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FEEDBACK ACTIVE CONTROL FOR REDUCING ROTOR VIBRATIONS WITH MAGNETIC ACTUATOR

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Abstract: *In this work a theoretical analysis of the range application of an active control system will be developed using a magnetic bearing as no-contact actuator for reducing rotor vibrations. The analysis will be carried out as a function of the flexibility of the rotor. Feedback type architecture of control will be used to continually attenuate synchronous and sub-synchronous vibrations. The synchronous component is due to unbalance mass and the sub-synchronous is due to an instability which often appears in turbo machinery with high speed operation. It is known that the magnetic actuators are limited by the maximum magnetic force and by the maximum current electric (saturation current) and that these parameters can become significantly influenced by the flexibility of the rotors. Therefore, the analysis will be accomplished regarding the performance, the electric current and the control force necessary to be applied to reduce vibration levels of a rotor in extreme rigidity or in extreme flexibility condition. The analysis will be carried out using model of rotor developed by the impedance matrix method.*

Keywords: *feedback control, rotor dynamic, magnetic actuator*

1. INTRODUCTION

Active magnetic bearings are feedback mechanisms that support a spinning shaft by levitating it in a magnetic field. Besides several other applications, they also have the added capability for active vibration control allowing for the reduction of rotor vibrations, Kasarda (2000). A considerable amount of literature exists on control algorithms used for reduction of rotor vibration using magnetic actuator. Shi *et al.* (2004) used a new adaptive vibration control algorithms developed for minimizing selected vibration performance measures by adjusting the amplitude and phase of a synchronous signal injected at the summing junction of the magnetic bearing feedback control loop. Two methods have been investigated, i.e., the filtered-x adaptive filtering techniques and the method that minimize the magnitude of the magnetic bearing system error signal, both for synchronous disturbance attenuation.

An interesting work was published by Piper *et al.* (2005). They present a novel approach to reducing blade-rate noise of axial-flow fans. By using magnetic bearings as a noise control actuator, it was possible to create the anti-noise source for the disturbance noise source. This approach allows for global noise reduction throughout the sound field. Also, it can be mentioned other works that use magnetic bearings as actuator (Jang *et al.* 2005 and Kasarda *et al.* 2004).

There are several interesting publications where the authors use active control systems with magnetic actuator together with systems of fault control or fault diagnosis in turbomachinery or rotor system, for instance (Zhu *et al.,* 2003 and Aenis *et al.* 2002).

The performance of a vibration control system can be different depending of the rotor shaft flexibility. Thus, the contribution of this work is analyzing the performance of an active control system using magnetic actuator to reduce synchronous and sub-synchronous vibration of a rotor considering several shaft diameters simulating different flexibilities. It was analyzed the performance of the actuator respect to its position along of rotor, the electric current of control and the load of control for reducing the vibrations.

2. THEORETICAL FUNDAMENTALS

2.1. Rotor modeling

In this work the rotor model is developed employing two methods together. First, the rotor shaft is modeled as a free-free beam without any discrete mass, stiffness and damping elements connected to it. Only the continuous mass and stiffness of the beam, as well as the structural damping are regarded at this time. The free-free beam model is carried out using the theory that establishes the relation between the external forces applied on the beam and its vibratory motion through the computation of the effect of a finite number of natural frequencies and mode shapes, Johnson *et al*., (2003). In the following step the effects of stiffness and damping of the magnetic actuators, as well as the discrete mass of the disks are added onto the free-free beam model employing the impedance matrix method, Bonello and Brennan (2001). From this whole theory the general equation (1) can be obtained, which calculates the velocity at any point on the rotor due to a force applied at any other points. The detailed procedure of that rotor modeling methodology can be seen in the work published by Nascimento and Hipólito, (2006).

$$\mathbf{u}_i = \hat{\mathbf{T}}_{ij}\mathbf{f} \tag{1}$$

2.2. Feedback control system of magnetic actuators

The magnetic actuators operate using an electronic feedback circuit to provide the necessary electric current to the actuator to control the rotor vibrations at point where it is located. Thus, a global transfer function expressing the relationship between the output control current and the input shaft position must be established, Guiráo and Nascimento (2004). This relationship can be given by the equation,

$$i_c(j\omega) = G(j\omega)x(j\omega) \tag{2}$$

where $i_c(j\omega)$ is the electric current of control, $G(j\omega)$ is the global transfer function obtained from the characteristic of all components of electronic circuit and $x(j\omega)$ is shaft displacement. The transfer function can be expressed as,

