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Rotating Machinery Vibration Control Based on Poles Allocation using Magnetic Actuators

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1. Abstract
The recent years have seen the appearance of innovative systems for acoustic and vibration attenuation, most of them integrating new actuator technologies. In this sense, the study of algorithms for active vibrations control in rotating machinery became an area of enormous interest, mainly due to the countless demands of an optimal performance of mechanical systems in aircraft, aerospace and automotive structures. Also, many critical machines such as compressors, pumps and gas turbines continue to be used beyond their expected service life despite the associated potential for failure due to damage accumulation. In this way, this paper presents an developed approach for active vibration control in a rotor using Active Magnetic Bearings (AMB) that is numerically verified. The feedback technique is used for this controlling device and the controller gain is obtained by the pole allocation. The AMB uses electromagnetic forces to support a rotor without mechanical contact. It offers many advantages compared to fluid film and rolling element bearings, such as no wear, the ability to operate in high temperature environments, and no contamination of the working fluid due to the absence of lubricant in the system. By monitoring the position of the shaft and changing the dynamics of the system accordingly, the AMB keeps the rotor in a desired position. This unique feature has broadened the applications of AMB and now they can be considered not only as a main support bearing in a machine but also as dampers for vibration control and force actuators.
2. Keywords: Active Control Vibration, Active Magnetic Bearing, Rotating Machinery, Pole Allocation

3. Introduction

In the past, rotordynamic instability problems were not very common. However, in the last few decades, the appearance of rotordynamic instability has increased due to the development of turbomachines rotating at high superficial speeds, having more stages to manage higher pressures, and closer seals clearances for increased efficiency, designed to meet industry challenging requirements [1].
To date, many kinds of instabilities have been identified. These include oil whirl and oil whip due to fluid forces in hydrodynamic bearings. Aerodynamic instabilities caused by the effect of forces due to variations in blade-trip clearances in axial flow compressors were quantified by [2]. [3] described the cases of load dependent instability observed in three centrifugal compressors. One of the compressors presented a small sub-synchronous amplitude when running at 11,000 rpm and when the load was increased, the instability was triggered preventing the compressor to reach the design discharge pressure.
Design specifications suggest that a rotor can operate with acceptable levels of instability as they do with acceptable levels of imbalance. [4] describes and acceptance specification for centrifugal compressors that requires that the average sub-synchronous vibration level be bounded and below 40% of the overall vibration specification.
Mathematical modeling has been performed allowing the development of computer that predict the threshold of instability and often allow the designer to change the rotor-bearing system characteristics to avoid the appearance of instabilities within the operational range of the machine. [5] studied the influence that high-pressure oil seals have on turbo-compressors stability, introducing some design guidelines to avoid the appearance of sub-synchronous vibrations. [4] developed a reliable compressor design taking into account the interaction of labyrinth seals with the rotor and addressed the importance of the design of labyrinth seal clearances to enhance stability. However, limitations associated with models and understanding of other instability mechanisms results in a periodic building of turbomachines with exhibit unstable behavior even though they passed full speed, no-load shop tests before being installed for operation.
The AMB is a feedback mechanism that supports a spinning shaft by levitating it in a magnetic field. Shown in Figure 1 is one quadrant of a radial AMB consisting of a position sensor, a controller, a power amplifier and an electromagnetic actuator. For operation, the sensor measures the position of the shaft and the measured signal is sent to the controller where it is processed and then, the signal is amplified and fed as a current into the coils of the magnet, generating an electromagnetic field that keeps the shaft in a desired position. The strength of magnetic field depends on the air gap between the shaft and the magnet and the dynamics of the system including the design of the controller.

