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Design, simulation and validation of a viscoelastic damper for structural vibration reduction

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Abstract. Passive dampers are used to reduce structural vibration levels in a wide range of applications. Rubber is still one of the most widely used materials. However, to simulate the viscous-elastic properties, complex dynamic models are frequently required. In this project, we designed, synchronized, and built a viscous-elastic damper for use on a steel frame structure. FRF was used to identify the structure's initial modes, and vibration levels were measured using a shake table and an impact hammer. To describe and simulate the entire system, Finite Element Models were developed. To linearize the viscous-elastic damper behavior we built a sample and tested it to estimate equivalent proprieties values. Based on those equivalent properties, the mass and shape of a new sample were adjusted to synchronize the frequency and optimize the damper effect. The resulting geometry was fabricated and applied at the structure. The structure was excited over again and the vibration levels were compared to previous results. The results show a 48.7 percent reduction in vibration levels with only a 8.3 percent increase in mass (damper mass). The experimental tests and simulation comparison also demonstrates good agreement.

Keywords: Experimental modal tests, Simulation comparison, Finite Element Method, Dynamic viscoelastic Damper, Passive Vibration Control.

1 Introduction

Designer engineers are common dealing with an environment full of vibration sources, the most diverse structures spread around the world are constantly working under those excitation. The structure's response is directly related to its natural frequencies, and as the excitation approaches one of these frequencies, the structure may enter into resonance, causing large vibration amplitudes [1]. This effect may cause serious damage on the structure, loosening of screws, excessive bearing wear, crack formation and even structural and mechanical failure [2]. In this context, the application of dynamic passive vibration absorbers corresponds to a method commonly used to prevent structural damage from vibrating sources [3]. Between the different passive control options the mass damper with viscoelastic material are large used [4]. Although viscoelastic materials have good damping capacity, it is necessary to determine and characterize properly the properties of elastomers. According to Mainardi [5], Bagley and Torvik [6] and Silva Neto [7] these properties depend on the excitation amplitude, frequency, temperature variation and static preload conditions. Hence the importance of knowing the environment in which the absorber will operate, as environmental factors, as mentioned above, interfere at its capacity and work efficiency.

In order to lower the vibration levels of the structure, this work aims to determine the initial mode dynamics of a simplified structure and develop, produce, and verify a viscoelastic vibration mass absorber. Using a straightforward approach for dimensioning and synchronization using Finite Element Models this work also compares the simulations to the experimental testing indicating acceptable levels, in addition to successful reductions in the structure's vibration level.

The study is divided into sections that include experimental setup and data analysis, a thorough explanation of the main structure's test and simulation processes, the characterization of the viscoelastic parts, and damper absorber tuning. At last a part where the findings and conclusions are described.

2 Experimental Set-up and data acquisition

To validate the efficiency of the dynamic vibration absorber, the main structure and a viscoelastic vibration absorber were used, built at the Laboratory of Vibrations and Instrumentation (LVI) at the Academic Unit of Mechanical Engineering at the Federal University of Campina Grande. Figure 1 illustrates an experimental setup, which highlights the main frame coupled to the absorber device and the instrumentation used in the modal tests with an impact hammer.

The values of excitation (N), acceleration response (m/s²/N), and the Frequency Response Function (FRF) of the Response/Excitation ratio for all tests were collected using a PCB Piezoelectronics 352C68 accelerometer, PCB impact hammer Piezoletronics 086D05, and the signals were acquired on an Agilent 35670A dynamic signal analyzer. The data collected by the dynamic signal analyzer were exported in universal format (.unv) and processed with the help of Altair®HyperGraph®.

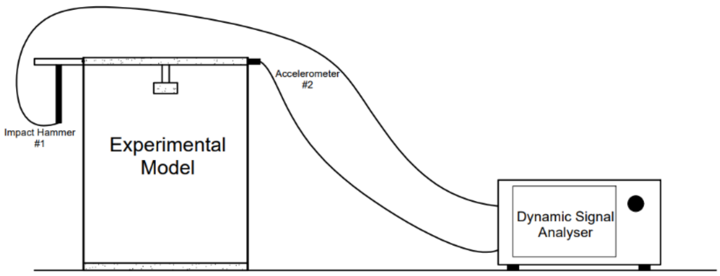

Figure 1: FRF test data acquisition system.

