Application of a Magnetic Actuator as an External Excitation Source in Fault Detection

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***Abstract*—Rotating machinery covers a broad range of industrial applications. Consequently, unexpected failures may lead to suddenly manufacturing break-off involving high maintenance costs. The most common faults in such machines introduce anisotropy to the system causing changes in its modal parameters and increasing the response backward component. Modal analysis methods can be used to identify changes in the machine behavior and foresee incipient faults. This paper presents the time domain Multiple Output Backward Autoregression (MOBAR) method and discusses its advantages over frequency domain methods on field tests in industry. For the MOBAR method, a tuned non-synchronous excitation called blocking test is applied to the shaft using a magnetic actuator, and when the excitation is turned off, the decay response is used to estimate natural frequencies of the system and the corresponding modal damping. The changing of these parameters can be used to predict system failures. To demonstrate the usability of electromagnetic actuators to apply this method for fault diagnosis and identification, a case study is presented in which the directional coordinates modal parameters are identified for a rotor supported by journal bearings using the MOBAR technique.**

I. INTRODUCTION

Heavy rotors, widely used in industries and power plants, have shafts rotating at high speeds that must have high trust levels. In order to obtain safety operating conditions and avoid sudden manufacturing break-off, an efficient maintenance program must be designed, once few hours out of the production line usually means considerable rising of the costs. Therefore, not only the maintenance process must be fast, but also diagnosis tests must not be time consuming.

In rotordynamics, many of the possible failures affect the anisotropy degree of the system; namely journal bearings and seals wear and oil-induced instabilities; and consequently the modal parameters of the system (natural frequencies and modal damping) change. Thereby the identification of the modal parameters change can be used in fault detection, isolation and identification (the three steps in Fault Diagnosis and Identification – **FDI** – according to [1]).

The parameter identification can be performed using one of the many existing modal analysis methods. Such methods are divided into two categories: frequency domain and time domain methods. The first group needs the acquisition of the frequency response function (**FRF**) of the rotor in the entire operational range in order to accurately identify the system modal parameters. The problem is that **FRF** measurements need excitation and response measurements, and it can be very time consuming, e.g. a stepped-sine measurement takes a long time to be made and sine-sweep or white noise techniques require many averages. As time domain methods are always based on transient response, the tests required are much faster; and there is no need to acquire excitation signals.

A remarkable family of time domain methods is the auto-regressive algorithms (**AR**). A backward **AR** method based in a single channel response data was developed in [2]; however methods which take into account multiple response signals achieve better accuracy, with less bias and deviation due to noise ([3], [4]). The method developed in [2] was extended to address multiple outputs in [4] and is referred to as **MOBAR** (Multiple Output Backward Autoregression). Multiple output techniques have better performance due to the richer modal information contained in the data set, presenting also better capability to identify close spaced modes, which are a common characteristic of rotating machinery (e.g. modes crossing due to the gyroscopic effect or a close pair of forward and backward modes); this capability is important in fault diagnosis once the anisotropy inserted by many failures affect both forward and backward modes, that must be identified for correctness of the **FDI** procedure.

Regarding external excitation source for modal analysis of rotating machinery, there is a practical limitation when using shakers or impact hammers, once due to friction, undesired tangential forces and noise can be acting on the system. Moreover, those procedures are hard to be applied on real machines. Hence, the use of an external magnetic actuator is very convenient when working with rotating systems. For this reason, a magnetic actuator has been a promising solution to apply the external excitation in the rotor, as it was done by [3], [5] and [6].

In this context, this article aims to present the application of a magnetic actuator to the **MOBAR** method and analyze its robustness, as well as provide a step-by-step guide to its application on field tests in industry, when short break-off times are mandatory. A case study regarding the identification of directional coordinates modal parameters in a journal bearing supported rotor is presented to demonstrate the potential applicability of this technique in **FDI**.

II. MOBAR

A. Test Procedure

As previously mentioned, time domain techniques are based in transient response measurements. In the **MOBAR** case, the recommended excitation is the blocking test ([3]). In this case, a tuned-sinusoidal excitation is applied to the system and then turned off. The response decay is used for the modal parameter estimation. This sinusoidal excitation is tuned in the frequency of the natural mode to be identified; making the contribution of that mode the dominant component in the system vibration (overcoming the synchronous unbalance and any other signals); i.e. the signal-to-noise ratio (**SNR**) of the mode is maximized and the decay observed in the response is relative to that mode only. It is also important to know the direction of the mode being excited (backward or forward), once that the blocking test is a rotating force that should have the same direction of the mode to guarantee the maximization of the mode **SNR**.