$$G(j\omega) = a_G(\omega) + jb_G(\omega) \tag{3}$$

where $a_G(\omega)$ and $b_G(\omega)$ are the real and imaginary part of the transfer function, respectively. From a mathematical model, the dynamic characteristics of a magnetic actuator can be given by,

$$K_{eq}(\omega) = K_x + K_i a_G(\omega) \tag{4}$$

$$C_{eq} = \frac{K_i b_G(\omega)}{\omega} \tag{5}$$

where K_{eq} and C_{eq} are the equivalent stiffness and damping of the magnetic actuator. K_x and K_i are defined as position stiffness and current stiffness, respectively, which are obtained from the geometric and constructive characteristics of the actuator.

3. ROTOR CHARACTERISTICS

The dynamic behavior of a rotor is influenced by its physical characteristics, such as its span, distribution of masses, stiffness of the supports, material of the rotor, geometric characteristics, etc. Considering that the objective of this work is analyzing the application of magnetic actuator to control vibrations in function of the flexibility of the rotors, a system was developed in which the diameter of the shaft was varied to simulate changes in its flexibility, maintaining unaffected the other characteristics of the system.

For this analysis was employed a simplified model of rotor supported by conventional bearings and with an unbalanced rigid disk fastened in it, as showed in Fig. 1. The rotor is made of steel and has 650 mm of length and the shaft has diameter variable to simulate different rotor flexibilities, as indicated in Tab. 1 (from case A to E). The conventional bearings are placed at positions x = 100 mm and x = 600 mm from left to right and the main properties of these bearings are indicated in Tab.2. The actuator assumes variable positions of 10 mm in 10 mm along the shaft to analyze the positions where the actuator reaches larger performance to control the rotor vibrations. The dynamic characteristics of the actuator are presented in Tab. 3 and were obtained using the theory presented in section 2.2, taking a set of parameters for the components of the electronic circuit of control. The rigid disk is placed at position x = 400 mm, has diameter of 150 mm, thickness of 11 mm and mass of approximately 1.5 kg.

The rotor was assumed operating at rotating speed of 6000 rpm, which corresponds to a synchronous frequency of 100 Hz and a sub-synchronous frequency of 50 Hz. The synchronous exciting force was calculated considering the amount of unbalance imposed to the rotor whose value is of 2 x 10^{-4} kg.m. The magnitude of the sub-synchronous exciting force was obtained as 25% of the magnitude of the synchronous exciting force. On the whole band of analyzed frequency was also considered a noise of unitary amplitude acting onto the rotor.

The rotor model was developed regarding 66 nodal points and the vibration in horizontal direction will not be considered and will be assumed to be independent of the vertical vibration. The simulations will be presented for exciting frequencies between 0 to 300 Hz. That frequency range is supposed to contain at least the first three vibration modes of the rotor, which have higher probability to be excited in the practice.

Table 1. Shaft diameters simulating different rotor flexibilities

Case	A	B	C	D	E
Shaft diameter (mm)	5	7.5	10	12.5	15

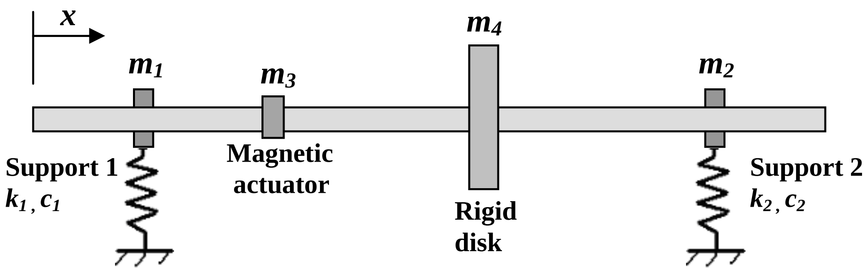

Figure 1. Sketch of rotor for analysis of active control of vibration with magnetic actuator

Table 2. Properties of the conventional bearings

Support mass m_1 and m_2	0.15 kg
Support stiffness k_1 and k_2	5 x 10^5 N/m
Support damping c_1 and c_2	5 N.s/m
Support positions (from left to right)	100 mm and 600 mm

Table 3. Properties of the magnetic actuator

Mass m_3	0.25 kg
Equivalent stiffness	6.5 x 10^3 N/m
Equivalent damping	160 N.s/m
Actuator positions	variable of 10 mm in 10 mm

