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APPLICATION OF ACTIVE MAGNETIC ACTUATOR FOR EXCITATION OF ROTATING SYSTEMS SUPPORTED BY TILTING PAD JOURNAL BEARING

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Abstract. *Application of excitation force in rotating system is commonly accomplished in order to identify the parameters of the rotor and bearings. For this reason, the use of Active Magnetic Actuator is a good option since it enables applying forces without contact with the rotor. In this work, the excitation of rotors by using active magnetic actuators is studied for the specific application of a flexible rotor supported by tilting pad bearings. For this purpose a desired magnetic force has to be applied independently on the position of the shaft inside the active magnetic actuator. A PID controller is used during the operation in order to guarantee that the magnetic force applied in the rotating system is the desired force. The excitation force is applied by an Active Magnetic Bearing (AMB) used as an active magnetic actuator. Nonlinear equation for the magnetic force is described in the paper and implemented in a test rig. The method is validated and the different estimations for the magnetic force are experimentally compared. Moreover, the dynamic behavior of the rotating system is verified for different forces, such as sine and constant force.*

Keywords: *active magnetic actuator, rotating system, PID controller, parameters identification.*

1. INTRODUCTION

The experimental identification of parameters is a task commonly used to verify the real dynamic behavior of rotating system. In this situation, the excitation force should be carefully applied in order to avoid noise and uncurtains in the experimental measurements. Differently of the tests accomplished in structures, the application of an accurate force is not a simple task due to the rotor rotation. For this reason, many researches used to apply the excitation force in the bearing house, due to the difficulty to excite the shaft (Cheng-Song and Pei-Yi 1985; Russo, 1999;). Other alternative is to use a shaker to excite the shaft, in which a bush is assembled in the interface shaker-shaft to reduce friction and noise (Okabe, 2007). However, in order to apply an excitation force reducing noise and friction in the shaft, the application of magnetic actuator became a good alternative. The use of magnetic actuators to excite rotating system presents advantages, allowing to apply excitation forces in the shaft with no contact. For this reason, many researches have applied magnetics actuators in studies of rotating machines as nonsynchronous exciters (Kwanka, 1992; Baumann, 1999; Moore *et al.*, 2002; Sorokes *et al.*, 2009; Bidaut *et al.*, 2009; Pettinato *et al.*, 2010; Cloud *et al.*, 2012).

Hoffmann *et al.* (1997) proposed the calculation of the active magnetic bearing (AMB) force by directly using the measured characteristics of the AMB. This characteristic is described by a polynomial. The mathematical description is used in a compensator, which is used to linearize the system behavior.

In 2007, a simple concept of magnetic actuator was developed for application in rotating system (Castro *el al.*, 2007). Differently of the conventional magnetic actuators, this magnetic actuator uses Hall sensor to measure the magnetic field and to control the applied magnetic force. The experiments accomplished in this work showed that the magnetic actuator presents good excitation capacity, obtaining good results to the frequency response function (FRF) of the rotor.

In order to improvement the controller for application of excitation forces, Zingerli and Kolar (2010) proposed an observer based on force control which accounts for different nonlinear effects like eddy currents, magnetization of material core, air gap induced current and stray flux. The experiments show good results regarding the position independent control of magnetic force.

In 2010, Baumann (2010) proposed an experimental identification of dynamic coefficients in cylindrical journal bearing using a test rig composed by magnetic bearings located in the extremities of the rotor and one journal bearing

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located in the midspan of the rotor. In this work, the dynamic coefficients could be determined in different positions inside the bearing due to the position controller of the magnetic bearing, which enables located the shaft before the application of the excitation forces.

Recently, Mendes *et al.* (2012) proposed an analysis of a complete model of rotating machinery excited by magnetic actuator system. In this work, a mathematical model was presented considering the rotor model with the dynamic coefficients of the bearings jointly with the magnetic actuator model and the controller model. The experimental results showed the excitation of forward and backward components of the rotor dynamic response.

In this context, this work aims to present an approach for application of magnetic actuators as nonsynchronous exciters in rotating system with tilting-pad bearings. For this reason, a PID controller is used to control the excitation force applied in the shaft. Firstly, static test was performed to guarantee that the desired force actually is the force applied on the rotor and, consequently, dynamic tests were accomplished in the rotor for different rotation. In the next sections the methodology and block diagram used in this approach are presented and, after that, the experimental results obtained in the static and dynamic tests.

2. METHODOLOGY

The methodology used in this work is divided in three sections: active magnetic actuator, force control and block diagram. In the first section, the equation that describes the behavior of the magnetic force in the actuator magnetic bearing (linear and nonlinear force equations) is presented. Afterwards, the force control used to set the desired force is presented and, finally, the block diagrams used in the complete approach.

2.1 Active Magnetic Actuator

A magnetic bearing is an electrodynamic actuator which is widely used in rotating machinery, avoiding the contact between shaft and bearing surface. Due to the absence of frictional force, this bearing is applied in high speed systems, such as centrifugal and high rotation engines. This magnetic bearing can be used as an external exciter (like a shaker), since a controller is used to guarantee the desired force in the excitation.

In general, there are at least four coils radially positioned about the rotation axis, performing magnetic forces in two directions. The resultant magnetic force obtained through two coils located on opposite sides can be written as:

$$F_{res} = F_+ - F_- = k_{MB}\left(\frac{i_+^{\ 2}}{(s_0+r)^2} - \frac{i_-^{\ 2}}{(s_0-r)^2}\right) \tag{1}$$

Where s_0 is the air gap in the magnetic actuator, i is the current in the coil, r is the displacement (position) of the shaft from the center of the magnetic actuator and k_{MB} is the magnetic actuator constant parameter defined by:

$$k_{MB} = \frac{\mu_0 \cdot A_L \cdot n^2 \cdot \cos(\alpha)}{4} \tag{2}$$

The k_{MB} constant of the magnetic actuator depends on the geometry and material, where μ_0 is the magnetic permeability of the air ($\mu_0 = 4\pi.10^{-7}$ N/A^2), A_L is the coil cross section area, n is the convolution number of the coil and α is the angle of the coil orientation. The symbol + denotes the coil in positive direction of the shaft displacement (positive r) and - denotes the coil in negative direction of the shaft displacement (negative r).

In order to linearize the Eq. (1) and avoid negative currents in the coils, meaning magnetic flux losses, a bias current I_v is added, as written in Eq. (3):

$$i_+ = I_v + i_s \,,\; i_- = I_v - i_s \tag{3}$$

The currents in the coils consist of the bias current and the control current i_s. Thus, the Eq. (1) can be written as:

$$F_{res} = k_{MB}\left(\frac{(I_v+i_s)^2}{(s_0+r)^2} - \frac{(I_v-i_s)^2}{(s_0-r)^2}\right) \tag{4}$$

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This equation neglects the influence of saturation effects of the material core, magnetic hysteresis of the material as well as stray flux (Hoffmann *et al.*, 1997), but is a first approach which inhibits the nonlinear relation to the air gap of the magnetic actuator.