The step-by-step guide to apply the MOBAR technique in a field test is described below (based in [3]).

1. *Determine the natural frequencies*. Before applying the blocking test, the frequencies and directions of the modes to be analyzed should be estimated. This can be done in several ways: based in a model simulation (if there is enough precision) or stepped-sine (needs averaging). Stepped-sine excitation was chosen in this work because it can be done only in the interest frequency range and it is a tuned excitation (presenting better results than white noise measurements), and has the advantage over the sine sweep because it does not have transient effects as excitation frequency changes.

2. *Apply blocking test*. After obtaining modes and frequencies, the blocking test can be applied to the system for each of the natural frequencies to be analyzed. It is recommended to perform more than one blocking test per frequency for averaging purpose (it is observed at this point that the blocking test is a fast procedure, once it takes only enough time to the response due to the excitation achieve permanent regime, and decay after the excitation being turned off). Reference [3] observed that the amplitude of the blocking test should be high to maximize the **SNR** of the mode, but low enough to avoid non-linear effects.

3. *Apply the MOBAR method*. After recording the free decay of the response, the **MOBAR** method can be applied off-line in order to identify the modal parameters of the system. Details of the **MOBAR** method are given in the following section.

One advantage of the **MOBAR** method over the frequency domain methods is that it does not need the **FRF** of the entire operational frequency range of the machine in order do the modal parameters estimation; only the natural frequencies need to be known. Thus the stepped-sine needs to be performed only in a narrow frequency range in the resonance region, significantly reducing the test duration. An indicative of this frequency range can be obtained using a numerical model of the rotor.

However the main advantage of using **MOBAR** is that it can independently identify forward and backward natural modes, even if they are close spaced; which is a known drawback of frequency domain methods or other time domain methods based on one output signal only.

B. MOBAR method

Autoregressive model is a family of algorithms in time domain widely used in system identification. The name of the algorithms are due to how the models are built, where the response for a given time instant is written as a function of the future time instants (backward autoregressive model - **BAR**) as given in (1), or as a function of the previous time instant[1] (forward autoregressive model - **FAR**) as in (2).

$$y_t = \sum_{i=1}^{q} a_i y_{t+i} = a_1 y_{t+1} + a_2 y_{t+2} + \cdots + a_q y_{t+q} \quad (1)$$

$$y_t = \sum_{i=1}^{q} b_i y_{t-i} = b_1 y_{t-1} + b_2 y_{t-2} + \cdots + b_q y_{t-q} \quad (2)$$

q represents the model order, a_i and b_i are the model coefficients and y_i is the measured time series data.

The **MOBAR** uses a backward autoregressive model and the presented development is based in [4]. The method can identify systems with many modes, even if they are close spaced, through the use of multiple measurement channels (which can be signals of different nature as displacement, velocity and acceleration). The terms y_i in (1) are column vectors of length p, where p is the number of sensors, and the coefficients a_i are square matrixes of dimension p. From N measurement points, the matrix equation (3) can be written using (1), and can be solved using least mean square or singular value decomposition ([2], [3], [4], [5]). Once the a_i coefficients were obtained, the backward autoregressive model can be written in the state space form as shown in (4) using the state vector $z_t=[y_t^T \ y_{t+1}^T \ \ldots \ y_{t+q-1}^T]^T \in R^{pqx1}$.

$$\begin{bmatrix} y_1^T \\ y_2^T \\ \vdots \\ y_{N-q}^T \end{bmatrix}^T = \begin{bmatrix} a_1^T \\ a_2^T \\ \vdots \\ a_q^T \end{bmatrix}^T \cdot \begin{bmatrix} y_2 & y_3 & \cdots & y_{N-q+1} \\ y_3 & y_4 & \cdots & y_{N-q+2} \\ \vdots & \vdots & & \vdots \\ y_{q+1} & y_{q+2} & \cdots & y_N \end{bmatrix} \quad (3)$$

$$\begin{gathered} z_t = A_d \cdot z_{t+1} \\ z_t = \begin{bmatrix} a_1 & \cdots & a_{q-1} & a_q \\ I_p & \cdots & 0_p & 0_p \\ \vdots & \ddots & \vdots & \vdots \\ 0_p & 0_p & I_p & 0_p \end{bmatrix} \cdot z_{t+1} \end{gathered} \quad (4)$$