4. ROTOR DYNAMIC VERSUS ROTOR FLEXIBILITY

A factor which controls critical speeds of the rotors is the shaft flexibility (or shaft diameter). Thus, we must look at this effect as a function of the rotor support stiffness. Also, the rotor flexibility normally modifies the performance of the active system and an analysis must be carried out in this case. Figure 2 shows the mode shapes of the rotor for different cases of shaft flexibility (different shaft diameters). Table 4 shows the three first critical speeds of the rotor for all the analyzed cases corresponding to the modes shapes of the Fig. 2. As it can be seen the first critical speed increases strongly until the case C. After that, only very small variations in this critical speed are observed. Similar tendency also is observed in the second critical speed. On the other hand, the critical speeds of superior order (as the third) keep constant until case B and after that increase continually in other cases. In the cases A and B (smaller diameters), the support stiffness is higher than the shaft stiffness and the first critical speed and mode shape are completely dependent of the geometry of the shaft. So, in those cases we can say that the rotor is in the “shaft dependent” region. In the cases D and E the first critical speed approaches of an asymptotic value and it will only vary as a function of the support stiffness while the third critical speed increases continually. In terms of first and second mode shapes (Fig. 2), the cases D and E indicate a tendency these modes work almost as modes of rigid body. Therefore, we can say the rotor is in the “support dependent” region. Finally, in the case C the system is said to be in “transition” zone since the critical speeds are simultaneously dependent on the shaft geometry and stiffness of the supports.

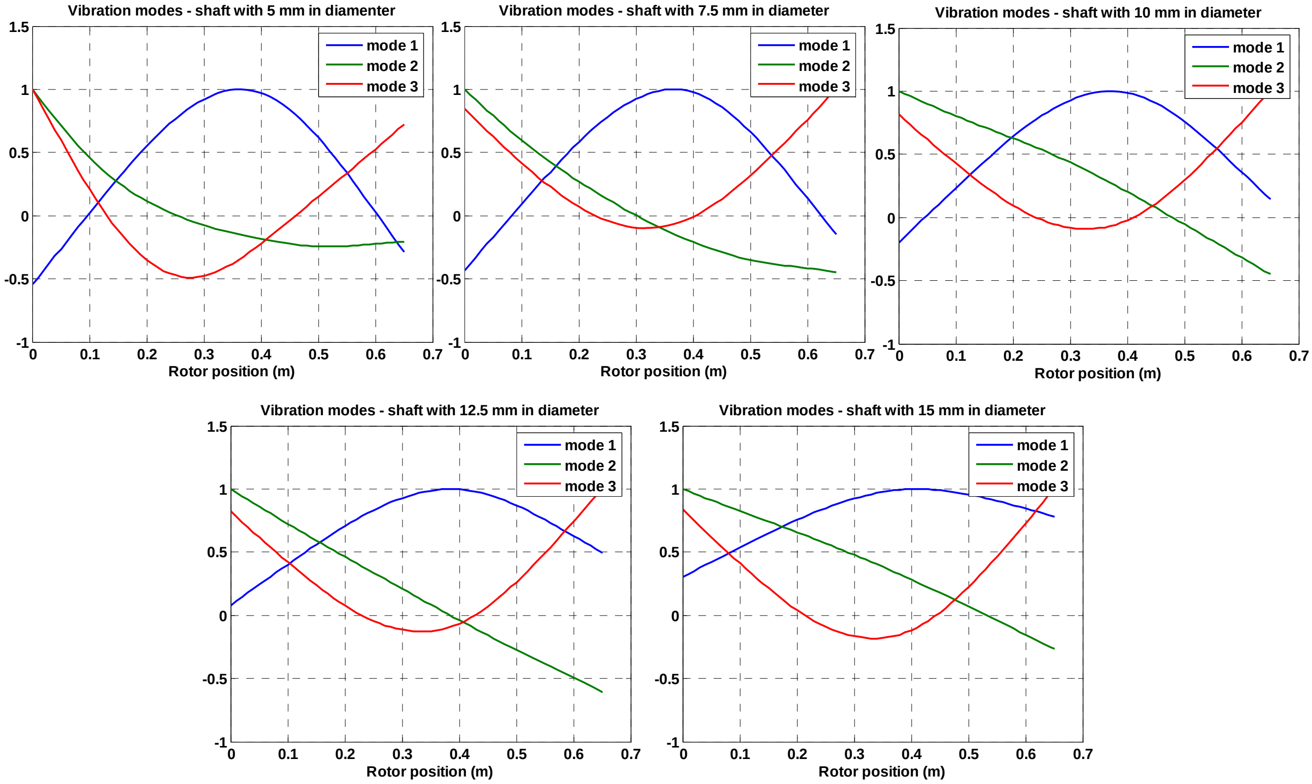

Figure 2. Mode shapes as a function of the rotor flexibility

Table 4. Critical frequencies (Hz) for the cases from A to E.

