FREQUENCY RESPONSE ATTENUATION OF ROTATING SYSTEM USING HYDRODYNAMIC BEARING WITH ELECTROMAGNETIC ACTUATORS

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Abstract. *Due to their capacity of modifying the dynamic characteristics of rotating machinery, improving their performance and increasing their availability, active bearings as tilting-pad hydrodynamic bearings and active magnetic bearings (AMBs) are largely studied and improvements in their design is an important topic of research in the field of rotating systems. In this sense, this work presents an active hybrid bearing, which is composed by a hydrodynamic bearing with electromagnetic actuators, taking advantages of the hydrodynamic lubrication (high load capacity) and those of the electromagnetic systems (high actuation capacity). The hybrid bearing is tested in a rotating system, and a proportional-derivative controller (PD controller) is employed in order to control the lateral vibrations of the rotor. Experimental tests are performed and the functionality of the hydrodynamic bearing with electromagnetic actuators is checked, showing its capacity of both supporting and attenuating the frequency response of the closed-loop system. The experimental results suggest that this kind of hybrid bearing are a promising design of active bearings in comparison to traditional hydrodynamic and magnetic bearings employed separately, overcoming their limitations and joining their advantages in a compact assembling.*

Keywords*: active bearings, active magnetic bearings (AMBs), hydrodynamic bearings, rotor dynamics, PD controllers.*

1. INTRODUCTION

Rotating machinery such as pumps, compressors, turbines and machine tools are largely employed by industry and productive processes and improvements in their development and design are largely studied, in order to increase their performance and availability. In this case, a challenging question is controlling lateral rotor vibration, which is defined as radial-plane orbital motion of the rotor spin axis, resulting in energy losses and even in fatigue and failure of the machine (Adams, 2001). Bearings constitute one of the most critical components of rotating machines, thus affecting the rotor dynamic performance, life and reliability of the machine (Vance, 2010). Based on these concepts, the idea of using active bearings like active magnetic bearings, tilting-pad hydrodynamic bearings and bearings coupled to piezoelectric and hydraulic actuators appears as an economic and effective way of controlling vibrations in rotors and improving the machine's behavior, modifying its dynamical characteristics according to the operation conditions, in order to satisfy desired requisites: stability, performance, higher availability, less energy consumption and longer lifecycle.

In this context, an active bearing whose application has been increasing in the last years is the active magnetic bearing. Its operation principle is based on the magnetic suspension of the rotor using electromagnetic actuators, being the actuators also responsible for controlling the lateral vibrations, in which a sensor measures the rotor displacement, and the measured signal is sent to a controller, thus feeding back the control system (Fig. 1) - Schweitzer and Maslen, 2009. Hence, due to their contactless nature, AMBs have several advantages over conventional bearings: they can operate over a wide range of temperature; high rotation speed can be achieved; there is no friction; and no lubrication is required (Duan *et al*., 2001).

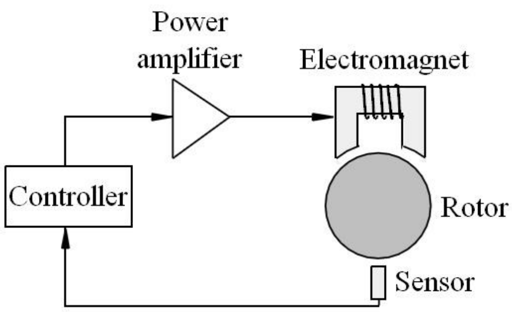
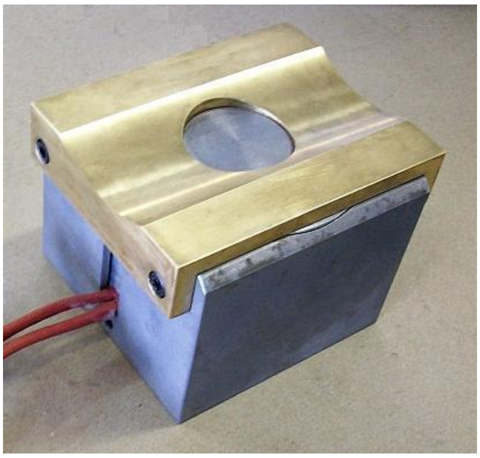


Figure 1. Active magnetic bearing operation principle (Schweitzer and Maslen, 2009).