The Eq. (4) can be solved to find the current i_s needed to apply the desired force F_{res}:

$$i_{s1,2} = \frac{I_v\left((s_0-r)^2+(s_0+r)^2\right)}{(s_0-r)^2+(s_0+r)^2} \pm \sqrt{\frac{I_v\left((s_0-r)^2+(s_0+r)^2\right)}{(s_0-r)^2-(s_0+r)^2} - I_v^{\,2} + \frac{(s_0-r)^2(s_0+r)^2}{k_{MB}\left((s_0-r)^2+(s_0+r)^2\right)}F_{res}} \quad (5)$$

Moreover, this equation can be simplified as:

$$i_{s1,2} = \frac{I_v \cdot k_{MB} \cdot s_0^{\,2} + I_v \cdot k_{MB} \cdot r^2 \pm \left(r^2\sqrt{I_v^{\,2}\cdot k_{MB}^{\,2} - F_{res}\cdot k_{MB}\cdot s_0 \cdot r} - s_0^{\,2}\sqrt{I_v^{\,2}\cdot k_{MB}^{\,2} - F_{res}\cdot k_{MB}\cdot s_0\cdot r}\right)}{2\cdot k_{MB}\cdot s_0 \cdot r} \quad (6)$$

Assuming that the value of the bias current is significantly higher than the control current, Eq. (4) can be linearize as written in Eq. (7):

$$F_{res} = \left(k_{MB}\frac{4I_v}{s_0^{\,2}}\right)i_s - \left(k_{MB}\frac{4I_v^{\,2}}{s_0^{\,3}}\right)r \quad (7)$$

The linear force equation is largely used, because it enables to represent the real situation and facilitate the implementation of the controller. However, this equation is not used in this work due to the difficulty to set the controller parameters in order to obtain satisfactory results to the excitation force. Thus, all results presented in this work were obtained from the nonlinear force equation.

2.2 Force control

In many applications a PID control is used to control of the shaft position inside the bearing. This way, the rotor can be supported or located in a specific position (Baumann, 2010). However, this work uses a force control only to guarantee the application of the desired force independently of the shaft position.

The goal of this approach is to estimate the magnetic force by only using the measured displacements of the shaft in the active magnetic actuator. Then, using the current applied in the coils and the displacements of the shaft, the magnetic force can be estimated. Therefore, the force is controlled by a feedback control using a simple PID controller.

Figure 1 shows the PID controller used in the force control.

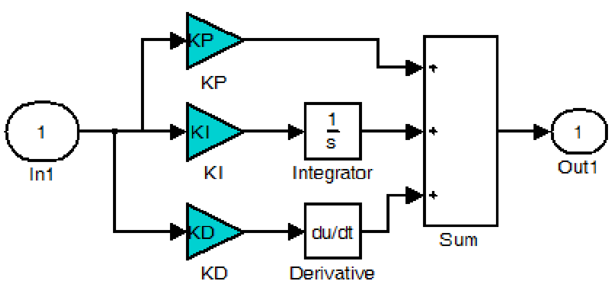

Figure 1. Block diagram used in the force control.

In this work, the parameters used in the force control were adjusted during the experimental tests. Although mentioned that the force control used a PID controller, in fact, the better results were obtained using a derivative parameter nil (K_D=0), leading to a PI controller to be used in the force control. Tab. 1 presents the parameters used in the force control

Table 1. Parameters of the PID controller for the force control.

Parameters	Values
K_P	0.1
K_I	50
K_D	0

2.3 Block diagram of the complete approach

In order to describe the complete approach used for the application of a desired force through an active magnetic actuator, Fig. 2 shows the Simulink block diagram of the system.

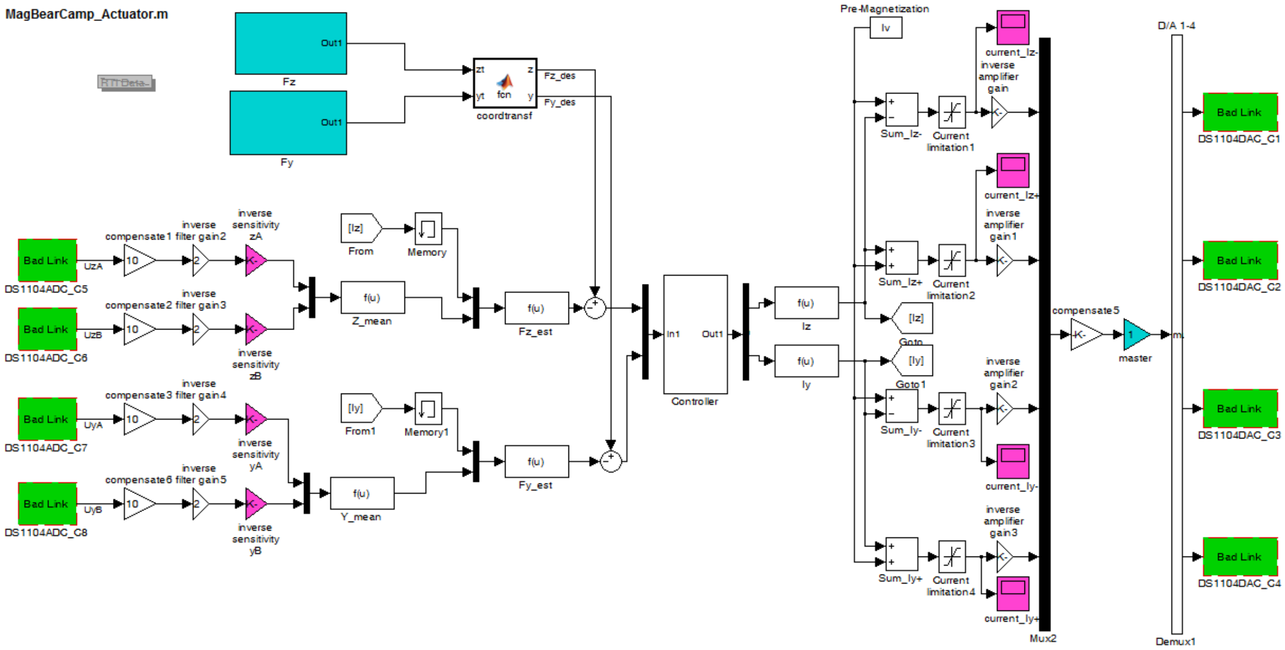

Figure 2. Block diagram used in the complete approach for the application of the desired force.

