DESIGN AND ANALYSIS OF A TUNABLE VIBRATION ABSORBER

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Abstract. *Vibration control is an essential topic in design and maintenance of machines and structures. The common classification of vibration control comprises four methods: passive, active, hybrid and adaptive-passive. Passive control is the most traditional and is accomplished by using reactive or resistive devices, commonly, springs and dampers. Active control consists of adding an applied force to the machine, part or structure under consideration to attenuate vibration level. Hybrid Control combines passive approach and active control, so that less amount of external power is necessary. Adaptive-passive control involves the use of a passive device with adjustable parameters and a specific tuning strategy, suitable for the application. The main subject of this study is the dynamic vibration absorber, DVA. This device combines inertia, stiffness and damping elements, in order to attenuate harmonic excitations, that is, tonal disturbance. The tuning is provided by matching the absorber natural frequency to the excitation frequency. A prototype of this device is built according to adaptive-passive control. The modification of the absorber natural frequency is provided by adjusting the stiffness value. Finally, experimental verification is presented to evaluate prototype performance.*

Keywords*: Vibration control; semi-active control; modal analysis; measurements*

1. INTRODUCTION

In the study of Dynamics and Vibration of machinery in general an area with increasing interest is the active vibration control. Among the possible classifications found in the literature about the controllers four main approaches can be mentioned: Passive Control, Active Control, Hybrid Control and Semi-Active Control, FRANCHEK et al.

The passive control scheme is traditional and involves the use of passive elements, reactive ones such as dissipative mechanisms and/or resistive ones, springs for instance, coupled to the target system. The most used materials, as far as passive isolation concerns, are metal alloys in different forms of springs, rubber, cork and felt, BERANEK. The obvious disadvantage lies on the lack of adaptation of the passive system to variations on the system excitation.

An active controller shows a proposed adaptation to the systems input conditions, making use of actuators applying a specified force or effort to the system, in order to minimize the vibration amplitudes. The absolute value and the sign (direction) ODF the force exerted are adjusted by a control law, considering the excitation to which the system is subjected.

The hybrid control is an association of the active and passive control schemes acting in parallel. Retaining the capability of adaptation, as in the active case, the parallel use of passive elements reduces the amount of energy of force necessary to control the system. This imposes less stringent restrictions on the actuators and turn the sytem more stable than the active control alone.

2. DYNAMIC VIBRATION ABSORBER: DEFINITION AND PRINCIPAL CHARACTERISTICS

A Dynamic Vibration Absorber, DVA, is a mechanical mechanism which allows the attenuation of vibration levels of a structure through the combination of stiffness, inertia and damping elements. The attenuation effect is a direct consequence of the relation between the system excitation frequency and the natural frequency of the absorber itself.

The simplest model of a DVA, (Fig. 1), consists of a mass-spring-system with only one degree of freedom and lumped parameters. To this system damping may be also added. This absorber, when coupled to a target system which shall have its vibration levels controlled, promotes the reduction of the vibration levels in a more or less restricted frequency range, being quite sensitive to small modifications on the excitation, and thus demanding a quite accurate design of the absorber itself. The efficiency is also related to the coupling position on the target system.

The addition of damping to the mass-spring-system reduces its sensitivity to variations on the forcing frequency, resulting in a broader frequency range where attenuation takes place, but at the same time reducing the attenuation efficiency. For applications where the excitation frequency shall experience considerable fluctuations due to the operational conditions of the system the inclusion of an adaptive scheme to the input conditions becomes necessary.

The simplest DVA operates in a passive way since its main parameters, inertia and stiffness are time invariant. For a semi-active controller an actuator is added to the structure of the DVA (Fig.1b), modifying these parameters according to a control algorithm for different excitations.

