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EXPERIMENTAL FUZZY MODAL CONTROL IN FLEXIBLE ROTOR USING ELECTROMAGNETIC ACTUATOR

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Abstract. *The present contribution is dedicated to the Electromagnetic Actuator (EMA). EMA uses electromagnetic forces to control the rotor vibration without mechanical contact. Due to the size of the system model, it was necessary to reduce the model of the rotating system. EMA is represented by a nonlinear model, which justifies the use of Fuzzy Logic Control. The FLCs were found to be well adapted for controlling structures with nonlinear behavior whose characteristics change considerably with respect to time. Since the modal states cannot be accessed directly during experimentation, they have to be rebuilt by using an estimator algorithm. Consequently, for determining the modal states a Kalman estimator is used. Finally, experimental results demonstrate the effectiveness of the methodology conveyed.*

Keywords: *Fuzzy Modal Control, Electromagnetic Actuator, Flexible Rotor*

1. INTRODUCTION

Currently an increase of research effort in engineering devoted to the development of new Active Vibration Control techniques (AVC) is observed. A number of types of actuators are available for active vibration control purposes, such as Piezoelectric Stack Actuators, Active Magnetic Bearings (AMB), and Electromagnetic Actuators (EMA). The Piezoelectric Actuators demonstrated to be very effective in rotor vibration control (Palazzolo et al., 1993). More recently (Simões et al., 2007) presented an active modal control strategy for flexible rotors by using two orthogonal PZT stack actuators installed at one of the supports of the machine, i.e., the PZT actuators were attached directly to the ball bearing, which allows the insertion of stiffness to the system. The AMB have been successfully applied in industrial turbo machines (Schweitzer et al, 2009). The AMB provide control effort through the application of lateral forces without mechanical contact between the rotor and the stator. However, they have some disadvantages, technical complexity and continuous power consumption (Horst et al., 2004), which is due to rotor support requirements.

The EMA uses the same physical principle of the AMB, but only in terms of the lateral force without contact, since the EMA is not used to support the rotor. The use of EMA results in a hybrid bearing. The active control using EMA was achieved successfully both numerically and experimentally in light structures (Der Hagopian et al., 2010). The advantage of EMA is due to the simple electromechanical structure associated with the control action, without mechanical contact. Koroishi et al (2014) used the hybrid bearing in active vibration control in flexible rotor.

Among well-known techniques for AVC, nowadays the Active Vibration Modal Control (AVMC) is a highlight, presenting successful results in various applications (Huberlaus et al., 2008; Li, 2011; Houlston et al., 2007; Yu et al., 2007). Modal control requires that the size of the structure model has to correspond to the number of modes to be controlled. The advantage of using AVMC is that this technique exhibits the most effective type of control for flexible structures, implementing a reduced numbers of actuators and sensors (Balas, 1978). The modal controllers can be designed by using many techniques, such as optimal (LQR, LQG) (Simões et al, 2007), robust controls (H_∞ norm) (Yu et al., 2007) and Fuzzy Logic Control (FLC).

Fuzzy Logic Controllers (FLCs) are also used in AVC problems. Zadeh [12] presented fuzzy logic as a technique to control a complex process based on expertise based rules of type "IF…THEN". This type of controller is adapted to taking into account slight nonlinearities and uncertainties due to the experimental procedures [7]. A new technique based on FLC is the Fuzzy Modal Control, which combines the advantages of FLC with those from the Modal Control, thus joining simplicity of implementation and robustness.

The aim of this work is to study both the effectiveness and the robustness of the controller to attenuate rotor vibration by using FLC for active modal vibration control. For this aim, experimental test are performed to analyze the control approach performed. This work is divided in five sections. After a brief introduction, the rotor model is defined by using the Finite Elements Method (FEM); the characteristics of the Electromagnetic Actuator (EMA) are presented; the control approach is discussed; experimental investigations are performed; finally, conclusions are summarized.

2. FLEXIBLE ROTOR SYSTEM

The rotor´s model was developed based on the rotor test rig presented in Fig. (1).

