Rotor-bearing vibration control system based on fuzzy controller and smart actuators

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ABSTRACT
Most rotating machines, especially those mounted on flexible shafts and bearings when it starts operating, tend to pass through critical speeds, ie speeds that can cause the system to resonate the mechanical structure. Hence it is a constant concern for finding effective methods to mitigate the effect of vibration when passing through such speeds. Currently, it has been studied applications of materials made from special alloys as actuators in dynamic systems, in order to reduce the vibrations in a frequency range related to the resonance region. In this direction, it is the use of components made of active materials such as Shape Memory Alloys (SMA), considered "smart", able to recover its original shape when the change in temperature and/or mechanical stress, and as the main characteristics, its high damping capacity, due to the increased levels of vibration. This paper presents a rotor-bearing vibration control system based on actuators SMA coil springs. A fuzzy controller has been used for control the vibration of the system based on the measuring of critical speeds. The experimental results of the operation of the system shown their effectiveness being obtained reductions of up to 60% in amplitude, during the passage through the resonance region.

Keywords: Vibration control, Smart materials, Shape memory alloy actuators, Fuzzy controller

1. INTRODUCTION

The most of rotation machines, overall those built in axis and flexible rotor-bearings, in the machine start up, it can pass through critical speed which may cause the system to resonance. The resonance works as mechanical amplifier, it may lead the system to collapse and induce seriously materials and/or humans damages. Therefore, there is the constant worry by looking for effectives methods that attenuate the vibration effect when mechanical systems pass through by such critical speeds.

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The common methods for attenuation of increase of acceleration when it pass through the critical speeds. Although, this approach become an inefficient fashion because it requires more power delivery to the machine to provide a high acceleration rate [1]. Other method, according with these authors, is to raise the relation the machine damping, which is not an easy task. Many works have been proposed in literature using passive or active-adaptive control.

The wide applicability of SMA in vibration control of machines and structures is greatly enhanced in [2], is indicated as the main factor for potential use in applications involving large forces and/or deformations.

In [3] is presented the design, numerical analysis and optimization of an adaptive vibration absorber. The authors have adopted as the basis of their studies, the model assumed transformation kinetics, which is considered, in addition to deformation, and temperature T, a scalar internal variable, which represents the volume fraction of martensitic phase.

Another control device was proposed by [5], which made use of the principle of the switching state absorption, known as SSA (State-Switched Absorber), which is a system that can quickly switch between the resonance frequencies when compared with TVA's classics.

In [1] is proposed a theoretical model of a system consisting of SMA springs for use in rotating systems whose equation is based on the control model for dynamic vibration absorber.

In this direction is the use of Shape Memory Alloy (SMA) actuators, considered a smart material (metallic alloy) able to return in the original shape with the change of temperature and/or mechanical stress, having as one of the main characteristics, its high damping capacity and stiffness increase in function of the temperature (heating). This paper shows the results of use SMA actuators (springs) to decrease vibration, in an active rotor-bearing (pedestal bearing) using a fuzzy controller.

2. THE ROTOR-BEARING SYSTEM

The representation of the rotor-bearing system is presented in Figure 1(a) can be considered a vibration absorber (Figure 1b). In this work, the system was considered as a vibration absorber in which the mass of the pedestal bearing corresponds to the secondary system. On the other hand, the mass of rotor-bearing is equivalent to the primary system. The values of mass and stiffness of absorber are chosen to motion of mass of primary system be minimum.

The dynamical equation of the system written in a matrix form [6], for the situation shown in Figure 1 is:

$$\begin{bmatrix} m & 0 \\ 0 & m_a \end{bmatrix} . \begin{bmatrix} \ddot{x}(t) \\ \ddot{x}_a(t) \end{bmatrix} + \begin{bmatrix} c + c_a & -c_a \\ -c_a & c_a \end{bmatrix} . \begin{bmatrix} \dot{x}(t) \\ \dot{x}_a(t) \end{bmatrix} + \begin{bmatrix} k + k_a & -k_a \\ -k_a & k_a \end{bmatrix} . \begin{bmatrix} x(t) \\ x_a(t) \end{bmatrix} = \begin{bmatrix} F_0 \\ 0 \end{bmatrix} .sen(\omega .t) \qquad (1)$$

where:

$x_a(t), \dot{x}_a(t), \ddot{x}_a(t)$	=	magnitude of the displacement, velocity and acceleration of the absorber mass
$x(t), \dot{x}(t), \ddot{x}(t)$	=	magnitude of the displacement, velocity and acceleration of the primary mass
m	=	primary mass
m_a	=	absorber mass
c_a	=	damping of the absorber
c	=	damping of the primary system
k_a	=	stiffness of the absorber

