Optimization of the constructive parameters of a tuned mass damper for the control of a cantilever beam

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Abstract. Beams are structural elements widely used in engineering. During their useful life, they are subject to different forms of static and dynamic loading. When dynamic actions are applied, they can assume a state of vibration, causing large amplitudes that can compromise the structure from the point of view of environmental comfort and also structural safety. These problems can be solved by inserting into the main structure a device called a tuned mass damper (TMD). From the dynamic characteristics of the main structure, the TMD is scaled and modeled as a mass-spring-damper system. In this context, the objective of this work is to present the effectiveness of the use of TMD for the control of vibrations in a set beam. The TMD used was of the cantilever beam type with a concentrated mass at the tip. This was modeled and fixed at the end of the main structure. Its optimization parameters were scaled from the mass ratio and damping. The simulation responses were obtained using the finite element method (FEM) using the computer program Ansys®. The results were satisfactory, for a mass ratio equal to 0.05 and damping factor of 0.1273.

Keywords: Tuned Mass Damper, Mass Ratio, Damping.

1 Introduction

Beam-type structural elements are commonly used in civil construction, as well as in other areas of engineering. During their useful life, these structures are subject to different forms of loading, both static and dynamic, which, under specific conditions, can assume a state of vibration that can compromise the entire structural system.

When subjected to dynamic actions, structures are likely to present a response to the applied load. This response can present large displacement amplitudes which, in turn, can compromise the physical integrity of the structure, cause discomfort and compromise the safety of the occupants.

Problems related to oscillations and large displacements in structures can be solved by inserting, in the main structure, a passive energy dissipation device called a Tuned Mass Damper (TMD). A secondary device, which can act on any frequency, whose vibration you want to control.

The study related to the use of TMD in beams is wide. An example is the Rio-Niterói Bridge. Due to the strong oscillations caused by the actions of wind and vehicular traffic, it was necessary to implement passive control devices to reduce vibration amplitudes. 32 dampers with 2.2 tons of mass were installed, which together correspond to 1.0% of the modal mass associated with the first flexural oscillation mode of the structure. After the implementation of the TMD system, the efficiency was proven with a reduction of around 75% of the displacement amplitudes [1].

Wu [2] proposed the use of dynamic damper to attenuate beam vibration when subjected to dynamic loads. The damper, coupled to the central part of the beam, was designed considering the mass of the primary structure.

The behavior of the beam was studied according to the dynamic characteristics for the first modal coordinate. The model was used to obtain the optimal values for the stiffness and damping of the system. Holanda et al [3] designed several TMDs to reduce oscillations by simultaneously controlling three natural frequencies of a bi-supported beam. The TMDs were sized so that their frequencies coincided with the beam frequencies. The results were satisfactory and showed a significant reduction in the region of frequencies of interest.

Ari and Faal [4] investigated a problem of optimizing dynamic vibration absorbers to suppress thin plate vibrations over a wide frequency range. TMD stiffness and mass adjustments were made to control vibrations. Yoon, Choi and So [5] proposed a kind of passive dynamic absorber to attenuate the first three modes of vibration. TMD was applied to three different types of beams. The results showed that the absorber is capable of controlling the three frequencies, attenuating two flexion modes and a torsional mode simultaneously.

In this context, this study presents the optimization parameters of TMD for application in beams subject to vibration. From numerical analysis via Finite Element Method (FEM), the answers are determined and point out that for a mass ratio equal to 0.05 the vibration level reduction of approximately 99%.

2 Methodology

In order to control resonance on a cantilever beam, a tuned mass damper (TMD) was designed for a mass ratio of 0.05. From the dynamic characteristics of the main structure, the TMD was modeled and coupled to the structure.

The primary structure has no damping. For TMD, damping was calculated from the mass ratio of the TMD point mass with the primary structure mass. The system responses were obtained by modal, harmonic and transient analyses and compared for the structure with and without TMD.