Mode shape	case A	case B	case C	case D	case E
1st	6	14	21	26	25
2nd	80	71	63	55	49
3rd	87	87	91	102	120

5. VIBRATION CONTROL ANALYSIS

The magnetic actuators possess design parameters that limit their use to control vibrations of rotors. For each use condition it is important to analyze the maximum magnitude of vibration at point of the shaft where the actuator is installed and the maximum values of the electric current and force of control. All those variables cannot overcome the maximum values of design specification. In this work it was used a magnetic actuator with nominal gap (distance between the rotor and stator) of 3.81 x 10^{-4} m. The actuator produces a maximum electromagnetic load of 53 N and admits a maximum current electric of control is 1.5 A. This current corresponds to the saturation current in the coils.

5.1. Optimal position for actuator installation

This section shows the procedure employed to obtain the optimal place to install the magnetic actuator to get the maximum vibration reduction. This analysis is necessary because in general there are several places for the actuator where we have amplification instead reduction of vibration. In this work it was calculated the RMS global vibration reduction placing the actuator at each nodal point of the rotor, for the two deterministic frequencies (synchronous and sub-synchronous) and for all cases of shaft diameter (simulating different rotor flexibilities). The results for the case B (diameter of 7.5 mm) and the case E (diameter of 15 mm) are show in Fig. 3. The regions with negative values indicate that occur amplifications of vibration. As we can see, large amplifications at wide band of frequencies can happen depending on the actuator is installed. For that analysis it was found that the optimal places for best actuator performance are: 410 mm for cases A, B and D, 380 mm for case C and 390 mm for case E. Thus, all subsequent results were obtained with the actuator installed at these optimal positions.

The first point observed in these results is that there is no evolutionary pattern of the reduction (or amplification) curves as the rotor leaves the “shaft dependent” condition to the “support dependent" condition. Thus, each case produces a particular curve of behavior, for the synchronous and the sub-synchronous exciting frequencies. That indicates that is very important to accomplish a previous analysis before adopting the point where the actuator will be set up. Another observed point is that unlike the expected, there are not significant attenuations of the vibration levels with the actuator positioned in the positive regions of the curves (optimal points), as can be observed in Tab. 5, which present the RMS global vibration reduction for synchronous and sub-synchronous frequencies. In addition, regarding that the synchronous and sub-synchronous reduction curves have quite different behavior, the decision in adopting the optimal position for the actuator which satisfies the two curves simultaneously is not simple task. Also, it is important to stand out that the optimal positions for the actuator are very close to the rigid disc, e. i., close to the point where the exciting forces are applied.

Table 5. Vibration reduction for synchronous and sub-synchronous frequencies.

Reduction	case A	case B	case C	case D	case E
Synchronous (%)	24	13.4	21.4	3.1	9.9
Sub-synchronous (%)	14.2	14.8	15.2	16.6	19

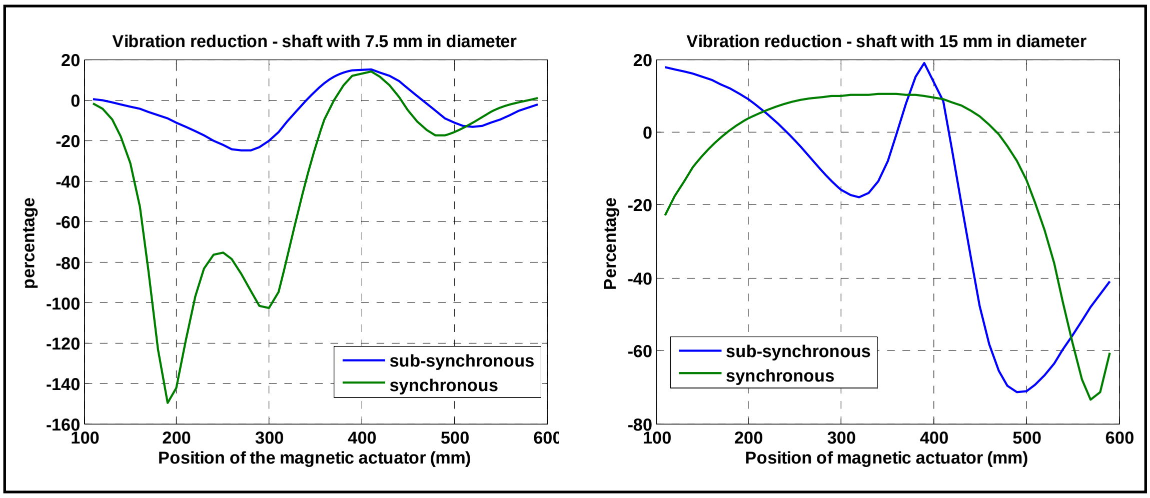

Figure 3. RMS vibration reduction as a function of actuator position along of the rotor (cases B and E)

