Dynamic Analysis of Wind Tower and Dimensioning of Tuned Mass Damper

Julliana Alves Guimarães [1] https://orcid.org/0000-0002-7363-4229

Marcelo dos Reis Farias [2] https://orcid.org/0000-0001-6994-0218

Manuel Teixeira Braz César [3] https://orcid.org/0000-0001-5640-0714

Rui Manuel Menezes Carneiro de Barros [4] https://orcid.org/0000-0002-9510-144X

[1] *Student (Double Degree Program)* - Instituto Politécnico de Bragança, Bragança, Portugal (IPB); Centro Federal de Educação Tecnológica Celso Suckow da Fonseca, RJ, Brasil (CEFET/RJ).
[2] *Professor* - Centro Federal de Educação Tecnológica Celso Suckow da Fonseca, RJ, Brasil (CEFET/RJ).
[3] *Professor* - Instituto Politécnico de Bragança, Bragança, Portugal (IPB).
[4] *Professor* - Faculdade de Engenharia da Universidade do Porto, Portugal (FEUP).

Main author's email: Julliana Alves Guimarães julliana.ja@gmail.com

Abstract

The present work sought to introduce concepts pertinent to the design of a passive damper, known as TMD (Tuned Mass-Damper), and to apply them to the case of a wind tower present on the IPB Campus. Therefore, it was necessary to perform a dynamic analysis for this structure through finite elements method, using the ANSYS software, in order to obtain its natural frequencies and vibration modes. This information is essential to the calculation of the optimal TMD's parameters, where it was proposed that it be used to attenuate the vibration of the tower in its first natural frequency. After that, some simulations were performed using MATLAB software in order to observe the structure response after the introduction of the TMD. The responses obtained showed a significant reduction of tower vibration, especially when greater mass values were used. In addition, it was observed that for cases where the optimal TMD's parameters are not used properly, the vibration attenuation occurs in a less satisfactory way. Through these results, it was possible to demonstrate the coherence of the calculations performed and the simulations made, presenting this work, therefore, important contributions to the future construction of the TMD.

Key-words: *Wind Tower; TMD; Vibration Control; Damping.*

1 Introduction

The wind generators are machines responsible for transforming the kinetic energy of the wind into mechanical energy by rotating the blades of a rotor that, connected to a generator, is capable of converting mechanical energy into electricity. Their increasing use in the global scenario have also led to the need for constant studies related to their control systems in order to guarantee the integrity of the tower, wind turbine and mechanical components, as well as to guarantee good operating performance. Among the different aspects existing in control systems, such as overcurrent control, short circuits, overvoltage,

DOI: 10.25286/repa.v6i2.1465

and others, there is a concern about the efficient control of the structure's vibration **[1, 2, 3]**.

The wind tower is an important component of the wind system, as it is responsible for sustaining the wind turbine. Over time, a notable factor is the increase in the height of these towers, subjecting the entire structure and mechanical components to greater vibration effects, thus demonstrating the importance of introducing an adequate structural vibration control **[4]**.

Devices such as TMD (Tuned Mass Damper), Controllable Fluid Dampers, TLD (Tuned Liquid Dampers, among others, can be used in order to promote structural vibration control. The dampers can also be classified as passive, active or semi-active, where the main difference is in the use or not of external power to absorb vibration energy. In a passive system this control occurs only due to the presence of springs and dampers, or even a pendulum, while in an active system the control occurs by adding an active element, being possible to find in the system controllers and actuators, responsible for calculating the necessary force to control the vibration and apply it to the structure. A semi-active system, in turn, uses external energy only in the modification of the stiffness and damping parameters of the device, not on direct application in the vibration attenuation as in the active control **[3, 5]**.

2 Tuned Mass Damper (TMD) and its Optimal Parameters

The TMD is characterized by the use of a mass m_t, a spring of stiffness k_t and a damper of coefficient c_t, where tuning its natural frequency to the frequency to be controlled, this device is able to control the vibration of the structure where it was implemented. An alternative geometry for the TMD is the pendulum, where it is modeled with a hanging mass that sways in the opposite movement to the excited structure. TMD has great applicability in tall buildings, bridges, towers and, also, in wind towers, and can be used as a passive, active or semi-active device. It is usually located at the point of greatest modal amplitude, in order to attenuate the vibrations related to the first mode of vibration **[3, 5]**.

