Control of vibrations in the frame structure using dynamic vibration absorber

Fabiana da Rosa Sufiatti[1], Paulo Rogério Novak[1], Giovanni Bratti[2], Francisco Augusto Aparecido Gomes[1]

[1]*Graduate Program in Civil Engineering, Federal Technological University of Paraná*
Via do Conhecimento, 85503-390, Pato Branco - PR, Brasil
fabianasufiatti@alunos.utfpr.edu.br, novak@utfpr.edu.br, franciscogomes@utfpr.edu.br
[2]*Department of Mechanical Engineering, Federal Technological University of Paraná*
Via do Conhecimento, 85503-390, Pato Branco - PR, Brasil
giovannibratti@utfpr.edu.br

Abstract. Structures are physical systems subjected to external actions that cause transmission of efforts. These actions can be classified as dynamic when they vary with time, in magnitude, direction, position, and/or direction and generate relevant loads. They are responsible for causing vibrations in the structure that can damage it, cause fatigue of the materials, and affect the users' comfort. The Dynamic Vibration Absorber (DVA) device aims to control these unwanted vibrations, increasing the life of the structure and providing comfort to users. This study aims to develop a dynamic analysis of a frame structure and design an DVA-type device to control the vibration of the structure. Initially, a numerical dynamic analysis of the primary structure was performed to identify the frequency response function and the first modes of vibration, using the Finite Element Method (FEM). From this analysis, the ideal parameters of the device were determined: stiffness, the mass ratio between the mass of the DVA and the primary structure, and the damping ratio to control the vibrations in the first natural frequency of the frame. The DVA was modeled as a cantilever beam with concentrated mass at the tip, using the Euler-Bernoulli equation. This DVA was coupled to the structure and the same previous numerical analysis were performed, where a reduction in displacement amplitude was observed. This result demonstrates that the device is efficient to control of vibration in the first mode of vibration of the frame under study.

Keywords: Dynamic analysis, dynamic vibration absorber, passive vibration control.

1 Introduction

With the evolution of civil construction provided by the increase in the resistance of materials and the control of executive methods, structures have been designed and built that are increasingly slender and covering greater spans. However, these structures are more subject to vibrations and have a lower capacity to dissipate energy. Therefore, studies are needed to mathematically characterize the dynamic external actions, analyzing the consequences of oscillations, for an assertive design of the components and connections of the structure [1]. Slender structures are more flexible and this flexibility leads to increased vibrations, which cause discomfort and can affect your safety when dynamic loads are generated by the action of earthquakes, winds, heavy traffic, machinery, among others, which makes it necessary to apply techniques for reducing vibrations [2].

According to Rossato et al. [3], vibration control is a technique widely used that aims to reduce the vibration amplitude of the installation of devices to the structure. This control can be passive, active, or hybrid and is done by inserting devices into the structure. Passive control is widely used, due to its low installation cost and the fact that it does not need external energy sources, it becomes reliable in the event of a power outage. Dynamic Vibration Absorbers (DVA's) are passive type vibration control devices, they consist of a secondary system added to a primary structural system, which can act at any frequency, whose vibration or radiated noise is desired to control (HOLANDA, 2018).

This paper aims to study the passives devices analyzing their influence and effectiveness in controlling the vibrations of a frame structure. For this, a numerical analysis was carried out using the Finite Element Method (FEM) with ANSYS® software, where the one-floor frame was analyzed, with the coupling of DVA to control the vibrations of the primary structure's first vibration mode. The DVA was modeled using the Euler-Bernoulli theory, as clamped at one end and with a concentrated mass at the other end, as the rush mass could not be negligible when compared to the point mass value. The DVA parameters such as stiffness, mass ratio, damping, rush length and concentrated mass at the tip were determined so that they meet the proposed equation for an optimally tuned damped DVA, according to Hartog [4].

2 Methodology

The object of study of this paper was a frame structure of a pavement numerically modeled using the Finite Element Method (FEM), using ANSYS®. The frame was composed of 4 (four) columns and 2 (two) plates, as shown in Fig. 1 and its properties are shown in Tab. 1.