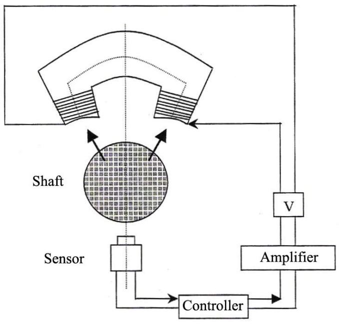

Figure 1. Magnetic levitation principle

Advanced control technology has made possible the use of AMBs to adjust the dynamics (stiffness and damping) of a rotor system, based on constant monitoring of the position of the shaft, this fact has introduced the concept of Active Magnetic Dampers (AMD), which no static load capacity, placed in a machine for vibration control purposes.
Therefore, vibration attenuation is an important goal in many engineering applications, particularly in aerospace industry. Active vibration control (AVC) distributed on structures is of practical interest because of the demanding requirement for stability and good performance. This is of particular importance in light structures, since, they, generally, have low degree of internal damping, [6]. Actually, many researches and text books have been published in this area as, for example, [7]; [8]; [9] and [10].
Although new applications continue to come on-line, two types of rotating machinery have historically represented the majority of equipment outfitted with AMBs on a commercial basis. The first type includes equipment under the category of turbomachinery (with nominal operating speed as high as 60,000 rpm), including centrifugal compressors, turboexpanders, and turbines, among others. The second type of rotating machinery is a turbomolecular pump, such as those used by the semiconductor industry to create ultra high vacuum environments. Additional commercial applications include machine tool spindles, X-ray tubes, neutron choppers, and cryogenic pumps, as can be seem in [11]
The rotordynamic analysis is becoming a previous phase of study to the design, due to the possibility of predicting problems during the operation of the system, as those caused by vibration amplitudes when a rotor, for example, is passing through a critical speed, see [2] and [12]. Therefore, the ability to monitor the structural health of these systems is becoming increasingly important. In this scenario, the application of the Active Magnetic Bearings (AMB) not only as a main support bearing in a machine but also as force actuators has become one of the useful devices in a control scheme for active vibration control in rotating machinery such as suspension systems for shafts or rotors. Several components of an AMB are characterized by nonlinear behavior and therefore the entire system is inherently nonlinear [13]. In addition to their use as support bearings, AMBs can also be used both as force actuators and sensors [11]. For example, in a rotordynamic system Humphris [14] utilized AMBs for both support and as a means to apply perturbation forces to the shaft, monitoring the response for health diagnosis. Likewise, Kasarda et al. [1] proposed a method utilizing AMBs for the non-destructive evaluation of manufacturing processes. Various methods have been developed to Active Vibration Control (AVC). Application of AVC in flexible structures has been increasingly used also to a solution for space structures to achieve the degree of vibration suppression required for precision pointing accuracy and to guarantee the stability. In this work, it is used pole allocation control to attenuate vibration signal in a flexible rotor using AMB and the feedback control technique.

4. AMB Theory concept

The Active Magnetic Bearing (AMB) theory is deeply linked with electromagnetism principles and the study of its theory is the bottom line in developing a magnetic actuator rotor system modeling. In analyzing the behavior of a magnetic actuator, the primary objective is to determine the forces generated by the actuator in response to voltages applied to its coils and motion of the actuated device. Once this analysis is well established, it can be used in the design of actuators, both in that it provides insight to the effects of the various design parameters and in that the analysis can be used to evaluate design choices, [13].
The analysis of the coil/geometry – force relationship and of the electrical properties is generally done using a fairly simple one–dimensional representation of the magnetic structure of the actuator. This approach is referred to as magnetic circuit analysis. It is known that the magnetic flux is generated in each pole of the actuator by a rolling up of N coils with an electrical current passing through it. It is good to highlight that in a magnetic bearing effects like forces lines diffusion and current flight are generally not taken into account in the electromagnetic force equation. Therefore, a geometric correction factor ε can conventionally be used to provide more precise results by taking into consideration these effects. Thus, from the magnetism physics principles, the force equation that the AMB can act in function of its geometric and build parameters can be reached, according to [12] is,

$$F = \varepsilon \frac{\mu_0 N^2 i^2 A_g}{4g^2} \tag{1}$$

where $\mu_0 = 4\pi x 10^{-7} (Hm^{-1})$ is the permeability of the free space (air), g is the gap between the rotor and the stator, Ag is the area face of each pole and the geometric correction factor ε is evaluated as 0.9 for axial bearing and 0.8 for radial bearing. These numbers are due to the electrical current flight effect and are more accentuated in bearings with radial geometry. Once the electromagnetic forces are only of attraction, the actuator must be positioned in both sides and diameterly opposite from the rotor, in a double action scheme, as shown in Figure 2, in a manner that the net force F_N in the bearing plane is given by

$$F_N = F_2 - F_1 = K_i i_p - K_x q_i \tag{2}$$

where K_i is the current stiffness, K_x is the position stiffness, i_p is the control current and q_i is the shaft displacement.