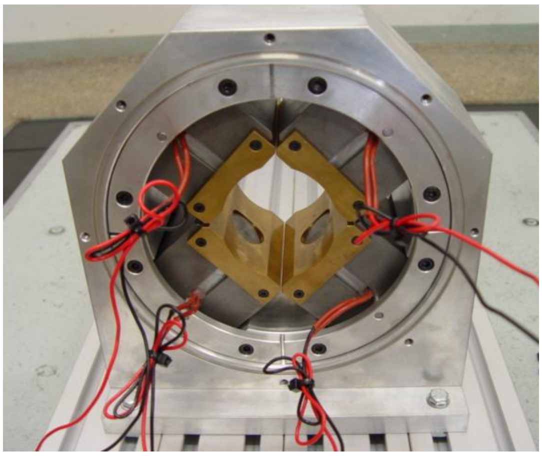
However, despite their advantages, AMBs become complex and voluminous as higher load capacity is required. Besides, in larger machines, protection systems such as back-up bearings must be mounted in the machine for preventing damages in case of electronic failure or bearing overloading (Kasarda, 2000). Hence, considering these limitations, there is the following idea of active hybrid bearing: a hydrodynamic bearing with electromagnetic actuators assembled in a same compact mounting, joining the characteristics of high load capacity and high actuation capacity. In this case, the magnetic system becomes smaller because the shaft is supported by the hydrodynamic forces, and the electromagnets are only responsible for controlling lateral vibrations, thus not requiring a back-up system, since electronic failure does not affect hydrodynamic lubrication. In this work, this proposed concept of hydrodynamic bearing with electromagnetic actuators is presented and the bearing's capacity of both supporting and controlling lateral rotor vibrations is evaluated. A PD control strategy is employed, and the frequency response functions (FRFs) of the uncontrolled and controlled system are compared.

2. HYDRODYNAMIC BEARING WITH ELETROMAGNETIC ACTUATORS

The main characteristic of the hydrodynamic bearing with electromagnetic actuators (Fig. 2) is joining the actuation capacity of active magnetic bearings, which are not appropriated for supporting high loads, with the capability of hydrodynamic bearings of sustaining heavy systems. In literature, the idea of joining the hydrodynamic bearing capacity with electromagnetic actuation is frequently related to electric motors, where the shaft is supported by hydrodynamic bearings and driven by magnetic bearings (Chen and Zhang, 2002). However, differently from the concept of active hybrid bearing presented in this work, the bearing and the actuators are usually apart and not integrated in a same mounting.

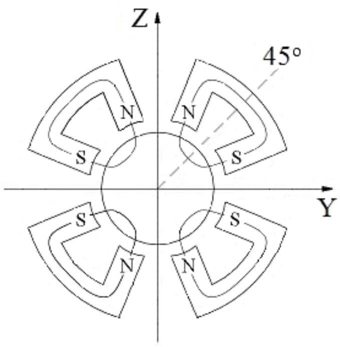


(a) pad with electromagnetic actuator

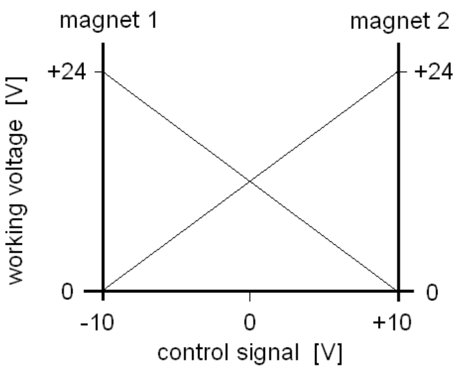


(b) assembling of the four tilting-pads

Figure 2. Hydrodynamic bearing with electromagnetic actuators.



(a) pole configuration



(b) working voltage balancing

Figure 3. Working principle of the hybrid bearing.