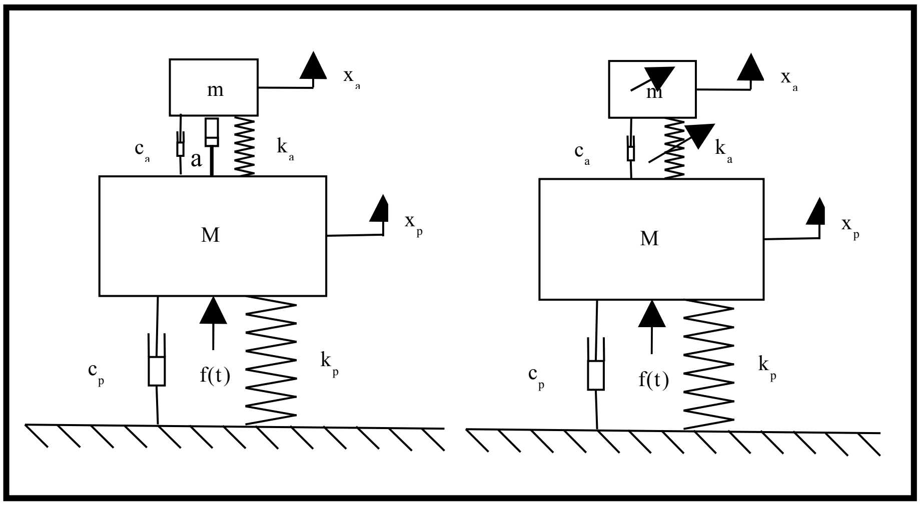


Figure 1. Vibration absorbers. A) Active Vibration Absorber, *a* represents the actuator; B) adaptive passive DVA

2.1. Equation of motion and Frequency Response

In order to analyze the frequency response of the DVA and the coupled target system one may start considering a system with two degrees of freedom with lumped parameters where M, k_p, c_p e m, k_a, c_a, represents the values of mass, stiffness and damping coefficient of the principal system and of the DVA, respectively; x_p e x_a are also the respective displacements. The motion equations of the system are:

$$\begin{bmatrix} M & 0 \\ 0 & m \end{bmatrix}\begin{bmatrix} \ddot{x}_p(t) \\ \ddot{x}_a(t) \end{bmatrix} + \begin{bmatrix} c_a + c_p & -c_a \\ -c_a & c_a \end{bmatrix}\begin{bmatrix} \dot{x}_p(t) \\ \dot{x}_a(t) \end{bmatrix} + \begin{bmatrix} k_p + k_a & -k_a \\ -k_a & k_a \end{bmatrix}\begin{bmatrix} x_p(t) \\ x_a(t) \end{bmatrix} = \begin{bmatrix} f(t) \\ 0 \end{bmatrix} \tag{1}$$

where the system is excited by an harmonic forcing mechanism with frequency ω_{exc} and constant amplitude F_0:

$$f(t) = F_0 e^{i\omega_{exc} t} \tag{2}$$

The behavior of the system can be represented and investigated, for instance, through Transfer Functions which describe how a specified input to the system is dynamically “transferred' to its output, AGUIRRE.

The transfer function can be defined as the Laplace transform of the impulse response, but may also be obtained from the ratio between the Laplace transform Y(s) of the output x(t) of the system (displacement) and the Laplace transform X(s) of the input y(t) (forcing) function as given by:

$$H(s) = \frac{\hat{Y}(s)}{\hat{X}(s)} = \frac{N(s)}{D(s)} \tag{3}$$

To obtain the Frequency Response Function of the system one must only make $s = j\omega$ on equation Eq. 3 thus:

$$\frac{\hat{X}_p(j\omega_{exc})}{\hat{F}(j\omega_{exc})} = \frac{(-m\omega_{exc}^2 + j\omega_{exc} c_a + k_a)}{D(j\omega_{exc})} \tag{4}$$

$$\frac{\hat{X}_a(j\omega_{exc})}{\hat{F}(j\omega_{exc})} = \frac{(j\omega_{exc} c_a + k_a)}{D(j\omega_{exc})} \tag{5}$$

where,

$$D(j\omega_{exc}) = Mm\omega_{exc}^4 - j[(M+m)c_a + mc_p]\omega_{exc}^3 - [(M+m)k_a + mk_p + c_a c_p]\omega_{exc}^2 + (k_p c_a + k_a c_p)j\omega_{exc} + k_a k_p \tag{6}$$

From equation Eq.9, and considering an undamped absorber ($c_a=0$), one can find the maximal attenuation of the vibration of the principal system, i.e., $\left|\hat{X}_p(j\omega_{exc})\right| = 0$, and the following condition must be verified:

$$k_a = m\omega_{exc}^2 \quad / c_a = 0 \tag{7}$$

From equation Eq. 7 it can be concluded that:

$$\omega_a = \omega_{exc} \tag{8}$$

where, $\omega_a = \sqrt{\frac{k_a}{m}}$ represents the natural frequency of the absorber and ω_{exc} is the excitation frequency.

