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ROTATING MACHINE CRITICAL SPEED SUPPRESSING BY USING ELECTROMAGNETIC ACTUATORS

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ABSTRACT

The possibility of suppressing critical speeds by using electromagnetic actuators (EMAs) is assessed numerically and experimentally in this paper. The system studied is composed of a horizontal flexible shaft supported by two ball bearings at one end and a roller bearing that is located in a squirrel cage at the other end. Four identical EMAs supplied with constant current are utilized. The EMAs associated to the squirrel cage constitutes the hybrid bearing. Results obtained, show that the constant current, when applied to the EMAs, produces a shift of the first critical speed toward lower values. Moreover, the application of constant current for a speed interval around the critical speed enables a smooth run-up or run-down without crossing any resonance.

INTRODUCTION

Modern rotating machinery becomes lighter and operates with higher speed due to the enhancement of its performance and efficiency. Consequently, the number of critical speeds included in the operating frequency range increases, and more sensitive to external excitations, also it requires more stable operating conditions and the attenuation of the generated vibrations. These vibrations are responsible for the performance degradations and also the excessive acoustic noises and fatigue-related damages.

Due to the rapid development in actuator technology, electronics, sensors, signal processing and machine design, the smart machine technology is an actual research topic for mechatronic and engineering (1, 2).

Rotating machines mounted on Active Magnetic Bearings (AMB) are representative of tendency of developing smart machine and have been successfully applied in industrial turbomachinery (3). Their main advantages are contact-less working environments, no sealing constraints, frictionless suspension, and their capacity to operate in active systems. They are well-suited for contactless operation as actuators and sensors in rotating machinery (4-9). AMB devices in conjunction with conventional support bearings are utilized either as active magnetic dampers (10), or for controlling the instability (11 and 12). In this case the AMB is considered as an ElectroMagnetic Actuator (EMA).

The work presented in this paper is part of a research program aimed at controlling the dynamic behavior of rotating machinery when the latter crosses critical speeds and instability zones. In a previous work (13), the influence of EMAs on the frequency response of a harmonically excited cantilever beam is investigated numerically and experimentally. The intensity of the current generating the EMAs force was varied and its effect on the dynamic behavior of the system was analyzed. Results indicated that EMAs produce a softening behavior in the system, the resonance curve shifts toward smaller values. The aim in this paper is to analyze the dynamic behavior of a rotating machine in the presence of EMAs supplied with constant current. The softening effects of EMA is well known, the objective here is to use those effects to make it simpler to cross a critical speed. The developed approach is not an alternative for the active control, which is necessary to attenuate the effects of the external excitations.

The paper is organized as follow: the test rig is first described then; the designed EMAs are presented and identified experimentally and finally the results stemming from numerical simulations and experimentations are discussed.

EXPERIMENTAL SET-UP

The system studied (Fig. 1) is a test machine composed of a horizontal flexible shaft of 0.04 m diameter containing two rigid discs. The rotor is driven by an electrical motor that can accelerate the shaft until the rotation of 10,000 rpm including 2 first critical speeds. In this study, only the first critical speed is considered. The shaft is supported by bearings located at its ends, as follows: a roller bearing (B2) at one end and two ball bearings at the other end (B1). The roller bearing is located in a squirrel cage attached to the framework of the test bench by three identical flexible steel beams. The Electro-Magnetic Actuators (EMAs) located on the external cage constitutes a smart active bearing and provides nonlinearity in the dynamics of the system.

