23rd ABCM International Congress of Mechanical Engineering
December 6-11, 2015, Rio de Janeiro, RJ, Brazil

MODELING AND OPTIMIZATION OF MAGNETIC ACTUATORS TO VIBRATION CONTROL

Rafael Pilotto
Ricardo Ugliara Mendes
Katia Lucchesi Cavalca
Laboratory of Rotating Machinery, Faculty of Mechanical Engineering, University of Campinas
P.O.Box 6122, 200 Mendeleiev Street, 13083-970 Campinas, SP, Brazil
rpilotto89@gmail.com; rumqld@gmail.com; katia@fem.unicamp.br

Abstract. *The use of magnetic actuators for vibration reduction aims to substitute, in some occasions, the magnetic bearings, since the magnetic actuator involves less complex project requirements in its configuration. The present study consists of evaluating how a magnetic actuator behaves while controlling vibration in a flexible metallic beam, modeled using the finite elements method. Due to design restrictions involving considerable flexibility of a thin beam, there are limitations in the positioning of the actuator in the beam optimal position (highest amplitude of vibration). Hence, the magnetic actuator is inserted into the model utilizing electromagnetic theory. Other components are added to the model through the theory of control of mechanical systems, such as: current amplifier, inductive position sensor and a PID controller (Proportional-Integral-Derivative). The actuator and the sensor are placed in different nodes in the computational model, in order to study the effectiveness of the actuator-controller-sensor system depending on its positioning in the beam. The parameters of the PID controller for the entire length of the beam were obtained using the Ziegler-Nichols optimization method for the locations of the beam where the sensor system was positioned and a LPV (Linear Parameter Varying) technique was applied to the deployment of these parameters as a function of the beam length. For the experimental validation, an electromechanical actuator (Shaker) was used to act as an external excitation source in the beam, which generated sinusoidal waves in the natural frequencies of the system, in order to verify the behavior of the actuator/controller in critical conditions. Thus, the efficiency of the magnetic actuator in vibration control is evaluated regarding the sensor position alongside the flexible beam.*

Keywords: *Flexible beam, Adaptive control, Vibration control, PID, LPV*

1. INTRODUCTION

With the emergence of new, and more complex, engineering problems, the development of new technologies becomes mandatory. The many knowledge fields of engineering, independent in the past, nowadays are interrelated in order to develop new solutions and suitable methods to new patterns of testing and analysis. Nevertheless, vibration is still present in mechanical systems and can lead to fatigue and wear, which may cause significant damage to the system. Thus, it is often necessary auxiliary devices to act in parallel to the main system in order to maintain its designed performance.

In this context, the use of electromagnetic actuators arises as a solution to the vibration control problem, once it enables the application of compensating magnetic forces in the system, which have no mechanical interaction with the structure in analysis (Chiba, et al., 2005). The concept of a magnetic actuator allows both external excitation without contact (Furtado, 2008; Mendes, et al., 2013) and the vibration control.

This work aims to evaluate the performance of a PID controller in suppressing the vibration of a beam with high degree of elasticity. The Linear Parameter Varying (LPV) method was used to interpolate the PID controllers obtained with the Ziegler-Nichols optimization technique, for different sensor positions alongside the flexible beam, and construct a function whose domain contains the entire length of the beam, creating an adaptive controller for the system.

The methodology used in this work has potential to be used in a wide range of applications such as in the vibration active control in flexible transmission systems.

2. THEORY

In this section, the working principle of the magnetic actuator is presented, followed by the finite element method of the beam and a review of the PID controller. Finally, there is an introduction to the LPV method used in this work.

2.1. Magnetic Actuator

The magnetic actuator consists of two coils symmetrically positioned regarding the axial direction of the beam as in Fig. 1. The so-called differential assembly is needed once a single coil produces only attraction forces. As described by many authors in the literature (Chiba, et al., 2005; Furtado, 2008; Mendes, et al., 2013), the non-linear magnetic force obtained can be linearized using a bias current and a Taylor series expansion, yielding to Eq. (1). In the resultant magnetic force F_m (Eq. (1)), K_m is a constant which depends on the characteristics of the actuator poles and coils, ε represent a geometric correction factor due to losses in the air-gap, g_0 is the nominal air-gap length, i_0 the bias current, i_x is the control current and x is the displacement of the beam. The force equation can be re-written in terms of the actuator gain (K_A), which relates the control current provided to the actuator and the magnetic force, and the open-loop stiffness (K_x) that correspond to a negative spring and represent the unstable behavior of the magnetic actuator, justifying the need for a closed loop control. The parameters of the magnetic actuator are presented in Table 1.