3. DESIGN AND CONSTRUCTION OF A DVA

In the present work a dynamic absorber for the passive-adaptive vibration control is designed, built and tested. Its tuning principle is based on the variation of the stiffness of the system through geometrical variations of the DVA. The prototype is basically a clamped beam on one end and a concentrated mass, able to slide along the beam. The stiffness of the absorber consists in the stiffness of the beam assembly.

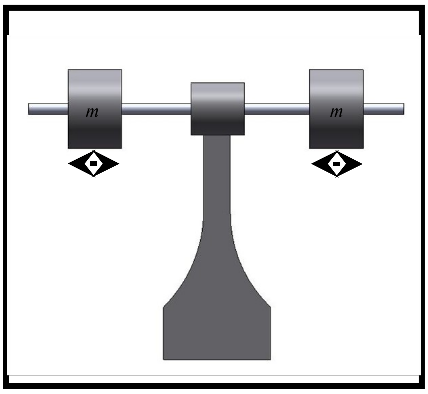


Figure 9. Schematic representation of the DVA

If the beam is likely to experience only small deflections, during the operation of the absorber, it can be assumed as a linear spring. Thus the calculation of the natural (circular) frequency of vibration leads to:

$$\omega_0 = \sqrt{\frac{k}{M}} = \sqrt{\frac{3EI}{a^3 M}} \tag{10}$$

where, $M = m + 0.23 m_{viga}$ $\quad I = \frac{\pi d^4}{64}$ and, a is the position of the mass, relative to the clamped end of the beam made from material with Young´s Modulus E, circular section with diameter d and inertia moment I.

The beam characteristics and the mass, concentrated in the two sliding blocks are constant. Their position relative to the main structure of the DVA can be varied through their sliding along the beams.

The design of the DVA can be split in three parts: the tunable DVA, the actuation mechanism and the electronic controller. The design of the prototype is shown in Figure 10. The DVA structure is made of two cylindrical blocks, acting as cursors, four beams, acting as longitudinal guides, and a base, used for the fixation to the target system. The actuator consists of a simple lever mechanism driven by a step-motor through a belt and pulley system.

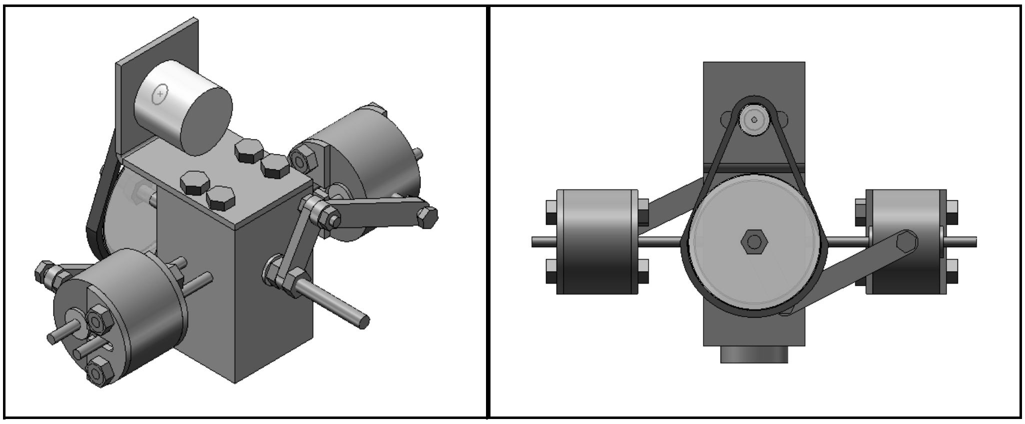


Figure 10. Prototype

The main parameters of the prototype are presented in Table 1.