Table 1: Characteristics of the numerical model [5]

Column material properties	
Modulus of elasticity – E (GPa)	215.70
Density - ρ (kg/m³)	7117.8
Poisson's Coefficient	0.30
Section H x B (mm)	1.60 x 12.70
Free span – L (mm)	267.00
Damping factor (ζ)	0.01
Floor material properties – Wood	
Modulus of elasticity – E (GPa)	3.776
Density - ρ (kg/m³)	682.02
Poisson's Coefficient	0.30
Mass - m0 (g)	400
Plate thickness 0 (mm)	19.55
Mass - m1 (g)	386
Plate thickness 1 (mm)	18.87
Dimensions (mm)	150 x 200

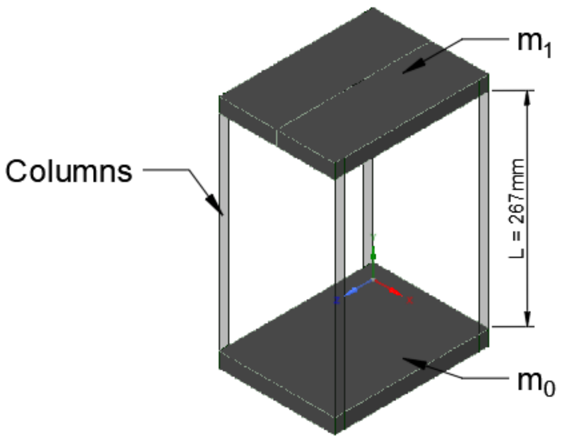

Figure 1: Frame structure

Floors were modeled as solid elements and columns were modeled as plate elements. The connection between floors and pillars is made through ANSYS® Node Merge and Contact Bonded features. For the use of the Node Merge feature, it is necessary that the nodes of the specific elements are indicated or determined at a distance smaller than the defined limit. Seams between columns and floors have been amended, with the objective of controlling how to obtain and be able to guarantee the alignment between nodes in the connection. The lower floor, used as the base of the structure, had its lower face fixed. The upper floor, where the DVA was attached, was divided into two equal ones. This consideration was carried out to ensure the alignment of the nodes between the pavement and DVA in the coupling, which was positioned in the central part of the pavement.

The mesh used was the regular hexahedral type, which was chosen due to the geometry of the structure. To obtain more accurate results and ensure that the applied mesh is reliable, the mesh convergence analysis was performed. For this, the sizes of the elements were changed and the values of the natural frequencies were simulated, until the values of these frequencies did not suffer any more variations, in order to obtain the best response with a good mesh resolution. The size adopted for the floor elements was equal to the width of the columns, to ensure the alignment of the nodes and to use the Node Merge feature. The size of the pillar elements was changed until reaching the convergence of the first six natural frequencies of the structure.

Harmonic analysis was performed in ANSYS® in order to identify a structure response in the frequency domain and determine the FRF (Frequency Response Function) of the system. Through this analysis, it was identified at which frequency the greatest displacement occurs, which is the first natural frequency of the system. For this, an impulsive force of 1N was applied in the center of the upper floor. The results were measured in the range of 5 Hz to 15 Hz and the number of points in the frequency was 300. Transient analysis was performed in order to identify the response of the structure in the time domain. The excitation force was applied in the center of the upper floor, it was variable and is given by:

$$F(t) = sin(\omega t) \tag{1}$$

where ω is the excitation frequency in rad/s, in this case, the first natural frequency of the frame was used; and t is the time, which ranged from 0 to 5 seconds.

The material used for modeling the DVA was structural steel with its properties in Tab. 2. The thickness and width of the DVA were kept constant for all analyses, as the objective was to analyze the behavior of the parameters, changing only the length of the DVA and the mass at its tip.

Table 2: DVA material data [6]

Módulo de Elasticidade – E (GPa)	200
Densidade – ρ (kg/m^3)	7850
Espessura – e (mm)	1.25
Largura – b (mm)	25

Undamped DVA's passives are efficient when the excitation frequency becomes constant, a situation rarely verified in practice [7]. When inserting an undamped DVA into the frame, it will eliminate the original resonance peak in the response curve but will introduce two new peaks. So when the excitation approaches these values it will still experience large amplitudes of vibration. For these lower amplitudes, a damped DVA can be inserted. To study the impact of DVA damping on the vibration amplitude of the structure, the optimal damping for the DVA was created through [4]:

$$\zeta^2_{ótimo} = \frac{3\mu}{8(1+\mu)^3} \tag{2}$$

where ζ is the damping factor of the DVA and μ is the ratio between the mass of the DVA and the mass of the primary structure.