3 Test Procedures

3.1 Main Structure

The simplified frame that will receive the vibration absorber is made up of two columns of thin stainless steel blades, that give the system's stiffness, and two rectangular steel bars that serve as the structure's base support and a the main mass, respectively. Initially, to determine the dynamic parameters of the main structure, FRF tests were performed, considering a frequency range of 0-100Hz. The Figure 2(a) presents the dimensions of the main structure and Figure 3(b) presents the respective excitation points and data collection of the dynamic movement of the structure. The upper mass is 0.760 kg.

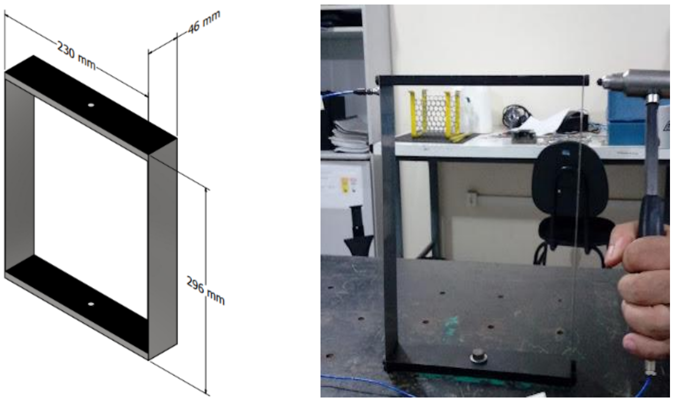

Figure 2: (a) Dimensions of the main structure model. (b) Schematic of the experimental setup of the FRF structure.

3.2 Characterization of the viscoelastic element

To characterize the viscoelastic element, a sample was constructed, with a square section (30x30mm) rubber rod (57mm) fixed to a steel block (block mass = 0.560 kg). We performed FRF using impact excitations and read the acceleration in different directions to estimate the equivalent viscoelastic properties such as modulus of elasticity (E), density (ρ), and structural damping (GE). Similar to the main frame structure, we performed the frequency response functions, but in this case all the 3 directions was characterized. Figure 3(a) shows the dimensions of this sample, and in figure 3(b) it is possible to visualize the setup assembled for the FRF tests.

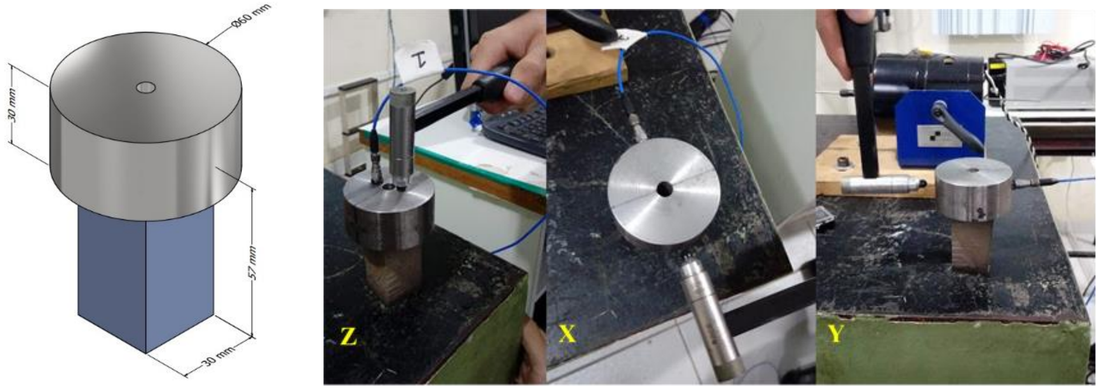

Figure 3: (a) Dimensions of the pendular sample; (b) Modal tests to obtain the FRFs in the sample.

3.3 Tuned Dynamic Absorber

With the rubber properties properly characterized, a finite element model was built and from that, it was possible to find the ideal dimensions for the construction of a tuned absorber, as showed at figure 4(a), in figure 4(b) we can see how the experimental test was carried out to confirm the tuned frequency. Tuned Absorber mass is 0.08 kg.