Table 1. DVA parameters

Parâmetro	Valor
f_a	150 – 250 Hz
m_{viga}	0,0038 kg
m_{cursor}	0,09345 kg
k_a	1,333 x 10^4 – 3,704 x 10^4 N/m

From the relation between the motion of the lever and the corresponding angular rotation of the driving motor one can obtain the control parameter for the automatic tunning of the absorber. The actual implementation of the control algorithm involves the communication between controller and absorber, i.e. the commands input to the step-motor and the measurement of the target system in terms of its vibration. The controller used is a PIC micro-controller with serial data transfer, A/D-sampling and 12V power supply.,

5. SIMULATION AND EXPERIMENTAL RESULTS

In order to evaluate the performance of the prototype in an actual application, it is coupled to a simple experimental structure, where the excitation can be separately controlled, Figure 11. Also simulations with the use of MATLAB© and ABAQUS© software were developed in order to obtain the natural frequencies of this system, its mode shapes and then help chose appropriated coupling positions for the DVA. The Figure 13 to 15 shows some of the results obtained from the FEM simulations, related to bending vibrations of the beam in a transversal direction.

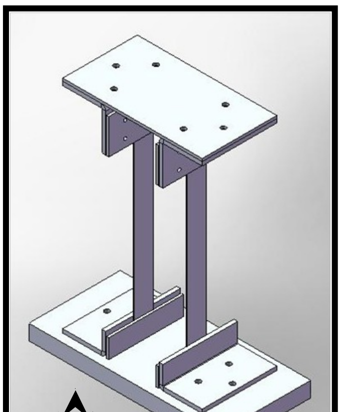


Figure 11. System subjected to vibrations

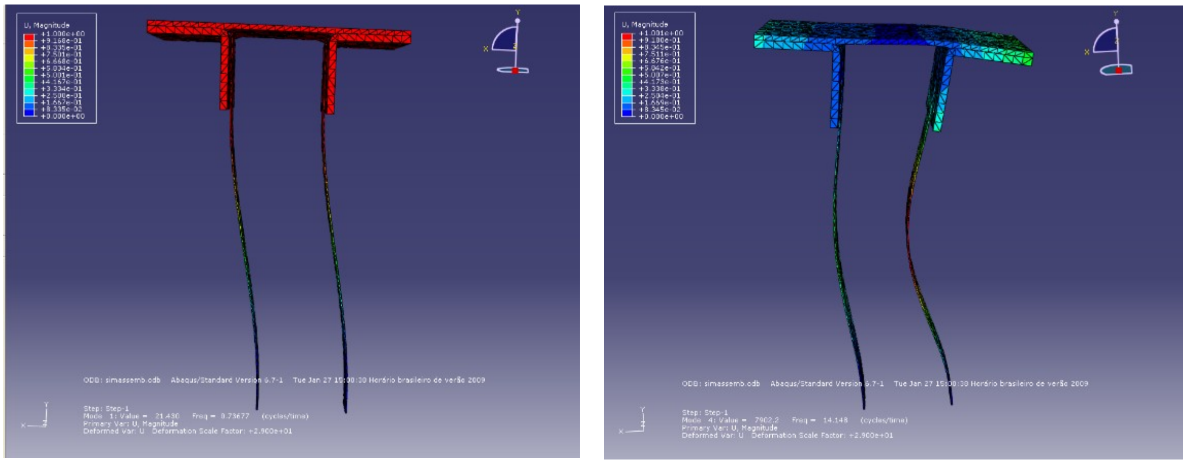


Figure 12. First and fourth modes, 0.74 Hz and 14 Hz, respectively

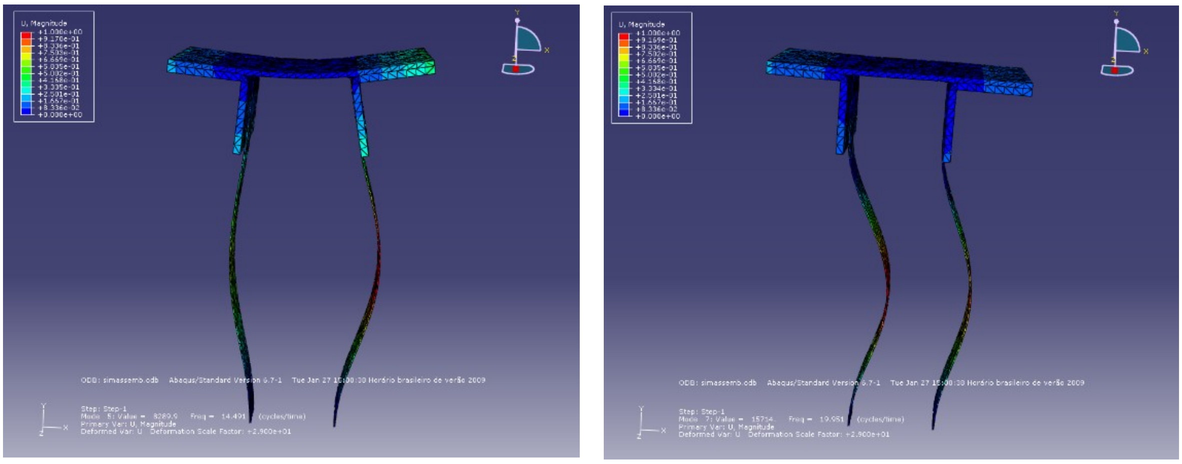


Figure 13. Modes 5 and 7, 14.5 Hz and 19.95 Hz, respectively

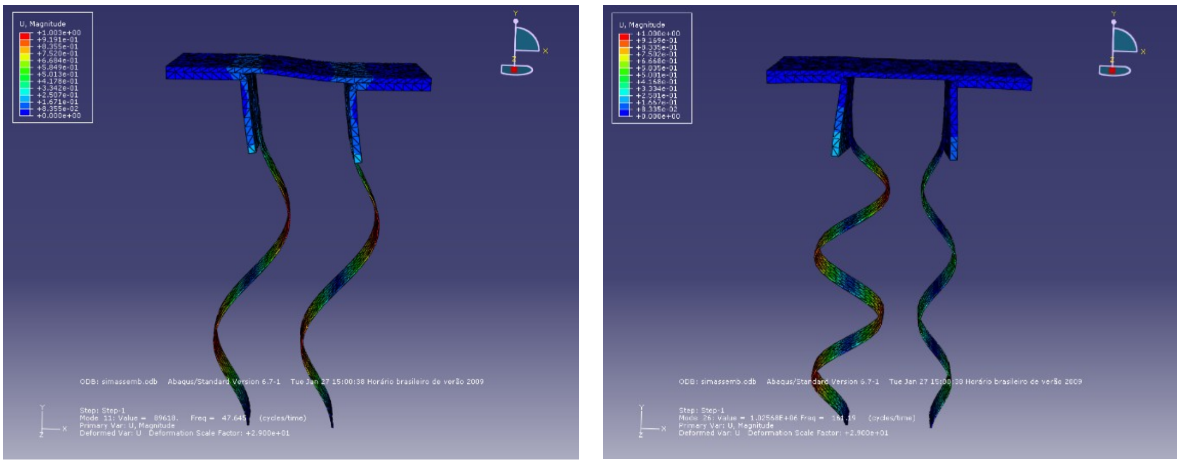


Figure 14. Modes 11 and 26, 47.65 Hz and 161.2 Hz, respectively

To validate the results obtained from the simulations a experimental modal analysis of the system was performed, in the frequency range of interest. As excitation a shaker fixed to the top of the structure was driven by a white noise signal, band-pass filtered between 30Hz and 600Hz. The data acquisition and processing was done though the spectral analyzer NetdB with the respective software dBFaSuite.

The most interesting frequencies were of 65, 175, 197 and 202Hz, respectively. The corresponding mode shapes can be found through the interpretation of the complex values of the experimental frequency response function. The obtained results are shown in figures 15 to 18.

