A FUZZY LOGIC CONTROLLER FOR THE ACTIVE CONTROL OF ROTOR VIBRATION USING AN ACTIVE MAGNETIC BEARING AS AN ACTUATOR

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Abstract. *This paper presents a theoretical analysis of the active control of rotor vibrations using an active magnetic bearing. The control strategy involves the use of fuzzy logic. To determine the system parameters needed to construct the fuzzy controller, a rotor supported by magnetic bearings system is modeled using the finite element method. Simulations are performed to calculate the maximum displacement of the shaft and the maximum control forces that can be applied to it. SIMULINK® is used to determine the control parameters and the system response. It is seen that a fuzzy controller together with the active magnetic bearing is capable of controlling the rotor when the membership functions are well established. It was found that the effectiveness of the controller is related to the number of membership functions.*

Keywords: *Magnetic Bearing; Active Control; Fuzzy Logic.*

1. INTRODUCTION

The development of efficient ways to support rotors has been a concern for industry since the advent of modern machinery. The devices usually employed for this purpose are journal and roller bearings, which dissipate large amounts of energy due to friction. Recently, Active Magnetic Bearings (AMBs) have been developed, which have replaced conventional bearings in some applications, as they have considerable advantages (Rodrigues, et al 2006). These include the elimination of a lubrication system, friction free operation, low energy waste and high rotation speed, which can lead to self-balancing of the rotor, reduction of global vibration levels, high reliability and low maintenance. Such properties mean that AMBs can be attractive option, with commercial applications including turbomachinery, high speed machinery (over 60,000 rpm) and spacecraft.

AMBs present a new concept in bearing technology. These electromagnetic devices are designed to levitate a shaft in the bearings using only magnetic forces (Guiráo, 2006). Thus, there is no mechanical contact between the shaft and the static components, drastically reducing power loss. Another important characteristic of an AMB is that it can be used as an actuator to actively control rotor vibrations (Gonçalves, 2006). The active control of AMB, however, is complicated due to the inherent nonlinearities associated with the system, e.g., nonlinearity due to the gap between the rotor and the bearing, gyroscopic effects and vibration due to mass unbalance. Since a Multi-Input-Multi-Output (MIMO) AMB system is highly nonlinear, modern control techniques are natural choices as they can provide more complete consideration of the complex effects, and even permit greater use of available clearance during operation (Huang, *et al* 2003).

Many researchers, however, have worked with linearized models, using Proportional Integral Derivative (PID) and Linear Quadratic Regulator (LQR) controllers. In 2006, Guiráo studied the use of PID controllers and the influence of the number of error sensors to control the vibrations in a shaft supported by AMBs. He showed that this kind of controller doesn't results in steady behavior of the system. The same conclusion was reached by Cardoso in 2003. Using an LQR controller to control simultaneously speed and displacement of the rotor, it was noted that the system took some time to reach steady-state, and the controller performance could be increased with some changes. An alternative to this type of control, is the use Fuzzy Logic in the controller in an attempt to eliminate the complications created by the nonlinearities of the system. Huang *et al.* 2003, proposed the use of Fuzzy Logic to model and control a conical AMB, which is highly nonlinear, using Linear Matrix Inequalities. They showed that the method can simplify the controller with a small number of rules for the fuzzy set.

In the work presented here an AMB is used as an actuator rather than a bearing to control the vibration of a rotor system supported by conventional linear roller bearings operating near the critical speed of rotation. The main aim of the paper is to investigate whether an AMB can be used effectively as an actuator together with a fuzzy controller to control the vibration of a rotor close to the first critical speed. The principal purpose of this work is to investigate how the increasing of the number of membership functions affects the controller performance, then Two fuzzy controllers with different numbers membership functions are studied to achieve this goal.

2. PROBLEM DEFINITION

The system studied in this paper is shown in Fig.1. It consists of a steel shaft of length 1m and radius 0.1 m. There is a disc element with 15 kg of weight representing the AMB, which is used as the actuator to control the vibrations of the shaft. There are two conventional roller bearings supporting the shaft, one at each end, which are connected to a rigid foundation. The physical properties of the system are given in Tab.1.

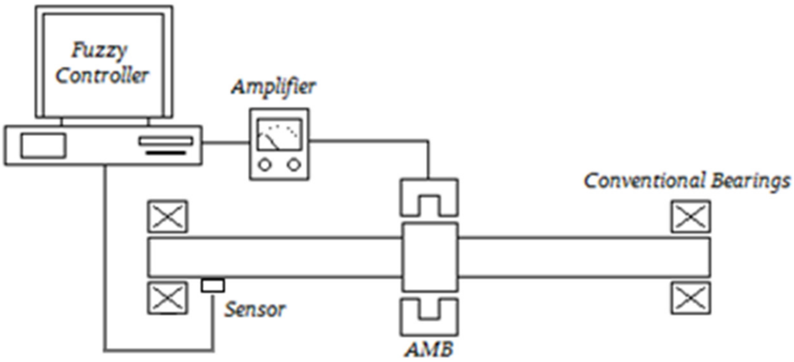

Figure 1. Diagram of the Rotor System and the AMB actuator, together with the control system.