The mass ratio was calculated using:

$$\mu = \frac{M_{eq,adv}}{M_{eq,pórtico}} \tag{3}$$

where M_{eq} is the equivalent masses, which were calculated through:

$$M_{eq,adv} = \frac{K_{adv}}{\omega_{n,adv}^2} \tag{4}$$

$$M_{eq,pórtico} = \frac{4 \times K_{eq,pilar}}{\omega_{n,pórtico}^2} \tag{5}$$

for the DVA and for the frame, respectively. Where an equivalent stiffness for a column is given by [8]:

$$K_{eq,pilar} = \frac{12 \times E_{pórtico} \times I_{pórtico}}{L^3} \tag{6}$$

considering the column as a fixed-fixed beam with end displacement.

Assuming a rod of uniform cross section and a point mass, the DVA was modeled as a cantilever beam and the stiffness of the DVA, K_{adv}, was calculated through [8]:

$$K_{adv} = \frac{3EI}{L_{adv}^3} \tag{7}$$

where E is the modulus of elasticity of the material of the rod; I is the moment of inertia of the rod in relation to its center of gravity; and L_{adv} is the length of the rod.

The relationship between natural frequencies, f, is given by:

$$f = \frac{\omega_{n,adv}}{\omega_{n,pórtico}}. \tag{8}$$

For the sizing of an DVA without damping the value of f is equal to 1 [8]. However, to achieve an optimal design of a damped DVA, Rao [8] DVA that the relationship between frequencies be given by:

$$f = \frac{1}{1 + \mu} \tag{9}$$

Once the value of the mass ratio is defined based on the value of the natural frequency of the primary structure in which the DVA will act, the values for the length of the rod and the concentrated mass at the tip are calculated.

In order to achieve greater precision in the design of the DVA, the mass of the rod (beam) of the DVA was also taken into account. Therefore, the DVA was modeled using the Euler-Bernoulli theory as a two-dimensional beam, with parameters of mass and stiffness, and the first natural frequency of the free beam with mass concentrated at the end is given by [9]

$$\omega_{1,adv} = \sqrt{\frac{3EI}{L_v^3\left(M + \frac{33}{140} M_v\right)}} \tag{10}$$

where: $\omega_{1,DVA}$ is the first natural frequency of the free-fixed beam; M_v is the mass of the beam; M is the mass concentrated at the free end of the beam; EI is the bending stiffness modulus of the beam; and L_v the total length of the beam.

As shown in Eq. (10), the equivalent mass of the DVA can be calculated through:

$$M_{eq,adv} = M + \frac{33}{140} M_v \tag{11}$$

and isolating the length L_v in Eq. (10) one arrives at:

$$L_v = \sqrt[3]{\frac{3EI}{\omega_n^2 M_{eq,adv}}}. \tag{12}$$

The equivalent mass of the DVA was calculated by multiplying the mass ratio by the equivalent mass of the frame. With this value, it was possible to define the length of the rod and calculate the mass of the beam through:

$$M_v = L_v \times b \times e \times \rho_{aço} \tag{13}$$

and the concentrated mass at the tip was then defined by:

$$M = M_{eq,adv} - \frac{33}{140} M_v \, . \tag{14}$$

Thickness and width were determined in such a way that the DVA could be modeled as a plate element. After modeling, their natural frequencies were calculated. The mesh convergence analysis was performed by varying the number of elements until the values of the natural frequencies converged to a constant value. Modal analysis, harmonic analysis, and transient analysis of the structure were performed again after coupling the DVA. To analyze the influence of the DVA parameters on the dynamic behavior of the structure, seven devices were dimensioned and inserted with different mass ratios, damping, rod length, and concentrated mass at the tip.

3 Results and discussions

Table 3 shows the values of the mass ratio (μ), damping factor (ζ), mass concentrated at the tip (M) and stem length (L) determined for the DVA's.

Table 3: Dimensioned DVA's parameters

Parameter	**DVA 1**	**DVA 2**	**DVA 3**	**DVA 4**	**DVA 5**	**DVA 6**	**DVA 7**
μ (%)	5.00	8.69	10.86	16.07	25.00	50.00	10.00
ζ	0.127	0.159	0.173	0.196	0.219	0.236	-
M (g)	7.39	27.39	38.54	64.58	108.18	228.25	34.98
L (mm)	283.68*	241.45	227.13	205.51	186.35	167.02	217.95

* The analyzes did not take into account the effect of gravity, so when the length of the rod resulted in a value greater than the free span between floors, the DVA was coupled inverted, externally to the upper floor.

Table 4 presents the natural frequencies of the DVA's for the first vibrating mode, calculated analytically for each corresponding mass ratio and the frequency of the DVA simulated in ANSYS®, in the free-fixed condition. The difference between these frequencies, for each DVA, is also presented in this table and it can be seen that the greater the mass ratio, the greater the error.