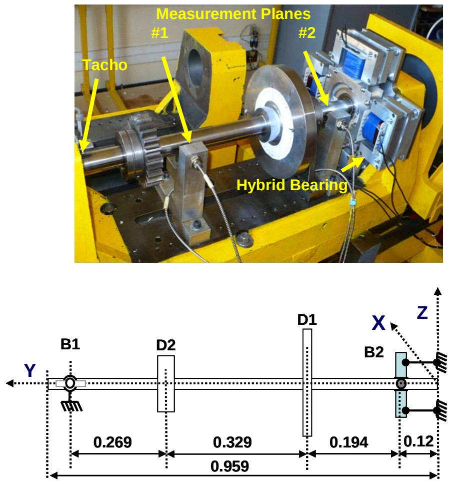

Figure 1. Experimental test rig

The displacements are measured by using four proximity sensors (Vibrometer TQ 103) arranged perpendicularly in two measurement planes located along the y axis, namely, measurement plane 1 and measurement plane 2 (Fig. 1). The sensors are labeled C1 and C4 for the horizontal direction and C2 and C3 for the vertical direction.

The data acquisition device used to collect experimental data was the SCADAIII interface of LMS® that enables real time data acquisition. Several codes of LMS® modal analysis software were used for data processing. The sampling frequency was set to 4096 Hz.

ACTUATORS IDENTIFICATION

EMAs are designed to deliver a maximum attraction force of 300 N, and the maximum possible current is 5.0 A. The control input could be either a current or a voltage. For practical reasons, aiming at simplifying the electrical EMA model, a current control configuration was utilized.

Since an EMA can only produce attractive forces, Four "identical" EMA supplied by constant currents are utilized. Each EMA is composed of a ferromagnetic circuit and an electrical circuit. The ferromagnetic circuit has two parts: an (*E*) shape, which receives the induction coil, and an (*I*) shape, which is fixed to the squirrel cage. Both parts are made of sets of insulated ferromagnetic sheets. The quality of the ferromagnetic circuit alloy is considered high enough and the nominal air gap between the stator and the beam is small enough to consider magnetic loss as negligible. The geometries of the actuators are summarized in figure 2.

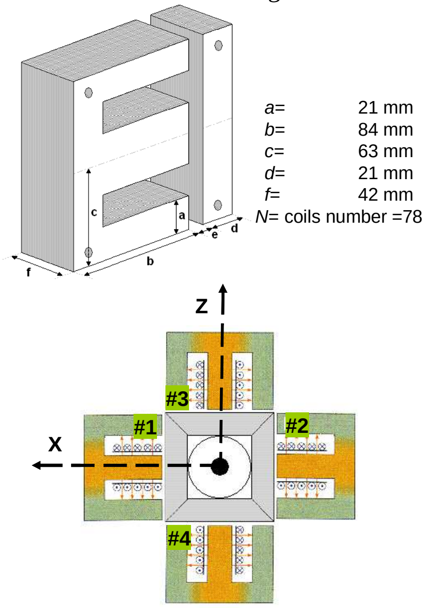

Figure 2. EMA details

Assuming negligible eddy current effects and conservative flux, the relationship between the electromagnetic force (F_{em}), air gap (e), gap distance (δ_a) and current (I) can be expressed as (14):

$$F_{em} = \frac{N^2 \mu_0 \; a \; f \; I^2}{2\left((e \pm \delta_a) + \dfrac{b+c+d-2a}{\mu_r} \right)^2} \tag{1}$$

where (a, b, c, d and f) correspond to the geometrical characteristics of the actuator and μ_0 is the magnetic permeability of a vacuum ($4\pi \times 10^{-7}$ H/m). μ_r is the relative magnetic permeability (dimensionless) that is a function of the air gap and can be varied according to temperature. Its value is based on manufacturer's specifications and is generally not known with great accuracy and has to be identified.

In this model, the inputs are the current and the gap distance; the output is the force. The geometrical parameters could be measured precisely; the only unknown is the relative permeability that has to be determined experimentally. The relative permeability is determined as:

$$\mu_r = \frac{\beta \, (b + c + d - 2a)}{1 - \beta e} \qquad \beta = \sqrt{\frac{2 F_{em}}{N^2 \mu_0 \; a \; f \; I^2}} \tag{2}$$

A specific experimental arrangement was realized (Fig. 3) in order to measure the force generated by the actuator for several air gaps and for increasing and decreasing values of the input current.