Table 1. Physical properties of the rotor system (Guiráo, 2006)

Properties	Values
Shaft Length	1 m
Shaft Radius	0.1 m
Shaft Material Elasticity Modulus	209 GPa
Shaft Material Density	7850 kg/m^3
Shaft Moment of Inertia	4.908 x 10^{-6} m^4
Disc Weight	15 kg
Conventional Bearing Stiffness	5 x 10^8 N/m
Conventional Bearing Damping	2 x 10^4 N.s/m
Foundation Stiffness	5 x 10^8 N/m
Foundation Damping	2 x 10^4 N.s/m
Foundation Mass	100 kg

The uncontrolled force that excites the system is due to rotating unbalance of the disc element of 0.0015 kgm. It acts as an external centrifugal force applied into the disc node in the model. The model of the AMB considers only linear dynamic behavior, and the system of interest is a 2D model of the rotor shown in Fig.1. The AMB applied to the system is a double action actuator with characteristics shown in Tab.2.

Table 2. AMB Properties (Guiráo, 2006)

Properties	Values
Geometry Factor for Radial AMBs	0.8
Air Magnetic Permeability	1.256 x 10^8 H/m
Number of Spirals	228
Clearance Area	64.426 x 10^{-6} m^2
Clearance Size	0.381 x 10^{-3} m

The controller was designed for steady-state operation at the running speed. It employs a fuzzy set of rules which uses the displacement and the velocity which is assumed to be acquired from a sensor(s) placed at the conventional bearing to produce a current that is fed back to the AMB which finally produces the control forces.

3. MODELLING OF THE ROTOR

The finite element method is used here to model the rotor. The equation that describes the rotor displacement can be obtained from Lagrange equation which provides the energy expressions for each rotor element (Lallane, *et al* 1997). For this approach, it is necessary that the shaft length is much larger than the radius, the material is linearly elastic, the Poisson effect is neglected, the rotation angle is very small and the shaft is made of a homogeneous material with isotropic properties (Seon, *et al* 1999). The shaft is discretized into four Euler-Bernoulli beam elements, with the disc element at the centre (node 3) being considered as a rigid mass. This is the position where the control forces are applied using the AMB. This position has the maximum displacement of the shaft in the first mode of vibration at 165.5 Hz, which is the mode that is to be controlled. For the suggested system, a rotor supported by two conventional roller bearings, were attached the mass-spring model of the bearing and its foundation, as can be seen on Fig. 2.

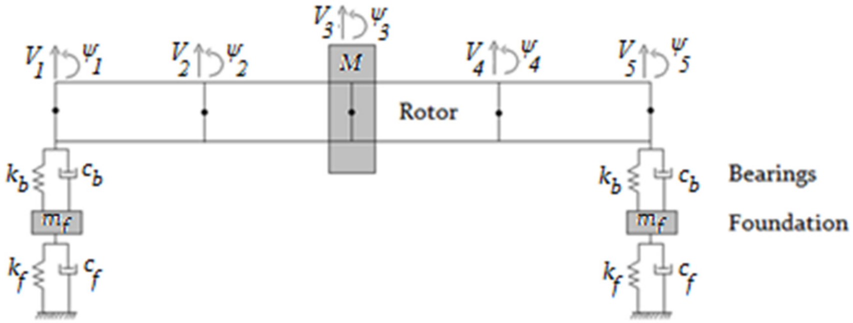

Figure 2. Scheme of the modeled rotor system used for the simulations

4. MAGNETIC BEARING MODEL

The AMB is based on the attraction force generated by a magnetic actuator. To lift a body magnetically, five components are needed, which here are the rotor made from ferromagnetic material, the magnetic actuator, the displacement sensor, the controller and an amplifier. This attraction force is directly proportional to the square of the magnetic flux density and the area of the actuator pole. Common values of magnetic flux density have the order of 1T (Tesla), allowing the production of forces of around 40 N/cm^2 (Furtado, 2008). However, some materials such as silicon steel, have a maximum flux density that approaches 1.4 to 1.7 T, as shown in Fig.3, which is a plot of magnetic flux density as a function of magnetic field. To obtain this flux value, a huge magnetic field is needed, which demands a great number of coils and a high electrical current, requiring an external cooling system. Thus, engineers generally consider the more practical option of enlarging the pole area rather than increasing the magnetic flux density value for higher force requirements.

