VIBRATION ATTENUATION ANALYSIS IN A STRUCTURAL SYSTEM APPLYING PENDULUM DYNAMIC ABSORBER (DVA) WITH SMA SUPERELASTIC WIRE AND ROD

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Abstract. *Mechanical vibrations are important phenomena in the physical world. Such oscillations can become undesirable, causing temporary damage or collapse of systems. Among the control methods, passive control stands out for different excitations, such as permanents in the machines, or transient ones, such as earthquakes and wind. Its main feature is low cost, no need for electrical power and little maintenance. The objective of this work was to characterize a Dynamic Vibration Absorber (DVA) of an alloy SMA superelastic (NiTi). The device consists of pendulum attached to a structure (1DOF), with the pendulum (wire and rod) made steel or SMA. Initially, the design was developed in a computational model to estimate geometric dimensions of the primary system. The pendulum rod was dimensioned so that natural frequency of absorber was tuned to the primary system. In the experimental analysis for structure without/with the passive controller, it was concluding the need for changes in structural parameters, such as increasing the excitation and stiffness of pendulum in order to optimize the DVA, allowing a greater deformation material and generation of mechanical hysteresis. As a result, greater energy dissipation and increased damping can lead to a reduction in the system response amplitudes in subsequent works.*

Keywords: Dynamic Vibration Absorber, Shape Memory Alloys, Structural Dynamics, Passive Vibration Control.

1. INTRODUCTION

A technique developed in order to control, reduce or even eliminate unwanted vibrations is the use of Dynamic Vibration Absorbers (DVAs) or Tuned Mass Dampers (TMDs). These are secondary systems consisting of elements of mass, stiffness and damping (secondary structure) that coupled to a mechanical structure (primary structure) are capable of attenuating its vibrations in a frequency band (Sadek, 1998).

In general, there are already several devices and techniques that aim to attenuate vibration effects in engineering structures, either through the implementation of passive dynamic absorbers or even by adopting active control systems. In a new line of study, the application of new materials and alloys to reinforce structures has been investigated, such as smart and functional materials (Moraes, 2017 and 2018). These materials, usually used as sensors and actuators in so-called intelligent structures, can have their stiffness, natural frequencies, among other mechanical properties altered, through the imposition of electric, electromagnetic, temperature or voltage fields, as an example, it can be mentioned Shape Memory Alloys (SMA). Another feature worth mentioning in the context of reducing undesirable vibrations is the great ability of these alloys to dissipate energy (Lagoudas, 2008).

Shape Memory Alloys are metallic alloys that undergo transformations between solid phases induced by temperature changes and/or appropriate stresses and that can recover apparently permanent deformations (Otsuka and Wayman, 1998). One of the characteristic properties of SMAs is superelasticity, characterized by the recovery of deformations resulting from martensitic transformations induced by stress. This phenomenon occurs after the applied load is removed and is observed in SMAs subjected to temperatures in which the alloy is in the austenitic phase. In this load application and removal cycle, the material has a high energy dissipation capacity due to the presence of a hysteresis loop related to the material stiffness (DesRoches, 2003). In addition to applications in static regime, superelasticity is very useful to act in dynamic regime thanks to its ability to dissipate mechanical energy in each stress-strain hysteretic loop. In this context, numerous devices using superelastic SMA as vibration and impact absorbers have been developed, both in macro and microscale (Oliveira, 2014; Chang, 2016).

Therefore, selecting a more efficient geometry for the application of the smart material is a challenge, considering that the dissipative power of the metallic alloy is directly related to the geometric shape of the mechanical element and the way in which it deforms and generates loops hysteresis. The main objective of this work is to verify the efficiency of vibration attenuation in a simplified building structure that applies two types of geometries for the pendulum of DVA

system, initially testing the use of circular wire and prismatic steel rod and subsequently comparing with the SMA material. The conclusions obtained from the dynamic behavior results will serve as a basis for future design and optimization of these devices.

This paper is structured into seven main topics. The first section contextualizes the study area with the respective problems, objectives and expected results of the research. Section 2 presents the methodology, equipment used in experimental tests and the mathematical formulation used in the analytical calculation of structural parameters. In section 3, the results and discussions of the hypotheses are explored, followed by conclusions corroborated by the observed results (section 4). Sections 5, 6 and 7 describe the acknowledgments to supporters, references and notice of responsibilities for the copyright of the work, respectively.