$$F_m = \varepsilon \frac{4K_m i_0}{g_0^2} i_x + \varepsilon \frac{4K_m i_0^2}{g_0^3} x = K_A \cdot i_x + K_x \cdot x \qquad (1)$$

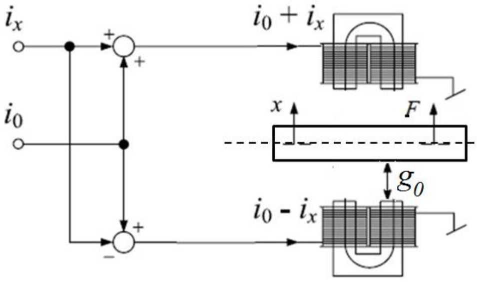


Figure 1. Schematics of the magnetic actuator differential assembly.

Table 1. Values of the magnetic actuator parameters.

Parameter	Value
Correction fator (ε)	0.8
Actuator Gain (K_A)	27.17 N/A
Open-Loop Stiffness (K_x)	243.1 N/m
Bias Current (i_0)	0.8 A
Air-gap (g_0)	$3.5\text{x}10^{-3}$ m

2.2. Finite Element Model of the Beam

The mathematical model of the beam was obtained using the finite element method (Eq. (2)), where **M** and **K** represent, respectively, the global matrices of mass and stiffness (based on the work of Nelson and McVaugh, 1976), and **C** is the structural damping matrix proportional to **K** by a factor of $2\text{x}10^{-4}$. **F** is the external force vector, which contains the magnetic force F_m and the *Shaker* external excitation force F_e. As mentioned before, the objective of the magnetic force is to compensate the effects of the external excitation.

$$\mathbf{M}\ddot{\mathbf{x}} + \mathbf{C}\dot{\mathbf{x}} + \mathbf{K}\mathbf{x} = \mathbf{F} \qquad (2)$$

The 1000 mm length beam was divided into 40 elements (41 nodes) of 25 mm each and a transversal area of 33 mm by 1.3 mm. The pinned ends of the beam were represented by external springs of very high stiffness. Due to the slenderness of the beam a journal of 50 x 55 x 7 mm is placed in the magnetic actuator position to avoid the magnetic field to cross the beam from one coil of the actuator to the other. The beam and the journal are both made of stainless steel, which has a Young modulus of $2{,}1\text{x}10^{11}$ N/m^2 and a specific mass of 7800 kg/m^3.

2.3. PID Controller

The well-known PID controller is composed by three components: proportional, integral and derivative. The proportional term (K_p) is often related to the stiffness of the system, the integral term (K_i) is related to the duration of the error and its magnitude, eradicating the residual stationary error derived from the pure proportional controller. Finally, the derivative term (K_d) is associated with the damping of the system and helps to predict its behavior, improving the settling time and stability. The derivative term has a drawback once it amplifies high frequency measurements, which compromise the controller performance. This limitation can be overcome by adding a filter coefficient (N_f) that acts as a low-pass filter in order to remove high-frequency noise components from the measurements. The controller with the added filter is given by Eq. (3).

$$PID(s) = \frac{\left(K_p + K_d N_f\right)s^2 + \left(K_p N_f + K_i\right)s + K_i N_f}{s^2 + N_f s} \tag{3}$$

The parameters of the PID controllers for each operating condition were obtained using the Ziegler-Nichols technique (Aströn e Hägglund, 1995).

2.4. LPV model using the S.M.I.L.E. technique

After the development of the PID controllers for several sensor positions throughout the beam, this group of controllers can be re-parametrized into a single PID Controller Function of the sensor position in the beam length. In order to accomplish this, a linear parameter varying (LPV) model was obtained using the S.M.I.L.E. (*State-Space Model Interpolation of Local Estimates*) technique. This technique consists in the interpolation of linear time invariant (LTI) systems obtained for the different operating conditions of the system.