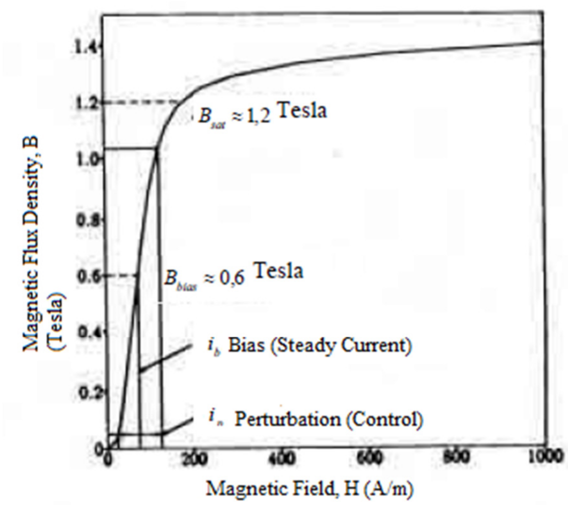

Figure 3. Magnetic Flux Density (B) versus Magnetic Field (H) for Silicon Steel (Allaire, *et al* 1993)

4.1. Double Action Actuator

Consider the sketch of an AMB, shown in Fig.4. The magnetic flux is produced at each actuator section (with a horseshoe shape) by a winding of N coils with an electrical current I flowing through it. A power amplifier produces the required current. The flux path, with length L, passes through the section of the actuator, crossing the clearances at the edge of each face of the actuator poles and flowing through the rotor.

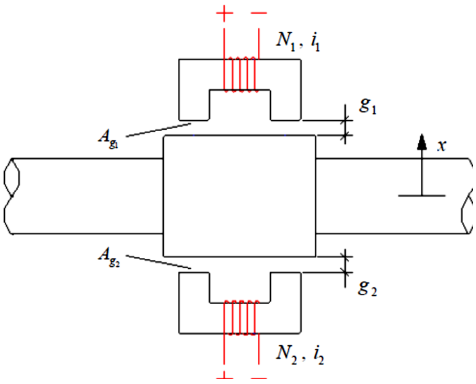

Figure 4. Basic geometry of a double action Magnetic Actuator

Once the flux density for each clearance in the magnetic bearing is known, it is possible to determine the force generated for a given electrical current. For a magnetic permeability μ_0, a clearance g and area A_g, the flux density of each clearance is given by (Clements, 2000)

$$B = \frac{\mu_0 N i}{2g} \tag{1}$$

and the force generated is given by

$$F_g = \frac{B^2 A_g}{2\mu_0} = \frac{\mu_0 N^2 i^2 A_g}{8g^2} \tag{2}$$

From figure 4, it can be seen that there are two clearances for each actuator. Because of the effect diffusion and leakage currents that are not considered at the electromagnetic force equation, a geometric correction parameter (ε) is induced so that for two clearances Eq. (2), becomes

$$F = \varepsilon \frac{\mu_0 N^2 i^2 A_g}{4g^2} \tag{3}$$

5. CONTROLLER

Fuzzy control involves at least 3 steps: Fuzzification, Logical Operation and Defuzzification, described at Fig. 5. The fuzzification transforms a numerical value into a linguistic value, which will be applied the rule base for the logical operations to determine the control action. Then, with a linguistic value of the control actions, the defuzzification transforms them into numerical values, so they can be applied to the system.

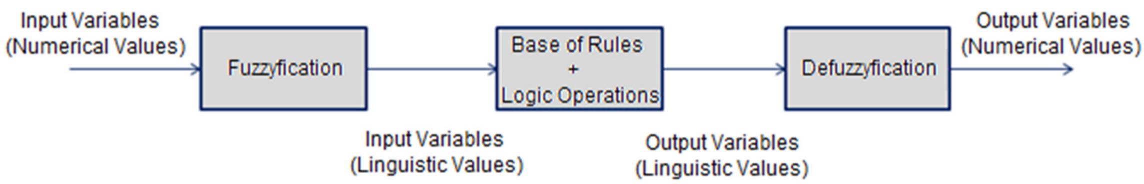

Figure 5. Basic Configuration of a Fuzzy Controller

In the first step, knowledge of the real system parameters is necessary so that the limits that will set the membership functions are known. In this case, the systems are not real, so those values have to be acquired from simulations with the system functioning free from any control forces and operating near the first critical speed of rotation (165,5 Hz). To do this the Bode plot was constructed as shown in Fig. 6. This shows the displacement at the bearing position where the sensor was located due to an input force at the AMB over the frequency range 100 Hz to 1 kHz. The displacement is highest at first critical speed.