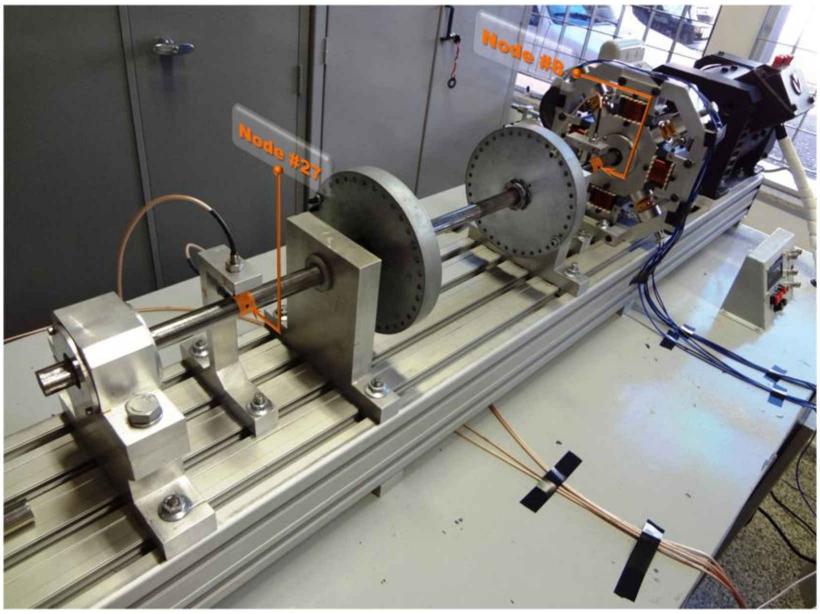


Figure 1. Rotor Test Rig (Koroishi et al., 2014).

Figure 2 presents the FE model in which 32 finite elements are used to represent the dynamic behavior of the rotor. It is composed of a flexible steel shaft with 80 mm length (E_{steel} = 2.05 X 1011 Pa, ρ_{steel} = 7850 kg/m^3, and □ steel = 0.3), two rigid steel discs located at the nodes #13, and #22, respectively, and two roller bearings (B_1 and B_2 located at the nodes #4 and #31, respectively). Displacement sensors are orthogonally mounted (along the horizontal and vertical directions) at the nodes #8 and #27 to collect the shaft vibration. The first twelve vibration modes were used to generate the displacement responses of the rotor model along the orthogonal directions. The Electromagnetic Actuators (EMAs) are located at the node #4, which corresponds to the position of the bearing B_1. The properties of the rotor are presented in Tab. (1).

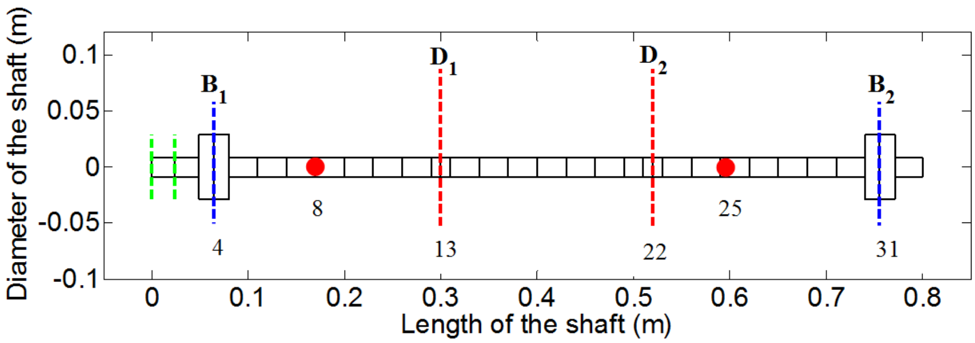


Figure 2. FE model of the rotor (- - - - Bearing; - - - - Disc; - - - - Coupling # node).

The dynamic response of the considered mechanical system can be modeled by using principles of variational mechanics, namely the Hamilton`s principle. For this aim, the strain energy of the shaft and the kinetic energies of the shaft and discs are calculated. An extension of Hamilton`s principle make possible to include the effect of energy dissipation. The parameters of the bearings are included in the model by using the principle of the virtual work. For computation purposes, the FEM is used to discretize the structure so that the energies calculated are concentrated at the nodal points. Shape functions are used to connect the nodal points. To obtain the stiffness of the shaft the Timoshenko`s beam theory was used and the cross sectional area was updated. The model obtained as described above is represented mathematically by a set of differential equations (Lalanne & Ferraris, 1998) as given by Eq. (1).

$$[M]\{\ddot{\delta}(t)\}+[C_b+\dot{\phi}C_g]\{\dot{\delta}(t)\}+[K+\phi K_g]\{\delta(t)\}=\{F(t)\}+\{F_{EMA}(t)\} \tag{1}$$

where $\{\delta(t)\}$ is the vector of generalized displacements; [*M*], [*K*], [C_b], [C_g] e [K_g] are the well-known matrices of inertia, stiffness, bearing viscous damping (that may include proportional damping), gyroscopic (with respect to the speed of rotation), and the effect of the variation of the rotation speed; $\dot{\phi}$ is the time-varying angular speed, and {*F(t)*} e {$F_{EMA}(t)$} are the forces due to the unbalance and to the electromagnetic actuator, respectively.