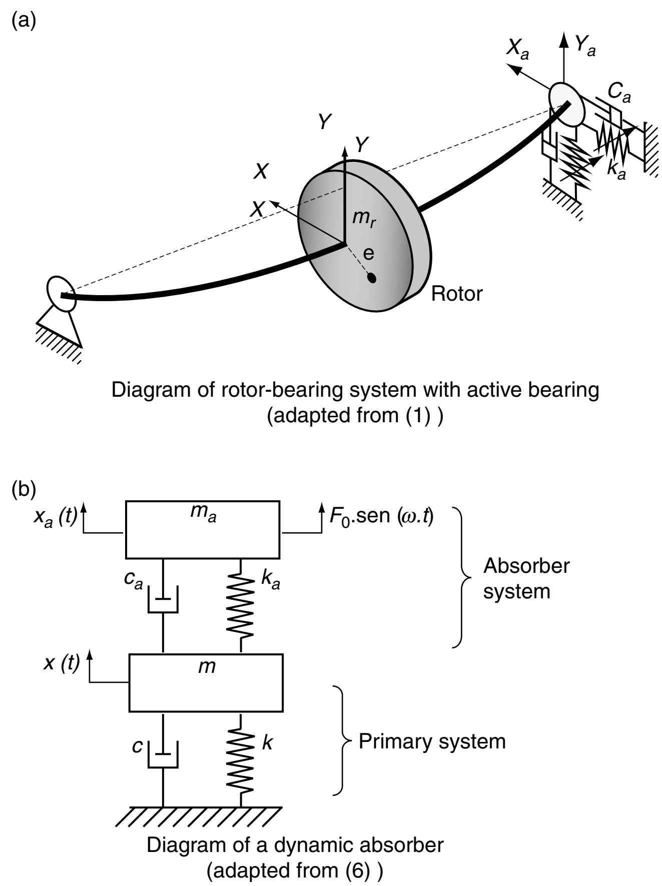


Figure 1 Representation of the dynamic rotor bearing system.

k	=	stiffness of the primary system
F_0	=	amplitude of the excitation signal
ω	=	excitation frequency
t	=	time

Equation (2) shows the solution in exponentially form when the system is in steady state, and X the vibration amplitude of the primary mass and X_a the amplitude of vibration of the mass of the absorber.

$$x(t) = X.e^{j.\omega.t} = \begin{bmatrix} X \\ X_a \end{bmatrix} .e^{j.\omega.t} \tag{2}$$

Substituting (2) into (1) and solving the system one can find as frequency response:

$$\frac{Xk}{F_0} = \sqrt{\frac{(2\xi r)^2 + (r^2 - \beta^2)^2}{(2\xi r)^2 (r^2 - 1 + \mu r^2)^2 + [\mu r^2 \beta^2 - (r^2 - 1)(r^2 - \beta^2)]^2}} \tag{3}$$

where:

$\xi = \dfrac{c_a}{2m_a\omega_p}$ (damping ratio) $r = \omega/\omega_p$ (ratio of the excitation frequency to the primary natural frequency)

$\omega_p = \sqrt{k/m}$ (natural frequency of the primary system) $\mu = m_a/m$ (ratio of the absorber mass to the primary mass)

$\omega_a = \sqrt{k_a/m_a}$ (natural frequency of the absorber) $\beta = \omega_a/\omega_p$ (ratio of the decoupled natural frequencies)

3. THERMOMECHANICAL BEHAVIOR OF THE SMA SPRINGS

There are several proposed models to adequately describe the thermomechanical behavior of SMA springs, both in line microscopic approach (metallurgical aspects) and macroscopic (phenomenological aspects) [3].

The thermomechanical model used in this work was proposed by [7] and it has been recently applied by [8]. Thus, it is possible to establish a relationship between stiffness and temperature that lies on a SMA spring.

In order to validate the model a spring of Nickel (Ni) and Titanium (Ti) was subjected to a test for determining the stiffness as a function of temperature, in an universal testing machine Instron® (Figure 2).

Figure 3 shows the result of simulation and experimental data of the variation of SMA spring stiffness with temperature.