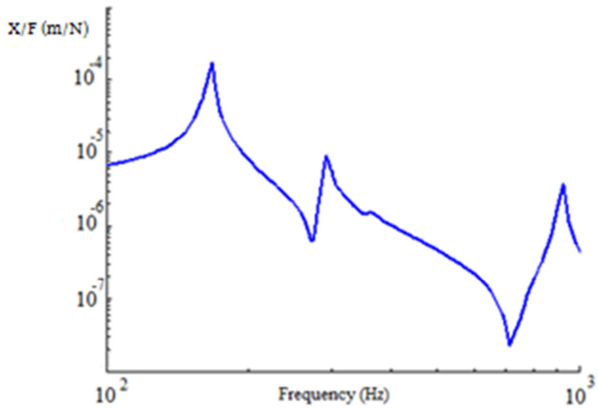

Figure 6. A Bode plot between a force input at the AMB position and the displacement of the rotor at one of the roller bearing positions

The parameters input to the fuzzification sets were displacement and velocity, which are of the third node into the vertical direction. The displacement limits were set in the interval [-3.6. 10^{-4}, 3.6. 10^{-4}] (m), and the limiting values of velocity were set between [-2.75.10^{-5}, 2.75.10^{-5}] (m/s), which were found with the simulation of the system vibrating without any control at a rotation near the critical speed.

Two different groups of rules (logical operators) were used in the simulations; one considering that in the fuzzification step the controller used 3 membership functions to obtain the linguistic values, and in the other there were 5 membership functions. The linguistic values mimic the human sense for how those variables change their values, and in this case, for the 3 membership functions set, the linguistic values for displacement were Large (L), Normal (N) and Small (S), for speed they were Fast (F), Normal (N) and Slow (S). For the 5 membership functions set, the linguistic values were: Large (L), Nearly Large (NL), Normal (N), Nearly Small (NS) and Small (S) for displacement, and Fast (F), Almost Fast (AF), Normal (N), Almost Slow (AS) and Slow (S) for velocity. In the 5 membership functions set the linguistic values that stand between the extreme values and the middle values were imposed to be near the boundary values, which is an option of the fuzzy design, knowing that most part of the signal has that amplitude.

Both groups used only the logical operator "and", which means that the combination of the inputs of linguistic values at the second step considers only the cases that happens an action of input at the velocity and the speed. These logical operations turn the linguistic value of input into a linguistic output. The output returns a linguistic value of current, which stands between High (H), Normal (N) and Low (L) current for the 3 membership functions set, and High (H), Almost High (AH), Normal (N), Almost Slow (AL) and Low (L) current for the 5 membership functions set.

Knowing that later the linguistic values will be turned into numerical values again, the relation of the rules can be expressed by a surface graph which shows the input – output relations as shown in Fig.7.

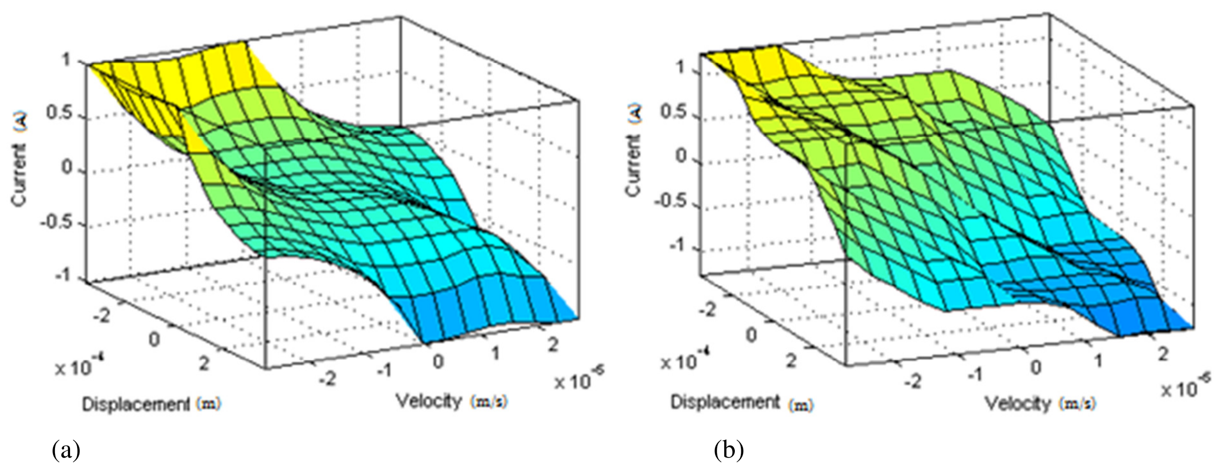

Figure 7. Rule configuration for 3 membership functions (a) and for 5 membership functions (b)

As seen in the graphs in Fig.7, there is a limit for the output values also. Those used in this paper are in the interval [-1, 1] A, which respects the interval of [-1.5, 1.5] A that are the maximum values of current that the AMB can accommodate safely (Gonçalves, 2006).

