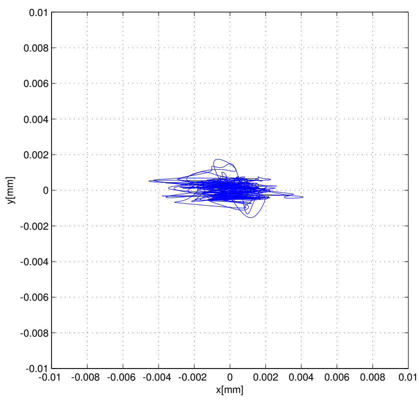
This section presents and discuss the experimental results obtained in the prototype of the AMB system. Tests were performed with different situations for the system, evaluating the following response characteristics: static behaviour (without rotation), transient and dynamic (rotation).

5.1 System response in steady state

The dynamic behaviour of the system for an operation with zero rotation speed, that is, for the static system, will be presented. In this way, the maximum displacement of the shaft about the central position of the bearings will be observed. Figures 4a and 4b shows orbits to AMBs. The rotor remains for most of the time in a range of $0,004mm$ about the zero points, but it is not completely static. This was expected since the system has a decentralized control structure, that is, a controller for each bearing direction, acting independently to keep the rotor position stable. However, the vibration produced by vertical control also influences the horizontal direction. Even with these small displacements, the system presents a satisfactory response, indicating an operation with an adequate performance of the control system.



(a) AMB A.



(b) AMB B.

Figure 4: Rotor orbits (in mm) at radial AMBs.

5.2 System response to impulsive forces

A system disturbance in operation with a dynamic force of 248.4N was applied to the shaft through an impact hammer, in the negative vertical direction at a midpoint of the total shaft length. The purpose of applying this type of external disturbance is to analyze the transient response of the magnetic bearing with the controller, which can be destabilized by making contact between the rotor and the bearings. Note in Figure 5 that after the disturbance, the controller stabilized the system quickly, taking about 0.12s (settling time). Figure 6 shows the movement of the rotor during the applied disturbance, keeping it within the limit.

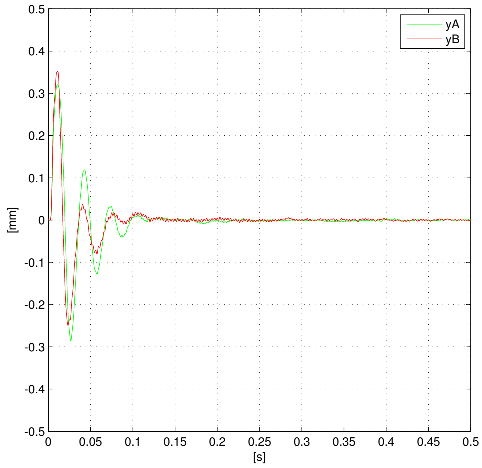
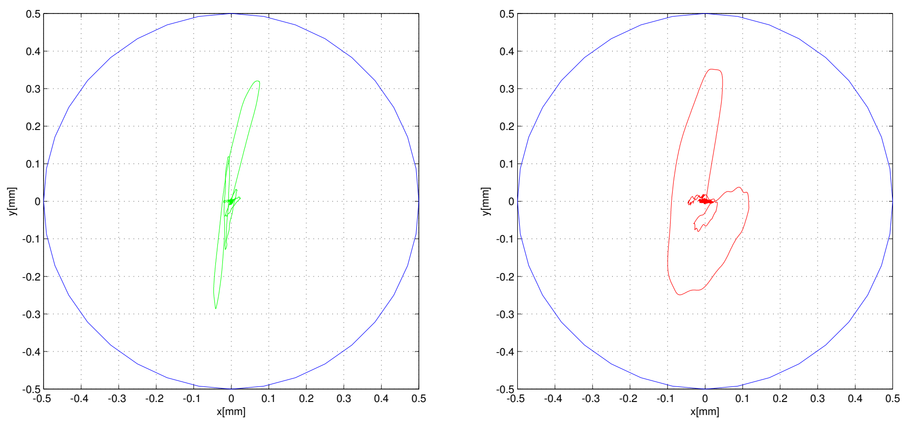


Figure 5: Response to impulsive force of AMBs.