5.2. Analysis of RMS global vibration

In the previous section the analysis of vibration reduction using the control system was restricted to the frequencies synchronous and sub-synchronous. However, to know the performance of the control system with more details taking the set of frequencies inside of the analyzed range, a more general analysis also was accomplished, in this case, for the actuator in the optimal positions obtained in the previous analysis. Figure 4 shows the RMS vibration levels with and without active control for the specific cases B and E, corresponding to the 7.5 mm and 15 mm rotor shaft diameters.

As already analyzed, the curves of the Fig. 4 show the low reduction of the RMS global vibration amplitude for synchronous and sub-synchronous frequencies (50 Hz and 100 Hz, respectively). However, the control system can significantly attenuate the vibration level of the first critical frequency for all cases, however, it is not effective to attenuate the critical frequencies of superior order.

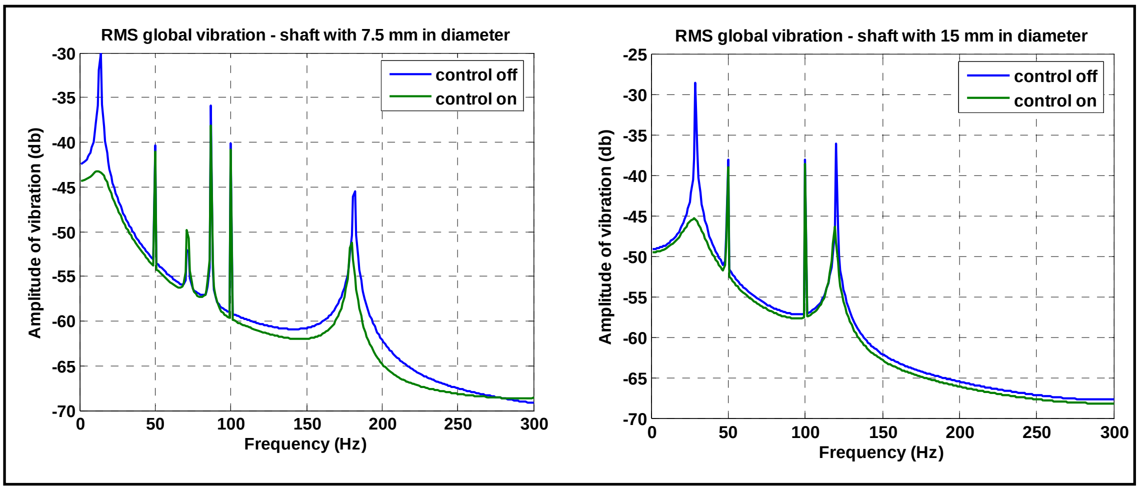

Figure 4. RMS global vibration with and without active control (cases B and E)

5.3. Analysis of local vibration

The analysis of the local vibrations allows observing the real vibratory behavior of the rotor to each point of interest (nodal points) with the magnetic actuator located at a given position and to a certain frequency. In this analysis the reductions of the local vibrations were verified with the actuator at the optimal positions previously obtained. Figures 5, 6 and 7 show the vibrations (displacements) along of rotor with and without control for the synchronous and sub-synchronous frequencies, corresponding to a rotor with shaft of 5 mm, 10 mm and 15 mm, respectively. It is important to stand out that in Fig. 5 the rotor is considered “shaft dependent”, in Fig. 6 the rotor is in “transition zone” and in Fig. 7 the rotor is “support dependent”, according to the flexibility classification.

The results of this analysis show that can happen vibration amplification at some areas along the rotor after application of the active control at synchronous and sub-synchronous frequencies, even if the reductions RMS is positive, as indicated in the Tab. 5. In general the vibration levels close to the support bearings are very low and the performance of the active control practically is not affected by the high flexibility or high stiffness of the rotor.