In other words, the main objective of this technique is to obtain a continuous and adaptive PID controller. The S.M.I.L.E technique consists in five steps (De Caigny, et al., 2009):

1º - For each operating condition, calculate the poles and zeros of the controller and sort them in a consistent order;
2º - Divide each controller into a gain multiplied by a series connection of first and second order sub-systems;
3º - For each series connection, calculate the gain of the zero-pole-gain representation;
4º - Solve the optimization problem to obtain the matrices of each sub-system LPV model.
5º - Built of the LPV controller model through the series connection of the LPV sub-systems models.

It is important to note that the first step is necessary to ensure that the controllers are consistently represented, so that the optimization problem can be solved. A diagram representing how the SMILE technique was applied in the context of this work is presented in Fig. 2, and a complete description of the method is found in the work of De Caigny, et al. (2009).

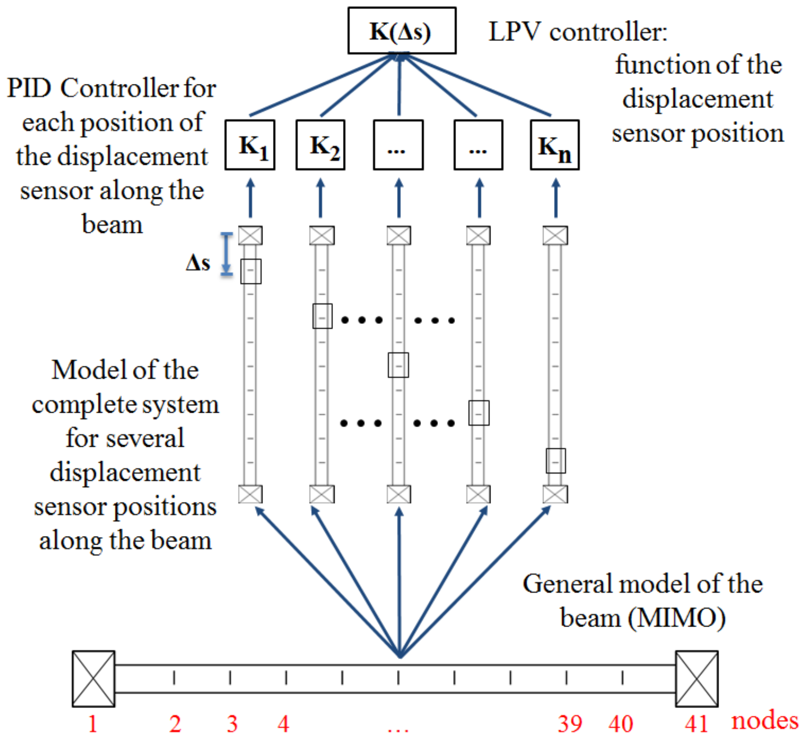


Figure 2. Summary of the S.M.I.L.E Technique for the flexible beam system.

3. NUMERICAL SIMULATIONS

The integration of the components of the system (beam, magnetic actuator, power amplifier and displacement sensor) was performed using a Matlab/Simulink diagram, illustrated in Fig. 3. The sensibility of the displacement sensor is 987 V/m and the power amplifier gain is 1.2231 A/V. The LPV controller was designed from a set of PID controllers calculated via Ziegler-Nichols optimization, as mentioned. In order to obtain the set of PID controllers, the actuator was fixed at the 15 cm mark in the beam and a different PID was obtained for three different positions of the displacement sensor: 15, 35 and 55 cm marks. With the LPV controller synthesis a single controller was obtained as a function of the entire range of possible positions of the displacement sensor.