2.1 Theoretical model of primary structure and TMD

The primary structure, also called base beam, consists of a 1020 steel beam, cantilever at one end and free at the other, Figure 1. The proposed TMD is an aluminum rod, with uniform cross section and a point mass at its end.

The beam has a constant rectangular cross section along its length. For the TMD to be sized to act on the first beam frequency for a mass ratio (μ) equal to 0.05, it is essential to know this frequency. According to Euller-Bernoulli's theory, the first natural frequency of a free set beam is given by [6]:

$$f_{beam} = \frac{2{,}04}{2\pi}\sqrt{\frac{3EI}{L^3M_1}} \tag{1}$$

Where:
E: Modulus of Elasticity;
I: inertia;
L: Beam length;
M_1: Beam mass.

The first frequency of TMD can be calculated by equation [7]:

$$f_{TMD} = \frac{1}{2\pi}\sqrt{\frac{3EI}{\left(M_h\frac{33}{140}+M_2\right)L^3}} \tag{2}$$

Where M_h refers to the mass of the rod and M_2 the point mass of the TMD.

2.2 TMD Sizing

To determine the parameters of a TMD it is necessary to know the characteristics of the base beam. From the physical characteristics of the primary structure, the mass of the M_1.

The ratio

$$\mu = \frac{M_2}{M_1} \quad (3)$$

provides the ratio of concentrated mass to the mass of the primary structure.

For a TMD to be optimally tuned the ideal is that the two frequencies of the system have equal or close amplitudes. For this it is necessary to determine the optimal values for the ratio of frequencies (f) and optimal damping factor (ζ) [8] and [9]:

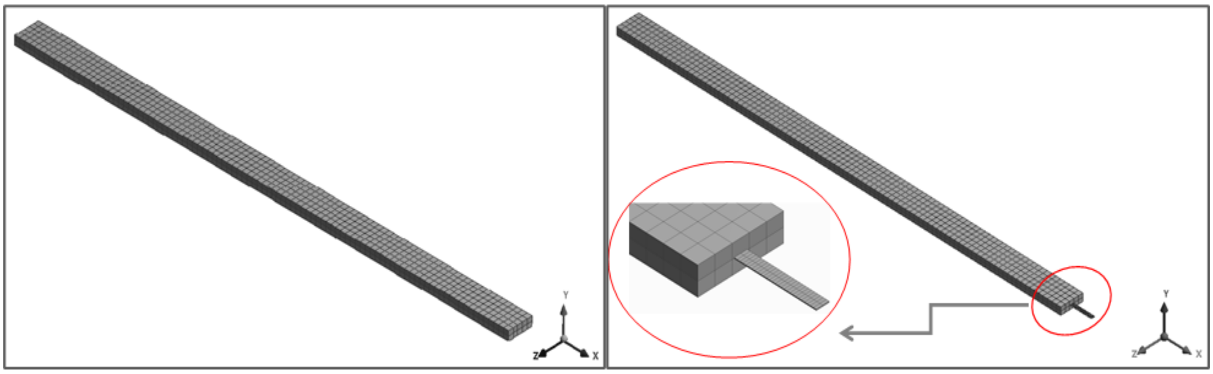
$$f = \frac{1}{1 + \mu} \quad (4)$$

$$\zeta = \sqrt{\frac{3\mu}{8\,(1 + \mu)^3}} \quad (5)$$

2.3 Numerical model of primary structure and TMD

The modeling of the systems was carried out using the finite element method by the computer program Ansys®.

The TMD was inserted at the free end of the primary structure, near the neutral line, through the contact between the face of the free end of the beam and the face of the TMD rod, as shown in Figure 1.



(a) (b)

Figure 1: Numerical model of the primary structure: (a) Primary structure; (b) TMD coupled to the primary structure.

The boundary condition used is the crimping of the surface with the smallest cross-sectional area of the beam and the coupling point of the TMD in the primary structure. The connection was made at the free end of the beam through the contact between the face corresponding to the free end of the beam and the face of the TMD rod.