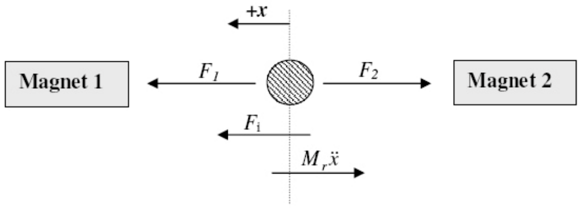

Figure 2. Layout of a controlled shaft using a radial bearing.

In Figure 2, F_1 and F_2 are attraction forces that act in one control axis, F_i is a harmonic external force applied upon the system and M_r is the rotor modal mass.

5. Structural Modeling

Before proceeding to the theoretical development of this work, three assumptions are employed in this paper: (i) an Euler–Bernoulli beam is used as a target structure; (ii) only flexural vibration of a beam is considered; (iii) feedback control is applied. Then, an equation of motion of an Euler–Bernoulli beam lying along the x-axis is written as

$$\rho\frac{\partial^2 v}{\partial t^2}+\frac{\delta^2}{\delta x^2}\left(EI\frac{\delta^2 v}{\delta x^2}\right)=q(x,t) \tag{3}$$

where $v(x,t)$ is the transverse displacement of the beam, ρ is mass density per volume EI is the beam rigidity, $q(x,t)$ is the externally applied pressure loading, and t and x indicate time and spatial axis along the beam axis [15]. We consider shape functions for spatial interpolation of the transverse deflection, v, in terms of nodal variables. To this end, we consider an element which has two nodes, one at each end. The deformation of a beam must have continuous deflection at any two neighboring beam elements. To satisfy this continuity requirement each node has both deflection, v_i and slope θ_i as nodal variables. In this case, any two neighboring beam elements have common deflection and slope at the shared nodal point. This satisfies the continuity of both deflection ands slope. The Euler-Bernoulli beam equation is based on the assumption that the plane normal to the neutral axis before deformation remains normal to the neutral axis after deformation. This assumption denotes $\theta = dv/dx$, i.e. slope is the first derivate of deflection in terms of x.

The main idea of this research work is to show a control formulation at the time domain to attenuate vibration in rotating machinery using active magnetic actuators. The structural dynamic model was obtained by the Finite Element Method (FEM). The disc linked to the shaft is represented by a concentrate mass and the bearings are represented by a damping-spring-mass system, as shown in Figure 3. It is used two magnetic bearings. Next, the magnetic force modeling is presented as well as the strategy used to reach it.

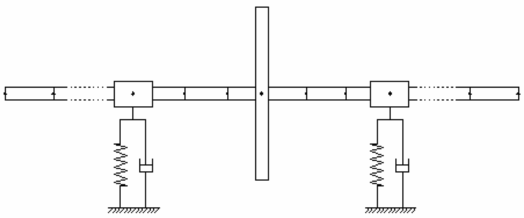

Figure 3. Modeled rotor scheme

The magnetic actuator force in a linearized model is given by the following expression, [16]:

$$F_N=\left(\frac{\varepsilon\mu_0 A_g N^2 i_b}{g_0^2}\right)i_p-\left(\frac{\varepsilon\mu_0 A_g N^2 i_b^2}{g_0^3}\right)q_i \tag{4}$$

where F_N is the magnetic force at the bearing plane, q_i is the shaft displacement where the magnetic force will act, i_p and i_b are, respectively, the disturbance and the permanent (bias) electrical current, g_0 is the gap between the rotor and the stator, μ_0 is the air permeability, N is the number of coils of the magnetic bearing, A_g is the gap area and ε is the geometric correction factor [12]. All those parameters can be seen in the following figure from [13], where the shaft displacement q_i is represented by the variable x.