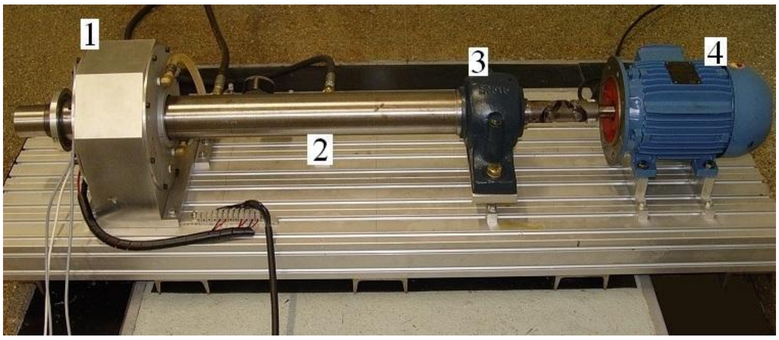
In Fig. 2, one can see a sliding surface with one electromagnet inside and the arrangement of all sliding surfaces in the active hybrid bearing. These surfaces are responsible for supporting the rotor: galleries in the bearing casing allow the injection of pressurized oil by a hydraulic system, and a thin oil film is formed between the sliding surfaces and the shaft (hydrodynamic lubrication), ensuring the support of the rotor (bearing mechanism). The working principle of the electromagnets is based on a four pole configuration, in such a way that they can attract the rotor along two orthogonal axes and in both directions (Fig. 3a). Each one of the two pairs of electromagnets is driven according to the working voltage balance shown in Fig. 3b (when the control signal is ±10 V, maximum control force is obtained).

3. EXPERIMENTAL TESTS

In this section, the active hybrid bearing is tested in a rotating system and its capacity of both supporting and controlling the lateral vibrations of the rotor is analyzed. In order to check the reduction of the vibration level, the frequency response functions of the uncontrolled and controlled system are measured and compared.

3.1. Test rig and experimental procedure

The hydrodynamic bearing with electromagnetic actuators and a self aligning ball bearing are responsible for supporting a rigid shaft connected to an electric motor by a universal joint (Fig. 4a). The system operates over a range of 540 rpm to 1800 rpm (9 to 30 Hz) and it is mounted on a ground vibration isolated platform, whose resonant frequency is ~2 Hz. A hydraulic unit supplies pressurized oil to the active hybrid bearing, and oil temperature is kept at a range from 38 to 41^{o}C by a tube fin heat exchanger, whose fan is driven by an on-off control system (Fig. 4b).



(a) test rig

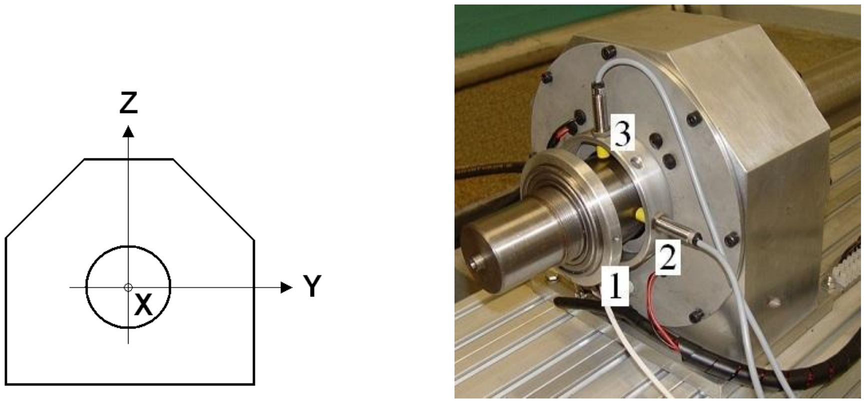
(b) hydraulic unit

Figure 4. Test rig of the active hybrid bearing: (1) hydrodynamic bearing with electromagnetic actuators, (2) shaft, (3) self aligning ball bearing, (4) electric motor, (5) oil reservoir and pump, (6) heat exchanger.

In order to obtain the frequency response functions of the system, the FRF estimator $H_1(\omega)$ is used (Maia and Silva, 1997; Ewins, 1984). A shaker is used to excite the system, whose stinger is connected to the rotating system by an excitation bearing mounted near the hybrid bearing (Fig. 5, item 1). Shaft displacement in the horizontal and vertical directions (Y and Z directions) is measured by inductive proximity sensors mounted in the active bearing (Fig. 5, items 2 and 3).

The control strategy consists in a PD-type controller whose proportional and derivative gains are determined by trial-and-error. As schematized in Fig. 6, the control signal and the excitation signal, which is a chirp signal (frequency ranging from 15 to 70 Hz), are sent from the data acquisition system at a sample frequency is 1 kHz. The control signal is sent to the drive of the electromagnetic actuators in the bearing. The excitation signal is amplified and sent to a shaker in order to produce the excitation force that is the input for the FRF calculation. The rotor displacements in both Y and Z directions (output) are measured and converted to a digital signal. In the case, the measured displacements are used to feed back the control system and to determine the FRF. Both control and the system identification procedures are implemented on MatLab Simulink environment via Real Time Workshop (RTW) (Fig. 7).