According to Fig. 2, the difference between the desired force and estimated force is calculated and used in the force control (PID control) to calculate the control current. As previously described, the control current is added to the bias current, resulting in the current to be applied in the coils. From this current and the gain of the amplifier, the voltage to be applied in the active magnetic actuator is determined in order to obtain the desired force.

Applying the magnetic force, the shaft displacement should be measured and used to recalculate the estimated force. This estimated force is also used to recalculate the control current and the iterative process is repeated.

It is important to point out that the output voltage should be multiplied by gain of the DSpace control board, while the input voltage should be multiplied by gains of the DSpace control board, of the analogic filter and of the proximity sensors.

3. EXPERIMENTAL RESULTS

The experimental tests were performed in order to verify the operational behavior of the active magnetic actuator. The active magnetic actuator was developed in the Laboratory of Structure Dynamic of the Technical University Darmstadt. In fact, this active magnetic bearing is normally used to support rotating system. In this case, a position control is necessary because the goal is to guarantee the position of the shaft.

Differently, the experiments performed at UNICAMP apply this bearing as a nonsynchronous exciter with no position control. In the test rig, the rotor is supported by two tilting pad journal bearings and the shaft position inside the bearing depends on the load and the rotational speed of the rotor (equilibrium position or locus of the shaft). The main goal is to apply a desired excitation force in the rotor independently of the shaft position inside the bearing.

Figure 3 shows the active magnetic bearing used in the experimental tests.

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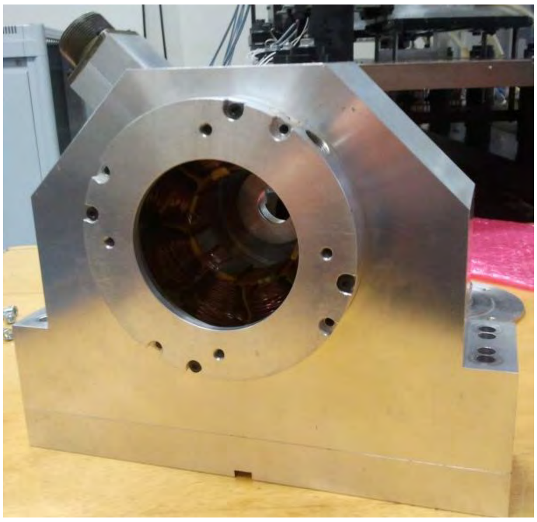
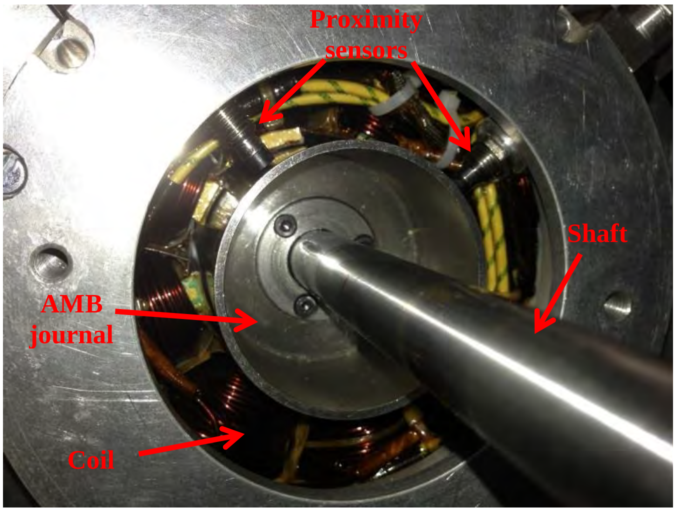


(a)



(b)

Figure 3. Active Magnetic Bearing (AMB): (a) general view of the Active Magnetic Bearing; (b) View of the Active Magnetic Bearing assembled in the test rig.

The geometric parameters and the operational data of the magnetic actuator are given in Tab. 2.

Table 2. Geometric parameters and the operational data of the magnetic actuator.

Parameters	Values
Magnetic permeability of the air (μ_0)	$4\pi.10^{-7}$ [N/A^2]
Coil cross section area (A_L)	0.00029 [m^2]
Number of turns in the coil (n)	208
Angular orientation of the coil (α)	22.5 [degrees]
Air gap (s_0)	0.8 [mm]
Bias current (I_v)	0.75 [A]
Maximum current ($i_{\max}$)	1.5 [A]
Minimum current ($i_{\min}$)	0 [A]

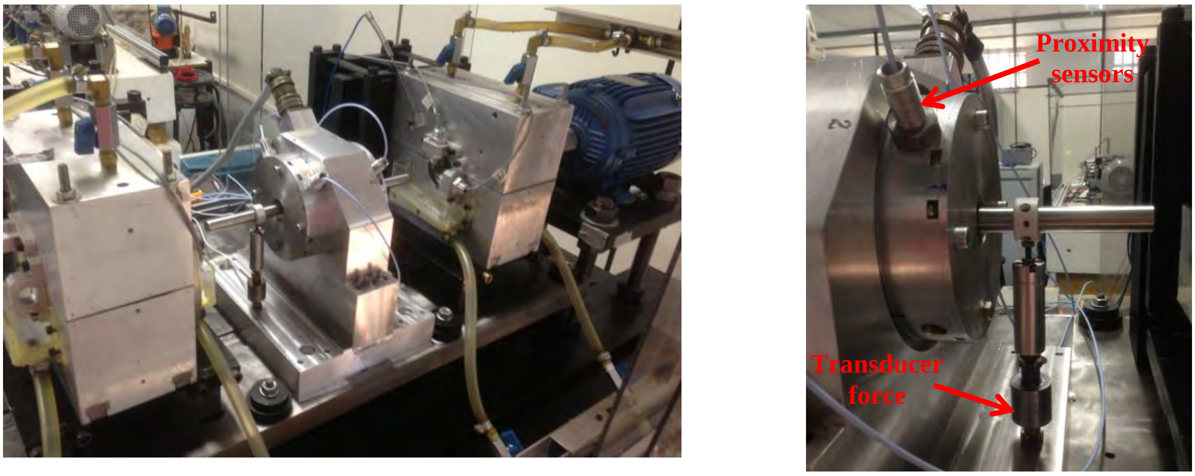
In the next sections, the experimental results obtained in the static and dynamic tests are presented.