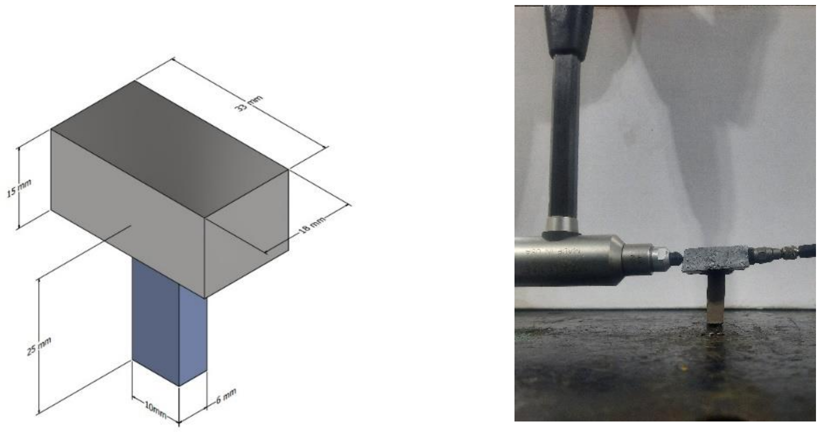

Figure 4: (a) Dimensions of the tuned vibration absorber; (b) Inertance test done for FRF

4 Simulation Procedure

Finite element models were built in Altair® Hypermesh® using elements most of the hexahedral type, as they better fit the geometry (figures 5 (a) and 5 (b)). From the data collected from the experimental tests, we built table 1 with the input data of the respective properties of the materials. Initially, models were built to represent the structure and the sample whose properties were calibrated from the test results (Table 1).

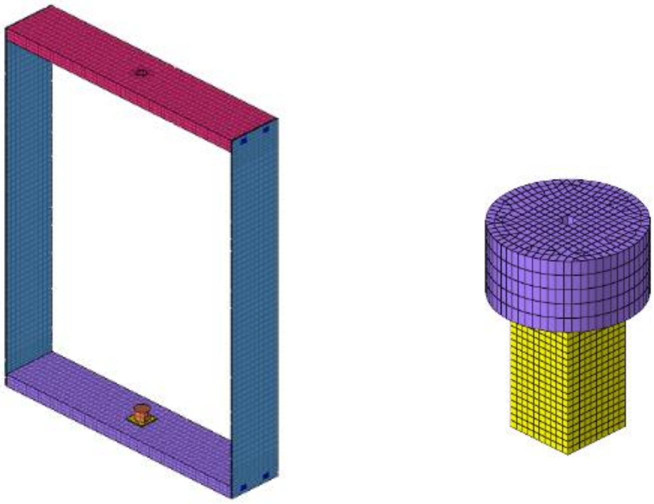

Figure 5: (a) Computational model of the structure; (b) Computational model of the sample.

Table 1: Properties used as input data for the computational model

Mainframe material properties	
Modulus of elasticity (MPa):	172000
Poisson's coefficient:	0.33
Density (kg/m³):	7790
Structural Damping:	0.0105
Vibration absorber material properties	
Modulus of elasticity (MPa):	2.1
Poisson's coefficient:	0.48
Density (kg/m³):	1790
Structural Damping:	0.045

Through the properly calibrated computer model in finite elements, we virtually tuned an absorber model, verifying that the dimensions presented in Figure 5 (a) would be the most adequate, considering that the absorber with these characteristics has the same resonance frequency as the main structure. A finite element model of the tuned absorber was also built (figure 6(a)), after which the structure model was assembled with the tuned vibration absorber attached to it (figure 6(b)).

