CONTROL OF MECHANICAL VIBRATIONS USING HELICAL SPRINGS OF SHAPE MEMORY ALLOYS

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***Abstract**: The shape memory alloys are materials that, upon receiving a mechanical strain are able to return the original dimensions by means of a specific temperature rise. Using these properties, these alloys are applied in various areas such as aerospace industry, automotive, biomedical, oil exploration, as well as those related to vibration control. Given this context the present work shows the implementation of a fuzzy temperature controller applied to a helical spring mounted on a system with one degree of freedom. Among the results, the study showed that it was possible to control the vibration amplitudes of the forced excitation system, by controlling the temperature of the helical spring shape memory alloy.*

***Keywords:** shape memory, vibration, control*

1. INTRODUCTION

Shape memory alloys (SMA) are metallic alloys that have well defined phase transformations when subjected to an appropriate thermal cycle. This results in a unique behavior, because when the material is deformed by a mechanical load, the original shape of the material is recovered when it is subjected to a specific warm. This shape change can happen only on heating (Single Memory Effect) or shape change occurs in both heating and cooling equipment (Two Way Shape Memory Effect).

This behavior is not found in the materials most commonly used in engineering. Thus, research is being conducted in order to better describe their behavior as well as where and how they can be applied.

Among the various forms of use of shape memory materials, the vibration control of structures is one of the most studied (Lagoudas, 2008). This vibration control is mainly based on change in structural stiffness or vibratory system (Choi, 2000; Holdhusen, 2000; Tao, 2006).

The use of SMA helical springs to reduce vibration levels, may present different results, desirable or not. The use of springs with wire LMF thin, the heat load on / off is low showing a short response time, and may be satisfactory. However, very fine wires limit the strength of performance of shape memory alloy. If the solution to the problem above is to increase the diameter of the wire, this will result in a greater requirement for activation of the thermal load to maintain the same response time.

This work shows the experimental results of a mass-spring system with a coil spring of Nickel-Titanium. Initially, the work consists of a theoretical and experimental characterization of the properties of a helical spring made of Nickel-Titanium wires (SMA). Then the simulated results are compared with experimental results of mass-spring system.

2. MASS-SPRING SYSTEM

Consider a damped primary system of one degree of freedom (1 dof), with mass m, damping c_{SMA} and stiffness k_{SMA} subject to harmonic excitation force F. Consider also that the rigidity of the system consists of a spring nickel-titanium with shape memory properties. The system described was assembled as seen in Fig 1.

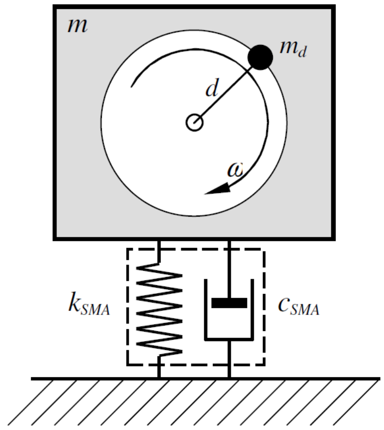


Figure 1 – Mass-spring-damper system

Equation (1) describes the system in question:

$$m.\ddot{x} + c_{SMA}\dot{x} + k_{SMA}.x = F_0.e^{i.\omega.t} \tag{1}$$

In this case, F is the force excitation system caused by unbalanced mass m_d. The amplitude this excitation force is defined as:

$$F_0 = m_d.d.\omega^2 \tag{2}$$

Where:
d is the value of the unbalance mass m_d;
ω is the angular velocity.

The displacements of mass over time, are obtained from Eq. (3):

$$x(t) = Xo.e^{i.\omega.t} \tag{3}$$

The variable Xo is the magnitude of the amplitude given by:

$$Xo = \frac{F_0}{\sqrt{\left(k_{SMA} - m.\omega^2\right)^2 + \left(\omega.c_{SMA}\right)^2}} \tag{4}$$

2.1 Stiffness of the System

The stiffness of a helical spring depends on parameters such as geometry and modulus of the material involved in manufacturing. In the case of materials with shape memory Khajepour et al (1998) showed that the modulus of elasticity of shape memory alloy depends on the fraction of martensite and austenite, has been found that the modulus of shape memory materials can increase up to three times with increasing temperature (Srinivassan, 2001).