The **BAR** model (4) can be transformed in a **FAR** model by isolating z_{t+1} as in (5). The eigenvalues of the transformed **FAR** model (**TFAR**) have an interesting property: the

[1] A **FAR** model can be transformed in a **BAR** model by isolating the term y_{t-q} and adding q to the sub-indexes (for notation purpose only); the inverse transformation is analogous.

eigenvalues located inside the unit circle are attributed to the system modes and the others are spurious modes [4]. Therefore, the system modes can be easily identified[2] and the reduced model (A_r) can be converted to a continuous model (A_s) using (6); where dt is the sampling time. The modal information of the physical system (natural frequencies and damping) is obtained from the eigenvalues of A_s.

$$z_{t+1} = A_d^{-1} \cdot z_t$$
$$z_{t+1} = \begin{bmatrix} 0_p & I_p & \cdots & 0_p \\ \vdots & \vdots & \ddots & \vdots \\ 0_p & 0_p & \cdots & I_p \\ a_q^{-1} & -a_q^{-1}a_1 & \cdots & -a_q^{-1}a_{q-1} \end{bmatrix} \cdot z_t \quad (5)$$

$$A_s = \ln(A_r)/dt \quad (6)$$

Overspecify the model order q is an important step to improve accuracy of the algorithm ([2], [4], [5]). Even though the **MOBAR** has the inherent property of separating system modes from spurious modes, when the measurements are subjected to significant levels of noise or the system has high damping – fast decay response, the separation may not be done properly. Therefore, the stabilization diagram may be used to help the system modes identification. The stabilization diagram is widely used in many modal analysis methods and it consists in applying the method for many different orders q and plotting the identified eigenvalues in a diagram as a function of q. In the diagram it is possible to analyze which eigenvalues do not change among the different model orders (in terms of frequency, damping and eigenvector), i.e. identify the system modes.

III. EXPERIMENTAL SETUP

A. Test Rig

The test rig setup was defined to reproduce some typical dynamic effects of a journal bearing supported turbomachinery; in this case similar to the real machine analyzed in [6]. Instability due to the presence of the journal bearings was chosen to validate the **FDI** method capability because it is directly related to the anisotropy changing of the bearings as the rotational speed of the rotor increases.

The test rig presented in Fig. 1a consists of a steel SAE 1030 shaft supported by two journal bearings and one rolling bearing. There is also a disc and the electromagnetic actuator journal, both made of steel SAE 1020. As previously mentioned, the actuator is usually inserted at one end of the shaft, however as the test rig shaft is thin, the rolling bearing at the shaft end was used in order to avoid the actuator journal to be overhung and present high amplitude vibrations. A schematic drawing of the shaft with the main dimensions is presented in Fig. 1b and its design parameters are found in Table I.

Displacement sensors measure the vibration amplitude at the journal bearings. Disc vibration, oil temperature and load on the bearings are also measured for monitoring purposes only.

[2] The matrices of the **TFAR** model are different from the common **FAR** model, which has all of its eigenvalues located inside the unit circle.