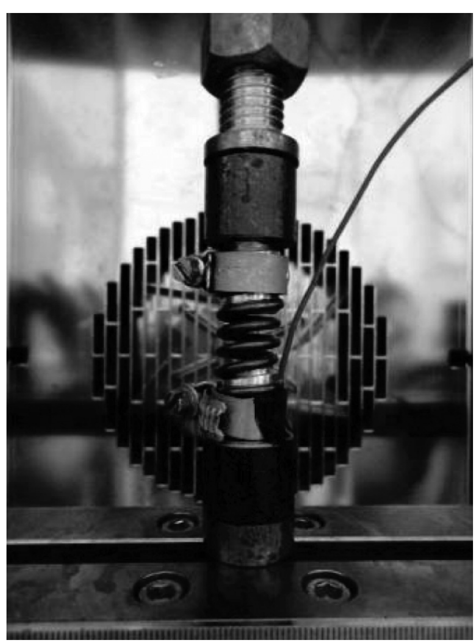


Figure 2 SMA spring during characterization test in a Instron machine.

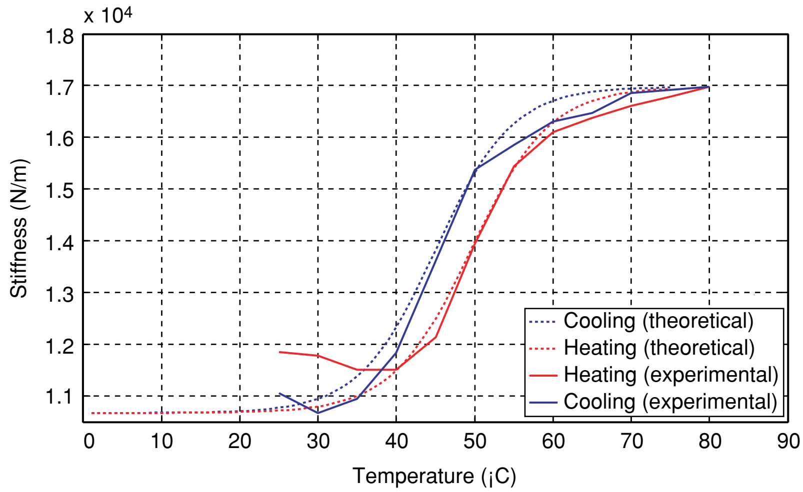


Figure 3 Variation of SMA spring stiffness with temperature.

Based on the results of Fig. 3, one can verify that mathematical model is valid for the SMA spring used in labaratory. Otherwise, if the percentages of Ni and Ti in the alloy are different, other values of transformation temperatures and, consequently, the modulus will be obtained.

Furthermore, by comparing the experimental results with the theoretical model, it is observed that there is an approximation of the curve behavior during cooling, which did not appears clearly during heating. The reason of this effect is the material behavior of the alloy when the austenite to martensite transformation.

The theoretical model proposed in [7] emphasizes just changing fractions of martensite and austenite between the temperatures of initial and final austenite phase, and between the temperatures of initial and final martensite phase.

4. CONTROL SYSTEM

The main idea in a vibration control system is to prevent the state from resonance, and this can be done in several ways. In the case of systems which vary the stiffness and damping as a function of temperature, it is obvious that the vibration control of these systems is directly related to temperature control elements that confer stiffness and damping.

For the system studied, using SMA springs, the temperature of the spring can be associated to the natural frequency. Therefore, controlling the temperature of the SMA spring provides the adjust of the natural frequency of the system. The control of the natural frequency results in low amplitudes of vibration of the system, i.e., it is necessary to control the temperature of the spring to reduce the vibration of the main mass.

The diagram of temperature control of the SMA spring is shown in Figure 4.

In the diagram, the controller aims to send information that will allow the activation of actuators for the change in temperature of the springs, with the goal of changing the stiffness of the same when passing through resonance, causing vibration reduction. The control strategy was based on the information coming from a vibration sensor placed in the SMA bearing, which from an admissible value of amplitude vibration, sent over information to

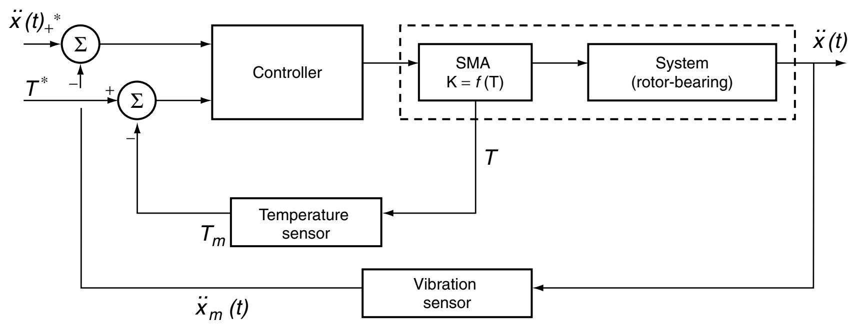


Figure 4 Representation of the control system.

drive pulse heating (electrical current through springs) or cooling (fans with airflow over the springs). The logic of control of the springs is based on a fuzzy controller implemented in LabVIEW.