Table 1. Physical characteristics of the bearings.

Characteristic	**Value**	**Characteristic**	**Value**
Rotor		**Bearings**	
Mass of shaft (kg)	4.1481	k_{x1} (N/m)	$7.73X10^5$
Mass of disc D_1 (kg)	2.6495	k_{z1} (N/m)	$1.13X10^5$
Mass of disc D_2 (kg)	2.6495	k_{x2} (N/m)	$5.51X10^8$
Thickness of D_1 (m)	0.1000	k_{z2} (N/m)	$7.34X10^8$
Thickness of D_2 (m)	0.1000	C_{x1} (N.s/m)	5.7876
Diameter of shaft (m)	0.0290	C_{z1} (N.s/m)	12.6001
		C_{x2} (N.s/m)	97.0231
		C_{z2} (N.s/m)	77.8510

The bearing located in node #4 corresponds to the hybrid bearing that contains the electromagnetic actuator, showed by fig. (3).

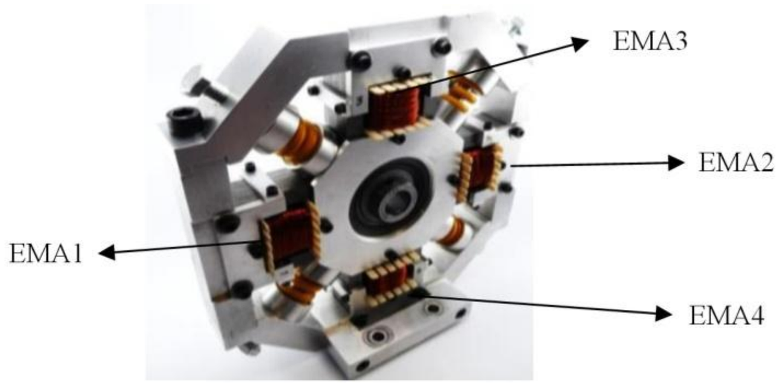


Figure 3. The structure of actuator in the rotor.

Four EMAs are used in the system, i.e., two for each control direction (x and z). The EMA applies only attraction force and each actuator acts separately. The ferromagnetic circuit used by each actuator is presented in Fig. (4). The geometry and properties are presented in Tab. (2).

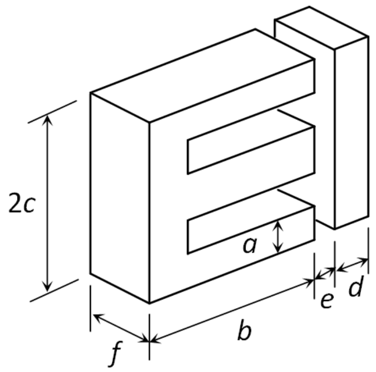


Figure 4. Ferromagnetic circuit (Koroishi et al., 2014).

Table 2. Parameters of the coil.

μ_0 (H/m)	$1.26 X10^{-06}$
μ_r	700
N (spires)	250
a (mm)	9.50
b (mm)	38.0
c (mm)	28.5
d (mm)	9.50
f (mm)	22.5
e (mm)	0.5

The EMA is used to apply the control force to the rotor system. The forces provided by the EMA are inversely proportional to the square of the sum of nominal gap and displacement. With these characteristics, each coil applies a force that is given by Eq. (2).

$$F_{EMA} = \frac{N^2 I^2 \mu_0 a f}{2\left[(e \pm \delta) + \frac{b+c+d-2a}{\mu_r}\right]^2} \tag{2}$$

3. FUZZY MODAL CONTROLLERS

In the present contribution, two membership GBELL Matlab@ functions "Positive" and "Negative" are used (Mahfoud et al., 2011). The inference engine implements the "minimum" function w_j. The rules are given in Tab. 3.

Table 3. Fuzzy controller rules.