(a) AMB A.
(b) AMB B.

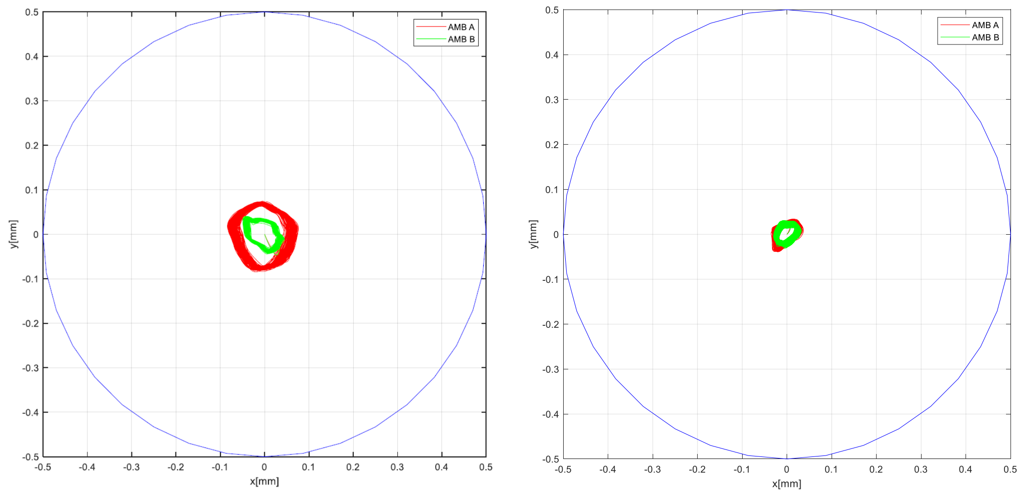
Figure 6: Rotor orbits (in mm) at radial AMBs to impulsive force.

5.3 Rotating system response

The analysis of the dynamic behaviour of the system was carried out with the rotor rotating at a rotation speed of 2000 RPM. The system is driven by a DC motor coupled at the shaft end close to bearing A. The values in the time and orbit graphs indicate the maximum and minimum limits given by the bearing support structure with $0.5mm$ radial clearance. Before actuation, the rotor remains supported on the support bearings with the magnetic bearing control system turned off. Then, the system is turned on and the rotor is brought to zero position stably without rotating. Then, the system is driven with the DC motor creating a constant rotation speed. In this way, wear due to friction during contact between the rotor and the support bearing is avoided.

Figure 7 shows orbit at AMBs for spin speeds of 2000 and 4000 RPM. An increase in amplitude is noted concerning the case of the static system. These results show the control implemented in the system was satisfactory, keeping the rotor within the limit of the support bearing $(0.5mm)$. Figure 7b for a spin speed of 4000 RPM, it was observed there is a decrease in the vibration amplitude of the rotor in the bearing in relation to the system with a rotation speed of 2000 RPM

(Fig. 7a). As the system always presents a residual unbalance, with an increase in the spin speed the amplitude change. In this case, it presents a decrease in position amplitude due to the increase in stiffness related to the increase in rotation frequency.



(a) 2000 RPM. (b) 4000 RPM.

Figure 7: Rotor orbits (in mm) at radial AMB locations at different spin speeds.

6. CONCLUSIONS

In this work, a PID control structure has been presented to perform rotor stabilization. Four independent PID controllers have been used for each direction, two for each bearing. An advantage of using such a structure is that its parameters (stiffness and damping) can be adjusted to the desired behaviour. The controller implementation was performed on the National Instrument CompactRio platform through the LABVIEW software in a reprogrammable device (FPGA) with parallel execution in real-time. It was verified this controller presented satisfactory results in the bearings with the device used.

The experimental results showed a stable magnetic bearing system is obtained with the PID controller for rotor levitation, responding satisfactorily to the disturbances imposed during the system operation and to the drive-through a DC motor, even at rotational speeds. Furthermore, it was verified low oscillation amplitudes were reached for the two bearings, being the same within the limits of the support bearing, as seen in the graph of the orbits for the rotor movement. Thus, the objective of obtaining stable rotor levitation was achieved, with low vibration amplitudes and within the established limit.

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