In all analyzed cases (from case A to case E), the displacement at the position where the actuator is installed didn't overcome 1.3 x 10^{-4} m at both analyzed frequencies, which is a value very inferior to 3.81 x 10^{-4} m, corresponding to the nominal gap between the rotor and the stator of the magnetic actuator. Therefore, there is not risk of happening contact between the mobile and stationary part of the actuator and the control system is applicable with respect to that limitation factor.

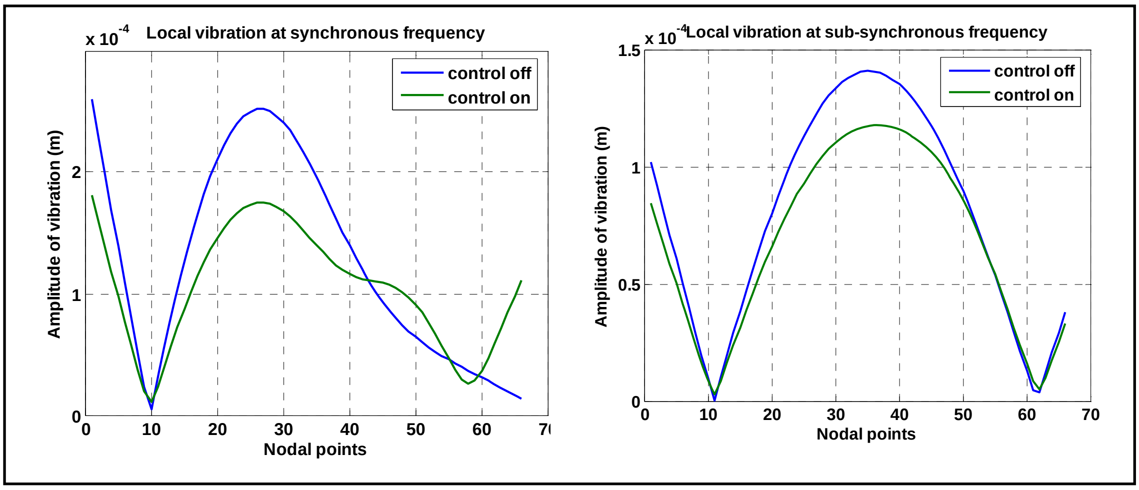

Figure 5. Vibration along of rotor before and after control - shaft with 5 mm in diameter (case A)

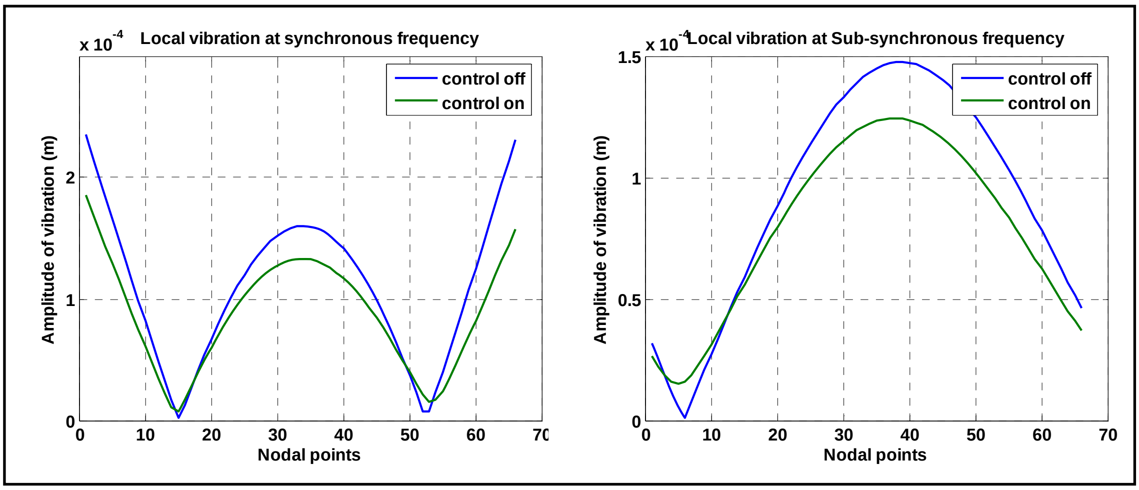

Figure 6. Vibration along of rotor before and after control - shaft with 10 mm in diameter (case E)