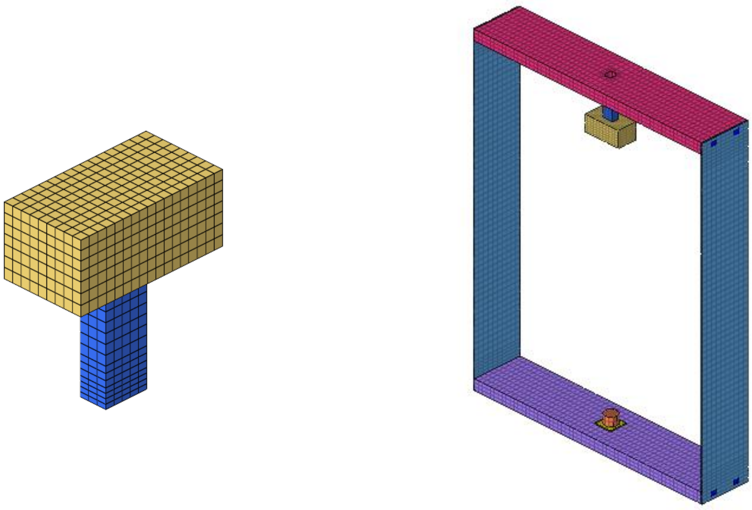

Figure 6: (a) Computer model of the tuned vibration absorber; (b) Computational model of the structure.

5 Results and discussions

The experimental results and the computer simulations showed a relatively satisfactory coherence, where in figure 7 the frequency response functions (FRF) are presented in the longitudinal direction for the test and simulation of the main structure without the vibration absorber

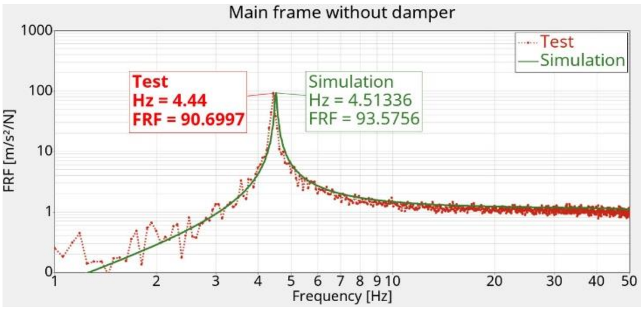

Figure 7: FRF of the main frame

Analyzing Figure 7, we can observe a peak in the natural frequency at ft =4.44 Hz with an amplitude of At =90.70 m/s²/N in the experimental test results and a peak at fs =4.51 Hz with an amplitude of As =93.58 m/s²/N in the simulation, whose values served as parameters for the construction of the tuned vibration absorber, which must be as close as possible to these values obtained for the main structure.

The graphs with the results of the inertance tests in the X and Z directions with the sample manufactured to determine the properties of the viscoelastic element can be seen in figure 8. The results for the Y direction were suppressed due to the similarity with the X direction, having since the part presents symmetrical geometry in these axes, these results served as a guide for determining the input data for the finite element model of the vibration absorber.

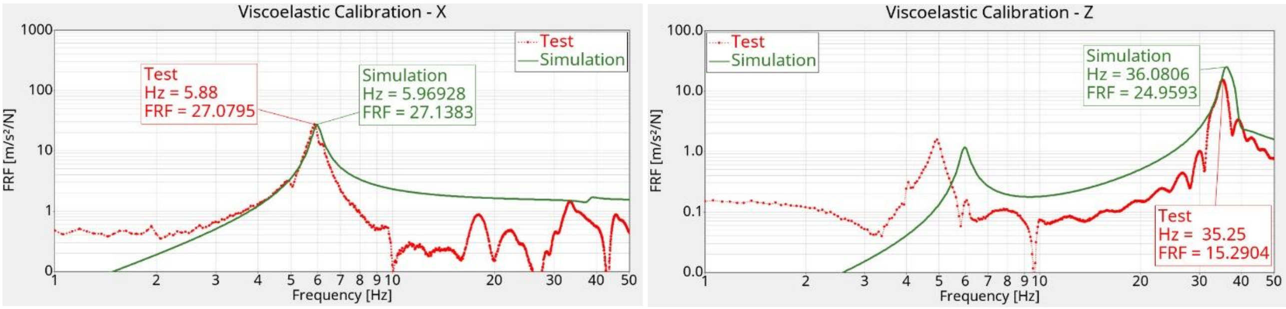

Figure 8: Results of the FRFs for the sample in the X and Z directions.

The FRF results obtained in the tuned absorber are shown in figure 9 where we observe a resonance peak at 4.81Hz with an amplitude of 13.06 m/s²/N.