The physical and geometric properties of the beam structure and the TMD, for f = 0. 952, are presented in Table 1.

Table 1: Primary Structure and Tuned Mass Damper Properties

Properties	**Primary Structure**	**TMD**
Length (m)	2.00	0.084
Width (m)	0.10	0.015
Thickness (m)	0.03	0.0012
Density (Kg/m³)	7850	2710
Mass (Kg)	47.1	0.56750 [(1)] + 0.0086[(2)]
Young's Modulus (GPa)	200	70
Poisson's Coefficient	0.30	0,33
Damping Factor	0	0,1273

[(1)] Point mass inserted into the free end of the TMD; [(2)] Rod mass

To obtain the answers of the system, a convergence analysis was performed in relation to the refinement of the mesh of the primary structure and the TMD. The initial mesh was refined by changing the size of the elements, until reaching a constant value of the results in order to obtain the best response with a good resolution and the shortest computational time. For the beam structure, elements of 20 mm were used and 7.5 mm elements were used for the TMD.

The system responses were obtained through simulations. Modal analysis to determine natural frequencies and vibration modes, harmonic analysis to obtain the frequency response curve at the free end of the beam and TMD, and transient simulation to determine the responses in the time domain.

3 Results

In this section, the results of the dynamic responses of the structure with the TMD inserted in the main structure are presented.

The natural frequency for the first vibration mode of the primary structure was 6.1315 Hz, Figure 2.

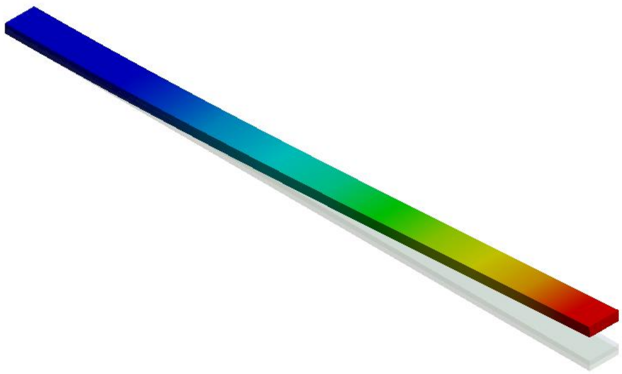


Figure 2: First beam vibration mode.

Table 2 shows the natural frequencies for the first six beam vibration modes with the TMD coupled.

Table 2: Natural girder frequencies with coupled TMD

Vibration mode	Frequency (Hz)
1	5.3498
2	6.7414
3	19.798
4	38.414
5	72.135
6	107.32

The offset at the free end of the beam, with and without the coupled TMD, is shown in Figure 3. For the structure without the TMD, a amplitude of 118.6 mm and the frequency of 6.13 Hz are observed. It is noticed the appearance of two new resonances around the original resonance in relation to the frequency of 6.13 Hz due to the presence of TMD. However, this type of combination should be avoided, because if the system operates with an external force at this frequency, when on and off, it will always pass through the first resonance peak. Therefore, for TMD with optimal damping calculated, with damping factor equal to 0.1273, the amplitude reduction was around 99.7%.