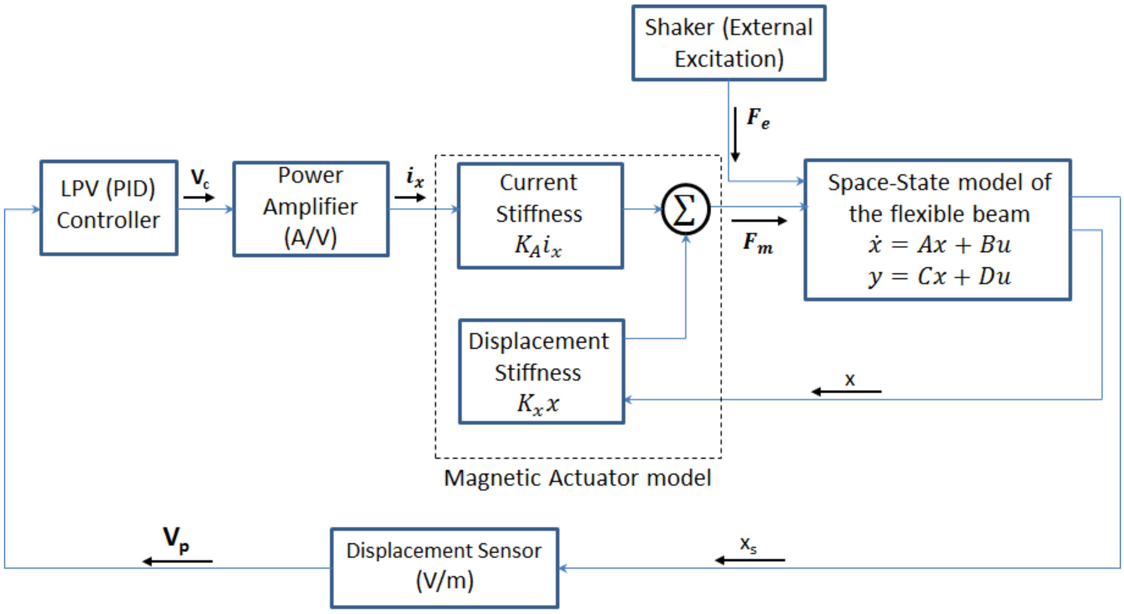


Figure 3. Schematic of the complete system (beam, magnetic actuator, power amplifier and displacement sensor).

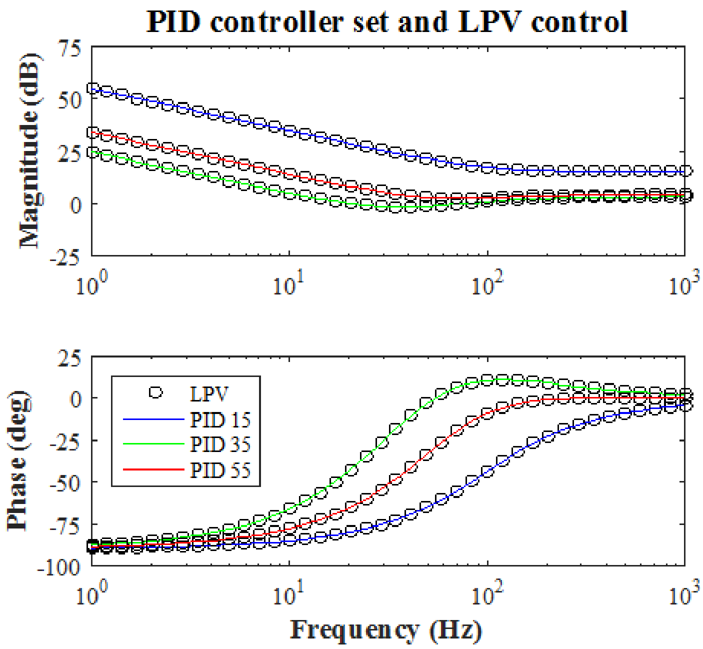


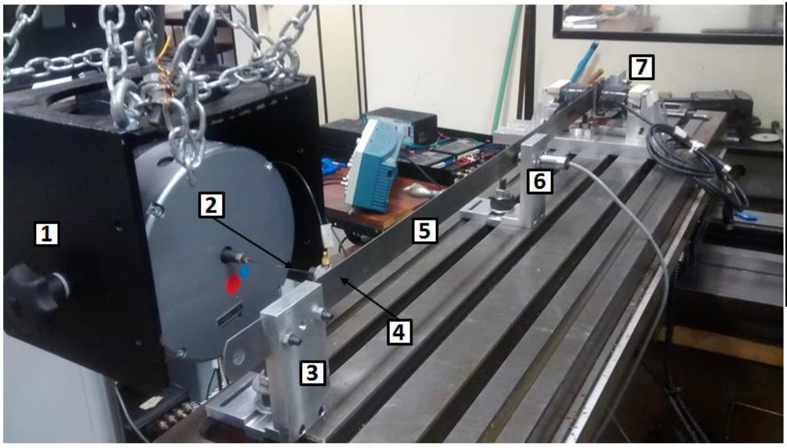
Figure 4. Bode diagram comparing the set of PID controllers obtained and the synthesized LPV controller for different displacement sensor positions (15, 35 and 55 cm marks) along the beam.