2. METHODOLOGY

The methodology used in this study consists of practical experimentation of a structural prototype using a bench with a data acquisition system. The designed test system simulates a one-story building (primary system), and a pendulum system (secondary system) is mounted on its upper floor. The design of the primary system was limited in its structural parameters (mass and stiffness) to meet excitation and sensing system requirements, such as natural frequencies and displacement amplitudes. The design of the system was carried out computationally using software such as @MatLab and @SolidWorks, as well as previous simulations of the dynamic behavior (modal analysis) in an analytical and numerical way, in order to validate and compare the results. In the analytical method the solution of dynamic matrix was used as an eigenvector problem in which the determined eigenvalues correspond to natural frequency of vibration the structure without and with the implementation of DVA. The numerical simulation carried in @SolidWorks aims to visualize the modal behavior of the system with the implementation of secondary pendulum system, observing the material deformation in the respective modal forms of vibration.

The framed structure behaves as a system with a single degree of freedom under the influence of an external excitation. The DVA is installed in its upper floor in order to provide vibration reduction in the amplitudes at critical points, where the resonance phenomenon occurs. In this work, two (02) geometries will be applied to the SMA tuned pendulum, a wire with a circular section and a rod with a prismatic section, and one (01) steel rod with a prismatic section with the objective of analyzing and comparing their efficiencies in controlling the structure's vibrations.

2.1 Experimental procedure

The work was developed at the Mechanical Engineering Academic Unit (UFCG), and had the support of the Vibration Laboratory (LVI). To accomplish the lateral excitation in the framed structure, a shake table model Shaker II from the Quanser manufacturer was used. To acquire the response of the structure and the force excitation two LVDT's were used, a WI/10mm LVDT was used to monitor the shake table movement, while a WA/20mm-L was used to capture the response of the upper floor of the structure.

The data acquisition device model HBM MX840B captured all points in the experiment using a sine wave sweep, in a frequency range of 1 to 10 Hz, for 1 minute and 40 seconds, with a 2400 Hz sample rate.

The experimental results will be presented in the complete version of the work. The main equipment and measurement chain used in the experiment are listed below (Figures 1 and 2):

• Amplifier for high power systems UPM 180-25B;

• Shake Table II Quanser with a mass of 27.2 kg, 400 W servomotor, accelerometer with sensitivity of 9.81 m/s²/V;

• Inductive Displacement Sensor WI/10MM-T and WA/20mm-L;

• QuantumX HBM MX840B 8-Channel Universal Data Acquisition System

• Computer (Workstation)

The structural design was developed in a computational environment with the aim of estimating the geometric dimensions of the system (Figure 3a), such that it met the criterion of having a first natural frequency below 5 Hz, due to limitations of the excitation system (Shake Table).

The model was built with type SAE 1020 carbon steel bars for the pavement composition, type 304 stainless steel rectangular plates for the columns composition, and as connecting elements, common steel M6 screws. To connect the corner bracket to the upper mass, an M5 screw was used and the corner pieces were joined using two M4 screws, totaling 970 grams that will be used for the experiment calculations.

In this work, a pendular dynamic absorber with a Nickel-Titanium (Ni-Ti) SMA wire attached to the mass was used. It has a superelastic effect at room temperature. The system assembly is shown in Figure 4 and its components are shown in Table 1.
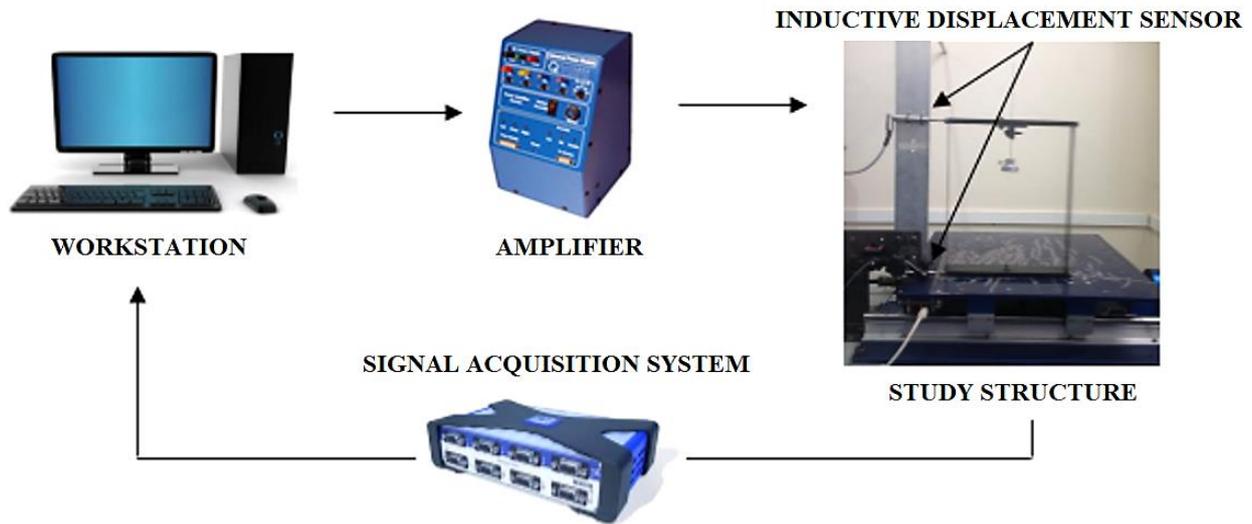