A mathematical model that describes the behavior of the change in the martensitic state to austenitic (Ikuta et al, 1991) is shown in Equation (5). You can then make a relationship between stiffness and temperature spring which is an alloy with shape memory.

$$\xi_A = 1 - \frac{1}{1 + \exp\left[\frac{6,2}{A_F - A_S}.\left(T - \frac{A_F + A_S}{2}\right)\right]} \tag{5}$$

Where:
ξ_A : austenite fraction;
A_F: finish temperature for the austenite phase;
A_S: start temperature for the austenite phase;

T: spring temperature.

Like Shape Memory Alloys, during cooling, show a hysteresis in temperature after heating to a fully austenitic state, similarly, the fractions of rhombohedral martensite phase can be determined during cooling according to Eq. (6):

$$\xi_R = 1 - \frac{1}{1 + \exp\left[\frac{6,2}{R_S - R_F} \cdot \left(T - \frac{R_F + R_S}{2}\right)\right]} \tag{6}$$

where :

ξ_R : volume fraction Rhombohedral phase;
R_F: final temperature for the R phase;
R_S: start temperature for the R phase;
T: spring temperature.

With the complete phase R (ξ_R =1 ou ξ_A=0), the spring stiffness has a minimum value, which will be denoted by k_{min} This phase occurs at temperatures lower than AS, in order that the internal structure of the material in this temperature range provides a minimum modulus.

For the completely austenitic state (ξ_R=0 ou ξ_A=1), the spring stiffness has a maximum value, which will be denoted by k_{max}. This phase occurs at temperatures above the AF, resulting in a maximum modulus of elasticity. Therefore, it becomes that the change of the stiffness of the material behaves differently between heating and cooling, leading to Eq. (7) for the heating phase and Eq. (8) for the cooling phase of the alloy:

$$k_{SMA-A} = k_{\min} + \left((k_{\max} - k_{\min}) - \frac{(k_{\max} - k_{\min})}{1 + e^{\left[\frac{6,2}{A_F - A_S} \cdot \left(T - \frac{A_F + A_S}{2}\right)\right]}} \right) \tag{7}$$

$$k_{SMA-R} = k_{\min} + \left((k_{\max} - k_{\min}) - \frac{(k_{\max} - k_{\min})}{1 + e^{\left[\frac{6,2}{R_S - R_F} \cdot \left(T - \frac{R_F + R_S}{2}\right)\right]}} \right) \tag{8}$$

For verification of Equations (7) and (8), it was necessary to manufacture a helical spring with 12 mm diameter and 9 active turns as shown in Fig. 2. The material used was a Nickel-Titanium wire (50.2 to 49.8) with 2.59 mm in diameter. The alloy is identified as the manufacturer Alloy M (Memory-Metalle GmbH ©).

Figure 2 – SMA spring

It was also observed in Equations (7) and (8) that stiffness depends on the transformation temperatures of austenite and rhombohedral phases, these temperatures found by the method of differential scanning calorimeter known as DSC. During this measurement process are checked exothermic peaks (during cooling) and endothermic (for the reverse transformation during heating) that could determine the beginning and end of transformation, responsible respectively for the release or absorption of heat. The calorimeter test result is shown in Fig. 3.

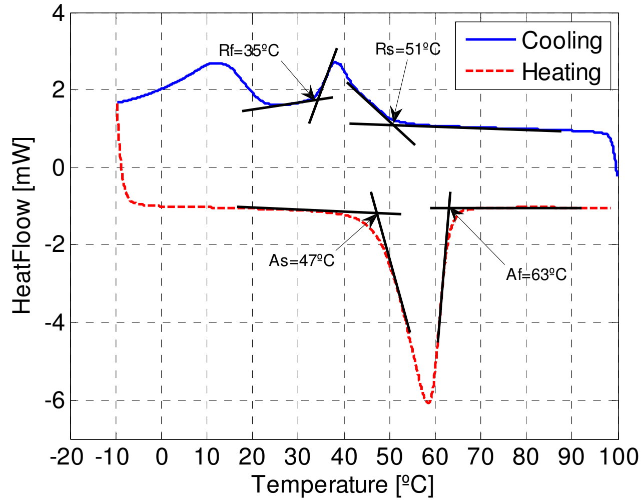


Figure 3 - Transformation temperatures obtained by DSC

The spring was subjected to a test to determine the stiffness as a function of temperature. For this we used a universal testing machine Instron ®. In each test, the spring was compressed into 10 mm, where the machine recorded the force in Newtons for each spring displacement in mm.