In Fig. 7, one can see that the Y and Z displacements are filtered by a DC level filter, in order to change the average value of these signals to zero, reducing the control voltage sent to the actuators - this procedure is especially useful to avoid drive overloading (±10V). In addition, the actuator's drive has two channels (Ch1 and Ch2), each one corresponding to a pair of actuators. Since the actuators are placed in 45^{o} directions (Fig. 3a), the Y and Z signals from the same pair of actuators are summed. The Z signal is multiplied by -1 before being summed with the Y signal in the channel 1 in order to follow the coordinate system adopted (Fig. 5a).



(a) coordinate system (b) excitation bearing and sensors

Figure 5. Excitation system and proximity sensors in the test rig.

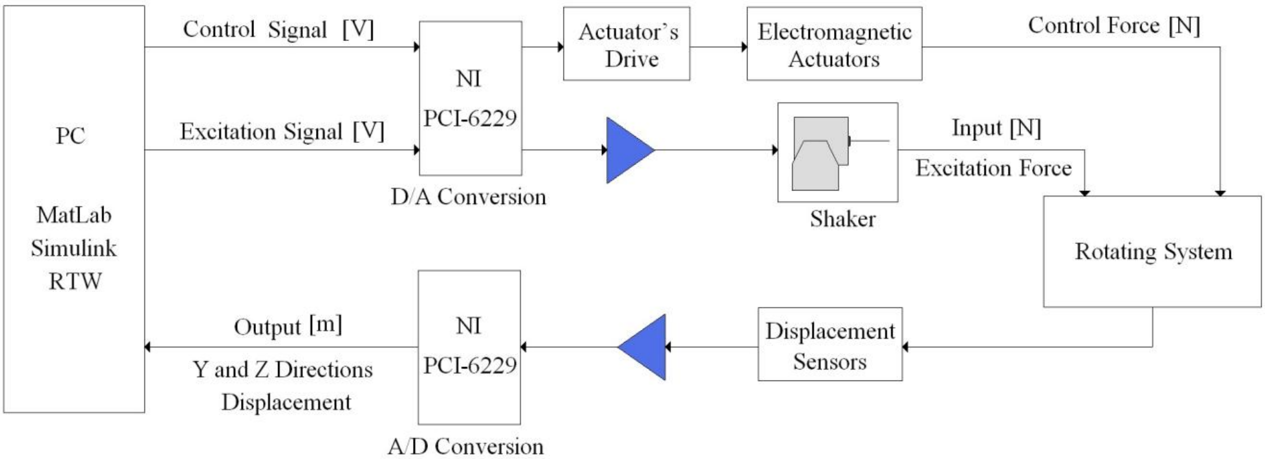


Figure 6. Scheme of the identification procedure of the rotating system with the active hybrid bearing.