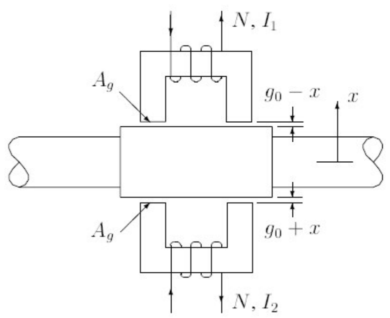

Figure 4. Opposing magnets: a push-pull arrangement.

In order to have a simpler notation, this force can be rewritten as

$$F_N = B_c \mathrm{i}_c - K_q q_i;$$

$$\text{where } B_c = \left(\frac{\varepsilon \mu_0 \mathrm{A_g N^2 i_b}}{\mathrm{g}_0^2} \right) \tag{5}$$

$$\text{and } K_q = \left(\frac{\varepsilon \mu_0 \mathrm{A_g N^2 i_b^2}}{\mathrm{g}_0^3} \right)$$

Since the force F_N will be used as the control force to suppress the vibration signal, the aim is to calculate an electrical current i_p which makes F_N becomes the control force. Therefore, in this work this electrical current is obtained by using the pole allocation technique and them the system gain is calculated. Once it is considered the shaft split in n degrees of freedom and two magnetic actuators though the shaft (one at each shaft end point), there are two magnetic forces $F_{N1} = B_{c1}i_{c1} - K_{q1}q_{i1}$ and $F_{N2} = B_{c2}i_{c2} - K_{q2}q_{i2}$. The shaft displacement differential equation with the influence of the magnetic forces shown in Eq.(4) can be expressed as continuous in time in the matrix form as follow,

$$\ddot{\mathbf{q}}(t) + \mathbf{M}^{-1}\mathbf{D}\dot{\mathbf{q}}(t) + \mathbf{M}^{-1}\mathbf{K}\mathbf{q}(t) = \mathbf{M}^{-1}[\mathbf{B}_c\mathbf{i}_c(t) - \mathbf{K}_q\mathbf{q}(t)] + \mathbf{M}^{-1}\mathbf{B}_w\mathbf{w}(t) \tag{6}$$

where $\mathbf{B}_c = [\mathbf{B}_{c1}\ \mathbf{B}_{c2}]$, $\mathbf{B}_{c1}$ is a vector *n x 1* with the term B_{c1} at the d.o.f. i_1 and zeros at all the another terms. Similarly, $\mathbf{B}_{c2}$ is a *n x 1* vector dimension with the term B_{c2} at the d.o.f. i_2 and zeros at all the another terms. Also $\mathbf{i}_c = \{i_{c1}\ i_{c2}\}^T$, $\mathbf{K}_q$ is a n x n matrix dimension with K_{q1} and K_{q2} at i_1 and i_2 positions and zeros at all another positions. In this way, doing $\mathbf{K}_T = \mathbf{K} + \mathbf{K}_q$ and considering $\mathbf{B}_w$ as the disturbance signal matrix considerate as an harmonic force acting upon the system because the unbalanced rigid disc attach to the rotor, the following equation is obtained.

$$\ddot{\mathbf{q}}(t) + \mathbf{M}^{-1}\mathbf{D}\dot{\mathbf{q}}(t) + \mathbf{M}^{-1}\mathbf{K}_T\mathbf{q}(t) = \mathbf{M}^{-1}\mathbf{B}_c\mathbf{i}_c(t) + \mathbf{M}^{-1}\mathbf{B}_w\mathbf{w}(t) \tag{7}$$