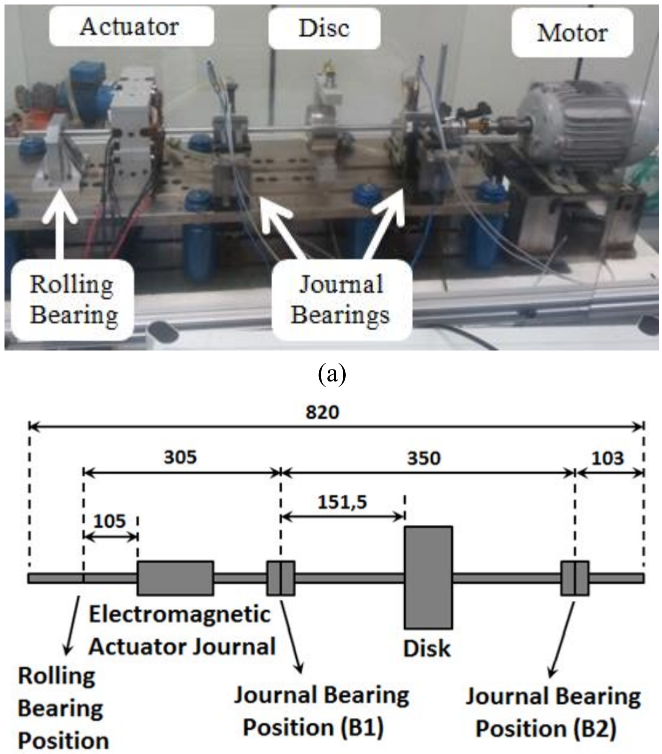


Figure 1. Experimental setup: (a) test rig; (b) schematic drawing.

TABLE I. DESIGN PARAMETERS OF THE TEST RIG

Parameter	Value
Shaft Length	820 mm
Shaft Diameter	12 mm
Disk Length	47 mm
Disk Diameter	95 mm
Electromagnetic Actuator Journal Length	80 mm
Electromagnetic Actuator Journal Diameter	40 mm
Journal Bearing Length	20 mm
Journal Bearing Diameter	30 mm
Lubricated Oil	ISO VG 32
Journal Bearing Radial Clearance	90 µm
Lubricated Oil Temperature in the Tank	25-27 °C

B. Electromagnetic Actuator

The electromagnetic actuator in Fig. 2a was the external excitation source used to apply the blocking test and the stepped-sine in the shaft during the tests. It was designed and built at the Laboratory of Rotating Machinery (LAMAR) at UNICAMP [7]. Having an homopolar configuration, the actuator is able to apply an excitation of 150 N in a frequency range up to 100 Hz, when operating with an air-gap (radial clearance) of 2.5 mm and a current of 3 A. Therefore, it is capable of exciting all the inertia involved in the test rig throughout the entire operating frequency range of the rotor. The first natural mode of the actuator was evaluated through a finite element model simulation (Fig. 2b) and was found to be at 576 Hz, which is far above the operating frequency of the system, as desired.

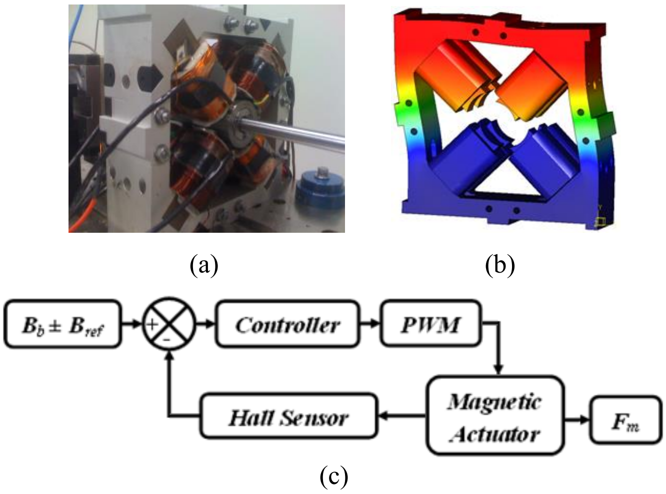


Figure 2. Electromagnetic Actuator: (a) test rig assembly; (b) first natural mode at 576 Hz; (c) actuator system diagram.

The force (F_m) produced by an electromagnetic actuator in equation (7) is proportional to the pole area of the actuator (A_g), the air magnetic permeability (μ_g) and the magnetic flux density (B), which is given by equation (8) and proportional to the current (i), the number of coils (N) and the air-gap length (l_g), according to [8] and [9]. For a differential assembly, the magnetic force, equation (7) becomes equation (9), where B_1 and B_2 are the magnetic field of the opposite coils. Using a bias magnetic field (B_b) added to the reference field (B_{ref}), the linearized force given in equation (10) is obtained [10].

$$F_m = \frac{B^2 \cdot A}{\mu_g} \tag{7}$$

$$B = \frac{\mu_g \cdot N \cdot i}{2 \cdot l_g} \tag{8}$$

$$F_m = \frac{A_g}{\mu_g}\left(B_1{}^2 - B_2{}^2\right) \tag{9}$$

$$F_m = \frac{A_g}{\mu_g} 4 B_b B_{ref} \tag{10}$$

Therefore, the actuator force control loop can be designed in terms of magnetic field ([7], [11]) as shown in Fig. 2c, where B_{ref} is the magnetic flux density necessary to obtain the desired reference force. The reference magnetic field is compared to the actual magnetic field, measured by the hall sensor, and the result is the input of the proportional controller. The output voltage of the controller is converted to current by the PWM amplifier (operating in current control mode) and sent to the actuator coils, generating the magnetic force F_m.