In the stiffness test, the spring was subjected to 10 compression cycles at each temperature measurement. The initial test temperature was 20°C with 3°C difference between each temperature check. The result of the stiffness test is shown in Fig. 4. The values of maximum and minimum stiffness in this temperature range were equal to k_{max}=6103 N/m (63ºC) and k_{min}=4097 N/m (34°C).

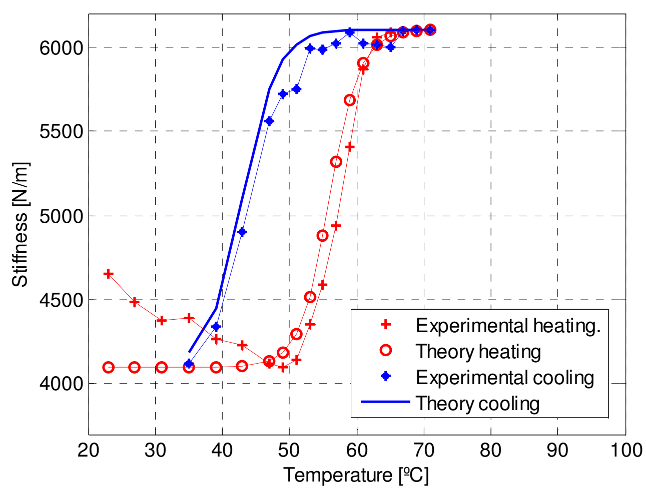


Figure 4 – Stiffness SMA spring

2.2 Damping system

Shape memory alloys show a change in damping due to the movement of martensitic plates in the processes of heating or cooling of the alloy. Experimental results show that both the martensitic phase as the R phase have high damping due to movement between plans or martensitic plates. Given this, it is expected that the mass-spring system under study may also have a similar behavior with a need for investigation on the damping.

The procedure to find the value of damping was to subject the system to an impulse input by hammer impact, and collect the displacement along a time scale. With these data is possible to find the damping coefficient ζ, which is described by the equation:

$$\zeta = -\frac{1}{2.\pi.n}.\mathbf{ln}\left(\frac{A_n}{A_1}\right) \tag{9}$$

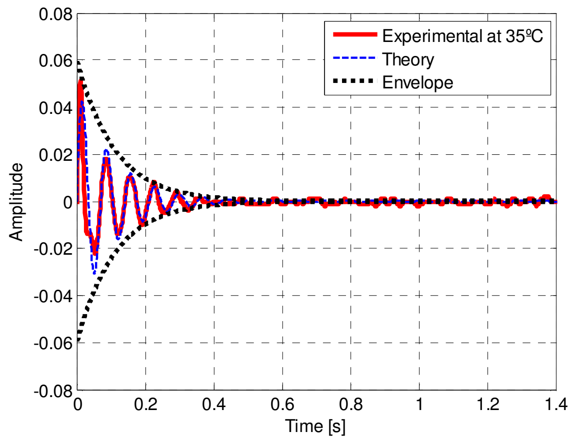
Where:

n: number of periods;
A_1: peak amplitude of the first measurement;
A_n: peak amplitude of the period n.