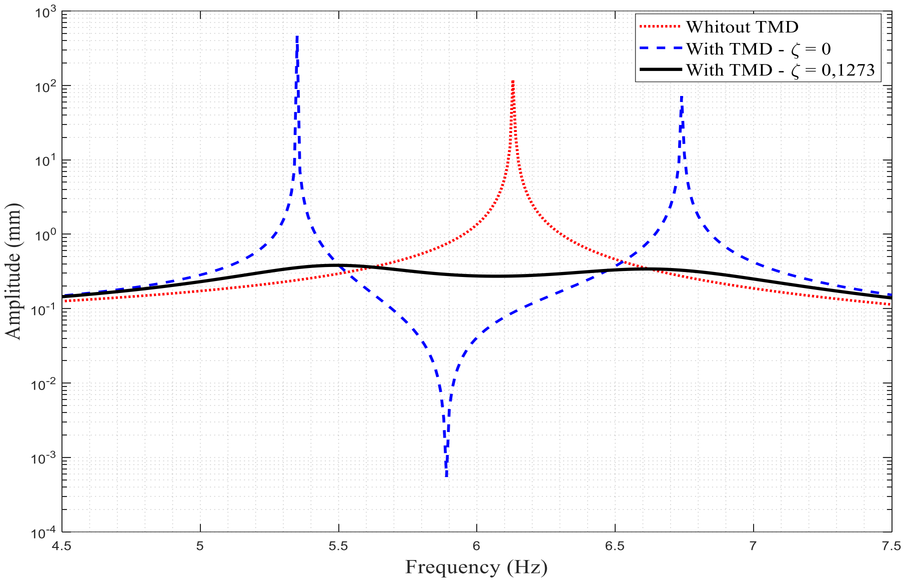


Figure 3: FRF beam with and without TMD.

The amplitude at the end of the AMS was simulated, Figure 4. For an undamped structure, the amplitude of 2807 mm for the first peak is observed for a frequency of 5.35 Hz and for the second peak the amplitude of 244.4 mm for a frequency of 6.74 Hz. Therefore, with optimal damping of 0.1273 an amplitude of 1.48 mm for the first peak and 1.09 mm for the second peak was observed.

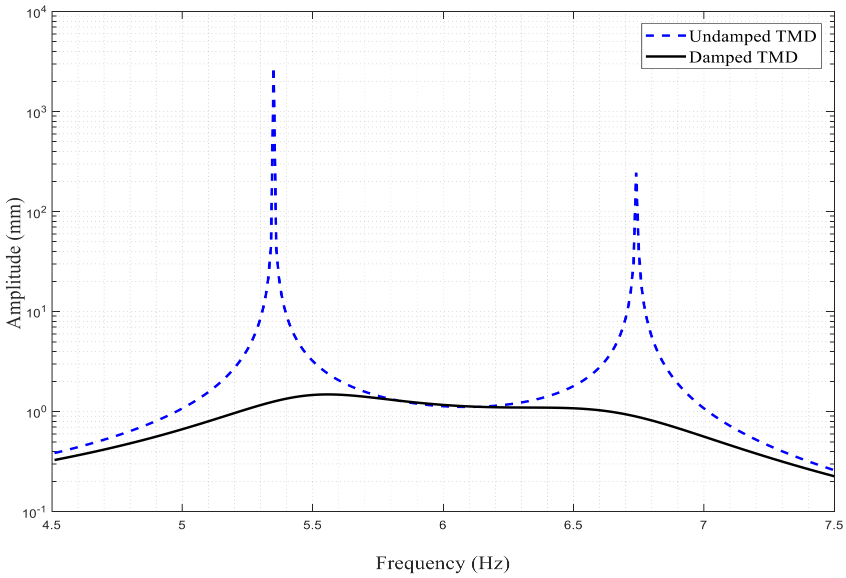


Figure 4: FRF at the end of the TMD with and without damping.

Figure 5 presents the answer, in the time domain, in relation to the displacement of the free end of the beam without the TMD and with the TMD coupled. A decrease of approximately 91.6% of amplitude was observed with the insertion of the dampened TMD, demonstrating that it was able to control the vibratory system.