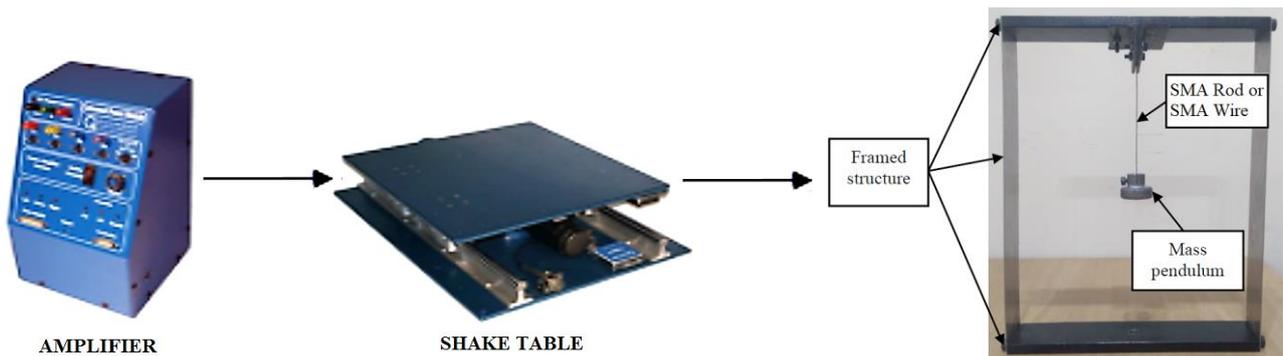Figure 1. Measurement chain used in the experiment. Source: Own authorship.

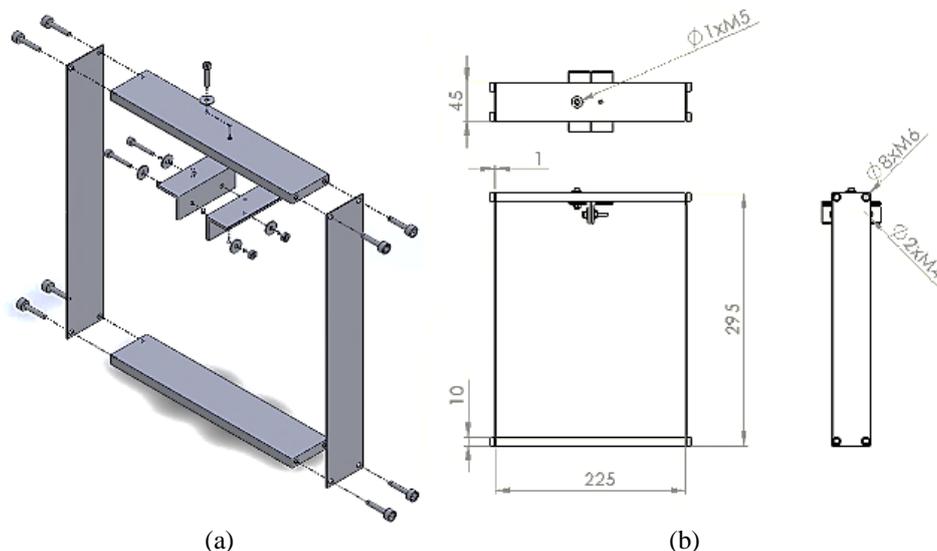Figure 2. Instrumentation chain for base motion analysis. Source: Own authorship.

(a)   (b)
Figure 3. Simplified technical drawing of the structure. a) Isometric view. b) Front view and dimensions.
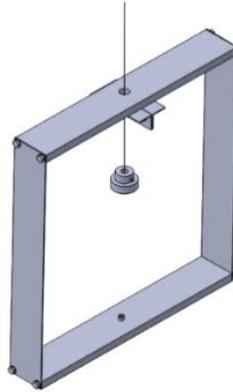Source: Own authorship.