Since the Eq.(7) can be rewritten into the state-space notation, it is possible to reach the control electrical current solving the Diferential Equation. Thus,

$$\dot{\mathbf{x}}(t) = \begin{Bmatrix} \dot{\mathbf{q}}(t) \\ \ddot{\mathbf{q}}(t) \end{Bmatrix} = \begin{bmatrix} \mathbf{0} & \mathbf{I} \\ -\mathbf{M}^{-1}\mathbf{K}_T & -\mathbf{M}^{-1}\mathbf{D}_a \end{bmatrix} \begin{Bmatrix} \mathbf{q}(t) \\ \dot{\mathbf{q}}(t) \end{Bmatrix} + \begin{Bmatrix} \mathbf{0} \\ \mathbf{M}^{-1}\mathbf{B}_{ow} \end{Bmatrix} \mathbf{w}(t) + \begin{Bmatrix} \mathbf{0} \\ \mathbf{M}^{-1}\mathbf{B}_c \end{Bmatrix} \mathbf{i}_c(t) \tag{8}$$

$$\mathbf{y}(t) = \mathbf{C}\mathbf{x}(t)$$

Provided that,

$$\mathbf{A} = \begin{bmatrix} \mathbf{0} & \mathbf{I} \\ -\mathbf{M}^{-1}\mathbf{K}_T & -\mathbf{M}^{-1}\mathbf{D}_a \end{bmatrix}; \quad \mathbf{B}_w = \begin{Bmatrix} \mathbf{0} \\ \mathbf{M}^{-1}\mathbf{B}_{ow} \end{Bmatrix}; \quad \mathbf{B}_i = \begin{Bmatrix} \mathbf{0} \\ \mathbf{M}^{-1}\mathbf{B}_c \end{Bmatrix} \tag{9}$$

the following equation can be reached

$$\begin{aligned} \dot{\mathbf{x}}(t) &= \mathbf{A}\mathbf{x}(t) + \mathbf{B}_w\mathbf{w}(t) + \mathbf{B}_i\mathbf{i}_c(t) \\ \mathbf{y}(t) &= \mathbf{C}\mathbf{x}(t) \end{aligned} \tag{10}$$

6. Poles allocation control

The mathematical modeling was numerically verified using the feedback pole allocation control. As pointed out by [17], the goal of linear feedback control is to place the closed loop poles on the left half of the complex plane of the eigenvalues, so as to ensure asymptotic stability of the closed loop system. One approach consists of prescribing first the closed-loop poles associated with the modes to be controlled and then computing the control gains required to produce these poles. Because this amounts to controlling a system by controlling its modes, this approach is known as modal control. The algorithm for producing the control gains is known as pole allocation, pole assignment, or pole placement.

According to [25], the state equations shown in Eq.(10) can be written in the vector form without the disturbance matrix as

$$\dot{\mathbf{x}}(t) = \mathbf{A}\mathbf{x}(t) + \mathbf{B_i} i_c(t) \tag{11}$$

where $\mathbf{B_i}$ is a constant n-vector and $i_c(t)$ is the control input, in this system given by the control electrical current. From [17], the open-loop eigensolution consists of the eigenvalues $\lambda_1, \lambda_2, \ldots, \lambda_n$ and the right and left eigenvectors $\mathbf{i_{c1}}, \mathbf{i_{c2}}, \ldots, \mathbf{i_{cn}}$ and $\mathbf{v_1}, \mathbf{v_2}, \ldots, \mathbf{v_n}$, respectively. The two sets of eigenvectors are assumed to be normalized, so that they satisfy the biorthonormality relations $\mathbf{v}_j^T \mathbf{i}_{ci} = \partial_{ij}\ (i, j = 1,2, \ldots, n)$.
It is considered control of m modes and assume that the control force has the form

$$i_c(t) = -\sum_{j=1}^{m} g_j \mathbf{v}_j^T \mathbf{x}(t) \tag{12}$$

where the g_j $(j=1,2, \ldots, m)$ are the modal control gains. Inserting Eq.(12) into Eq.(11), it is obtained the closed-loop equation

$$\dot{\mathbf{x}}(t) = C\mathbf{x}(t) \tag{13}$$

where

$$\mathbf{C} = \mathbf{A} - \mathbf{B}_i \sum_{j=1}^{m} g_j \mathbf{v}_j^T \tag{14}$$