3.1 Static tests

Firstly, experimental tests were performed in a stationary shaft supported by transducer forcers, as can be seen in Fig. 4. From the tests, desired forces are applied by magnetic actuator in the shaft and the resultant forces are measured by transducers forces. This way, a comparison between the desired and measured forces can be accomplished, aiming to verify the performance of the magnetic actuator. The geometric parameters of the shaft and the AMB journal (see Fig. 3(b)) are given in Tab. 3.

Table 3. Geometric parameters of the shaft and the AMB journal.

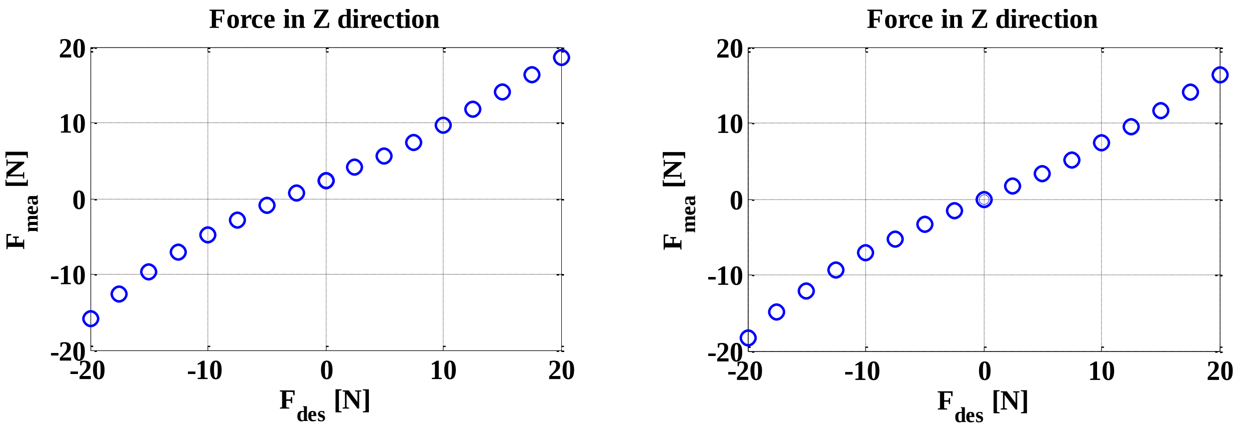
Parameters	Values
Shaft diameter (d_{shaft})	15 [mm]
Shaft length (l_{shaft})	220 [mm]
AMB journal diameter ($d_{AMB\ journal}$)	56 [mm]
AMB journal length ($l_{AMB\ journal}$)	120 [mm]



(a) (b)

Figure 4. Configuration in the static test: (a) general view of the test rig; (b) view of the transducer force assembly.

In this experimental test, desired forces and measured forces were compared. Fig. 5 shows the results obtained for the application of constant forces.



(a) (b)

Figure 5. Constant force applied in the static test: (a) Resultant force considering the residual force; (b) Resultant force removing the residual force.

According to Fig. 5(b), the relation between the measured force and desired force is practically linear, although the measured forces are lower than the desired forces up to the highest value of applied static force (20N). This behavior can be consequence of the saturation of the coil core, in which the actuator is not able to apply the magnitude of the desired force. Moreover, Fig. 5(a) shows that there is a residual force when the desired force is nil. This residual force occurs due to the misalignment of the shaft inside the magnetic actuator. For an ideal case, if the shaft is perfectly centered in the actuator, the resultant force is nil when the bias current is applied. However, despite the careful alignment of the shaft in the actuator, a residual misalignment is still present in the test rig, causing this residual force in the shaft.

After the experimental tests with constant forces, sinusoidal forces were applied in the shaft through the magnetic actuator. Fig. (6) shows the desired force, the estimated force and the measured force when a sinusoidal force is applied.

As can be verified in Fig. (6), all the forces have a good correlation, mainly in sinusoidal force of low frequency. As the frequency increases, higher differences are observed between the measured force and the desired force. This behavior can be explained due to the saturation of the coil core and the controller used in this approach. When the coil core is in saturation condition, the nonlinear effects are predominant in the magnetic actuator and the controller is not able to accurately generate the desired force. Changing in the parameters of the controller can improve the performance in this condition, but affects the performance in other frequency range. Hence, it is important to adjust the parameters of the controller for the application frequency range.

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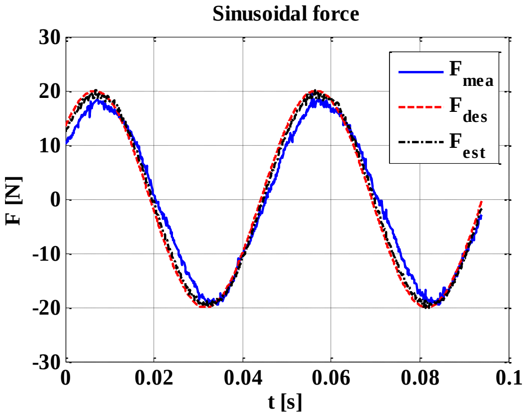
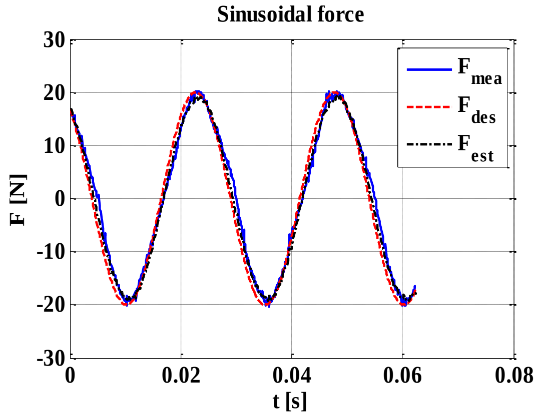
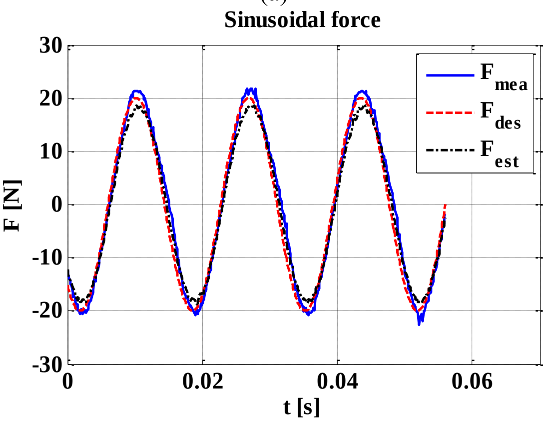