Figure 4. System with wire pendulum absorber. Source: Own authorship.

Table 1. Absorber components. Source: Own authorship.

Description	Dimensions (mm)	E (GPa)	Mass (kg)	Amount (Und.)
Mass (pendulum)	Ø30 x 10	200	0,0807	1
Phillips screw	4 x 20	193	0,0047	1
SMA Wire	Ø 0,9	45	-	1
SMA Rod	0,5 x 8	45	-	1
Steel Rod	0,6 x 10	193	-	1

2.2 Mathematical formulation

With the mass of the absorber system defined by Table 1, it must have a transverse stiffness k_2 such that its natural frequency is equal to that of the primary system, therefore:

$$\omega_{n1} = \omega_{n2} \rightarrow \sqrt{\frac{k_1}{m_1}} = \sqrt{\frac{k_2}{m_2}} \rightarrow k_2 = \frac{m_2 k_1}{m_1} \tag{1}$$

Where m_1, k_1, m_2, k_2 are the mass and stiffness of the primary system and the mass and stiffness of the absorber system respectively.

The transverse stiffness k_2 is a function of the length of the absorber wire, therefore, as this has already been defined through Eq. (1), the ideal wire length will be given by Eq. (2).

$$l = \sqrt[3]{\frac{3EI}{k_2}} \tag{2}$$

In the system of Figure 5, similar to the gantry used in this work, the length L, height h, modulus of elasticity E and moment of inertia I_b for the main mass and I_c for the support columns of the gantry to which they are attached to the base are represented. Eq. (3) is used to calculate the stiffness of each column.

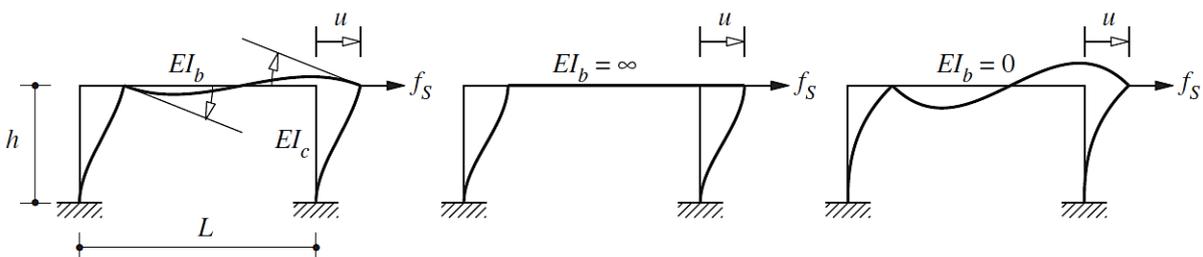$$k = \sum_{Colunas} 12\frac{EI_c}{h^3} = 24\frac{EI_c}{h^3} \tag{3}$$

Figure 5. Free body diagram of a frame subject to lateral force fs. Source: Adapted from Chopra, 1995.

In this method, the damping measurement is based on the frequency response curve. Bandwidth (at half power) is defined as the width of the frequency response curve when the magnitude (Q) is $\left(\frac{1}{\sqrt{2}}\right)$ times the peak value. This value is denoted by $\Delta\omega$, as can be seen in Figure 6 (Rao, 2011).

The value of $\Delta\omega$ can be related to damping as follows:

$$\Delta\omega = \omega_2 - \omega_1 = 2\zeta\omega_r \tag{4}$$

And therefore, the damping can be estimated across the bandwidth, using the relation:

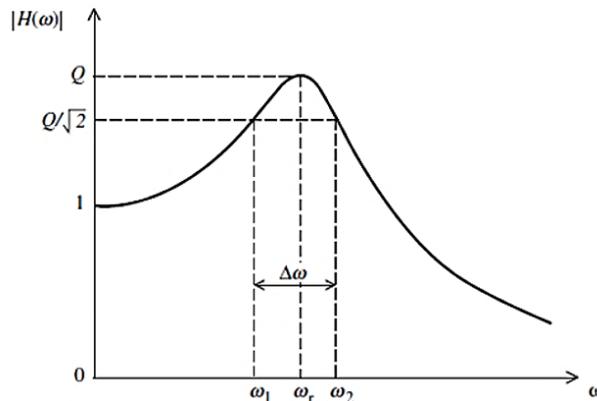$$\zeta = \frac{1}{2}\frac{\Delta\omega}{\omega_r} \tag{5}$$

Figure 6. Bandwidth method in a system with 1 DOF. Source: Adapted from Rao, 2011.