Figure 4 presents a comparison, through bode diagram, between the set of PID controllers obtained using the Ziegler-Nichols optimization and the synthesized LPV controller for the same displacement sensor position. It can be noted that the LPV control (black circles) can successfully represent the PID set for the three displacement positions evaluated (15 cm mark – blue line, 35 cm mark – green line, 55 cm mark – red line).

Once the LPV controller was obtained using the complete model of the system, its performance was evaluated via simulations and compared with measurements in the test rig described in section 4, and the results are presented in section 5.

4. TEST RIG

The test rig assembly is presented in Fig. 5. The tests consist of applying a sinusoidal force with the *Shaker* on the beam, and measure the vibration with a proximity sensor from Turck (Bi6-M18-LiU). A National Instruments NI-PCIe-6259 board acquires the sensor signal, which is the PID controller input. The PID controller is simulated in real-time using MATLAB/Simulink and sends the control signal (a voltage) to the PWM power amplifier (via the same NI board). The power amplifier provides the correspondent current to the magnetic actuator coils, which generates the force to control the beam vibration (equivalent to Fig. 3).



1. *Shaker*
2. *Stinger*
3. Beam support
4. Force transducer
5. Flexible beam
6. Magnetic Actuator
7. Displacement Sensor

Figure 5. Test rig assembled with the *Shaker*.

Figure 6 present a detailed view of the beam where the numeric marks were used as a reference for positioning of the magnetic actuator and the sensor. As the center of the beam present high vibration levels due to its flexibility, the actuator and its journal had to be placed at the 15 cm mark of the beam. The beam vibration at the journal position must be smaller than the air-gap for the correct operation of the actuator. The *Shaker*, responsible for the external excitation, was connected at the 90 cm mark.

Figure 6. Details of the numeric marks on the beam surface.

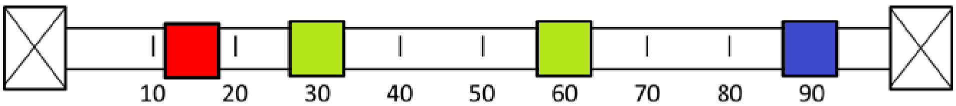


Figure 7. Position of the system components along the flexible beam on the experimental tests: magnetic actuator/journal on 15cm mark (red square); *Shaker* on the 90 cm mark (blue square); position sensor was placed on two different locations for different tests, 30 cm and 60 cm marks (green squares).