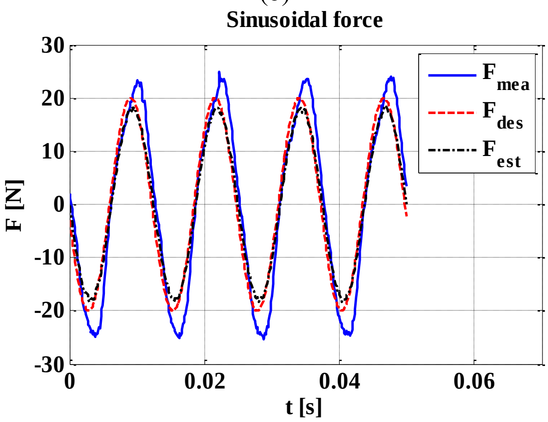
(a)



(b)



(c)

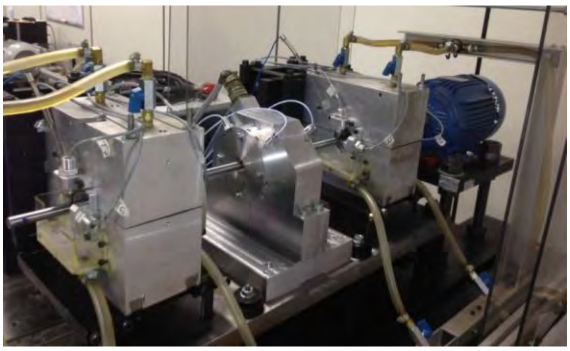


(d)

Figure 6. Sinusoidal force applied in the static test: (a) Frequency of 20Hz; (b) Frequency of 40Hz; (c) Frequency of 60Hz; (d) Frequency of 80Hz.

3.2 Dynamic tests

After the static analysis, experimental tests were accomplished in a non-stationary shaft. In this case, excitation forces were applied in a rotating shaft. The test rig used in the experiments is presented in Fig. 7.


(a)

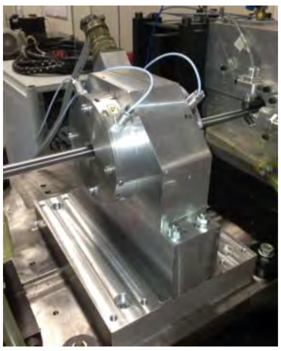

(b)

Figure 7. Configuration in the dynamic test: (a) general view of the test rig; (b) view of the active magnetic bearing (AMB).

From the tests, a response signals evaluation of the shaft is accomplished. Therefore, comparing the response signals of the shaft with and without excitation, it is possible to verify if the magnetic actuator can correctly excite the rotating system. In these tests, the shaft is supported by tilting pad journal bearings and its length is 900 mm. Also important to note is that the distance between bearings is 570 mm and the magnetic actuator is positioned at the central position.

The geometric parameters of the tilting pad bearing are given in Tab. 4.

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Table 4. Geometric parameters of the tilting pad journal bearing.

Parameters	Values
Number of pads	5
Pad angle	63.5 [°]
Angle between pads	8.5 [°]
Pad thickness	13.5 [mm]
Pad width	50.0 [mm]
Bearing configuration	LBP
Pad pivot	0.5
Bearing diameter	50.0 [mm]
Radial clearance	90 [μm]
Preload	0.4
Oil viscosity (ISO VG 32 at 25°C)	0.07 [Pa.s]

Firstly, the dynamic behavior of the shaft is analyzed with no excitation force for a rotational speed of 7 Hz. The displacement signals in this analysis were obtained in the tilting pad journal bearing located in the free extremity of the shaft, trying to avoid the influence of the coupling in the dynamic response. According to Figs. 8(a) and 8(c), the shaft displacement in the Y direction occurs practically around the bearing center, while the shaft displacement in the Z direction occurs around a negative position (below the horizontal axis) due to the load of the shaft. Moreover, it is possible to verify that the harmonic component (unbalance mass) is predominant, although the sub-harmonics components (possible misalignment effects) can be observed in Figs 8(b) and 8(d).