3. RESULTS AND DISCUSSIONS

This section presents the experimental results of the two tests applied to the SMA tuned pendulum geometries. The first test applying a wire with a circular section and the second test applying a rod with a prismatic section. Finally, a common steel rod with a prismatic section is applied in the second test with the aim of analyzing and comparing the efficiency of the smart material in controlling the structure's vibrations.

3.1 Experimental results for DVA with SMA wire

The respective graphs of the frequency spectrum are illustrated in Figure 7(a) and (b). The dynamic behavior of the structure is remarkable, with the system amplitude peaks and the anti-resonance region being able to be visualized for the system configurations with attenuator pendulum.

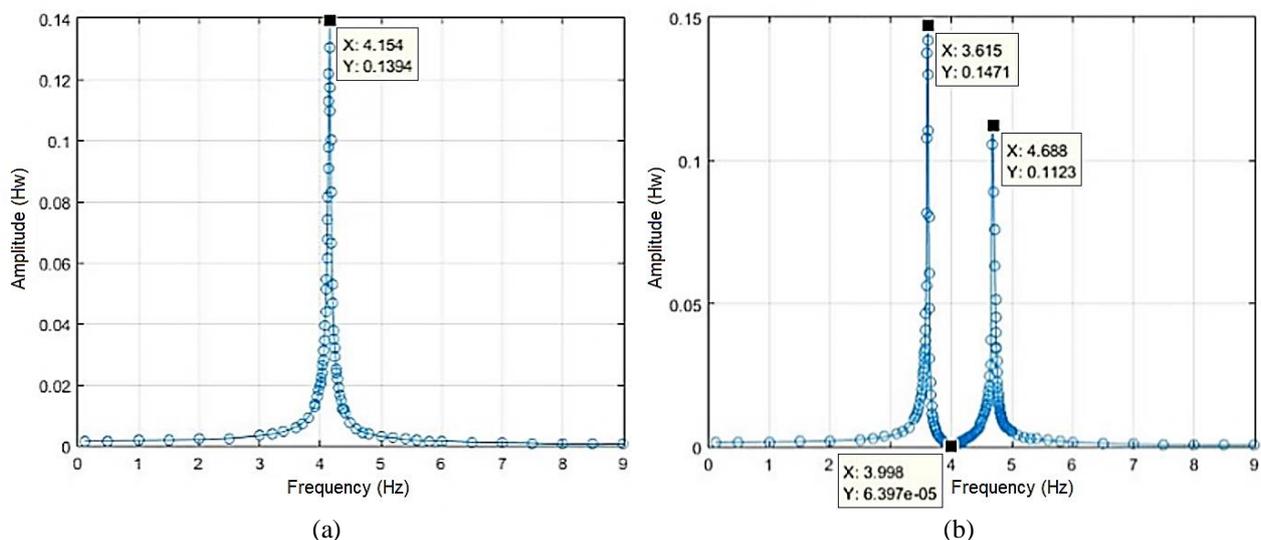(a)   (b)

Figure 7. Experimental frequency response function (FRF) of the system. a) Structure with 1 DOF. b) Structure with DVA and 2 DOF. Source: Own authorship.

Tables 2 and 3 present the values of the natural frequencies obtained, where it is possible to verify small relative errors for the natural frequencies, validating the mathematical modeling in the characterization of dynamic systems of this type.

Table 2. Comparison of the natural frequencies of the structure of 1 DOF. Source: Own authorship.

Natural Frequencies	Experimental	Analytics	Relative Error (%)
1st natural frequency (Hz)	4,15	4,14	0,24

Table 3. Comparison of the natural frequencies of the structure of 2 DOF. Source: Own authorship.

Natural Frequencies	Experimental	Analytics	Relative Error (%)
1st natural frequency (Hz)	3,61	3,58	0,83
2nd natural frequency (Hz)	4,69	4,78	1,88

In the graph of Figure 8 it is possible to observe the bandwidth method to calculate the damping involved in the FRF of the system with the absorber (2 DOF) of the system in forced vibration obtained in the tests with the Shake Table.

First, the amplitude value in resonance of mode 1 (Q_1) of 0.1471 was extracted from this graph. The half power value ($Q/\sqrt{2}$) will then be 0.1041 (first horizontal line). This half-power value is linked to two frequencies in the FRF that can be found by interpolating the points closest to this value. The frequency values shown below were found.