The defuzzification step can use many methods to transform the linguistic information into numerical values used to control the system. Here, the Centroid method, which is simple and accurate, was used. Figure 8 shows an example of the defuzzification step for a displacement of 1.98 x10^{-4} m and velocity of 1.43 x10^{-5} m/s, using the Centroid method for the 9-rule set (3 membership functions). As seen, the sum of the areas of the first 9 elements of the third column appear in the last element of the same column, where the centroid of this area results into the controller numerical output, a control current of -0.315 A. The red lines that cross the displacement and velocity rules show the value for them at the respective situation, and the yellow colored areas show which membership function is activated with those values. The blue colored areas are the activated rules for the output, and the red line at the last element shows the centroid value of the sum of those areas.

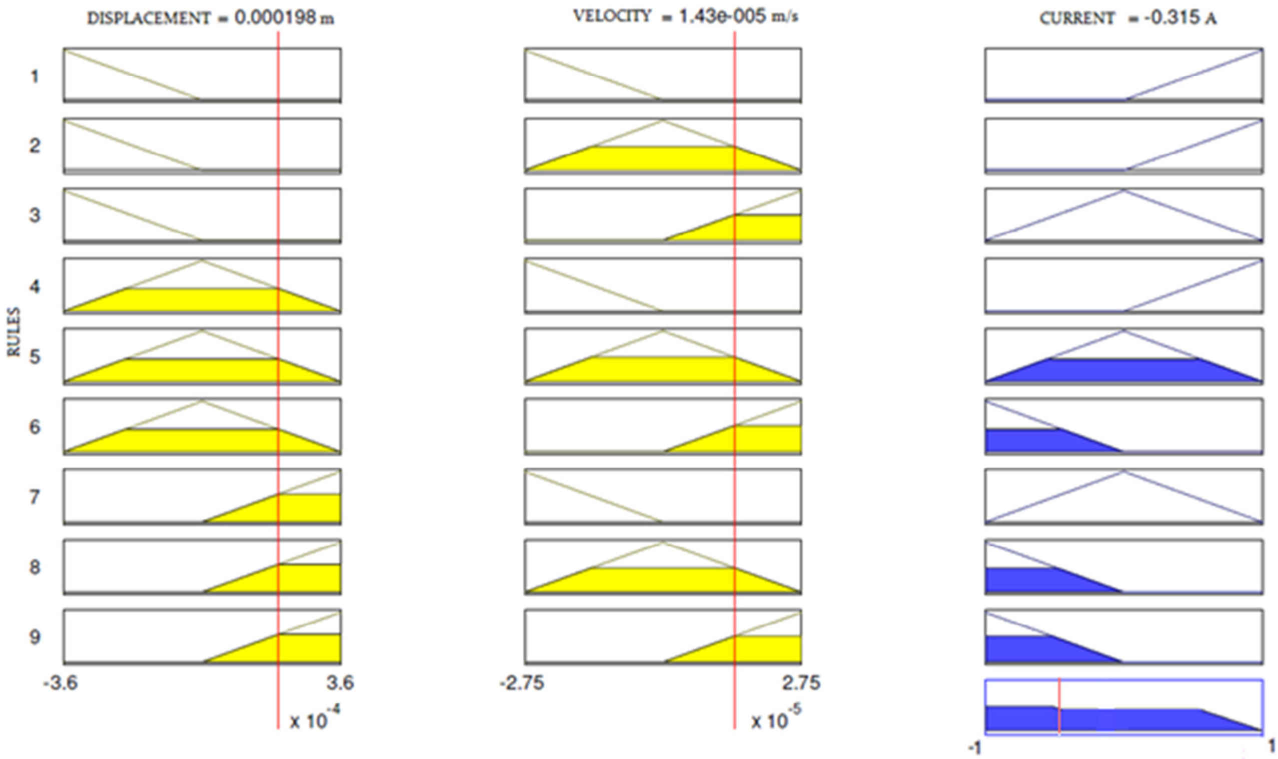

Figure 8. Defuzzification result for an example of a base of rules with 9 rules.

6. CONTROL SIMULATIONS

In this section, a fuzzy controller with different numbers of membership functions and the controller gain is discussed. The current and the displacements at points of interest are presented. The closed-loop controller is designed using SIMULINK® and is shown in block diagram form in Fig. 9.