Table 4: DVA's natural frequency

	μ (%)	f	**f_n (Hz) Calculated**	**f_n (Hz) ANSYS®**	$\lvert f_{n,calculated} - f_{n,ansys} \rvert$	**Error (%)**
DVA 1	5.00	0.952	10.670	10.662	0.008	0.075
DVA 2	8.69	0.920	10.308	10.358	0.05	0.485
DVA 3	10.86	0.902	10.106	10.164	0.058	0.574
DVA 4	16.07	0.862	9.653	9.718	0.065	0.673
DVA 5	25.00	0.800	8.963	9.030	0.067	0.748
DVA 6	50.00	0.667	7.469	7.531	0.062	0.830
DVA 7	10.00	1.00	11.204	11.270	0.066	0.589

Harmonic analysis was performed in the range of 5 Hz to 15 Hz. The maximum displacement amplitude without DVA was 22.65 mm and occurred at the frequency of 11.20 Hz the first natural frequency of the structure. It is noticed that after inserting the DVA's, the peak of the first natural frequency was divided into two, one with a lower value and the other with a higher value, showing that the DVA worked to change the value of the first frequency and insert a new peak in the FRF of the structure, as can be seen in Fig. 2.

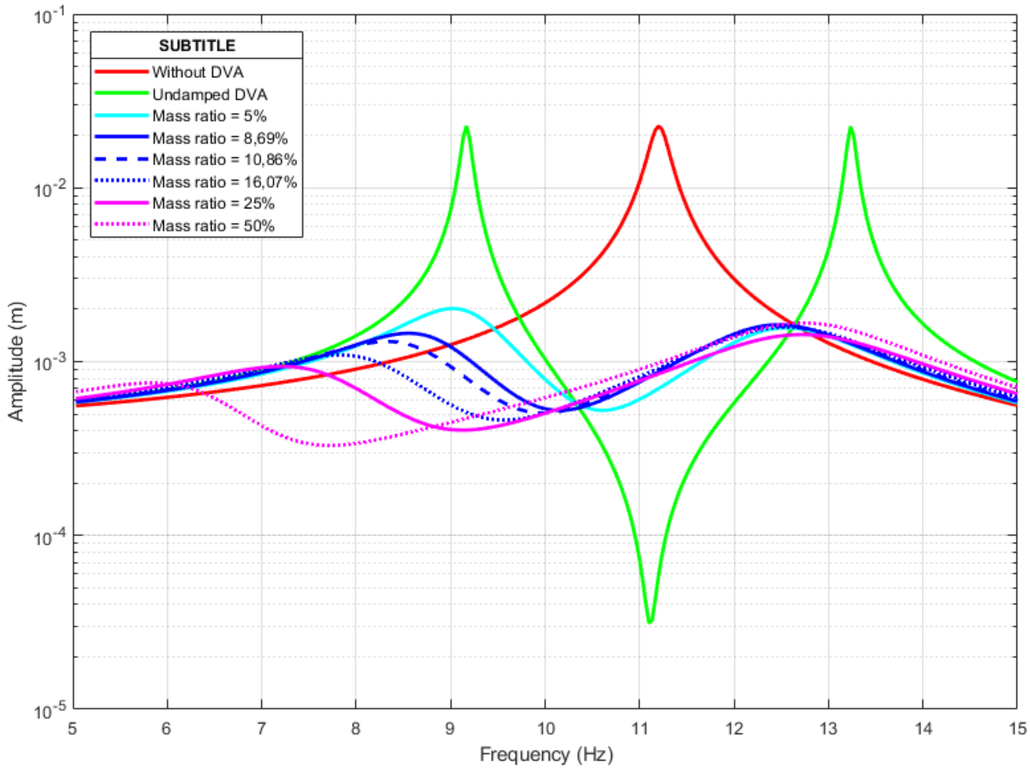

Figure 2: FRF with DVA

Table 5 shows the values of the maximum displacement amplitude with the DVA's. The amplitude reduction was above 90% showing that the damped device is able to significantly reduce the vibration of the structure. In Fig. 2, it is possible to observe that the curve of the structure of DVA 2, with μ=8.69%, presented the two peaks with similar amplitudes. For mass ratios greater than 8.69%, the amplitude of the first peak decreased with increasing mass ratio, and the amplitude of the second peak remained practically unchanged. Analyzing the error values presented in Tab. 4, it can be seen that the greater the associated error, the greater the difference in amplitude between the peaks of the response curve in the FRF. The error also increases with the increasing mass ratio. The higher the mass ratio, the lower the calculated natural frequency for the DVA will be and the difference between the DVA frequency and the first frame frequency will also increase. Note that the error between the calculated and the modeled interferes more in the result the higher the mass ratio and consequently the optimal damping.