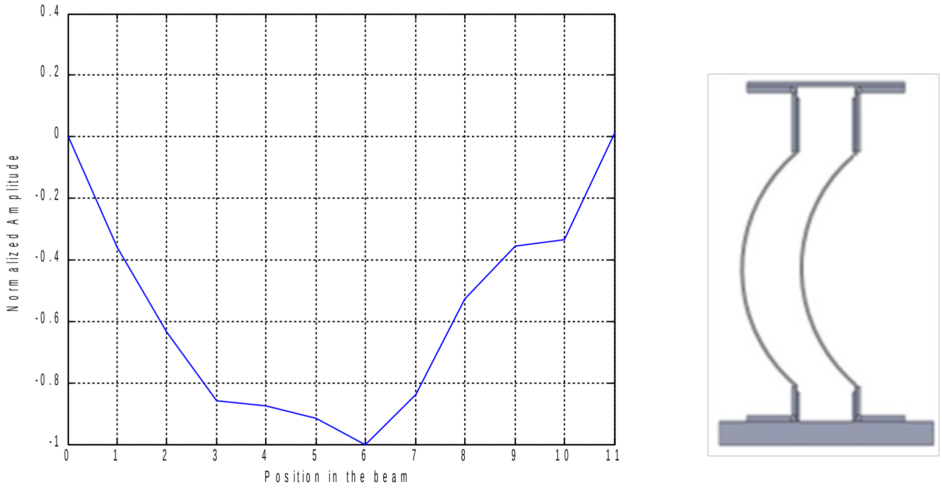


Figure 15. Mode related to 65 Hz

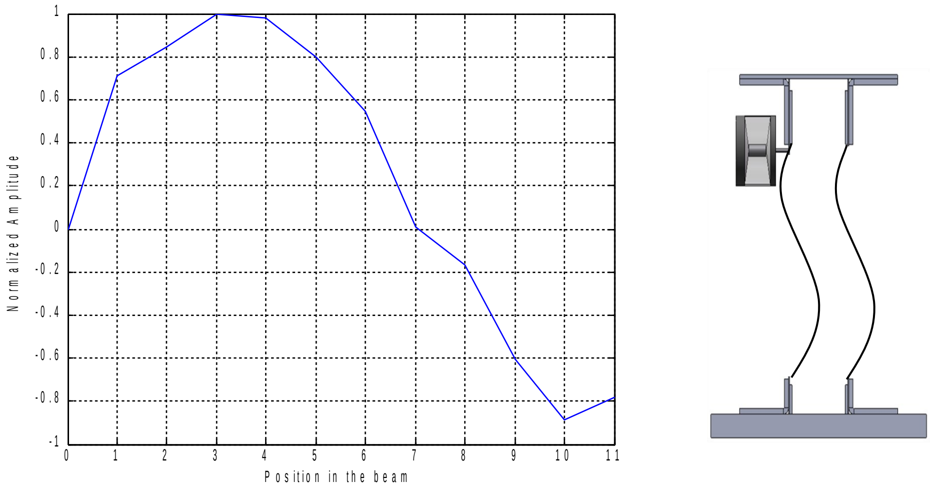


Figure 16. Mode related to 112 Hz

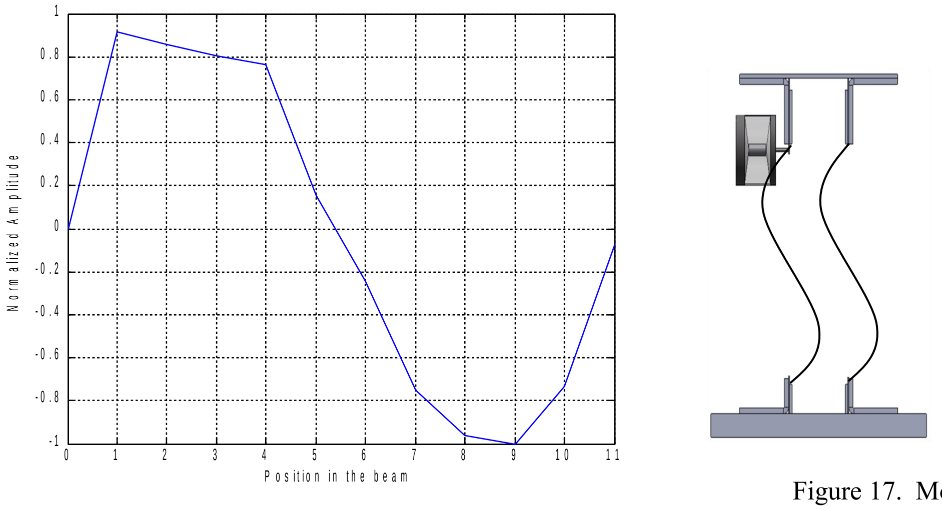
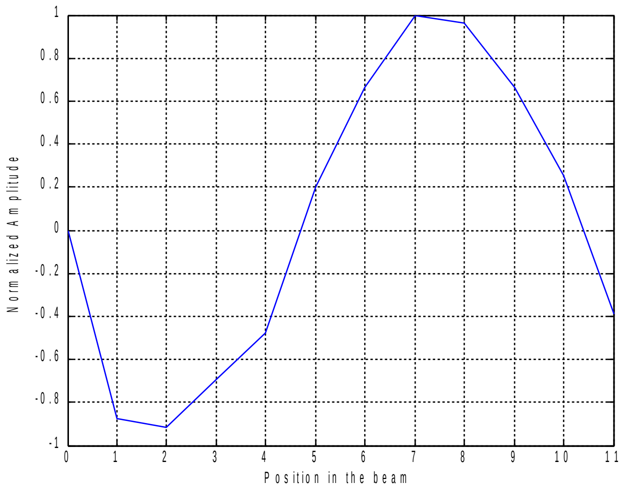