Doing a similar procedure for the second mode of vibrating, Q_2, we obtain the two natural frequencies related to the half-power value:

$\omega1,1 = 3,5983$ rad/s
$\omega1,2 = 3,6250$ rad/s
$\omega2,1 = 4,6648$ rad/s
$\omega2,2 = 4,7062$ rad/s

With these values it is possible to estimate the damping, in the form:

$$\zeta_1 = \frac{\omega_{1,2} - \omega_{1,1}}{2\omega_{n1}} = \frac{3,6250 - 3,5983}{2 \ x \ 3,615} = 0,0037$$

$$\zeta_2 = \frac{\omega_{2,2} - \omega_{2,1}}{2\omega_{n2}} = \frac{4,7062 - 4,6648}{2 \ x \ 4,688} = 0,0044$$
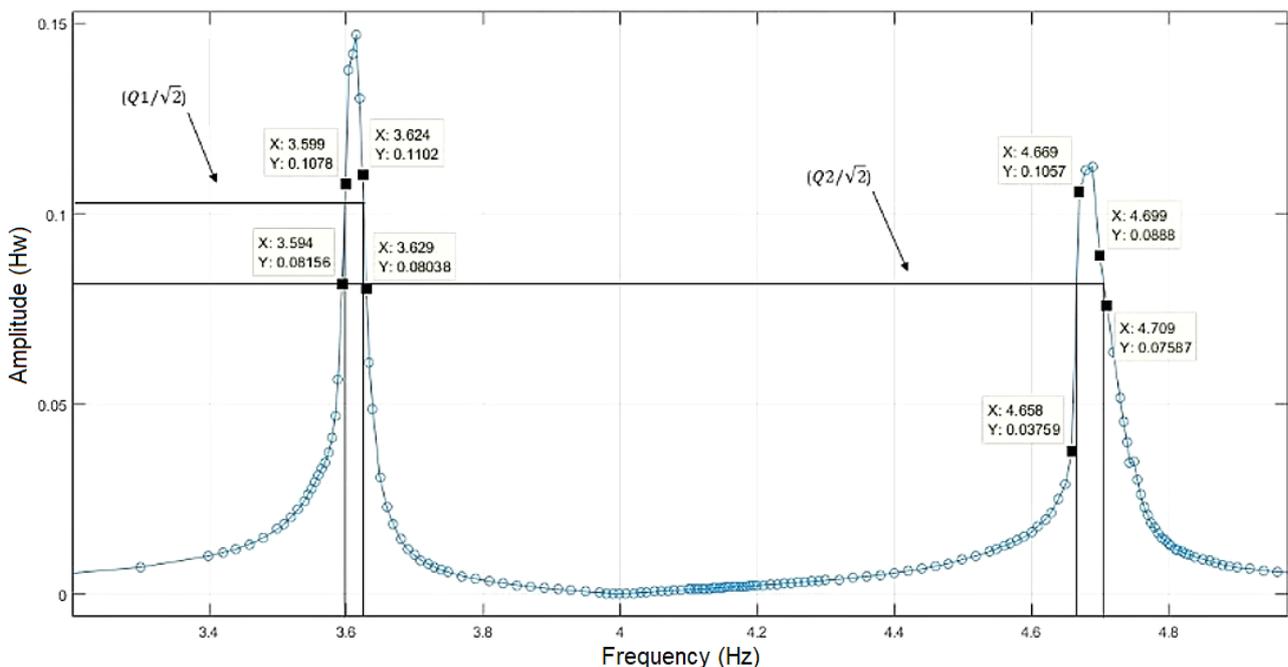

Figure 8. Bandwidth Method in the FRF of the structure (2 DOF). Source: Own authorship.

From the results found using the method above, it is possible to observe that the damping values are relatively low, this is due to the following factors:

• The wire diameter is small and did not allow a great deformation of it;

• The excitation force used in the Shake Table had to be reduced and controlled due to the use of LVDT's and the limitation of their strokes (10 mm).

Therefore, it was not possible to work completely inside the hysteretic loop of the SMA, taking full advantage of its phase transformations due to the low voltages suffered by the wire.