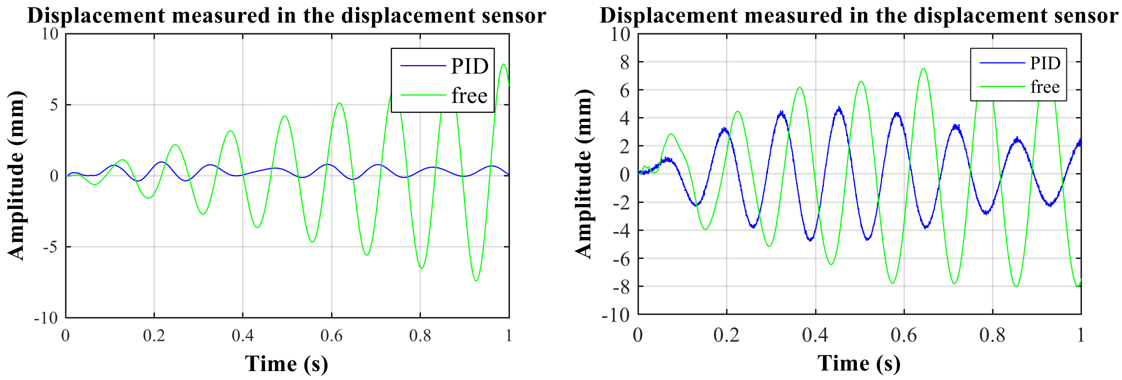
5. RESULTS AND DISCUSSION

In order to evaluate the performance of the LPV controller designed a series of tests were made. The *Shaker* generated an excitation sinusoidal signal in the frequency of the first three natural frequencies of the beam, and for each frequency, two cases were evaluated: with the sensor placed at the 30 cm mark and at the 60 cm mark; totalizing six tests. The disposition of the system elements in the tests are represented in Fig. 7, where the actuator is represented by the red square, the Shaker by the blue one and the sensor positions are the green squares.

Figures 11 to 16 present the results of the simulations and the measurements at the test rig for each of the six tests previously described. In each figure, the green lines correspond to the system response without the control (called "free" in the legend), and the blue line correspond to the system with the LPV PID control (called "PID").

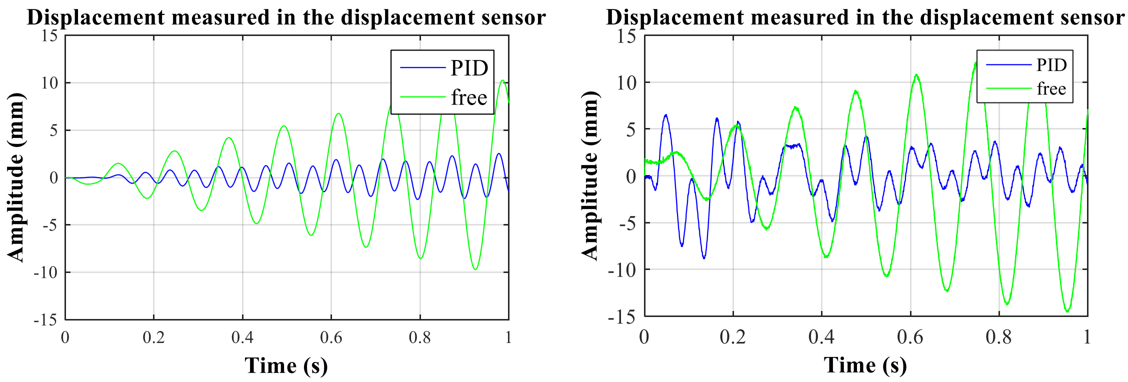
When the system was excited in the first two natural frequencies, the designed control could successfully reduce the vibration amplitude. However, in the third natural frequency, the result was the opposite, the control increased the vibration amplitude of the beam. The reason for the differences between the simulated results and the measurements is probably due to model limitations. The FEM model assume a small displacement condition, which may not be very suitable for the long flexible beam used; and this problem becomes even more significant when the system vibrates in a resonant condition as presented in this work, when the system is submitted to a harmonic excitation in its natural frequencies.

The fact that the sensor does not measure the vibration at the same point of the beam where the actuator applies the control force consists of a non-collocated control problem. An inherent disadvantage of such systems is that controller performance is limited and, depending on the combination of the sensor position, the mode shapes and controller gains, the controller can destabilize the system. In the system analyzed in this work, this phenomenon can be clearly seen in the third mode response with the sensor in the 60 cm mark (Fig. 16), where the control increase the vibration amplitude of the system instead of control it.



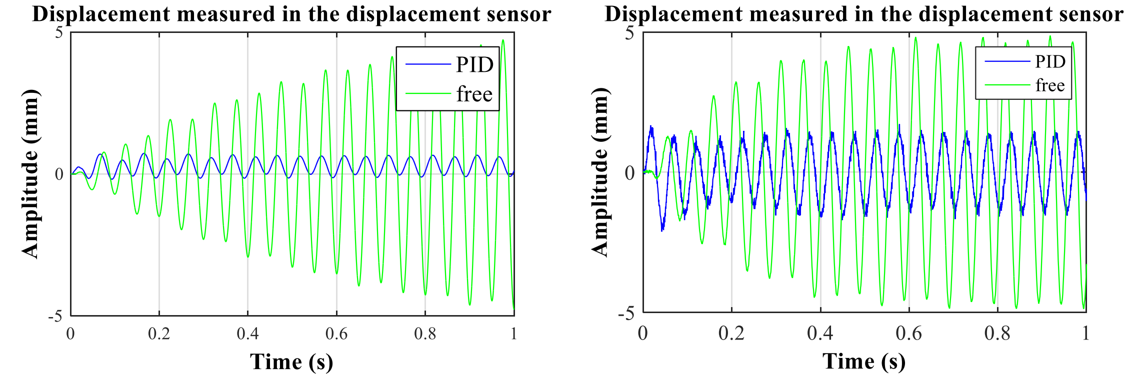
(a) (b)