Figure 17. Mode related to 175 Hz



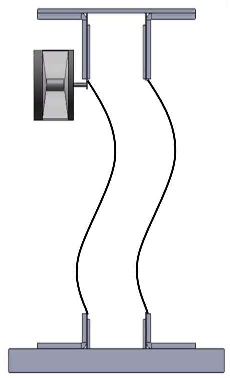


Figure 18. Mode related to 202 Hz

Based on the identified mode shapes the position for the fixture of the DAV on the structure can be chosen for the tests of the DAV itself.

A first measurement was done without the DAV to obtain the behavior of the structure itself. After that, with the DAV attached to the structure, four measurements were done for different tunning characteristics of the DAV, including the lowest and highest frequencies possible along with two intermediary values.

The input signal, fed to the shaker, was again white noise filtered for the same range of 30 to 600Hz. The results are shown in Figure 19.

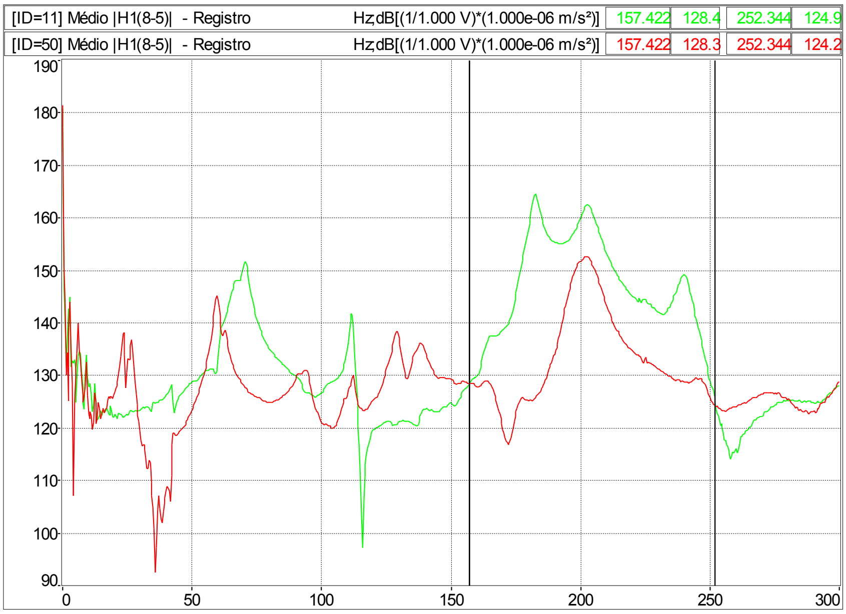


Figure 19. Experimentally obtained FRFs (RED – without DAV / GREEN DAV tuned at 190Hz)

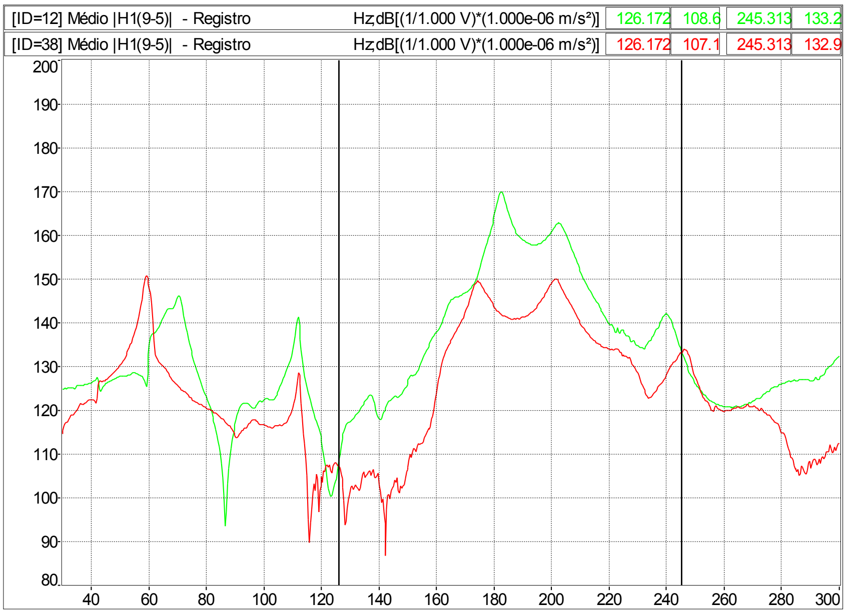


Figure 20. Experimentally obtained FRFs (RED – without DAV / GREEN DAV tuned at 175Hz)