Figure 3. Experimental arrangement for the actuator identification

The generated forces due to increasing and decreasing input currents are measured for several air gaps (Fig. 4).

The results obtained show that the hysteresis effect (due to electromagnetic flux) appears to be negligible and the generated forces are proportional to the current square value.

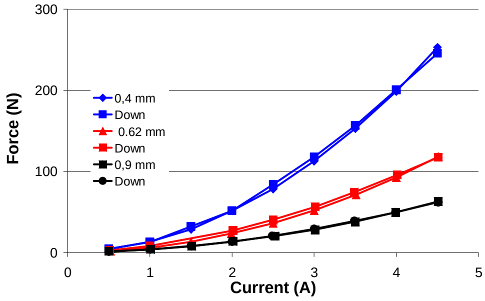

Figure 4. Measured forces versus current for several air gaps

In the model presented here, the relative permeability is assumed to be constant for low flux density. The mean value determined for the model is 950.

NUMERICAL SIMULATIONS

The rotor model is composed by the following elements: rigid discs that contribute kinetic energy only; flexible shaft providing both kinetic and strain energy; and bearings with elastic and dissipation characteristics. The shaft is modeled by beam elements with two nodes and 4 d.o.f. per node, namely two displacements (along the directions x and z) and two rotations (around the axes x and z), respectively (15). The electromagnetic actuators are considered by the simplified model of equation (1) with constant current of 3.5 A.

The model was discretized according to 43 nodes as shown in figure 5.The ball bearings (B1) are located at nodes # 4 and # 5 and the bearing containing the electromagnetic actuator (B2) at node # 39. The first disc (D1) is placed between the nodes # 12 and #16; the second disc (D2) is located between the nodes # 28 and #30. Finally, concentrated masses were included in the model at the position of the bearings and at the coupling between the shaft and the driving motor. It is worth mentioning that only the first eight vibration modes were taken into account in the calculation of the response.

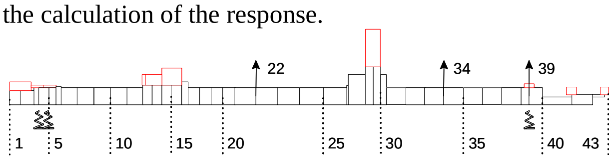

Figure 5. Finite element model of the test rig

Initial unbalances were added for simulation purposes as follow: 150 g.cm amplitude and 0° angular position at the node #15, and 120 g.cm amplitude and 0° angular position at the node # 30. The displacement amplitude for a run-up from 100 to 3500 rev/min in 30 seconds with linear speed variation is calculated and compared for the configurations 0 and 3.5 A constant current (Fig. 6).

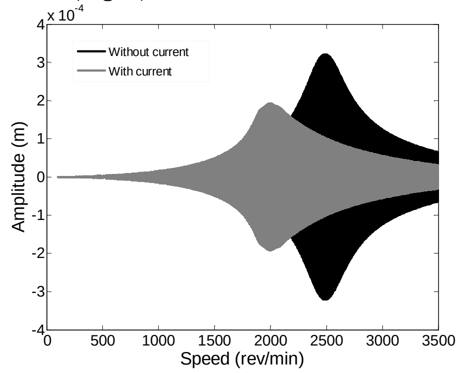

Figure 6. Displacement amplitude, run-up, node #22, Z direction, 3.5 A

It could be noticed that the constant current produces a shift of the first critical speed toward lower values. This constant current has a "softening" effect on the dynamic behavior. The same trends were observed for the run-down.

This behavior leads to the conclusion that a constant current applied on the electromagnetic actuators in the vicinity of the critical speed during a run-up could modify the dynamic behavior of the test rig such that the rotor will not cross any resonance.

EXPERIMENTATIONS

Experiments are carried out in order to assess the dynamic behavior with and without constant current. Actuators are mounted with the same air gap of 0.6 mm. Two configurations are considered in this paper: without current (0 A) and with constant current (3.5 A).