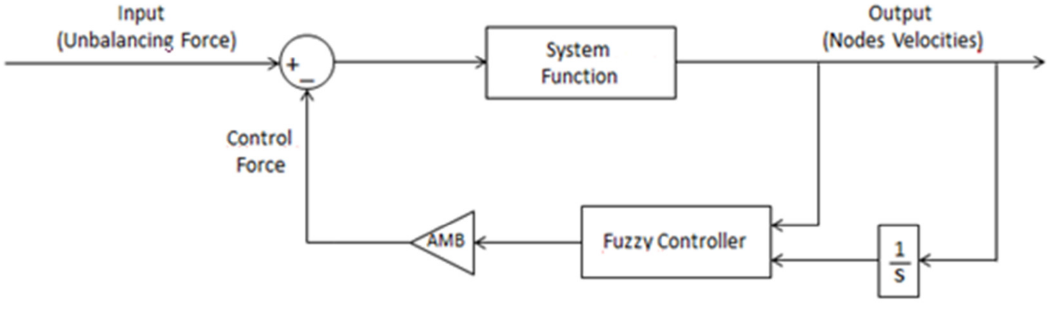

Figure 9. Closed-loop control for the rotor system

The displacements of the node that contains the magnetic bearing (the same as the disc element), and the nodes corresponding to the positions where the conventional bearings are attached are of interest. These are positions in which the vibration is required to be attenuated. For the uncontrolled system, the displacement values reach 0.310 ± 0.002 mm at the AMB and 0.361± 0.005 mm at the conventional bearing positions in the steady state, determined through 10 simulations for the critical speed (9930 RPM). Under control, initially with 3 membership functions, a gain value of 10 was found to be optimum for displacement reduction. The controlled displacements reduced to about 0.098 ± 0.003 mm at the disc element and 0.105 ± 0.005 mm at the conventional bearings, after going through a transient response and reaching the steady state of control. Those results were taken with 10 simulations using the control system. This corresponds to about a 70% reduction of the vibration level. The simulations can be seen in Figs. 10 and 11.

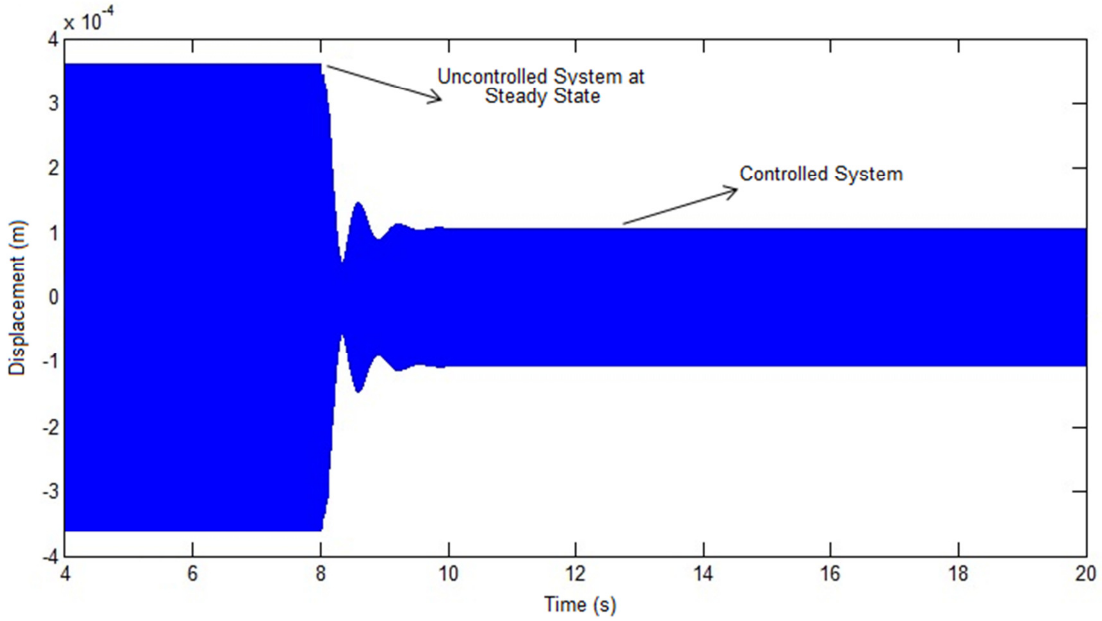

Figure 10. Displacement Signal for the Roller Bearing Position for a rotor speed of 9930 RPM
(Controller with 3 membership functions – 4th Simulation)