Rule	Condition	Decision
1	IF positive displacement AND positive velocity	Action
2	IF positive displacement AND negative velocity	No action
3	IF negative displacement AND positive velocity	No action
4	IF negative displacement AND negative velocity	Action

The Takagi-Sugeno method is used due to its well adaptation to the controller real time computation. The membership functions for the modes are similar ($z_1 = 0$ corresponds to "no action" and $z_2 = \alpha_i\, q_i + \beta_i\, \dot{q}_i$ to "action", where, α_i and β_i are, respectively, displacement and velocity constants for mode i.

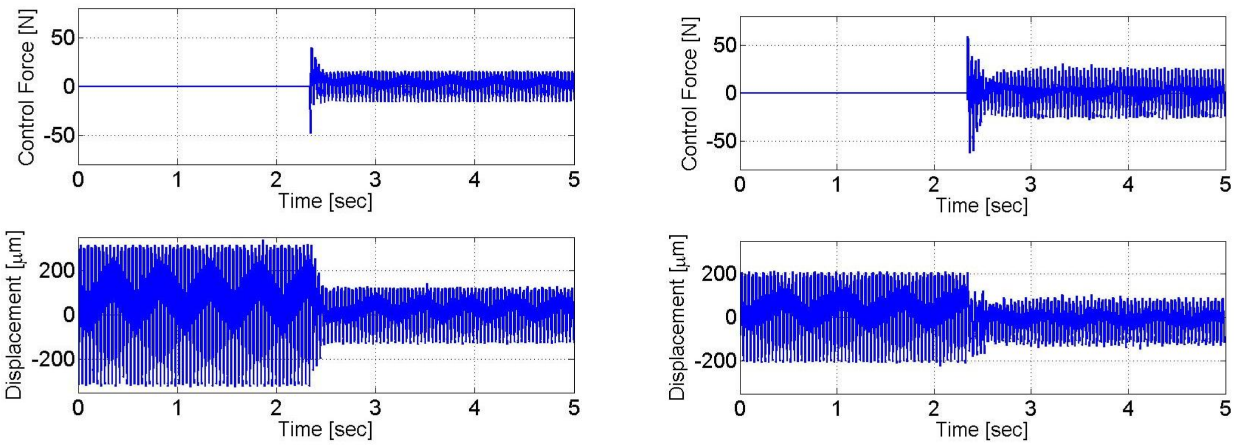
The control force F_i, for mode i, is given by the output of the controller and can be written as:

$$F_i = \sum_{j=1}^{2} w_j z_j \Big/ \sum_{j=1}^{2} w_j \tag{3}$$

4. EXPERIMENTAL RESULTS

The number of modes considered was analyzed in terms of their observability and controllability. For the 4 modes chosen, the system is observable and controllable. The Fuzzy Modal Controllers for the rotating system was performed by considering the unbalance response. For this purpose, the rotation speed of 1600rpm was considered; this rotation is close to the critical speed of the rotor.

Figure (5) presents the unbalance response for the steady state and the control force considering the system without variation in the Kalman estimator´s model.



(a) (b)

Figure 5. Unbalance response (1600rpm): (a) x Direction and (b) z Direction.

Figure (6) presents the comparison between the orbits of the system for the two following configurations: control-on and control-off.

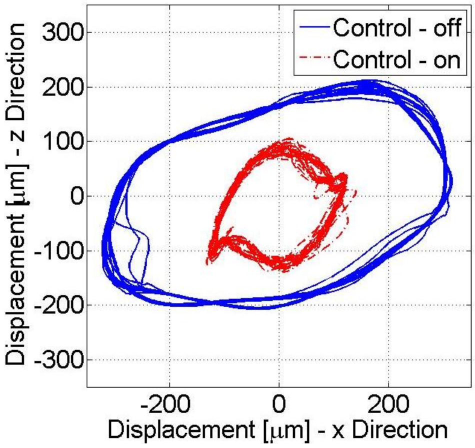


Figure 6. Orbit.

The plots of Fig. (6) demonstrate that the system was completely controlled when the control is on. Therefore, at the time instant of 2.5sec, the control was turned on, and the peak-to-peak amplitude was attenuated from 618.90μm to 246.54μm along the *x* direction and 416.43μm to 264.3μm along the *z* direction, corresponding to a reduction of 62.52% and 39.35%, respectively. In spite of the control force along the *z* direction was bigger than the one found along the *x* direction, the worst results occurred along the *z* direction, since it corresponds to the vertical plane (weight influence).

For analyzing the robustness of the control system, a non-parametric variation is considered in the models of the estimator. The variation of the estimator model corresponds to uncertainties in terms of system identification, for example. The uncertainty appears in the dynamic matrix $[A]$ by considering a variation from -20% to +20%. The natural frequencies of the estimator model are presented in Tab. (4).