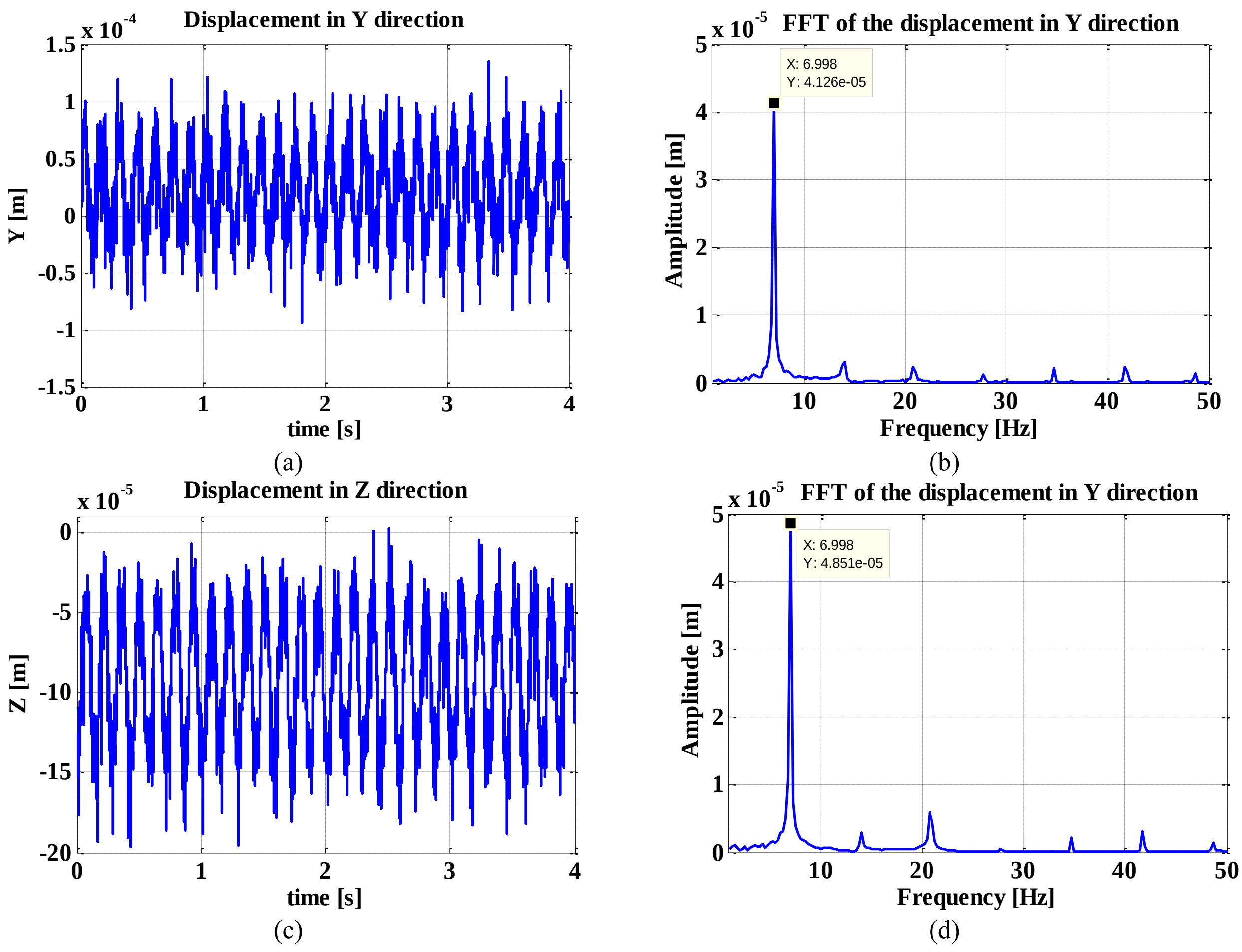

Figure 8. Dynamic test with no excitation force: (a) Displacement of the shaft in Y direction; (b) FFT of the shaft displacement in Y direction; (c) Displacement of the shaft in Z direction; (d) FFT of the shaft displacement in Z direction.

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In order to analyze the dynamic behavior under an excitation force, a sinusoidal force with amplitude of 10N and frequency of 20Hz is applied in the shaft in Y direction, as shown in Fig. 9.

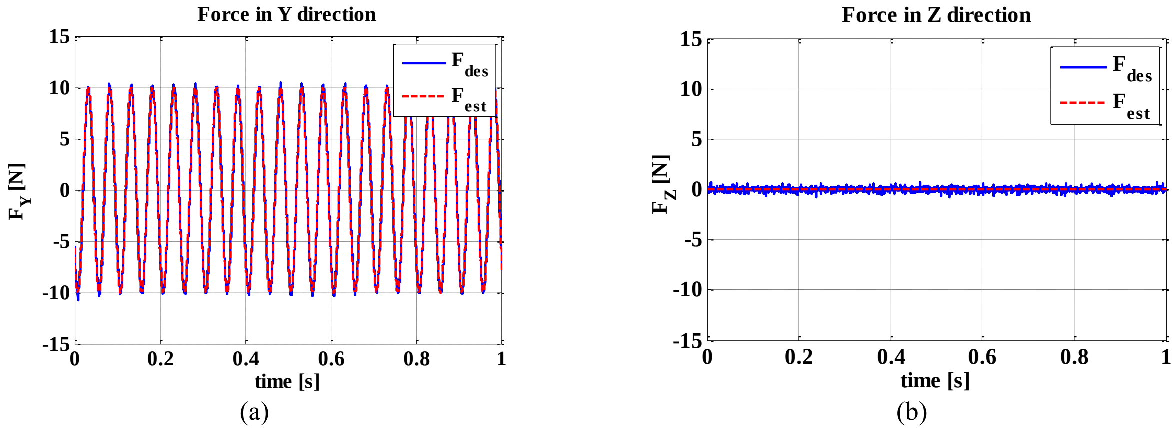

Figure 9. Sinusoidal force of applied in the dynamic test: (a) Amplitude of 10N and frequency of 20Hz in Y direction; (b) force in Z direction.

The shaft displacements in time domain and frequency domain are presented in Fig. 10. As can be verified, the shaft displacement increases in Y direction due to the excitation force, while in Z direction it presents a small decreasing. Moreover, it is possible to observe that the effects of the excitation force occur only in Y direction, since there is no significant component of 20Hz in Z direction. The component of the excitation force (20Hz) is clearly observed in the shaft displacement in Y direction.

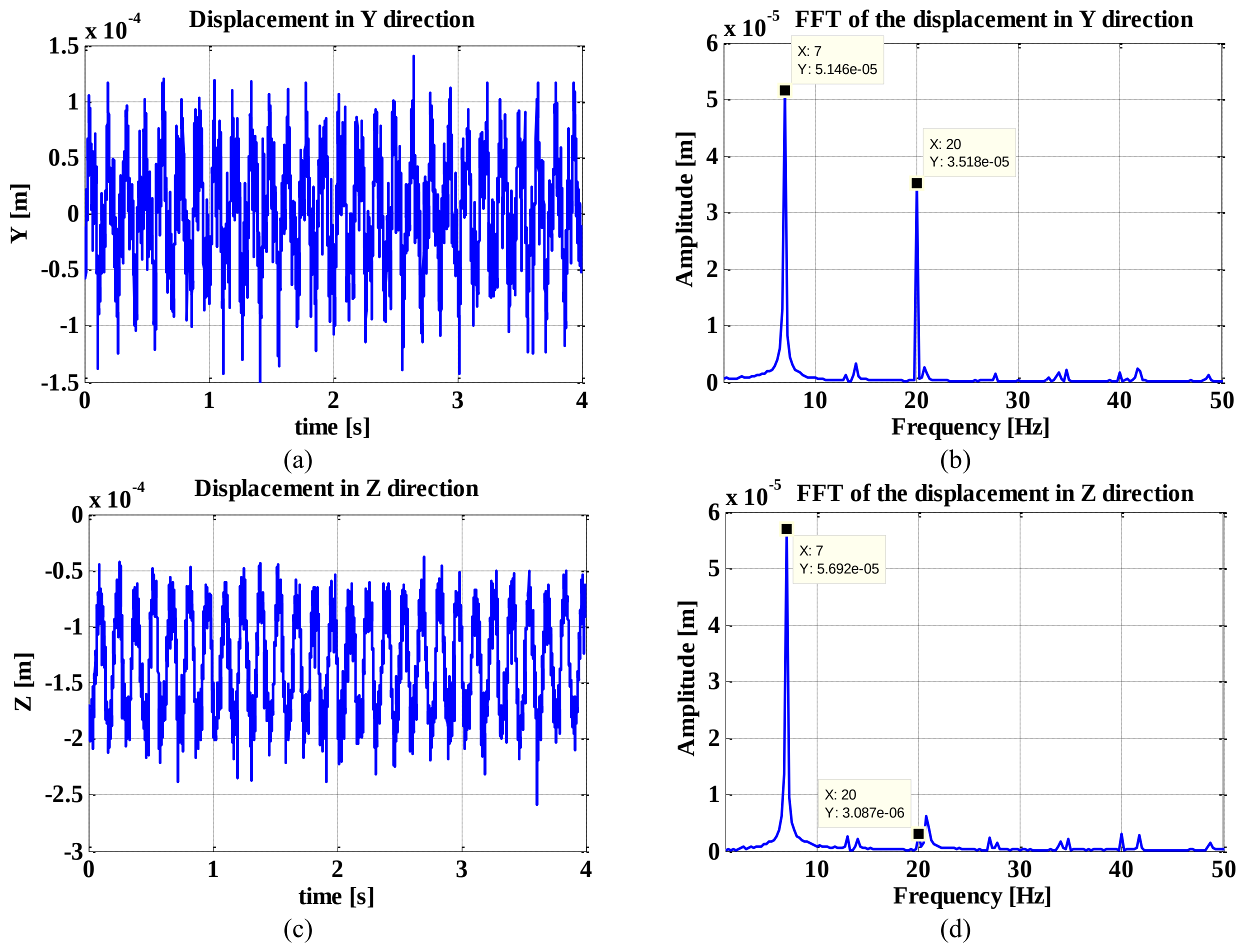

Figure 10. Dynamic test with sinusoidal force: (a) Displacement of the shaft in Y direction; (b) FFT of the shaft displacement in Y direction; (c) Displacement of the shaft in Z direction; (d) FFT of the shaft displacement in Z direction.