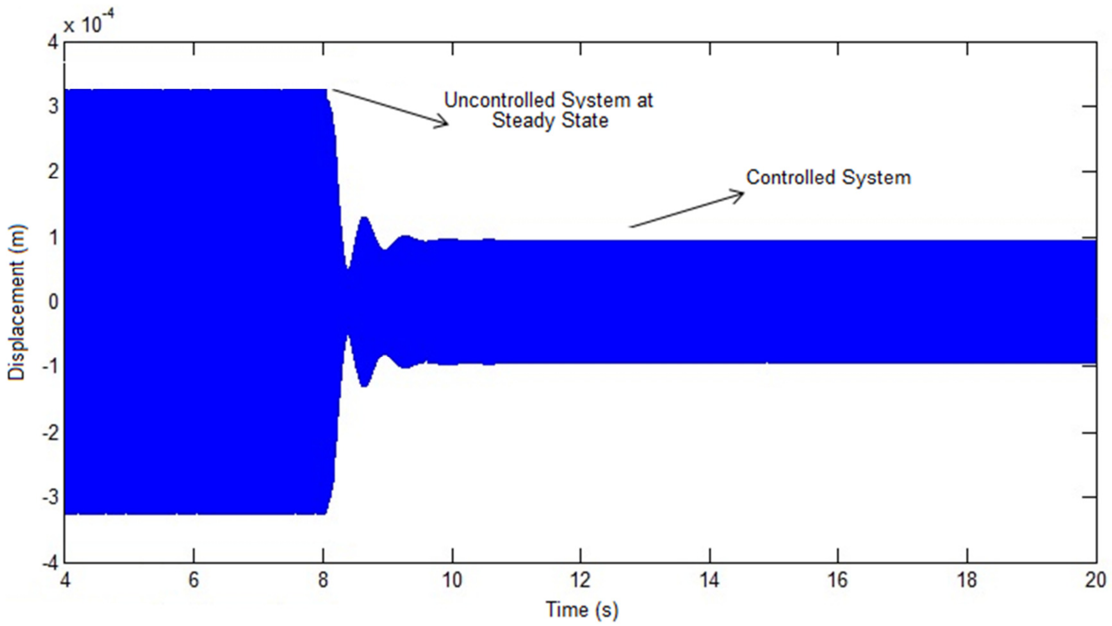

Figure 11. Displacement Signal for the AMB Position rotor speed of 9930 RPM
(Controller with 3 membership functions – 4th Simulation)

For the 5 membership function controller, an acceptable gain was found to be 10 determined by the comparison of 15 different values each one used just one time in a simulation. The maximum displacements with this gain are 0.013 ± 0.001 mm at the AMB and 0.00100 ± 0.00008 mm at the conventional bearings, corresponding approximately to a 97% reduction of the vibration level with respect to the uncontrolled system. These values were taken after 10 simulations. It could be noticed that the standard deviations of the maximum values of displacement at steady state of the controlled system do not represent great part of the mean values because the system is simulated, and the response obeys a model which does not change for each simulation, and it also justifies the great performance of the fuzzy controllers, which shows stable results. The dynamical behavior of the system under the action of this controller is shown in Figs. 12 and 13.