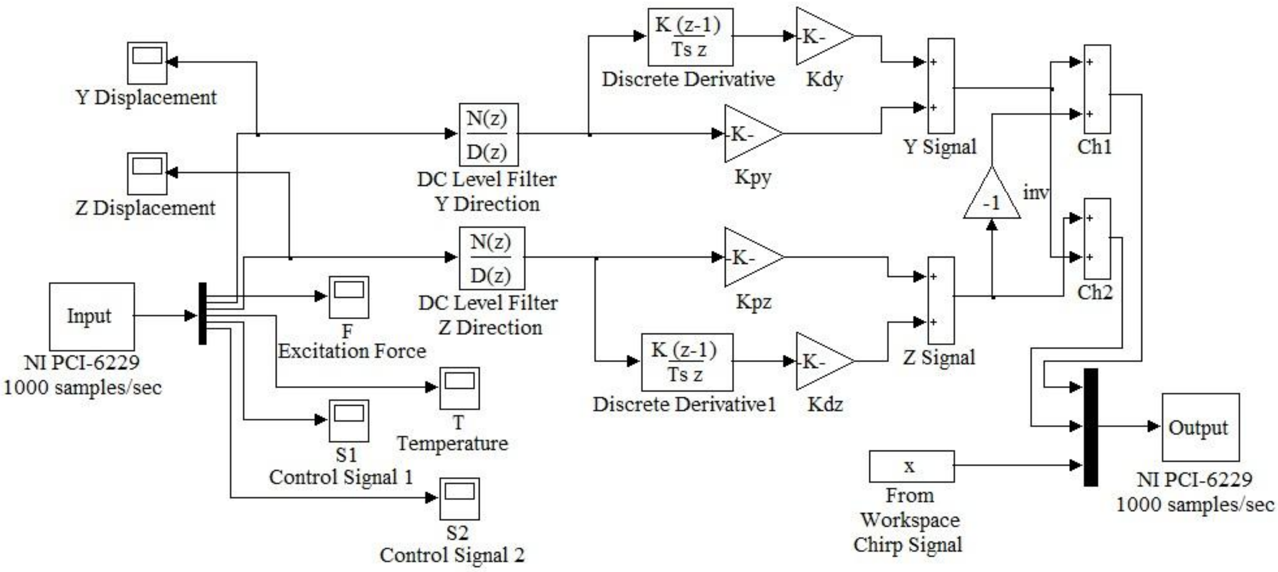


Figure 7. MatLab Simulink program used to excite and control the system.

3.2. Experimental results

First, the frequency response functions of the uncontrolled system for rotating speeds of 585, 1170 and 1760 rpm (9.75, 19.5 and 29.3 Hz) are measured (in this case, the gains K_P and K_D are zero). The shaker excitation (input) is applied in the Y direction and the displacement in this same direction is used in the calculations of the FRFs (Fig. 8). Due to a high noise-to-signal ratio, the FRFs measured for the displacement in Z are not clear and, even when the excitation is in the vertical direction, no information can be extracted from these data. Hence, all frequency response functions presented along this paper are calculated considering an excitation force in the horizontal direction and the shaft displacement in the same direction.

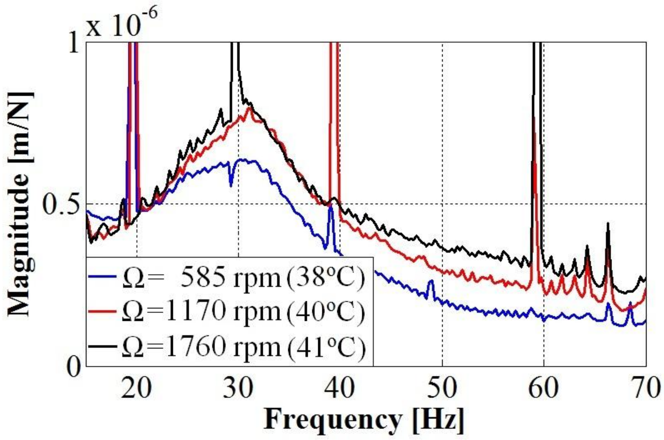


Figure 8. FRFs of the open-loop system for different rotating speeds (Y direction).

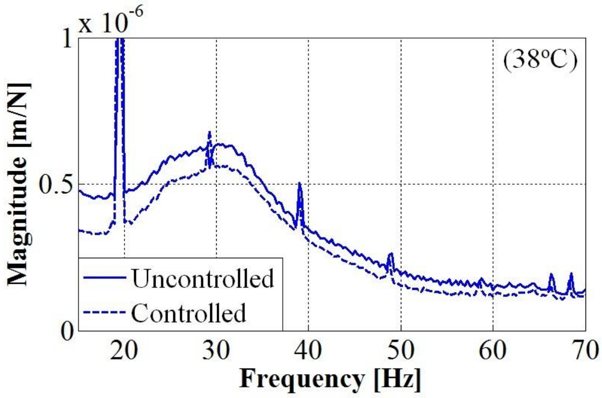
In Fig. 8, one can see that higher rotating speeds result in higher vibration levels with a slight shift in the resonance frequency. This is typical of rotating systems supported by lubricated bearings. As shaft speed increases, oil film equivalent stiffness increases and damping decreases. The spikes (high level peaks) in the FRF are due to shaft unbalance, characterized by frequency component at 1× the rotating frequency, and shaft misalignment, characterized by frequency component at 2× the rotating frequency.