5. EXPERIMENTAL RESULTS

The physical model assembled built of shaft-rotor system consist of 2 rolling bearing (one assembled in a rolling bearing housing (in the right) and another one in rolling bearing absorber with 4 SMA springs – in the left), one shaft of steel with one disc and mass attached in axis and electric motor of 0.5 cv supplied by a power converter (voltage-source), as show in Figure 5.

Table 1 shows the parameters of system.

It has been performed tests with the system being subjected to a variation in motor rotation, from 0 to 40 Hz, in a acceleration rate of approximately 1 rad/s^2 (with ramp time = 241s), with and without actuation of the fuzzy controller. The results achieved are presented in Figures 6 and 7.

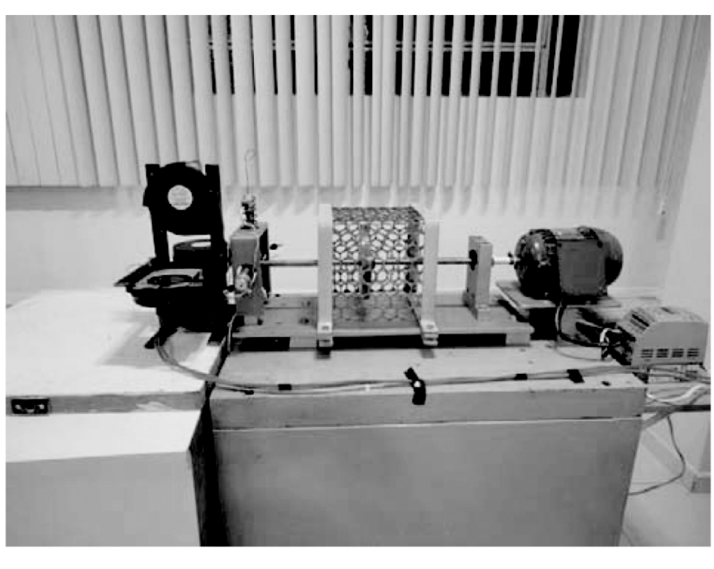


Figure 5 Experimental setup.

Table 1 System Parameters.

Phase	T (˚C)	k_a (N/m)	c_a (N.m/s)	m (kg)	m_a (kg)	k (N/m)	Shaft length (m)	Shaft diameter (m)
Martensite	22	11049	10.0	2.2	0.2	152380	0.42	0.012
Austenite	60	16976	5.0					