For the DAV tuned at 190Hz the most significant attenuation ranges from 157 to 250Hz, approximately. For a tunning at 175Hz the frequency range goes from 126 to 246Hz. It can be seen that the effedt of the absorber is extended in the vicinity of the chosen frequency due to the intrinsic damping in the system and due to a relative large mass ratio between system and absorber.

In a second test set accelerometers were fixed to both beams of the principal system with the DAV fixed to one of them, Figure 22. The graph shows the influence of the DAV in the corresponding beam, while the amplitudes of the other one were not strongly altered. This shows the effectiveness of the attenuation related to the position of the DAV, which is the point where the energy is drawn from the principal system.

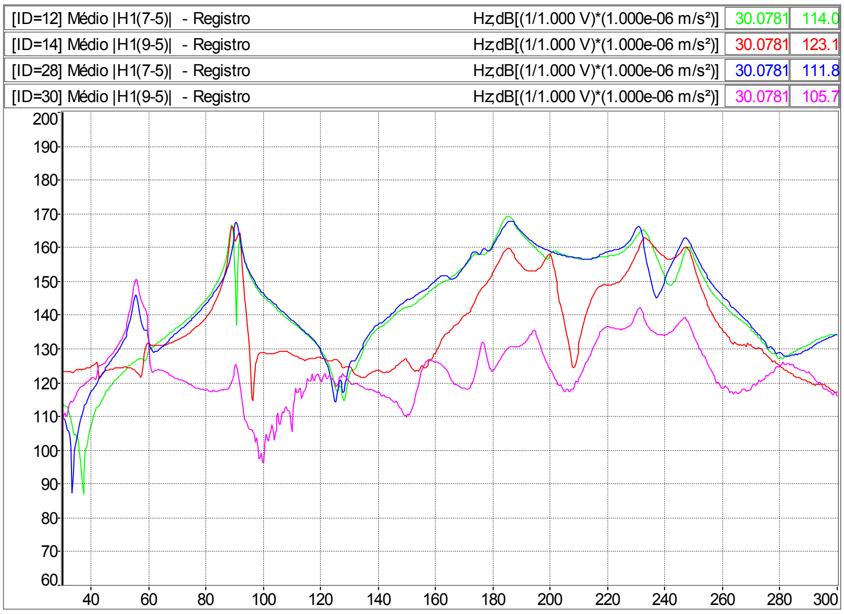


Figure 21. Experimentally obtained FRF-amplitudes.
(RED first beam without DAV / PINK first beam with DAV / GREEN second beam without DAV)
(BLUE second beam with DAV on the first beam)

5. CONCLUSIONS

This work shows the design and construction of a tunable Dynamic Absorber. The model consists of clamped beams with intermediary sliding masses. The modification of the stiffness of the system, i.e. the position of the masses, allows the adaptation of the operational frequency range of the absorber to the excitation.

The dimensions chosen for the prototype were based on vibration measurements, and corresponding frequency range, on an actual system in order to deal with a practical engineering case.

Tests were made for the evaluation of the performance of the system. To this purpose a simple structure and a controlled excitation with a shaker were used.

The input signal for the tests made were a band pass filtered white noise, allowing the study of the influence of the absorber on a broad frequency range, including but going beyond the operational range of the absorber. The results of the tests shows the expected attenuation of the system vibrations due to the absorber, in the vicinity of the tunning frequency. The small deviations from theoretical and experimental values are mainly due to the estimated and calculated values of the system parameters such as mass and stiffness, the linear approximation and due to a large mass ratio between system and absorber. It can be also seen in the results that the attenuation effect extends to neighbour frequencies due to the inherent damping of the system.

The implementation of a control algorithm is left as future work going beyond the scope of the present work. The tests were done with the absorber manually tunned.

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