In order to check the ability of the active hybrid bearing in controlling lateral vibrations of the rotor, a control signal is sent to the electromagnetic actuators (Figs. 6 and 7). A PD controller is adopted with K_P = 100 V/m and K_D = 0.8 V/m (determined by trial and error). The FRFs of the controlled system are shown in Fig. 9, and the resonant peak amplitudes are compared in Table 1.

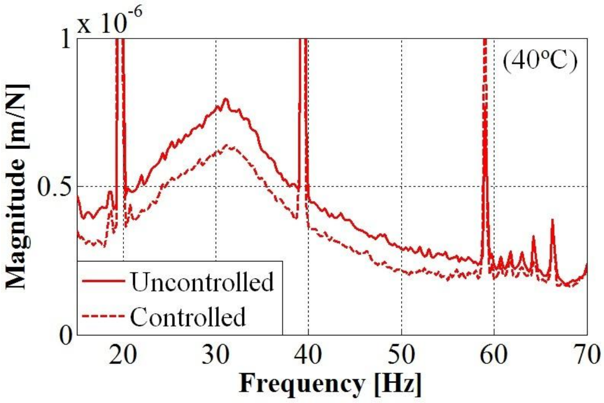
Figure 9 shows that it is possible to reduce the vibration level of the considered rotating system with electromagnetic actuators mounted inside the tilting-pad hydrodynamic bearing. Stronger reduction of amplitude is observed for higher rotating speeds, reaching up to 20% reduction for the 1760 rpm case (Table 1). These results suggest that the actuation capacity improves as a function of the rotating speed. Actually, if the actuation system is the same, this better performance at higher rotating speeds is explained by the decrease in oil film damping (hydrodynamic effect). Hence, the dampening effect of the hybrid active bearing becomes more pronounced at higher rotating speeds. Although not detected in the studied cases, it is expected that this performance of the actuation system reaches a limit due to the fact that oil film stiffness increases with rotating speed. As a result, the actuators will not be able to properly control shaft vibration because oil film is too stiff. This would be the case of high speed machines, running above 5000 rpm.

Table 1. Resonant peak amplitude comparison.

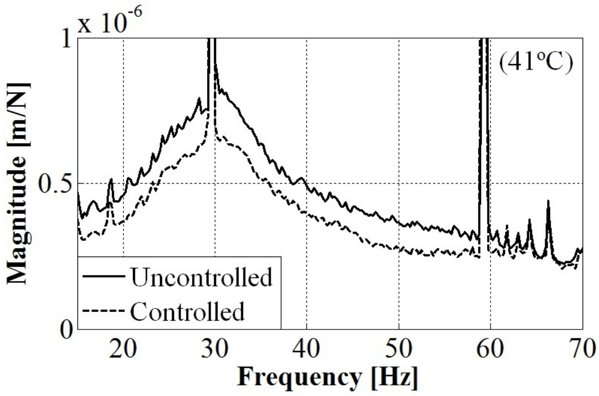
rotating speed (rpm)	*uncontrolled system* (m/N)	*controlled system* (m/N)	*peak reduction* (%)
585	0.634×10^{-6}	0.562×10^{-6}	11.4
1170	0.760×10^{-6}	0.623×10^{-6}	18.0
1760	0.810×10^{-6}	0.645×10^{-6}	20.4



(a) 585 rpm



(b) 1170 rpm



(c) 1760 rpm

Figure 9. FRFs of the uncontrolled and controlled system for different rotating speeds (Y direction).

4. CONCLUSION

In this work, the concept of hydrodynamic bearing with electromagnetic actuators was presented and experimental tests were performed. Experimental tests have shown the capacity of the active hybrid bearing of both supporting and controlling lateral vibrations of the tested rotating system. However, there is still room for improvements in the hybrid bearing design, in order to further enhance the control force produced by the electromagnetic actuators and, consequently, the actuation capacity.

The PD controller gains were obtained on a trial-and-error basis. From the point of view of performance, this approach seldom conduces to the best operation condition of the closed-loop system. In this sense, improvements in the controller gain calculation shall be considered. The authors believe that the control technique developed by Li *et al.* (2008), applied to rotating systems, as proposed by Buttini *et al.* (2011), can be useful. This technique is based on the frequency response of the system (no mathematical models are required), and results allow one to calculate the complete set of stabilizing gains of a PD-type controller and select the most appropriate ones in order to attenuate vibrations of the considered system based on frequency response measurements.

5. ACKNOWLEDGEMENTS

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