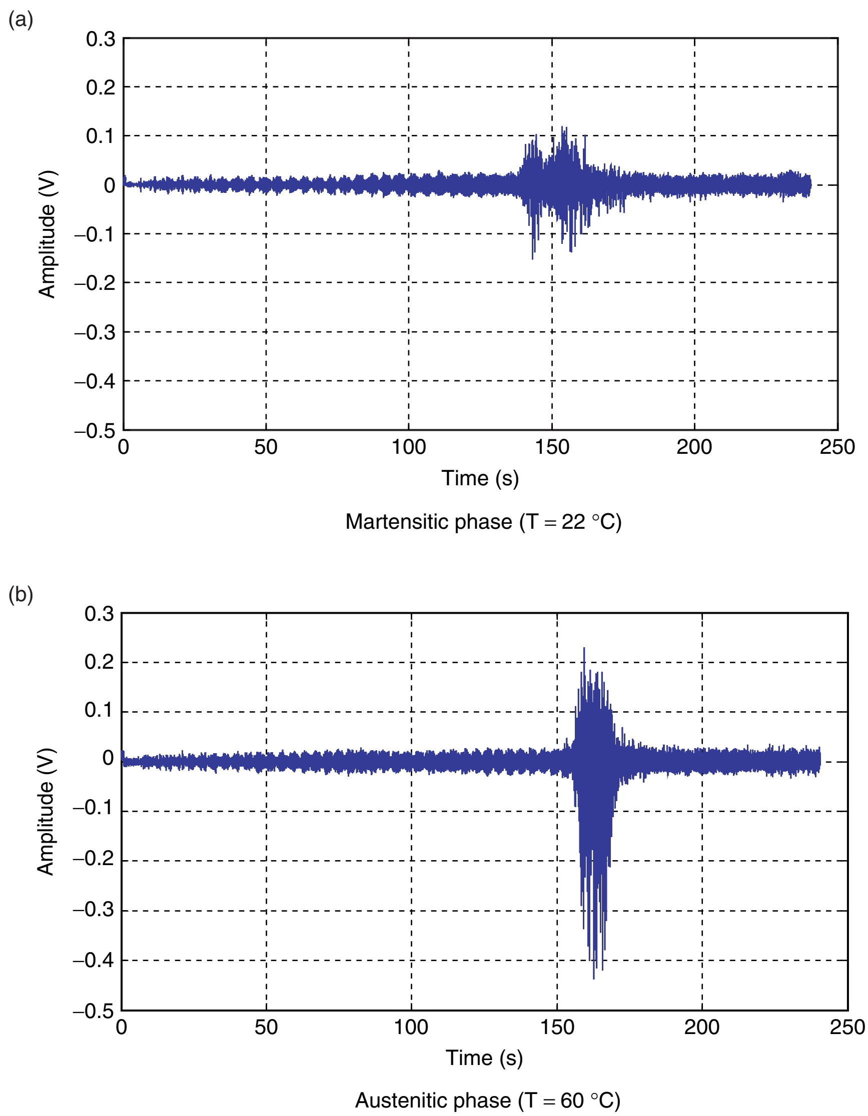
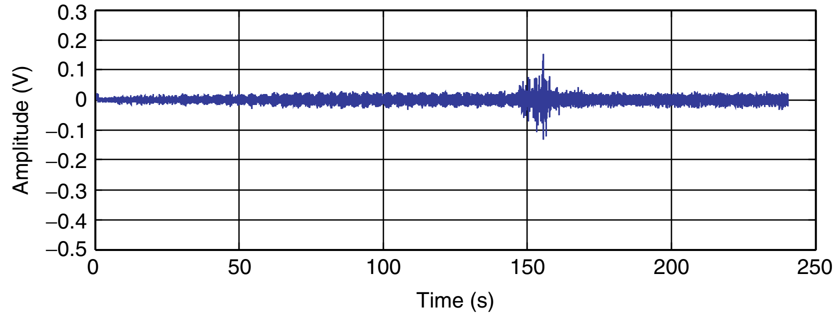
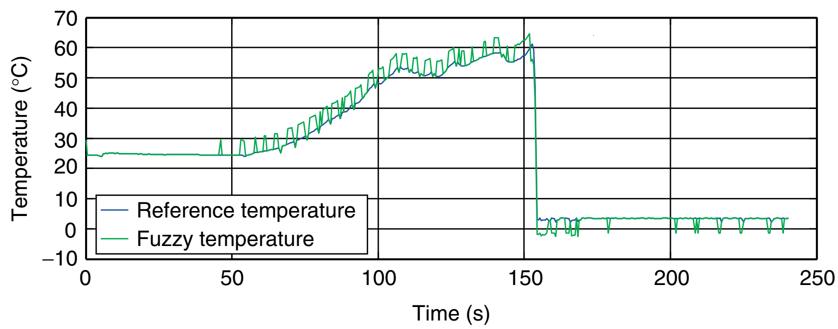


Figure 6 Vibration amplitudes, without control.

Figure 6(a) shows the results of amplitudes when temperature of the SMA springs were 22°C. It is observed an increase in amplitudes, when the system reaches the resonance at the martensite phase (fn_m= 27 Hz). In Figure 6(b), amplitudes are shown when the springs were



(a) Vibration signal (actuation of control system)



(b) Temperature signal (thermocouple sensor)

Figure 7 Performance of vibration control system: (a) waveform of vibration and (b) temperature signal.

heated to 60°C. It is also observed an increase in amplitude, when the system reaches the resonance at the austenite phase (fn_a = 30 Hz).

The actuation of fuzzy controller is shown in the vibration signal (Figure 7a), to a temperature variation description on the Figure 7b.

It has been found that there is a reduction of the order until of 60% in amplitudes when the passage by natural frequency in martensite phase (low stiffness springs) to austenite phase (high stiffness springs), in relation to the system without control. As shown in Figure 6a, the heating effect in springs in martensite phase and the suddenly cooling of the springs was found in 60°C (austenite phase), proving the efficiency of SMA spring actuators and the use of fuzzy logic in temperature control.

6. CONCLUSIONS

This work presents a basic theoretical-experimental study of vibration control in a rotor-bearing system with one rolling bearing absorber using SMA springs. The vibration control based in a fuzzy controller with stated rules in function of temperature variation, showed when the SMA springs were activated in resonance zone, the system was capable to reduce 60% of vibration amplitude in relation to normal state driving without any control.

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