Table 4. Natural frequencies (Hz).

Mode	-20%	0%	+20%
1	21.30	26.63	31.95
2	22.45	28.06	33.67
3	71.79	89.74	107.69
4	78.37	97.97	117.56

Figure (7) presents the obtained results for the present case.

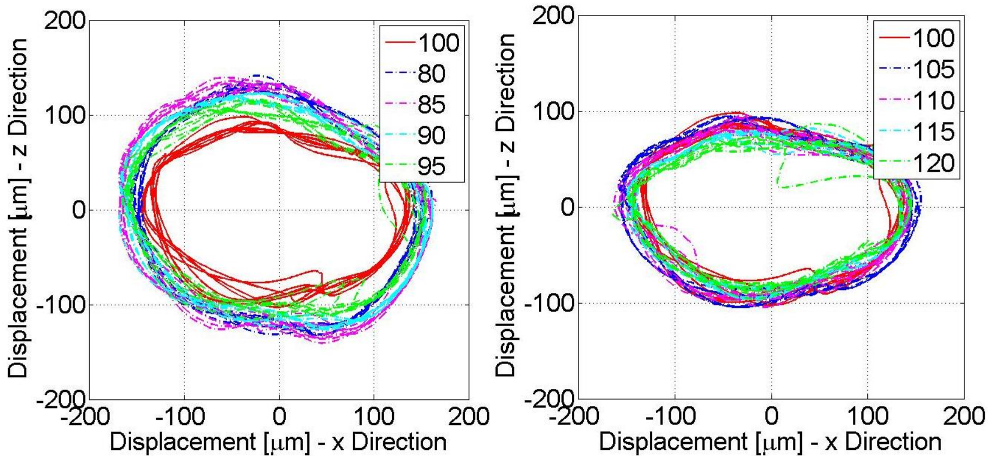


Figure 7. Variation of the orbit.

In Fig. (7), it is possible to observe a small change in terms of attenuation when the estimator has a variation from 0 to 20%. The response reduction is smaller when the model of the estimator changes from 0 to -20%. These results can also be seen in Tab. (5), where comparisons of percentage reductions of peak-to-peak values of the unbalance response are shown.

Table 5. Percentage reductions of peak to peak values of the unbalance response.

Variation of estimator's model	Control – Off		Control - On		Percentage reduction	
	x direction	z direction	x direction	z direction	x direction	z direction
80%	618.90μm	416.43μm	271.53μm	339.36μm	58.72%	22.00%
85%	618.90μm	416.43μm	324.22μm	355.13μm	50.71%	18.37%
90%	618.90μm	416.43μm	291.60μm	331.18μm	55.67%	23.88%
95%	618.90μm	416.43μm	287.22μm	310.62μm	56.34%	28.60%
100%	618.90μm	416.43μm	246.54μm	264.32μm	62.52%	39.25%
105%	618.90μm	416.43μm	266.90μm	280.33μm	59.42%	35.57%
110%	618.90μm	416.43μm	306.79μm	264.48μm	53.36%	39.21%
115%	618.90μm	416.43μm	253.95μm	239.96μm	61.39%	44.85%
120%	618.90μm	416.43μm	251.67μm	244.09μm	61.74%	43.90%

By analyzing the results presented in Tab. (5), it can be observed that the control was robust with respect to the variation of the model of the estimator, which is shown in terms of percentage reductions of the peak-to-peak amplitude values of the system response.

5. CONCLUSIONS

Fuzzy Modal Controllers were evaluated as an alternative strategy for the control of rotating machines. These controllers have shown to be quite efficient as demonstrated by the case studies conveyed. As expected, the system´s responses were completely attenuated when the control was turned-on in both directions (x and z). For the unbalance response analysis, a reduction of 62.52% along the *x* direction and 39.25%reduction along the *z* direction were observed. These results demonstrated the efficiency of the robust controllers in the application presented. It is important to observe that the best results were obtained along the *x* direction, since there is no influence of gravity along this direction.

The robustness of the controller was analyzed by considering a variation range from 0 to 20% in the dynamic matrix [A]. This variation was introduced only in the estimator model, i.e., it was introduced only in the rotor model. The Fuzzy Modal Controllers were found to be robust for both cases analyzed. Only small variations appeared in the vibration reduction for each case considered, which confirms the effectiveness and potential of the proposed technique.

6. ACKNOWLEDGEMENTS

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