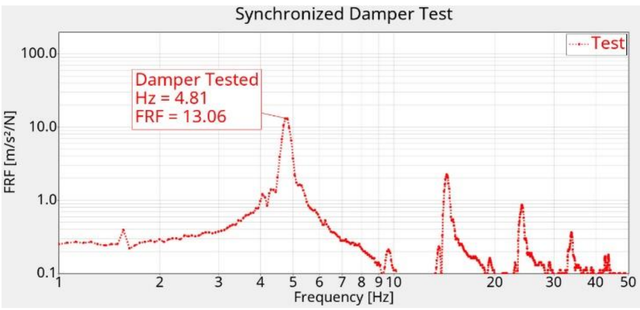

Figure 9: Results obtained for FRF's of the tuned absorber.

In figure 10, another comparison is made, still within the scope of comparing the experimental method with the computational one, where it is possible to visualize the results obtained from the structure with and without the vibration absorber, both experimentally and computationally.

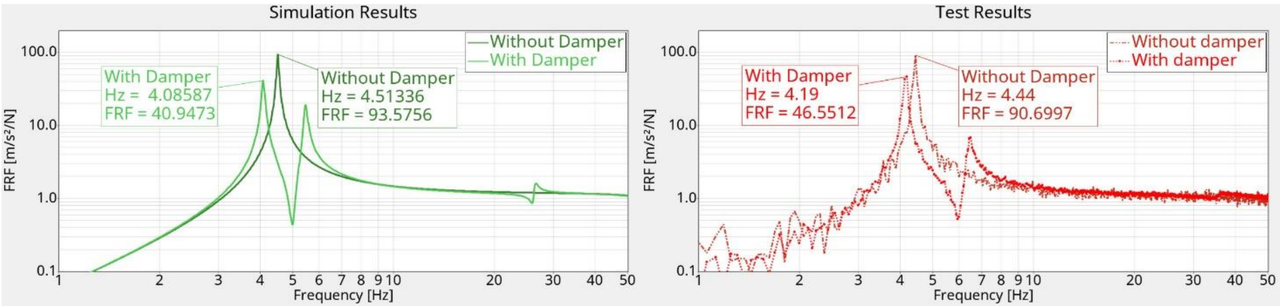

Figure 10: Comparison of the structure with and without vibration absorber

In the simulations, the observed results of the structure with the vibration absorber installed, show a reduction of 56.2% of the structure's vibrational response. As for the experimental results, the application of the vibration absorber showed a reduction of up to 48.7% in the vibrational response of the structure. From experimental tests and simulations, comparisons were made about the results obtained. Table 2 presents a comparison of the models with their respective calibrated properties, including percentage errors.

Table 2: Comparison between the results obtained

Description		Measurement	Simulation	Δ
Main Frame Without Damper		4.44 Hz	4.51 Hz	1.6%
		90.70 m/s²/N	93.58 m/s²/N	3.2%
Viscoelastic Calibration	X/Y Directions	5.88 Hz	5.97 Hz	1.5%
		27.08 m/s²/N	27.14 m/s²/N	0.2%
	Z Direction	35.25 Hz	36.08Hz	2.4%
		15.29 m/s²/N	24.96 m/s²/N	63.2%
Vibration Reduction Level		48.70 Hz	56.20 Hz	15.4%

The differences found in the structure without the vibration absorber differ by 1.6% in frequency and 3.2% in amplitude. For calibration of the viscoelastic element, preference was given to the X and Y directions (cross-section) because it is the frequency range of interest. In this aspect, we obtained satisfactory results in both directions with differences in the range of 1.5% for frequency and 0.2% for amplitude. In the longitudinal direction (Z) the difference was more relevant in amplitude, but with only a 2.4% difference for frequency. In the last comparison, the structure with the vibration absorber installed showed an acceptable difference in the results.

6 Conclusions

Through the investigation of the dynamic response of a structure with and without the vibration absorber, designed, built, and validated in this work we observed that the produced vibration absorber was able to lower the vibrational responses of the structure by 48.7 percent. This proved to be a highly effective solution and simple to design.

When the simulation and experimental results were compared, they are globally close. The minor deviation could be caused by to model simplification and geometric differences resulting from the handmade fabrication of the viscoelastic samples besides the representation of the boundary conditions.

More research into the experimental behavior of the viscoelastic element in terms of sensitivity to frequency, amplitude, and temperature is required. A comparison of the results obtained with a viscoelastic analytical model would also be done.

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