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Analogously Fig.8, the dynamic behavior of the shaft is also analyzed with no excitation force for a rotational speed of 25 Hz . Fig. 11 shows the shaft displacement obtained in the tilting pad journal bearing located in the free extremity.

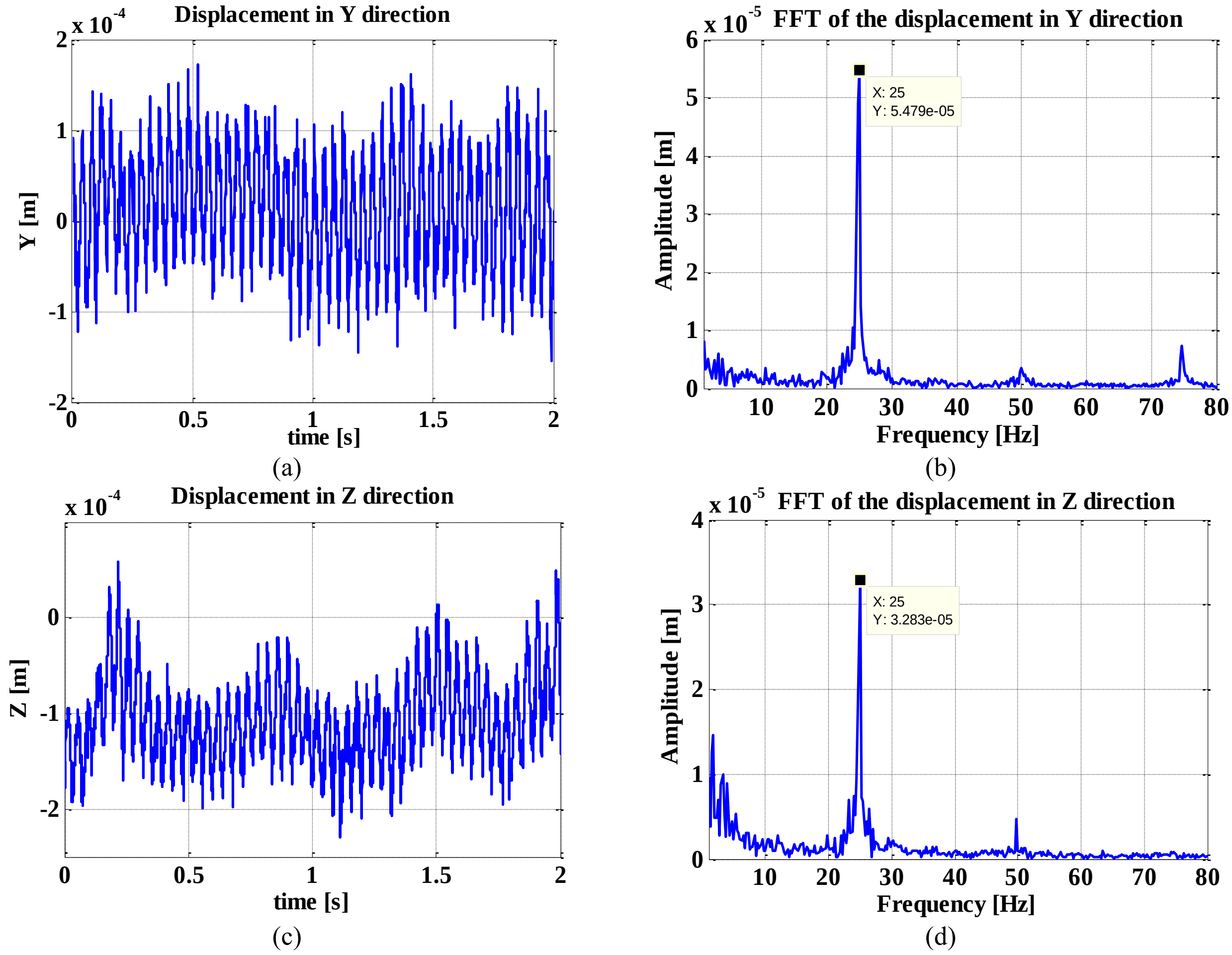

Figure 11. Dynamic test with no excitation force: (a) Displacement of the shaft in Y direction; (b) FFT of the shaft displacement in Y direction; (c) Displacement of the shaft in Z direction; (d) FFT of the shaft displacement in Z direction.

As in Fig. 8, the shaft displacement presents a harmonic response at 25Hz (rotational speed), due to the excitation for unbalance mass. Moreover, sub-harmonic components can be observed in the shaft displacement as shown in Figs. 11(b) and 11(d), possibly due to shaft misalignment or structural effects.

A sinusoidal force with amplitude of 10N and frequency of 20Hz is applied in the shaft in Z direction in order to verify the dynamic behavior of the shaft under excitation force. Fig. 12 shows the sinusoidal force applied.