Figure 11. Free system and the controlled system responses to *Shaker* excitation on the 1st natural frequency (displacement sensor at the 30 mark): (a) simulation; (b) measurements.



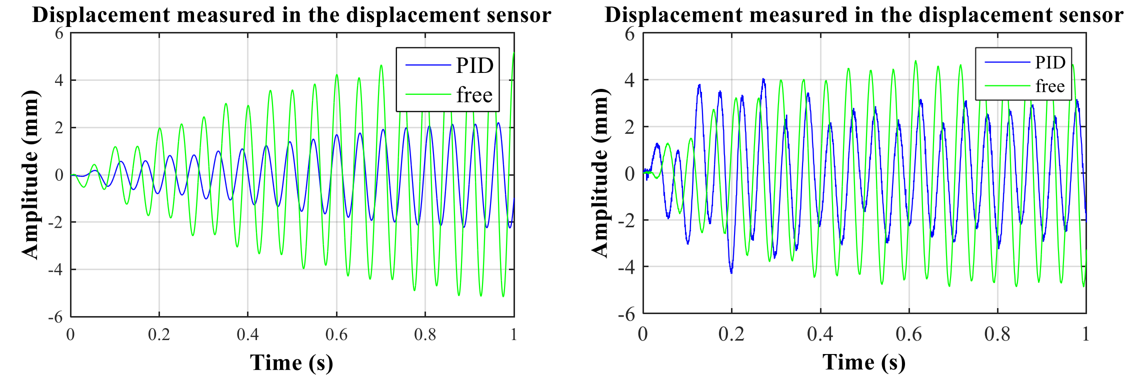
(a) (b)

Figure 12. Free system and the controlled system responses to *Shaker* excitation on the 1st natural frequency (displacement sensor at the 60 mark): (a) simulation; (b) measurements.



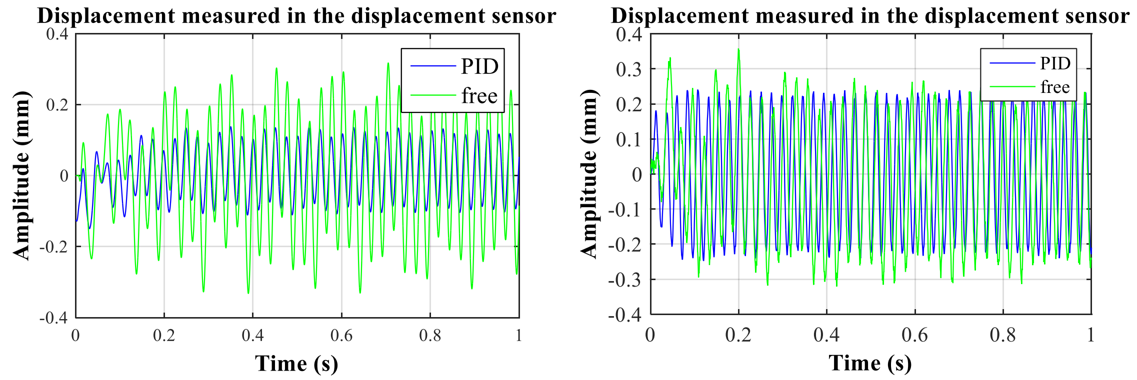
(a) (b)

Figure 13. Free system and the controlled system responses to *Shaker* excitation on the 2nd natural frequency (displacement sensor at the 30 mark): (a) simulation; (b) measurements.