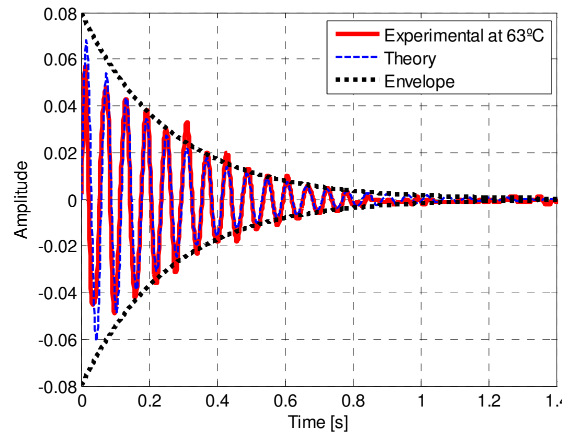
The equivalent damping is obtained by the expression:

$$c = 2.m.\omega_n.\zeta \quad (10)$$

Figures (5-a) and (5-b) illustrate some graphical results of responses to the impact applied to the system at temperatures of 35ºC and 63°C, respectively.



(a)



(b)

Figure 5 – Impulse response

It is observed that the SMA spring damping applies on the system varies according with temperature. For spring at 35ºC the damping was 10,3208 N.s/m for the spring to 63ºC the damping drops to 3,9403 N.s/m. The result graph for damping temperature range between 35ºC and 63ºC is shown in Fig. 6:

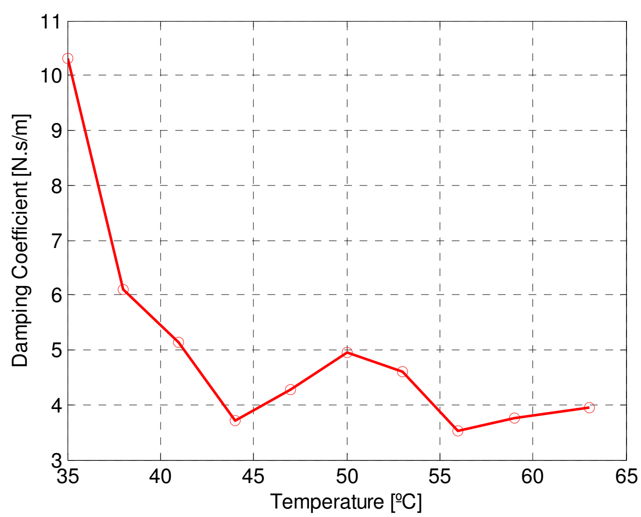


Figure 6 - Damping coefficient in function of temperature

3. CONTROL SYSTEM

The main idea in a vibration control is to prevent the system come in the resonance, and this can be done in several ways. For the case of systems where stiffness and damping change with temperature, it is obvious that the vibration control of these systems is directly linked to temperature control elements that provide stiffness and damping. One strategy that can be adopted to control this system is to prevent the approach between the excitation frequency and natural frequency of the system, ie, a great difference between these frequencies results in low vibration level. For the system under study, which uses SMA springs, the temperature of the spring can be associated with the natural frequency. Thus, controlling temperature SMA spring, has as a consequence the natural frequency control. This natural frequency control results in low vibration amplitudes of the system. In summary we need to control spring temperature to reduce main mass vibration.

The block diagram of spring temperature control is shown in Fig. 7.