3.2 Experimental results for DVA with SMA and STEEL rod

The Figure 9(a) illustrates the frequency spectrum graph. The dynamic behavior of the structure is typical, with a single amplitude peak for the framed structure and the presence of the anti-resonance region for the system with the DVA. It is possible to note that, in addition to the anti-resonance region, the steel rod DVA has created two new natural frequencies in the system, which is expected. However, these regions have lower response amplitudes than the non-DVA structure in its region of resonance. Table 4 illustrates the values of the different response amplitudes.
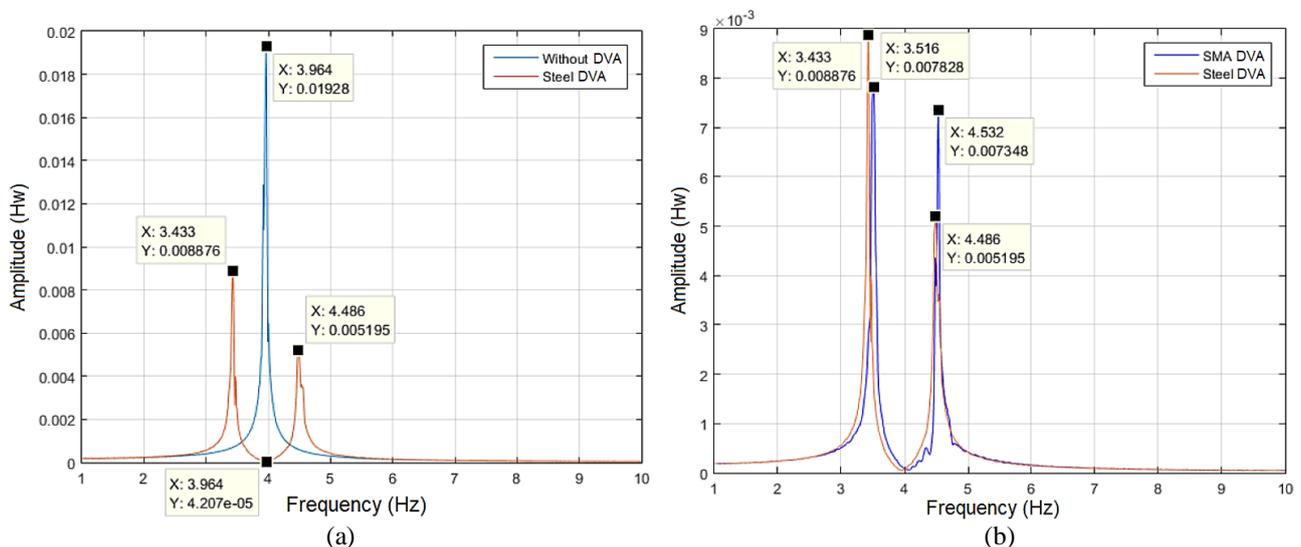

Figure 9. Experimental frequency response function (FRF) of the system. (a) Structure without and with the Steel DVA. (b) Structure with the SMA and Steel DVA. Source: Own authorship.

Table 4. Comparison of the response amplitudes. Source: Own authorship.

Configuration	Non-DVA Structure	Structure with Steel DVA		Structure with SMA DVA	
Frequency (Hz)	3,96	3,43	4,49	3,52	4,53
Amplitude (m/N)	0,0193	0,0089	0,0052	0,0078	0,0073

In the anti-resonance region, the amplitude reduction reached a level of 99.8%, which was once again expected, given that this is the function of the DVA use. In the regions of the two new natural frequencies, the structure response values are 54% and 73% lower, for 1st and 2nd natural frequencies, respectively, compared to the resonance region of the non-DVA structure.

The Figure 9(b) illustrates the frequency spectrum graph of the structure with the SMA DVA and Steel DVA. By making a comparison between the behavior of the structure with the SMA DVA and Steel DVA, there is a slight difference between the two natural frequencies, this is due to the nonlinear properties of the SMA material, changing phases during the experiment. It is also visible a decrease in the response amplitude of the 1st natural frequency of the structure (1st peak) as it was expected, due to the dissipative effects of the SMA material. However, the same behavior does not repeat itself in the 2nd natural frequency of the structure (2nd peak), thus having an increase in the response amplitude of the latter. Table 4 shows the values of the different response amplitudes.