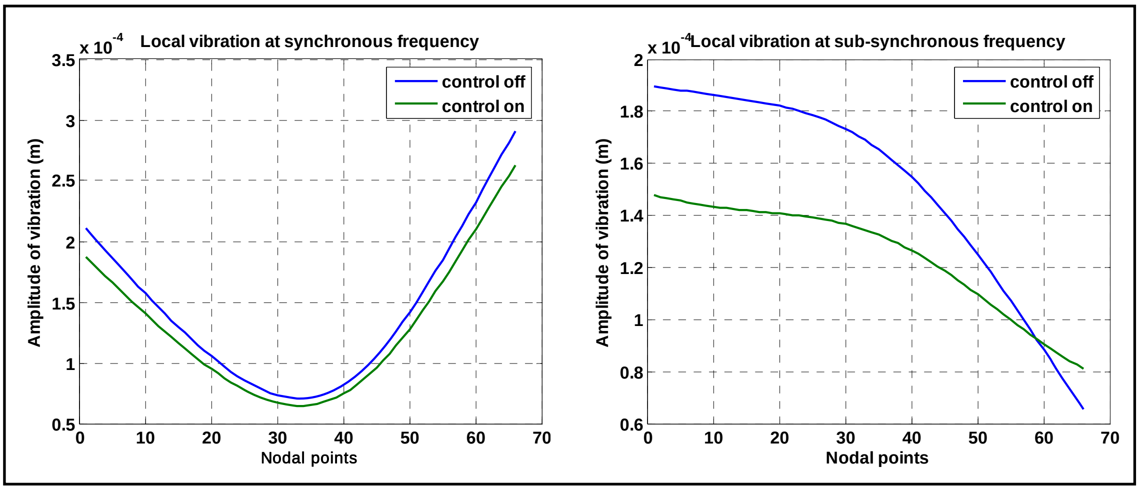

Figure 7. Vibration along of rotor before and after control - shaft with 15 mm in diameter (case E)

5.4. Analysis of electric current of control

Figure 8 presents the curves of control current calculated for the cases of rotor "shaft dependent" and rotor "support dependent" (case A and E, respectively). It can be verified the amplitudes of control current at synchronous and sub-synchronous frequencies clearly outstanding. At other frequencies, the low electric current of control is resulted of noise acting on the rotor in the whole band of analyzed frequencies. Table 6 indicates the exact values of the current of control at synchronous and sub-synchronous frequencies for all cases.

The results presented in the Tab. 6 indicate that the current of control at sub-synchronous frequency is practically independent of the flexibility of the rotor. The current of control of the synchronous exciting frequency demonstrates a tendency to decrease as the rotor becomes more rigid, e. i., when the rotor leaves the zone of "shaft dependent" and goes to the zone of "support dependent".

An interesting fact in these results is that, in general, the control current is relatively low for all the cases, very inferior to the value of the maximum allowed current of project, that is of 1.5 A. So, respect to this limitation parameter the magnetic actuator can be used without the risk of reaching the current of saturation. On the other hand, that fact indicates that the amount of rotor unbalance rotor could be higher that the control system could still operate in satisfactory conditions.

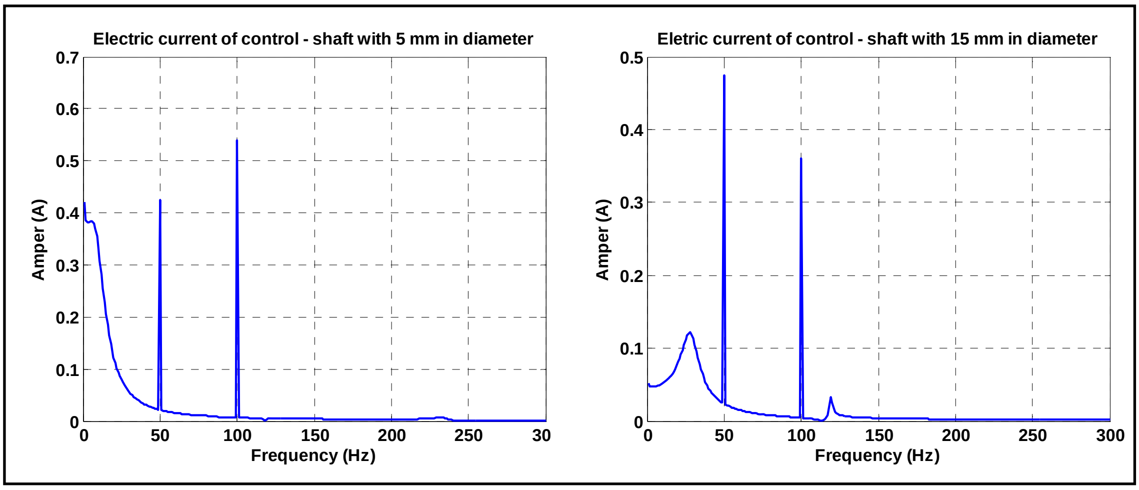

Figure 8. Electric current of control – shaft with 5mm and 15 mm in diameter

Table 6. Electric current of control at deterministic frequencies.