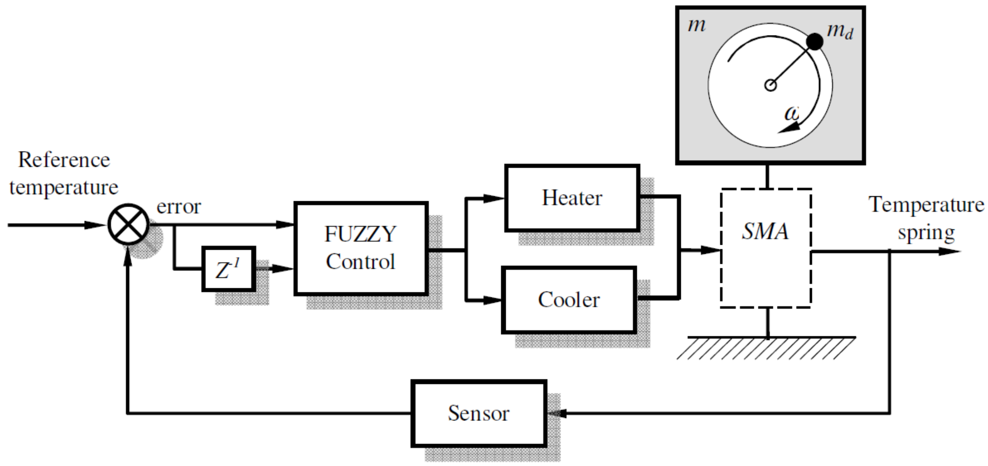


Figure 7 – Block diagram of spring temperature control

The natural frequency of mass-spring system in this study varied from 14,2 Hz (spring at 35ºC) to 17,3 Hz (spring to 63ºC), and this natural frequency must be keep as far away as the frequency of excitation. Figure 8 shows the comparison between frequency responses of experimental and theoretical system when the spring is at temperatures of 35°C and 63°C. It is observed that at frequency of 15,2 Hz, the amplitudes are theoretically equal to 1,8 mm.

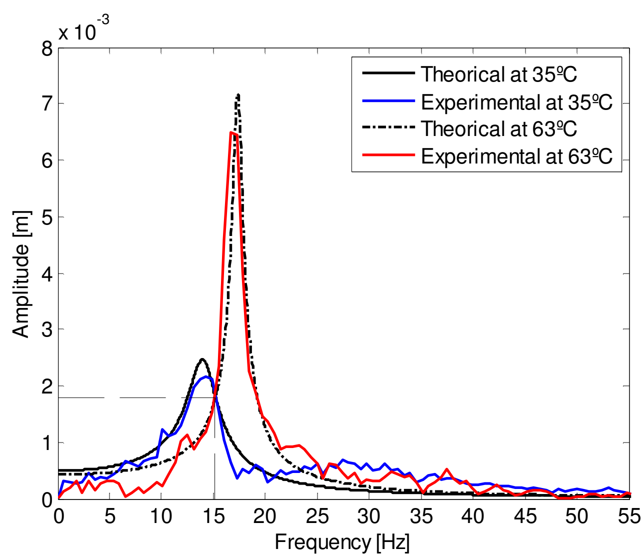


Figure 8 - Frequency Response theoretical and experimental

This excitation frequency of 15,2 Hz is a base to determine if the spring should be heated or cooled. The reference temperature SMA spring should be at 35 ºC (f_n = 14,2 Hz) when system excitation frequency present values above 15,2 Hz, making the system has a vibration maximum amplitude of 1,8 mm. If the excitation frequency present values below 15,2 Hz, the spring should be at a temperature of 63ºC (f_n = 17,3 Hz), which also results in a vibration maximum amplitude of 1,8 mm.

4. RESULTS

The system excitation was performed by rotation of an unbalanced mass coupled to a DC motor. The excitation signal spectrogram, shown at Fig. 9, presents frequencies ω_1=14,2 Hz and ω_2=17,3 Hz. This excitation frequency change occurs every 150 seconds, with a total time of signal of 750 seconds. The spring temperature was obtained through an NTC thermistor, and the mass acceleration with an accelerometer PCB Piezotronics 350B04 ™. To obtain the results of temperature, acceleration and also the motor control were used a data acquisition board USB6009 and software Labview, both products of National Instruments ®.

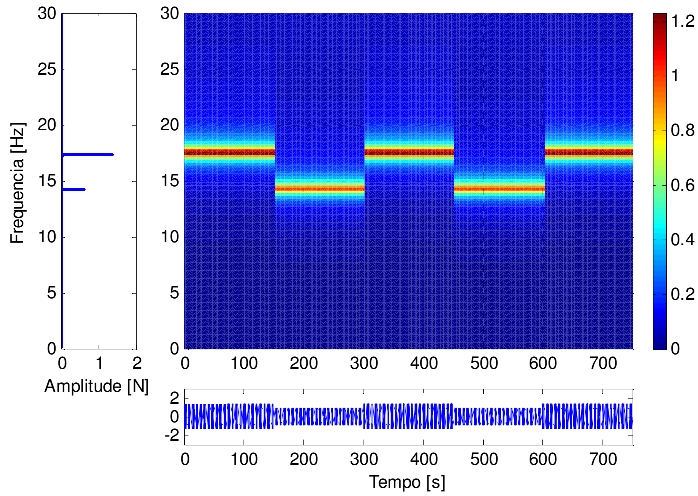


Figure 9 - Signal excitation at frequencies 14 Hz and 17 Hz.