IV. Results and Discussion

In order to investigate the robustness of the MOBAR method using an electromagnetic actuator as external excitation force, a series of tests were accomplished. For this paper purposes, the results are shown in two rotational speeds of the rotor: 65 Hz and 70 Hz.

The first step of the test procedure consists of determining the natural frequencies of the rotor to apply the blocking test. Different methods were briefly discussed, however, if the system presents high levels of noise or if the natural frequency appears as a flat surface in the **FRF** instead of a sharp peak, it may be difficult to precisely identify the natural frequency regardless of the method used.

In this work, the two natural frequencies associated to the first forward and the first backward bending modes were identified from the **FRF** obtained using the stepped-sine method, and are presented in Table II. To evaluate the robustness of the method, the blocking tests were applied for different frequencies around the natural frequencies to be identified in a range of ± 3 Hz (from 55 Hz to 61 Hz for the forward frequency and from 57 Hz to 63 Hz for the backward frequency); corresponding to approximately ± 5% of error in the first step of the test procedure, i.e., the natural frequency identification. The accuracy of the method was evaluated by repeating 10 times each blocking test in each of the defined frequencies.

TABLE II. NATURAL FREQUENCIES OF THE FIRST BENDING MODES IDENTIFIED FROM THE FRF USING STEPPED-SINE

Rotational Speed	Forward	Backward
65 Hz	58.1 Hz	60.9 Hz
70 Hz	56.8 Hz	60.2 Hz

It is important to remark that, although the disc vibration is measured, only the measurements in the bearings position will be used with **MOBAR**, once that is the common scenario when dealing with real machines in industry.

The results were obtained using a time interval of 0.17 s of the four signals (horizontal and vertical displacements in the position of both journal bearings) after the blocking test excitation was turned off. Fig. 3 presents a measurement of the vertical displacement of the first journal bearing where it can be seen the blocking test excitation been turned off at the first vertical line in 0.5 s, and the transient decay of the response until 0.67 s, at the second vertical line; the transient response between the vertical lines is the signal used as input for the **MOBAR** technique.

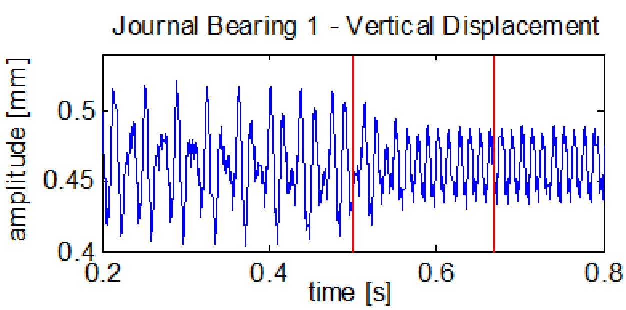


Figure 3. Vertical displacement of the first journal bearing.

After the measurements, as previously mentioned, the **MOBAR** method was applied using different model orders. For this rotor, it was notice that varying the model order from 20 to 50 presented good results. In the most of the measurements, higher orders implied in the method loss of ability to identify the natural frequencies. The authors believe this effect is due to the short time interval used to calculate a high order model, even though the natural frequency was well identified for lower orders, as exemplified in Fig. 4.

In the legend of Fig. 4: asterisk means an unstable pole, triangle is a pole which has its natural frequency close (inside a given range) from a pole of the previous model order, a square pole is the one which frequency and the modal damping have changed (again, inside a given range) from the previous model order, black circle represents a pole whose frequency and eigenvector (modal assurance criterion was used) were kept almost unaltered from the previous model order, and finally, white circle represents a stable pole whose three parameters (natural frequency, modal damping and eigenvector) present small changes from the previous model order. Therefore, in the stabilization diagram analysis, a pole which becomes stable for different model orders is very likely to be a real pole of the system. It should also be noticed that this stability concept is relative to the range defined by the user. In this work, the acceptable change from one model order to other for the pole to be considered stable was 2% for the natural frequency, 10% for the modal damping and 5% for the eigenvector. The modal damping has a higher tolerance once it is the most difficult parameter to be identified.