Next, it is denoted the closed-loop eigenvalues and right eigenvectors of C associated with the controlled modes by ρ_j and $\mathbf{w}_j$ (j=1, 2, …, m), respectively. But, because the open-loop right eigenvectors are linearly independent, they can be used as a basis for an n-vector space, so that the closed-loop eigenvectors can be expanded in terms o of the open-loop eigenvectors as follows:

$$\mathbf{w}_j = \sum_{k=1}^{n} d_{jk} \mathbf{i}_{ck},\ j = 1.2, \ldots, m \tag{15}$$

Recalling that $\mathbf{v}_j^T \mathbf{i}_{ck} = \partial_{ik}$, the closed-loop eigenvalue problem can be written in the form

$$\mathbf{C}\mathbf{w}_j = (\mathbf{A} - \mathbf{B}_i \sum_{l=1}^{m} g_l \mathbf{v}_l^T) \sum_{k=1}^{n} d_{jk} \mathbf{i}_{ck} = \sum_{k=1}^{n} d_{jk} \lambda_k \mathbf{i}_{ck} - \mathbf{B}_i \sum_{l=1}^{m} g_l d_{jl} = \rho_j \mathbf{w}_j = \rho_j \sum_{k=1}^{n} d_{jk} \mathbf{i}_{ck} \quad j = 1,2, \ldots, m \tag{16}$$

Moreover, letting

$$\mathbf{B}_i = \sum_{k=1}^{n} p_k \mathbf{i}_{ck} \tag{17}$$

Eq.(15) become

$$\sum_{k=1}^{n} (\rho_j - \lambda_k) d_{jk} \mathbf{i}_{ck} + \sum_{k=1}^{n} p_k \sum_{l=1}^{m} d_{jl} g_l \mathbf{i}_{ck} = \mathbf{0},\ \ j = 1,2, \ldots, m \tag{18}$$

which are equivalent to the n x m scalar equations

$$(\rho_j - \lambda_k)\, d_{jk} + p_k \sum_{l=1}^{m} d_{jl} g_l = 0,\ \ j = 1,2, \ldots, m;\ \ k = 1,2, \ldots, n \tag{19}$$

Also, [25]shown that Eq.(18) has a solution given by

$$\begin{vmatrix} \rho_j - \lambda_1 + p_1 g_1 & p_1 g_2 & \cdots & p_1 g_m \\ p_2 g_1 & \rho_j - \lambda_2 + p_2 g_2 & \cdots & p_2 g_m \\ . & . & & . \\ . & . & & . \\ . & . & & . \\ p_m g_1 & p_m g_2 & \cdots & \rho_i - \lambda_m + p_m g_m \end{vmatrix} = 0,\ \ j = 1,2, \ldots, m \tag{20}$$

From it, accordingly to [25], the following equation can be obtained,

$$g_j = -\frac{\prod_{k=1}^{m} (\rho_k - \lambda_i)}{p_j \prod_{\substack{k=1 \\ k \neq j}}^{m} (\lambda_k - \lambda_i)},\ \ j = 1,2, \ldots, m \tag{21}$$

Clearly, for g_j to exist, we must have $p_j \neq 0$. If any one of the p_j is zero, then the associated mode is not controllable. Finally, the control law is obtained by inserting Eqs.(21) into Eq.(12).
Eq.(21) can be used to compute the gains for control af any arbitrary number of modes, m = 1,2, …, n. If the interest lies in

controlling a single mode, say mode *j*, then the control gain reduces to

$$g_j = -\frac{\rho_j - \lambda_j}{p_j} \tag{22}$$

This assumes that the *j*th mode is controllable, which requires that the right eigenvector $\mathbf{u}_j$ be represented in the vector $\mathbf{B}_i$, Eq.(17).