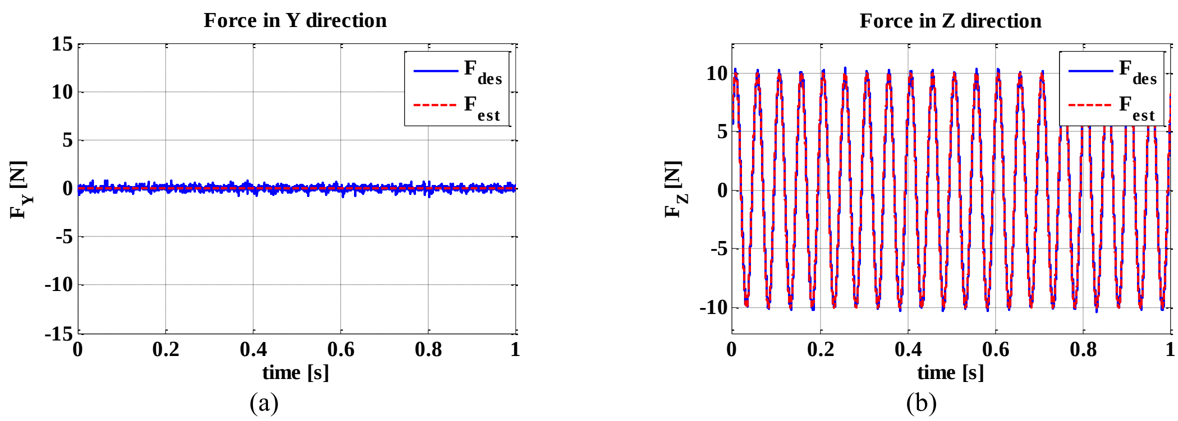

Figure 12. Sinusoidal force in the dynamic test: (a) force in Y direction; (b) Amplitude of 10N and frequency of 20Hz in Z direction.

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Figure 13 shows the shaft displacements in time domain and frequency domain when the sinusoidal excitation force is applied.

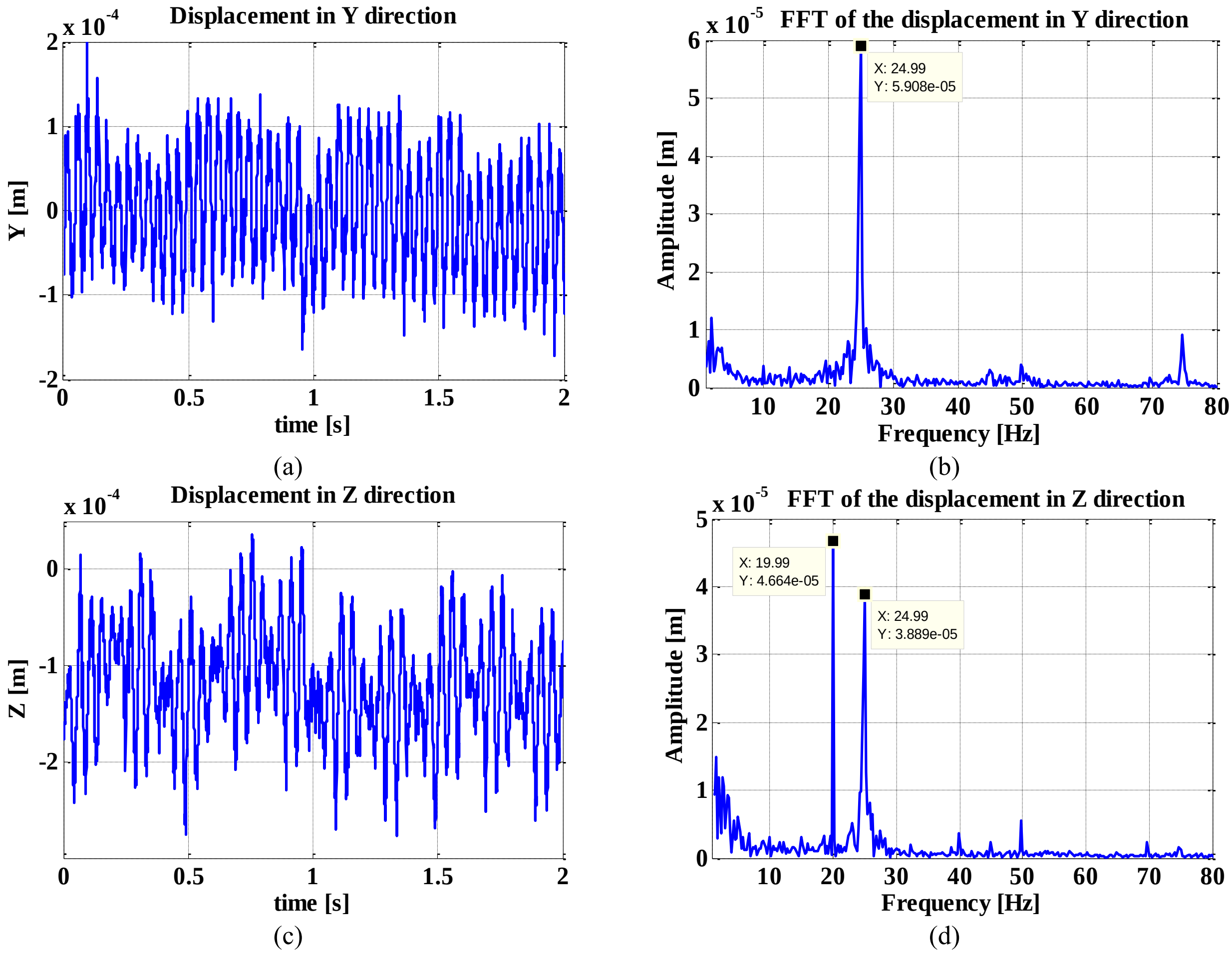

Figure 13. Dynamic test with sinusoidal force: (a) Displacement of the shaft in Y direction; (b) FFT of the shaft displacement in Y direction; (c) Displacement of the shaft in Z direction; (d) FFT of the shaft displacement in Z direction.

As previously observed, the shaft displacement increases in the direction in which the excitation force is applied. Also in this case, the effects of the excitation force in the shaft displacement are observed only in the direction of the excitation force. According to Fig. 13, the component of the excitation force (20Hz) is also evident in the shaft displacement in Z direction, while no component of the excitation force is observed in Y direction.

4. CONCLUSION

This work deals with the application of the active magnetic actuator for excitation of rotating systems supported by tilting pad journal bearing. An approach using a PID controller was proposed for application of the excitation forces. Static and dynamic tests were performed in order to verify the accuracy of the magnetic actuator as a nonsynchronous exciter.

The results obtained in the static tests showed that the relation between the desired force and the measured force is practically linear, although some differences are present due to the saturation of the coil core. Moreover, the results obtained for sinusoidal force show that the controller is not able to accurately control sinusoidal forces of high frequencies, because of the nonlinear behavior present in the magnetic actuator under saturation condition.

The capacity of the magnetic actuator to excite rotating system is investigated in the dynamic tests. The experimental results showed that the magnetic actuator is able to excite the rotor in both Z and Y directions separately. The influence of the excitation force is clearly observed in the shaft displacements, both at low (7Hz) and moderate (25Hz) rotational speed.

Therefore, the experimental results are promising to conclude that the approach proposed here is consistent and valid for some laboratory applications, since that magnetic actuator was successful applied for excitation of the rotating systems supported by tilting pad journal bearing.

5. ACKNOWLEDGEMENTS

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