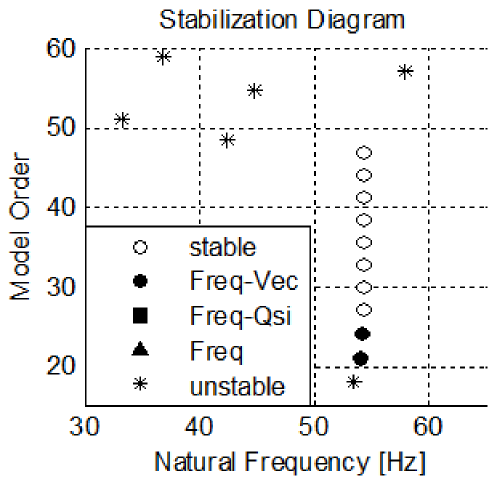


Figure 4. Example of stabilization diagram.

The results of the natural frequency identifications are presented in Figs. 5a and 5b for the rotational speeds of 65 Hz and 70 Hz, respectively. From the ten trials in each excitation frequency, not all of then resulted in a good identification (a stable mode), thus, the black numbers in the figures represent how many of the tests provided good identifications, the vertical bars are ± 1 standard deviation, and the lines connect the mean values. The black line represents forward natural modes and the backward natural modes are represented in grey.

Recalling that the forward natural frequency is around 57 Hz and the backward natural frequency is around 60.5 Hz, it can be noted that in both rotational speeds (Figs. 5a and 5b) analogous results were obtained. In the identification of the backward natural frequency, a higher excitation frequency (60 Hz to 63 Hz) presented better estimates than lower frequencies (smaller changes in the mean value); in both cases, a backward excitation of 58 Hz was not able to identify the backward mode, but the forward only. Otherwise, the forward natural frequency was also identified for lower excitation frequencies (56 to 58 Hz), still it was identified in the entire excitation range (except in the excitation frequency of 59 Hz for the rotational speed of 65 Hz).

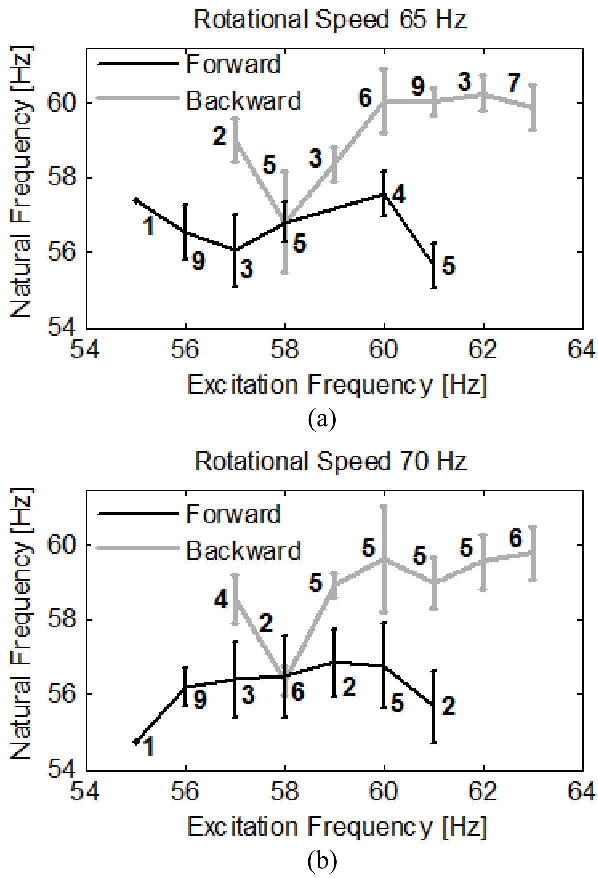
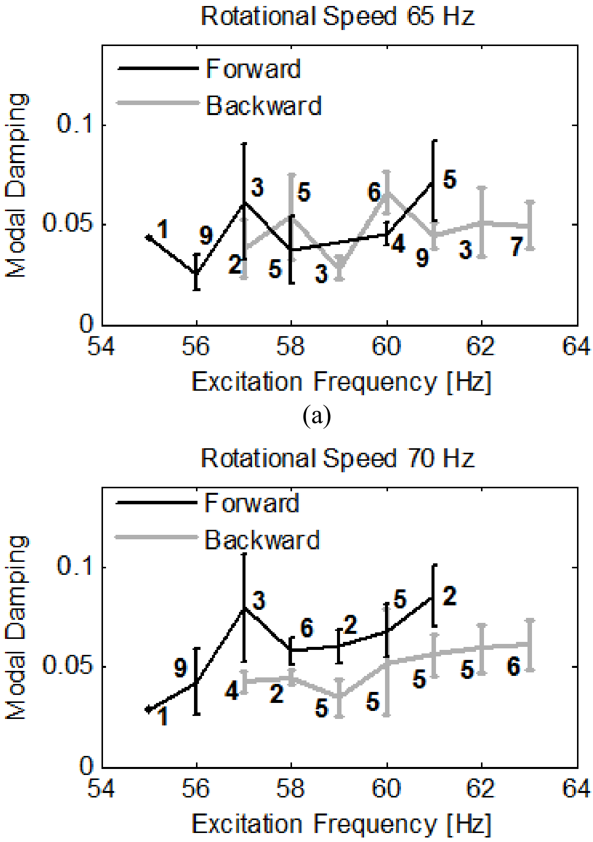