Figure 10 illustrates the mass displacement over time. It appears that the control system reduced the vibration amplitudes in changes excitation frequency. This figure is still possible to verify the influence of damping change depending on the temperature, in times of 150seg. and 450seg.

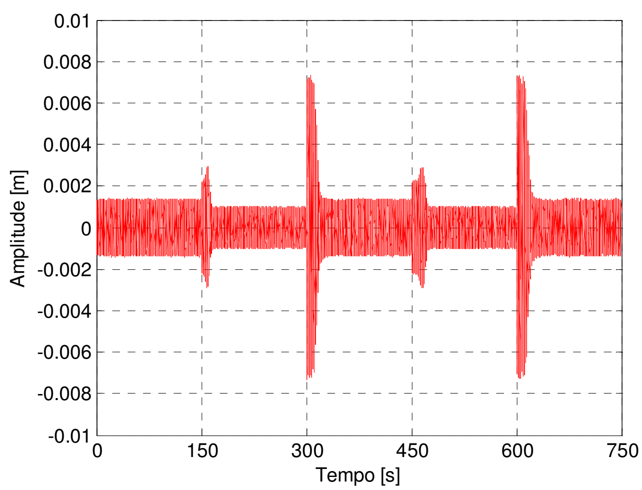


Figure 10 - Response mass displacement with system excited at frequencies 14 Hz and 17 Hz

Figure 11 shows the temperature variation over time. When the excitation signal has changed in 17 Hz to 14 Hz in the times in 150 and 450 seconds, the control system increased the spring temperature with the intention in change the natural frequency. The reference temperature in these periods increased from 35ºC to 64ºC. Note that the control initially tries to fix the beginning temperature around 35ºC. In times when the excitation signal has changed in 14Hz to 17Hz, the system again came into resonance, which forced the change of the reference temperature of 64 ° C to 35 ° C, being observed this happening at times in 300 and 600 seconds.

The time required to reach the reference temperature was 23 seconds, both for heating and for cooling. As experimental verification, the percentage of overshot during heating is 6%. However, during cooling, this percentage dropped to 1,5%.

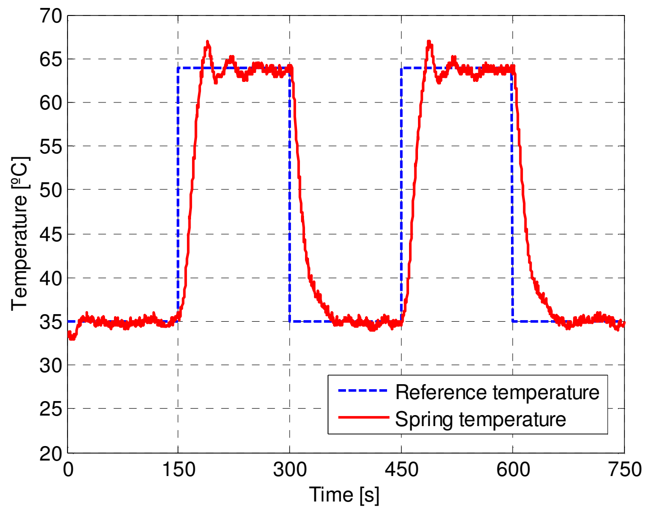


Figure 11 – Spring temperature variation with 14 Hz e 17Hz excitation

5. CONCLUSION

Despite the work it is the vibration control of a system with one degree of freedom, the controlled parameter was the temperature of a SMA helical spring, mounted on the system under study. Results of this temperature control showed that it was possible a reduction of up to 85% of the amplitudes of vibration;

As for the controller used to spring temperature control, was adopted a type fuzzy model, where there were two reference temperatures, 35°C and 63°C. The fuzzy controller showed positive results in reducing vibration levels under the conditions studied, using the strategy of choosing a reference frequency of 15,2 Hz, where there is a point of local minimum in terms of amplitude in the simulated curves;

6. ACKNOWLEDGEMENTS

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