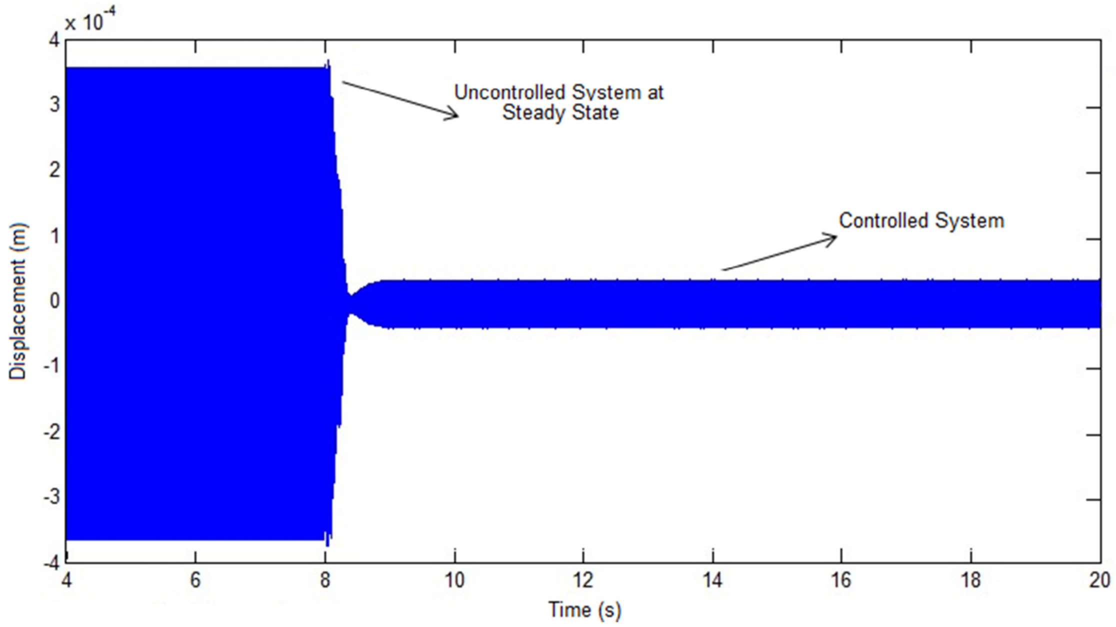

Figure 12. Displacement Signal for the Roller Bearing Position rotor speed of 9930 RPM (Controller with 5 membership functions – 7th Simulation)

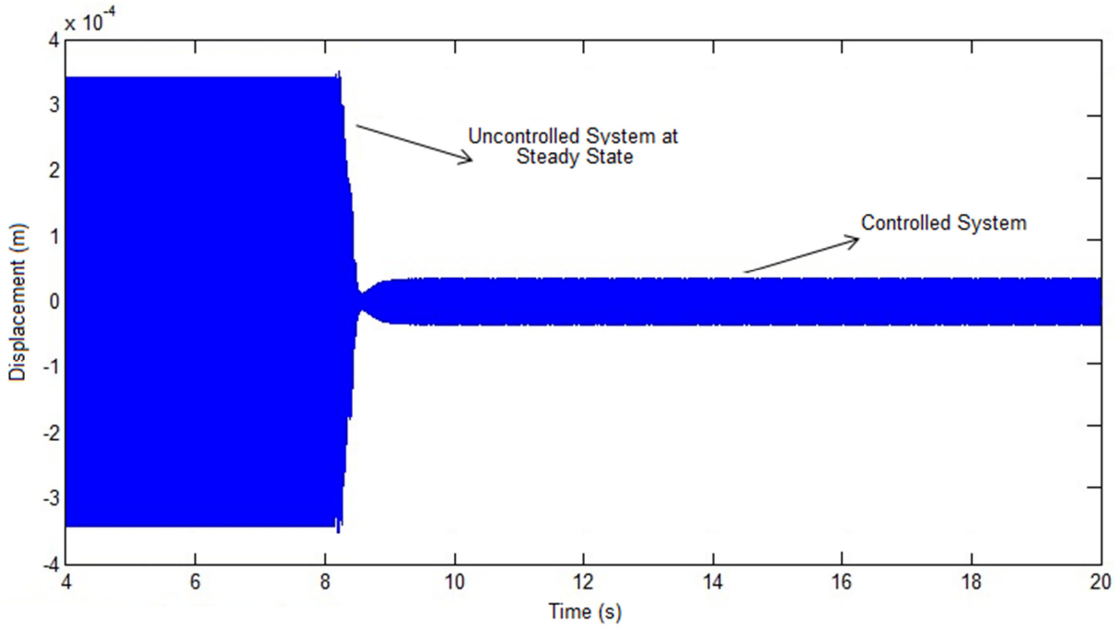

Figure 13. Displacement Signal for the AMB Position rotor speed of 9930 RPM (Controller with 5 membership functions – 7th Simulation)

It can be seen that the second controller has better performance with respect to the attenuation of the vibration, and also takes much less time to achieve steady-state. The fuzzy controller with 5 membership functions takes approximately one third of the time that the controller with 3 membership functions takes to achieve steady-state. This is because the first controller has a smaller number of membership functions and is thus not capable of reacting to the large range of displacements and velocities. Thus, some of the time there are no proportional actions to reduce the displacements in this range.

7. CONCLUSIONS

This paper has described a theoretical study into the use of a fuzzy controller coupled with an active magnetic bearing acting as an actuator to control the vibration of a rotor supported on roller bearings. Two fuzzy controllers were compared – one with 3 membership functions and a set of 9 rules and the other with 5 membership functions and a set of 25 rules – and it was found that the controller with more membership functions is better for this application, attenuating better the vibration at the interest points of the structure. With reference to the number of membership functions, it can be concluded therefore that increasing the number of functions results in better control. This is simply because with a rules set with more membership functions, there are more intermediary actions possible, increasing the performance of the controller and reducing the transient time.

The control technique was simple to implement, demanding low computational effort, and provided good results. Although the vibration reduction is not optimal when compared with other controllers, it has the advantage of being easy and quick to implement and demands much less effort to process the control. Moreover, the system to be controlled doesn't need to be modeled.

8. ACKNOWLEDGEMENTS

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