The amplitude of the displacements during a run-up from 800 to 4000 rev/min in 90 seconds with linear speed variation was measured; the displacements stemming from sensor C2 are presented in figure 7. Here also, it could be noticed that the constant current produces a shift of the first critical speed toward lower values. The same trends were observed for the run-down.

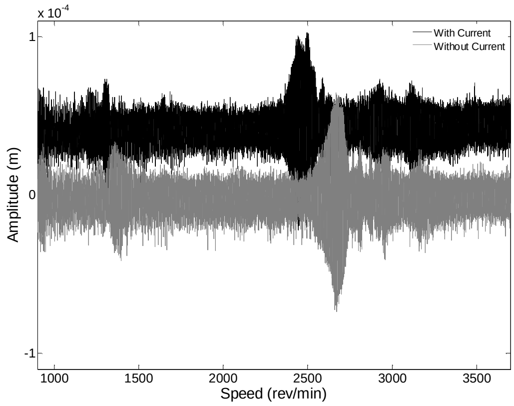

Figure 7. Displacement amplitude, C2, run-up, current 0 and 3.5 A

The second step consists on the application of the current only in the vicinity of the critical speed. The actuators are supplied with constant currents (3.5 A) when reaching the speed of 2400 rev/min until the speed of 2800 rev/min, where the currents are switched off.

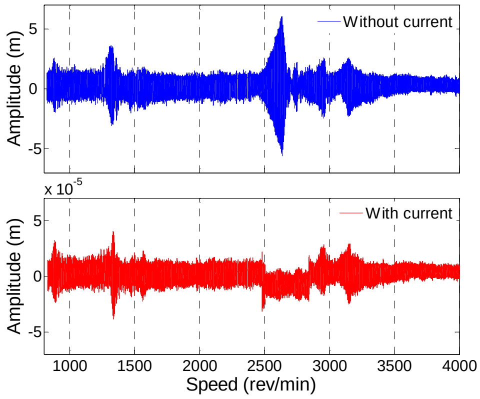

Figure 8. Displacement amplitude, C2, run-up, current 0 and 3.5 A

The displacements stemming from sensor C2 during the run-up from 800 to 4000 rev/min in 90 seconds with linear speed variation are presented in figure 8.

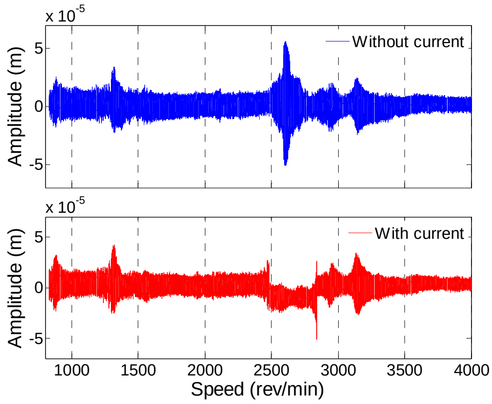

Figure 9. Displacement amplitude, C2, run-down, current 0 and 3.5 A

It could be noticed that the rotor does not cross the critical speed during the run-up. The same behavior was also noticed during the run-down (Fig. 9).

The electromagnetic actuators, utilized in this configuration, provide a "smooth" run-up or run-down and could enable the suppression of critical speeds. It is worth mentioning that only the first critical speed was monitored in this study. The position of the hybrid bearing corresponds to a vibration node for the second mode; consequently the electromagnetic actuators have no effects on the dynamic behavior.

CONCLUSIONS

The possibility of suppressing critical speeds by using electromagnetic actuators is assessed in this study. The approach consists on introducing "when needed" a softening behavior during operation so that the rotating machine does not cross any critical speeds during run-up or run-down.

The constant current, when applied to the EMAs, produces a shift of the first critical speed toward lower values. The application of constant current for a speed interval around the critical speed enables a smooth run-up or run-down without crossing any resonance.