This behavior may be a consequence of the different modal shapes the structure has in its two first natural frequencies, as illustrated in Figure 10. Since in the first modal shape (1st natural frequency) the DVA oscillates with the same phase (in the same direction) as the primary structure, as shown in Figure 10(a), it has more energy to strain (and stress) the SMA rod, thus generating the superelasticity phenomenon (the transformation between austenite and martensite due to a

mechanical stress) and, consequently, dissipation of vibrational energy. In the second modal shape (2nd natural frequency) the DVA oscillates in the opposite direction of the primary structure, as shown in Figure 10(b), hence it has less energy to strain the SMA rod and, therefore, less capacity to dissipate energy.
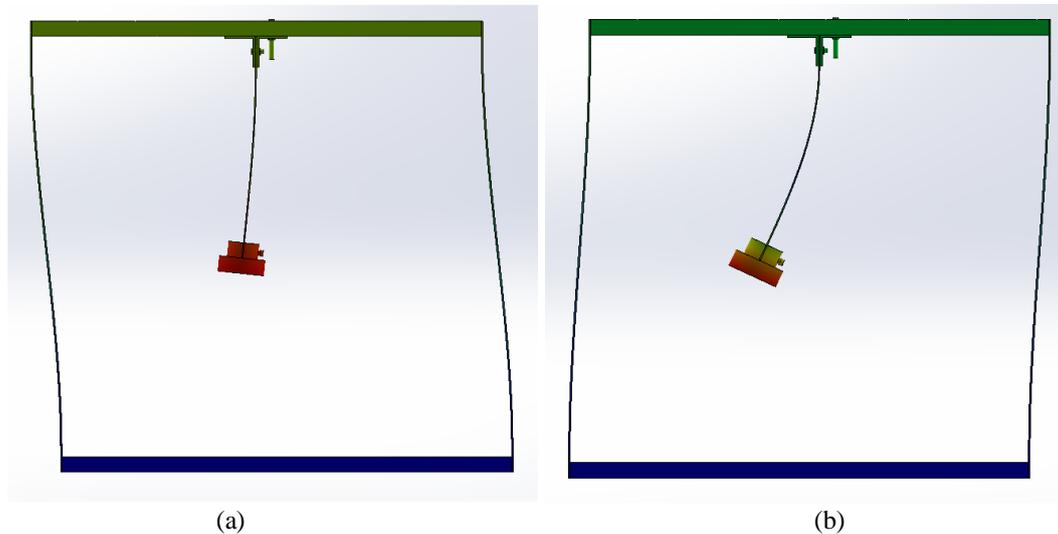

(a)   (b)

Figure 10. Modes of vibration. (a) 1st vibration mode. (b) 2nd vibration mode). Source: Own authorship.

4. CONCLUSIONS

In this work, the vibration attenuation efficiency of a simplified building structure that applied two DVA's systems was verified, analyzing the variation of the pendulum geometry with the use of steel (wire and rod) and SMA. Analytical and experimental modal parameters of natural frequencies, stiffness and damping estimates of a gantry-type system with 1 DOF (reference) and 2 DOF (when an absorber pendulum is incorporated) were found, based on impulsive lateral excitation.

In the first analysis, a good approximation of the experimental results of the natural frequencies was verified when compared with the simulation carried out through an algorithm developed in Matlab®, obtaining errors of the order of 0.83% for the first and 1.88% for the second frequency nature of the structure.

An experimental analysis was carried out for the structure without and with incorporation of SMA wires through a pendulum and verifying changes in important parameters, such as stiffness and damping variations, in order to make the necessary adjustments for a good performance of the absorber system. It was possible to observe an antiresonance region generated by the DVA, allowing the main system to be excited at its original natural frequency and not suffer any damage associated with the mechanical vibrations transmitted by the imposed excitation.

The study also verified the experimental parameters, such as natural frequencies and stiffness of the same system, using a lateral sinusoidal excitation with the incorporation of steel rods and the SMA material. Three different experimental analyzes were carried out for the structure without any DVA, with DVA steel rod and with SMA DVA rod, thus verifying changes in important parameters, such as stiffness variation, in order to make the necessary adjustments for a good performance of the o vibration absorber.

As in the previous system (DVA with SMA wire), the anti-resonance region generated by the DVA's allowed the main system to be excited at its original natural frequency and not suffer any damage from vibrations. It was possible to observe the energy dissipative properties of the SMA material reducing the response amplitude by up to 11.8%, increasing the damping of the system in its first modal form, but the same was not visible in its second modal form due to possibly insufficient transformations phase shifts in the SMA material generated by the smaller excitation amplitudes.

Analyzing the effectiveness of the two types of DVA's systems (wire and rod) it was not possible to identify more expressive results in the comparison of the geometry of these elements. It was possible to identify that the element deformation amplitude, related to the phase transformations of the material, regardless of its geometry, is much more relevant in the dissipative power and efficiency of the attenuator. In this way, a refinement in the design of the pendular system is necessary to increase the percentage of deformation of the SMA pendulum, increase the hysteresis loop generated and the material damping.

5. ACKNOWLEDGEMENTS

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7. RESPONSIBILITY NOTICE