Current (A)	case A	case B	case C	case D	case E
Synchronous	0.54	0.52	0.48	0.16	0.36
Sub-synchronous	0.42	0.43	0.46	0.47	0.47

5.5. Analysis of the control force

The control force is another parameter that can limit the use of the magnetic actuator. Figure 9 shows of control force highlighting mainly the amplitudes at synchronous and sub-synchronous frequencies. Table 7 presents the exact values of the force for all cases. This analysis indicates that magnitude of the control force of the sub-synchronous vibration is practically independent of the flexibility (or stiffness) of the rotor. Similar situation happens for the control force of the synchronous vibration, except when the rotor is in the zone of "support dependent", in which the control force decreases significantly, as it is indicated in Tab. 7.

Respect to this limitation parameter, we ca say that the perfect operating condition for the magnetic actuator is guaranteed, since it allows force until 53 N according to the manufacturer, which is a value higher than obtained in this analysis.

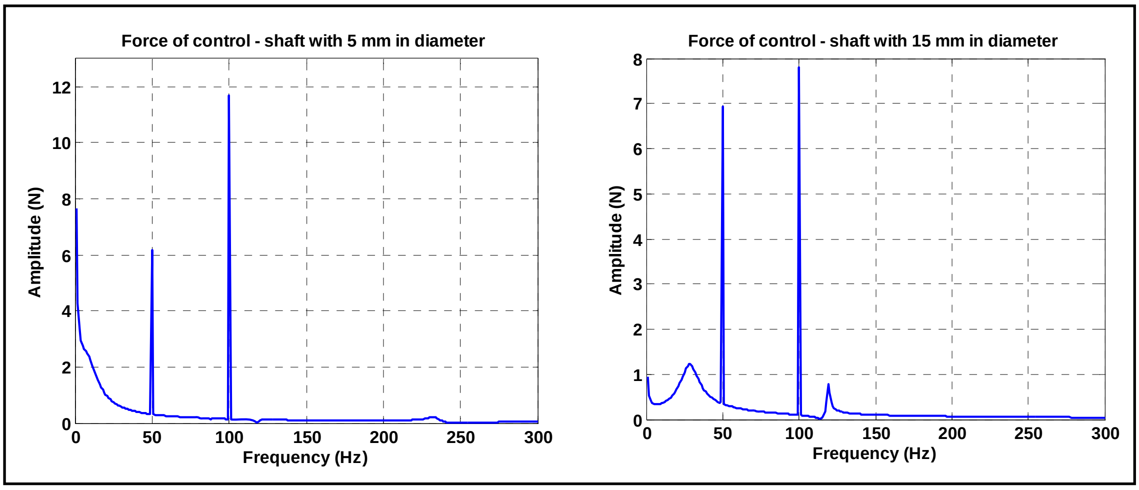

Figure 9. Force of control – shaft with 5 mm and 15 mm in diameter

Table 7. Force of control at deterministic frequencies.

Force (N)	case A	case B	case C	case D	case E
Synchronous	11.7	11.3	12.7	3.4	7.8
Sub-synchronous	6.2	6,3	6.8	7.0	6.9

6. CONCLUSIONS

The analysis of the global vibrations allowed determining the positions for the actuator which present larger performance to reduce the synchronous and sub-synchronous vibrations. This analysis also demonstrates that in all analyzed cases there are regions to install the actuator where the vibrations can be severely amplified, indicating the need of this analysis before installing the control system. A quite different vibratory behavior was also observed for each case of flexibility of the rotor, not happening notable tendency when the system passes from the "shaft dependent" condition to "support dependent" condition.

In general this control system didn't have the waited performance to reduce the synchronous and sub-synchronous vibrations, in accordance to RMS global and local vibration analyses. The local vibration indicates that even the actuator operating at positions of the best performance, it can still occur severe amplifications at some area along the rotor. With respect to the viability of application of this control system, none of the limitation parameters (current of control, forces of control and maximum displacement at position of the actuator) was reached. In general they were very below of the maximum value allowed by the design of the actuator. That indicates that the exciting forces acting onto the rotor could be larger and the control system would be still to be employed without overcome its limits of operation.

7. ACKNOWLEDGEMENTS

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