7. Numerical application

After had developed the approach showed above and had chosen a control technique, the modeled system needed to be validated with a numerical application. Basically, was used a theoretical steal rotor (Elastic or Young Modulus (E) of 209 GN/m^2, density of 7,800Kg/m^3, structural damping ratio (ξ) of 0.001) with a shaft of 0.171 *m* long, 12 *mm* diameter, supported by two active magnetic bearing (AMB) through the shaft, one at each end point and the unbalanced rigid disc with 8 mm thickness and diameter of 0.04 m. Table 1 shows the physical parameters of the AMB:

Table 1. AMB physical parameters

Parameter	Value
ε (geometric correction factor)	0.8
μ_0 (permeability of the free space (air))	$4\pi \times 10^{-7}$ Hm^{-1}
A_g (gap area between the rotor and the stator)	67.558×10^{-6}m^2
N (number of coils)	228
i (saturation electrical current)	3.0 A
i_b (permanent electrical current)	1.5 A
g_0 (gap between the rotor and the stator)	0.381×10^{-3}m
SS (space sensor sensitivity)	2,437 V/m
F (Maximum control force)	53 N
Magnetic Bearing Mass	0.250 kg

Figure 5 shows the frequency response function for the uncontrolled and controlled system for each sensor at each AMB. Out(1) refers to the right AMB and Out(2) to the left AMB.

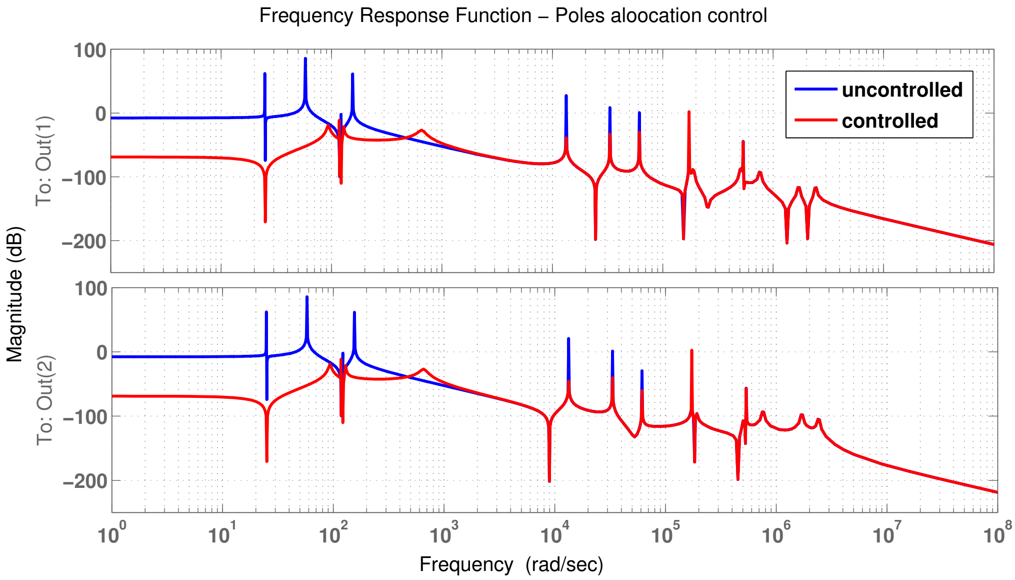

Figure 5. FRF of the system: uncontrolled and controlled behaviors.

By this frequency response function (FRF), it can be seen a large reduction at all amplitudes through the whole frequency range considered for this system. To obtain this control, the poles were allocated based on a previous control for the same system by the LQR optimal control technique solving the Riccati's Equation [18]. Also, the great reduction in the low frequencies range can not be considered as real, once the AMB changes the system stiffness and damping and these dynamic characteristics are only stabilized at frequencies upper than 100 Hz, as shown by [19]. Moreover, in turbomachinery scenario, the speed rotation is about 60,000 rpm when the vibration control needs to be achieved.

Figure 6 showns the response for uncontrolled and controlled system in the time-domain. Again, the output (1) and (2) are related to the fist and second AMB placed through the shaft. By this behavior, the symmetry of the system can be validated by also checking the electrical current behavior at each magnetic actuator, showed in Figure 7.