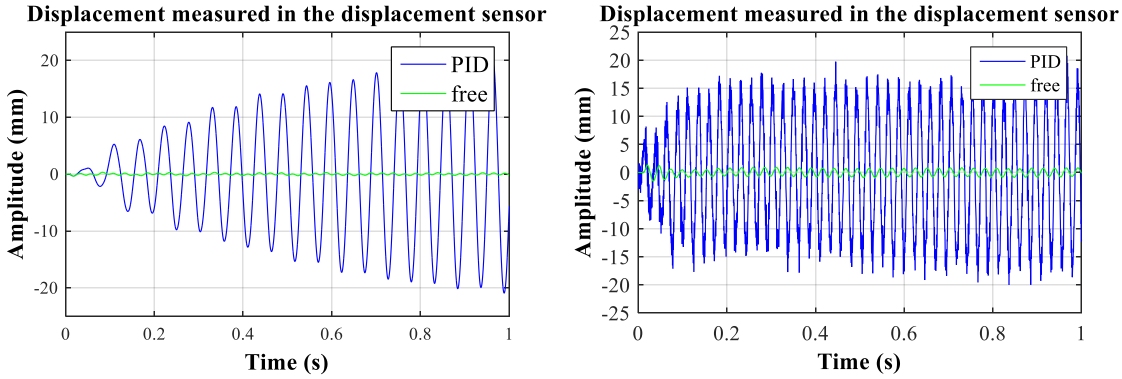
(a) (b)

Figure 14. Free system and the controlled system responses to *Shaker* excitation on the 2nd natural frequency (displacement sensor at the 60 mark): (a) simulation; (b) measurements.



(a) (b)

Figure 15. Free system and the controlled system responses to *Shaker* excitation on the 3rd natural frequency (displacement sensor at the 30 mark): (a) simulation; (b) measurements.



(a) (b)

Figure 16. Free system and the controlled system responses to *Shaker* excitation on the 3rd natural frequency (displacement sensor at the 60 mark): (a) simulation; (b) measurements.

6. CONCLUSIONS

The adaptive controller obtained through the combination of the PID set optimized via Ziegler-Nichols method with the LPV technique is promising for the vibration control of the flexible beam. It is noticeable that the magnetic actuator has its best results when the displacement sensor is closer to the actuator than when the sensor is closer to the point where the external excitation is applied; mainly because the system becomes closer to a collocated control condition when the actuator and the sensor are closer to each other.

Nevertheless, some improvements still need to be done such as to develop a more suitable model for the high vibration amplitudes present in the long and flexible beam and the study of using a sensor to measure the beam displacement in the actuator position to avoid the non-collocated condition. If a collocated condition is not feasible, a deeper study must be carried out and a different procedure, other than the Ziegler-Nichols method, must be used in order to adjust the PID gains before applying the LPV method.

7. ACKNOWLEDGEMENTS

The authors would like to thank SAE/Unicamp, CNPq and Schaeffler Brazil for the financial support during the project.

8. REFERENCES

Aström, K.J. and Hägglund, T., 1995. *PID Controllers: Theory, Design and Tuning*. Instrument Society of America, Research Triangle Park, 2nd edition.

de Caigny, J., Camino, J.F. and Sweyers, J., 2009. "Interpolating model identification for SISO linear parameter-varying systems". *Mech. Syst. Signal Process.*, Vol. 23, n. 8, p. 2395-2417.

Chiba, A., Fukao, T., Ishicawa, O., Oshima, M., Takemoto, M. and Dorrell, D.G., 2005. *Magnetic Bearings and Bearingless Drives.* Newnes, Burlington, 1st edition.

Furtado, R.M., 2008. *A magnetic actuator development for contactless excitation in rotor systems*. Ph.D. thesis –, Universidade Estadual de Campinas, Campinas. In Portuguese.

Mendes, R.U., Cavalca, K.L. and Ferreira, L.O.S., 2013. "Analysis of a Complete Model of Rotating Machinery Excited by Magnetic Actuator System". *Proceedings of the Institution of Mechanical Engineers, Part C: Journal of Mechanical Engineering Science*, Vol. 227, n. 1, p. 48-64.

Nelson, H.D. and McVaugh, J.M., 1976. "The Dynamics of Rotor-Bearing Systems Using Finite Elements". *Journal of Engineering for Industry*, Vol. 98, p. 593-600.

9. RESPONSIBILITY NOTICE

The authors are the only responsible for the printed material included in this paper.