Table 5: Maximum displacement amplitude Case 1 with DVA

	Without DVA	**DVA 7 undamped**	**DVA 1**	**DVA 2**	**DVA 3**	**DVA 4**	**DVA 5**	**DVA 6**
μ (%)	-	10.00	5.00	8.69	10.86	16.07	25.00	50.00
Maximum displacement (mm)	22.65	22.69	2.02	1.63	1.60	1.58	1.43	1.67
Frequency (Hz)	11.20	9.17	9.03	12.47	12.50	12.57	12.70	12.73
Amplitude variation (%)	-	+0.20	-91.1	-92.8	-92.9	-93.0	-93.7	-92.6

Transient analysis was performed in the interval of 0 to 5 seconds, with an excitation frequency equal to the first natural frequency of the structure, to evaluate a resonance situation. The result is shown in Fig. 3 and the displacement amplitude increases with time when the DVA is not coupled. Transient analysis was performed for cases in which DVA 5 and DVA 7 were coupled and the results are shown in Fig. 3. It can be seen that DVA 5 presents a better result compared to DVA 7 in controlling vibrations of the structure.

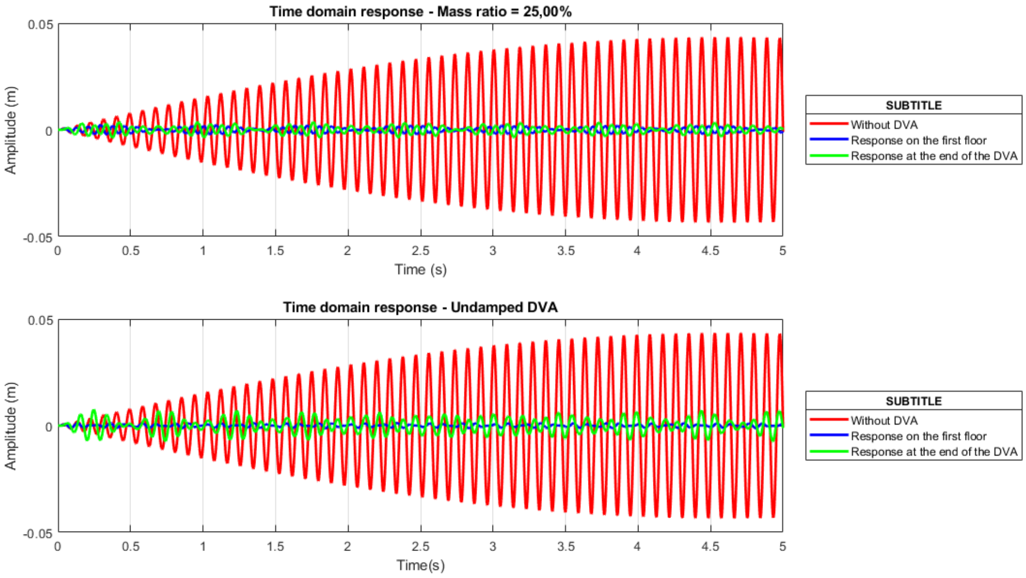

Figure 3: Time domain response

Undamped DVA significantly reduces vibrations when the excitation frequency is equal to the natural frequency of the vibrate mode for which it is sized. However, when analyzing Fig. 2, it can be seen that it has high displacement amplitudes in the first and second natural frequency, showing that the undamped DVA has a limitation in its operation, working in a limited bandwidth.

4 Conclusions

In the present study, the dynamic response of a frame structure of a floor was analyzed before and after coupling a Dynamic Vibration Absorber (DVA) device, with the objective of evaluating the ability of the DVA to control the vibrations of the floor. structure. The DVA's dimensioned and coupled to the structure were able to control the vibrations and reduce the displacement amplitudes of the first vibration mode. The best result found was for μ=25.00% with damping, this DVA reduced by 93.7% the displacement amplitude values in the harmonic analysis. In the transient analysis, the reduction was 94.7% with the excitation frequency equal to the first natural frequency of the structure without DVA. The objective was to dimension an optimally tuned damped DVA, but the natural frequency value of the DVA modeled through the MEF diverged from the calculated value and the FRF curve did not resemble the expected result. Through the results analyzed in this work, a high index of effectiveness of the device inserted in the control of vibrations of the studied structure was observed.

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