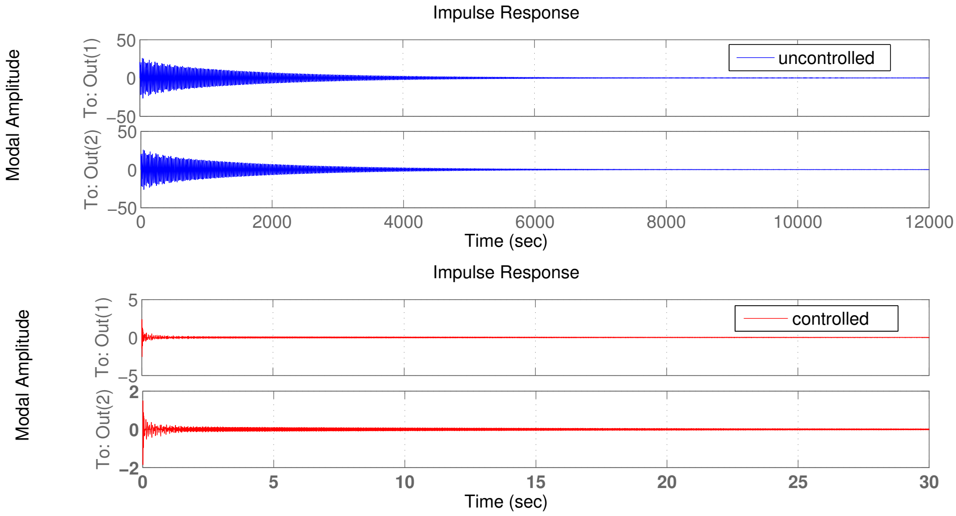

Figure 6. Uncontrolled and controlled impulse response system.

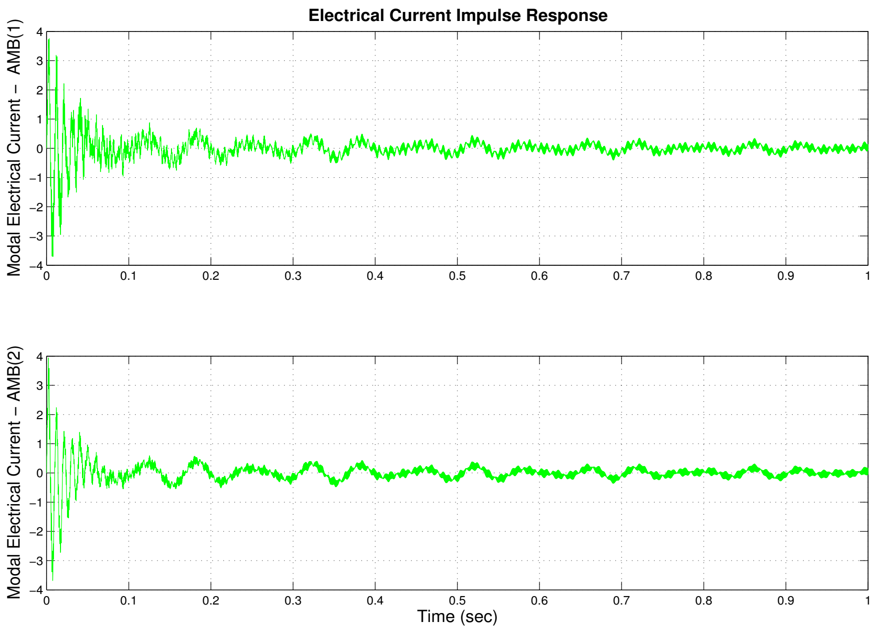

Figure 7. Impulse response, in modal amplitudes.

8. Final remarks

A pole allocation feedback control strategy associated with two AMB was used to actively control the rotor with 14 degrees of freedom. The system was described using the space state realization considering the dynamic equation of second order. By the figures shown before, the aim of validate the modeling proposed was achieved and the system symmetry showed at those results firm the numerical application used to test the system approach presented in this work. In order to follow with this work in the future, it is intended to transform the modal amplitudes in real amplitudes to check if the magnetic bearings selected are able to control this system or until what frequency they are able to control, once the control force depends on the frequency speed and the AMB selected has 53N of maximum control force and 1,5A of control electrical current (Table 1). After this, the following step is to elaborate low order controllers in order to carry out with their experimental verification.

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