Figure 5. Natural frequencies identified: (a) rotational speed of 65 Hz; (b) rotational speed of 70 Hz.

The modal damping identification for the rotational speeds of 65 Hz and 70 Hz are presented in Figs. 6a and 6b. The same symbology from Fig. 5 is used in Fig. 6: black numbers represent how many tests (from ten performed for each excitation frequency) resulted in stable poles, vertical bars represent standard deviation of the tests and the lines connect the mean values (black for the forward mode and grey for the backward mode).

In the identified modal damping, the backward mode damping presented smaller standard deviation and smaller changes in the mean value for higher excitation frequencies (60 Hz to 63 Hz).

The identification results of the forward modal damping presented relatively large standard deviations for the rotational speed of 65 Hz. However, it can be noted that the modal damping identified for both the forward and backward mode presented better results in the rotational speed of 70 Hz.



(a)

(b)

Figure 6. Modal dampings identified: (a) rotational speed of 65 Hz; (b) rotational speed of 70 Hz.

From all the 280 tests performed, nearly 43.5% produced valid identifications (black numbers in Fig. 5), showing the importance of performing repeated tests when using **MOBAR** to identify modal parameters.

The largest standard deviation presented in the frequencies identification was around 2.7% of the frequency value. The largest standard deviation in the modal damping identification was 70% in the worst case (rotational speed of 65 Hz and excitation of 57 Hz), but only three measurements resulted in stable poles in this set of tests. The excitation frequencies in which 5 or more tests resulted in stable poles presented a standard deviation around 40%. It is important to highlight that the system is highly damped, as shown in Fig. 3, which implies in added difficulty in tests performance and identification procedure.

V. CONCLUSIONS

It is concluded from the obtained results that the **MOBAR** method is very accurate and robust in the identification of natural frequencies, but it needs averaging in order to obtain accurate results in the modal damping identification.

The results obtained have shown that the **MOBAR** technique with blocking test excitation using an electromagnetic actuator as external excitation source can be successfully used to identify modal parameters of a rotating system. However it should be remarked that averaging is an important part of the process for highly damped rotors or rotors with a high signal to noise ratio (**SNR**). Conveniently, the blocking test is not time consuming since the duration of each test is about one second, thus one minute under test would provide 60 measurements, producing much better results. Since, modal damping identification is challenging for any identification method, the small time consumed in the blocking test is an important advantage over other methods.

Future works consist of two steps: first, use the MOBAR technique with the magnetic actuator as external excitation source in fault diagnosis and identification (**FDI**) when multiple incipient faults are present in the system; and second, extent the research towards fault tolerant control (**FTC**) using the same magnetic actuators/bearings.

The possible faults to be identified are those which change the characteristics of the system. For example, hydrodynamic bearing wear, cavitation, starvation and oil overheating; all of the mentioned faults change the stiffness and/or damping coefficients of the bearings, influencing the natural modes of the system and even, in more severe cases, making the system unstable. Once one of these conditions are diagnosed and identified (**FDI**), the fault tolerant control (**FTC**) can act through the magnetic actuator itself, allowing the system to safely shut-down for maintenance.

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