In this study, only the first critical speed was monitored. The position of the hybrid bearing corresponds to a vibration node for the second mode; consequently the electromagnetic actuators have no effects on the dynamic behavior. This approach requires a good knowledge of the dynamic behavior of the studied system in order to apply the current for the suitable speed range. It was noticed that the produced dynamic behavior depends on (and is sensitive to) the current intensity and the air gap value.

The amplitude response increases when applying the current (step input), on the other hand introducing the current with a given rate may introduce instability. Researches are going on in order to optimize the dynamic behavior as a function of the current intensity and the air gap values.

This study is an additional interesting utilization of EMA and is not an alternative for the active control, which is necessary to attenuate the effects of the external excitations.

REFERENCES

[1] Larsonneur, R., Richard, P., 2008, Smart Turbomachines using Active Magnetic Bearings, Paper GT2008-51299, ASME Turbo Expo, Berlin, Germany, June 9-13.

[2] Maslen, E. H., 2008, Smart Machine Advances in Rotating Machinery. IMechE, Exeter, UK, 8-10 September.

[3] Schweitzer, G., Bleuler, H., Traxler, A., 2003, Active Magnetic Bearings - Basics, Properties and Applications, vdf Hochschulverlag AG, ETH, Zurich.

[4] Kulesza, Z., Sawicki, J.T., and Storozhev, D.L., 2010, Smart Properties of AMB Supported Machines for Rotor Crack Detection: Experimental and Analytical Study, Proceedings of

the 8th IFToMM International Conference on Rotordynamics, September 12-15, KIST, Seoul, Korea.

[5] Lei, S., & Palazzolo, A., 2008, Control of flexible rotor systems with active magnetic bearings, Journal of Sound and Vibration , doi:10.1016/j.jsv.2007.12.028.

[6] Kasarda, M., Marshall, J. and Prins, R., 2007, Active magnetic bearing based force measurement using the multi-point technique, Mechanics Research Communications, 34, pp. 44-53.

[7] Mani, G., Quinn, D.D. and Kasaeda, M., 2006, Active health monitoring in a rotating cracked shaft using active magnetic bearings as force actuators, Journal of Sound and Vibration, 294, 454-465.

[8] Aenis, M., Knopf, E. and Nordmann, R., 2002, Active magnetic bearings for the identification and fault diagnosis in turbomachinery, Mechatronics, 12, 1011-1021.

[9] Kasarda, M., Mendoza, H., Kirk, R.G. and Wicks, A., 2004, Reduction of subsynchronous vibrations in a single-disk rotor using an active magnetic damper, Mechanics Research Communications, 31, pp. 689-695.

[10] Souza Morais, T., Steffen Jr, V., Mahfoud, J., Der Hagopian, J., 2010, Unbalance Identification in Nonlinear Rotors, Proceedings of the 8th IFToMM International Conference on Rotordynamics, September 12-15, KIST, Seoul, Korea

[11] El-Shafei, A., and Dimitri, A.S., 2007, "Controlling journal bearing instability using active magnetic bearings", Proceedings of ASME Turbo Expo, GT2007-28059, Canada.

[12] Souza Morais, T., Steffen Jr, V., Mahfoud, J., Der Hagopian, J., 2009, Monitoring Cracked Shaft By Using Active Electro-Magnetic Actuator - Numerical Simulation, 20th International Congress of Mechanical Engineering, COBEM 2009, November 15-20, Gramado, RS, Brazil.

[13] Der Hagopian, J., Belhaq, M., Seibold, Z. et Mahfoud, J., 2010, Maîtrise de Phénomènes de Saut dans les Réponses Fréquentielles des Structures avec Non Linéarité Localisée, Machines et Industries, DOI: 10.1051/meca/2010.

[14] Mahfoud, J. and Der Hagopian, J., 2010, Fuzzy Active Control Of Flexible Structures By Using Electromagnetic Actuators, ASCE's Journal of Aerospace Engineering, DOI: 10.1061/(ASCE)AS.1943-5525.0000067.

[15] Lalanne, M. and Ferraris, G., 1998, Rotordynamics Prediction in Engineering, John Wiley and Sons, Second Edition.