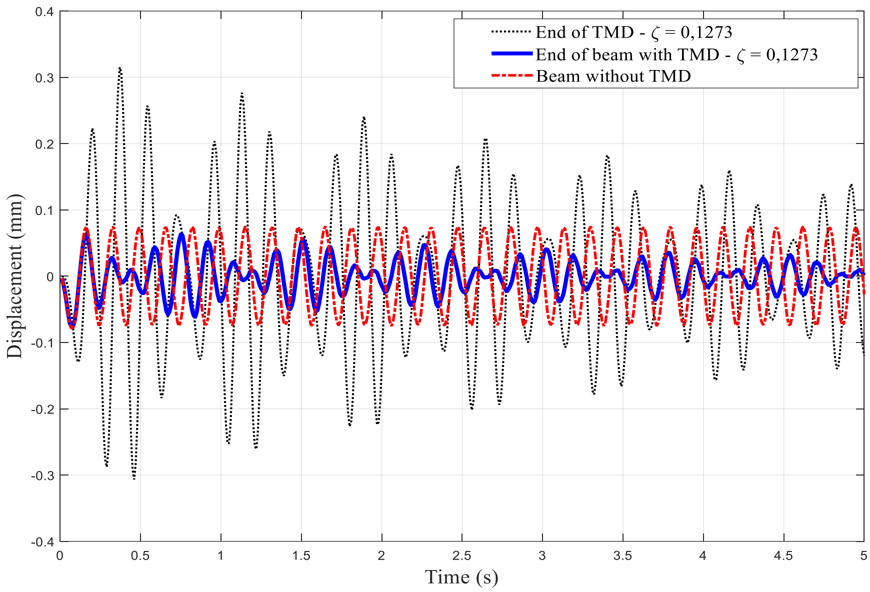


Figure 5: Response in the time domain to the beam with and without TMD

In Figure 5, in relation to the TMD, it is verified that the amplitude is greater than that of the beam, which justifies the fact that it is optimally tuned. Similarly, the phenomenon of the beat is perceived, in which the frequency is close to the natural frequency of the structure.

4 Conclusions

This work presented the vibration control of a set beam subjected to vibrations using TMD. Based on the constructive characteristics of the primary structure, the TMD was modeled. A dynamic analysis of the structure was performed before and after the coupling of the TMD in the base beam. The simulations were performed using the Finite Element Method.

A convergence analysis of the primary structure mesh and the ADV contributed to the accuracy of the results.

The results were satisfactory because the amplitude at the end of the beam showed a reduction of 99% in the frequency domain and approximately 92% in the time domain for a mass ratio of 0.05 and optimal damping factor of 0.1273, demonstrating that the dimensioned TMD model was able to control the vibration of the beam in a resonance situation.

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References

[[1] BATISTA R. C.; PFEIL, M. S. “Múltiplos atenuadores dinâmicos sincronizados para controle das oscilações induzidas pelo vento na Ponte Rio-Niterói”. *Revista Sul-Americana de Engenharia Estrutural*, v. 2, p. 73–95, 2005.

[2] WU J. J. "Study on the inertia effect of helical spring of the absorber on suppressing the dynamic responses of a beam subjected to a moving load". *Journal of sound and vibration*, v.297, p. 981-999, 2006.
[3] HOLANDA, RV DE et al. "Attenuation of sound radiation in concrete structure through the reduction of mechanical vibration". *Revista IBRACON de Estruturas e Materiais*, v. 11, p. 95-114, 2018.
[4] ARI, M.; FAAL, R. T. "Passive vibration suppression of plate using multiple optimal dynamic vibration absorbers". *Archive of Applied Mechanics*, v. 90, n. 2, p. 235-274, 2019.
[5] YOON, G.H; CHOI, H; SO, H. "Development and optimization of a resonance-based mechanical dynamic absorber structure for multiple frequencies". *Journal of Low Frequency Noise, Vibration and Active Control*, 2019.
[6] STRESSER J.F. NOVAK P.R. BRATTI G. GOMES FAA. "Aplicação de um amortecedor de massa sintonizado para o controle de vibração de uma viga engastada". *In: Iberian Latin-American Congress on Computational Methods in Engineering* (41.: 2020: On-line). Proceedings [recurso eletrônico]. São Paulo: ABMEC, 2020.
[7] BLEVINS, R. D. "Formulas for natural frequency and mode shape". *Van Nostrand Reinhold Company*. 1979.
[8] RAO, S. S. "Vibrações Mecânicas". 4. ed. São Paulo, SP: Pearson Prentice Hall, 2008, 424 p.
[9] DEN HARTOG, J.P. "Mechanical Vibrations". New York: Mcraw-Hill, 1956.