The parameters of stiffness, mass and damping of a TMD can be calculated from the dynamic equilibrium equations of the Structure-TMD system. It is commonly used to reduce the degrees of freedom of the structure, simplifying the system and facilitating the calculation procedures. It is also important to know the frequency at which the TMD should act and the structure's damping. TMD parameters must be dimensioned in order to obtain the best possible vibration control results, that is, minimizing the amplitude of movement of the structure. In the response of a 2 degree of freedom system, for example, in the frequency domain graph, there are two points where all curves pass, regardless of the value of the TMD damping factor, ξ_t. The so-called ideal curve is one where these two points are equivalent to the peaks of the curve (that is, zero slope), and whose height of both points tend to the same value. Then applying these conditions, the so-called optimal parameters of the TMD are obtained **[6, 7]**.

For the case of a mass subject to free vibration where a TMD is introduced, the result is a system of 2 degrees of freedom as shown in Figure 1, governed by the dynamic equilibrium equations shown in Equation (1), where the parameters with index t represent TMD, and with index s represent the main mass.

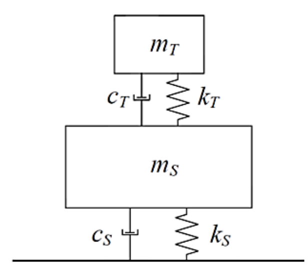


Figure 1: Theoretical model of 2 DOF representing a structure with damping and with a TMD installed, subject to free vibration [adapted].
Source: Paredes, M. M. **[6]**.

$$\begin{cases} m_s\ddot{x}_s + c_s\dot{x}_s + c_t(\dot{x}_s - \dot{x}_t) + k_s x_s + k_t(x_s - x_t) = 0 \\ m_t\ddot{x}_t + c_t(\dot{x}_t - \dot{x}_s) + k_t(x_t - x_s) = 0 \end{cases} \quad \textbf{(1)}$$

For this case, after application of the previously mentioned conditions and performing the proper mathematical calculations, it's possible to get the following equations of optimal parameters q and ξ_t of TMD shown in Equations (2) and (3), where q is the ratio of frequencies of the TMD and mass ($q = \omega_t/\omega_s$), ξ_t and ξ_s are the factor damping of TMD and mass respectively, and μ is the mass ratio ($\mu = m_t/m_s$) **[6]**.

$$q_{optimal} = \left(\frac{1}{1+\mu} - \sqrt{\mu}\frac{\xi_s}{(1+\mu)\sqrt{1+\mu-\xi_s^2}}\right) \quad (2)$$

$$\xi_{t,optimal} = \frac{\xi_s}{1+\mu} + \frac{\sqrt{\mu}\sqrt{1+\mu-\xi_s^2}}{1+\mu} \quad (3)$$

As mentioned, these equations correspond to the case of a mass subject to free vibration, but will vary depending on the conditions under which the system is subject, such as harmonic or random actions.

This work aims to propose a TMD to act on the first natural frequency of a wind tower subject to free vibration, dimensioning its optimal parameters and evaluating its influence on the system.

3 Methodology

3.1 Dynamic Analysis of Wind Tower

For this study, a small wind tower present on the Campus of the Polytechnic Institute of Bragança (IPB), with peak power equal to 1.4 kWp, was used as the object of study, as shown in Figure 2.

(a) (b)

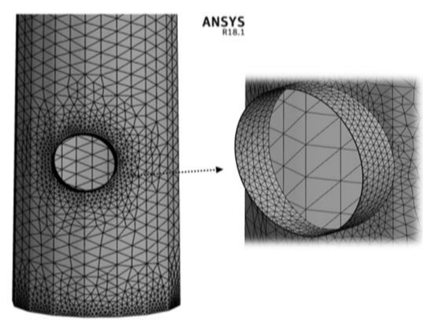
Figure 2: Horizontal-axis wind tower of the IPB Campus - (a) Wind generator and part of the tower, (b) bottom of the tower.
Source: Author.

The tower was modeled numerically using the finite element method in the ANSYS software. Then, from the dynamic analysis it was possible to obtain its natural frequencies and modes of vibration. For this purpose, simulations were made considering the upper end of the structure as being free and its base fixed, since this configuration is considered in the bibliography as a good initial approximation of a real structure base [9].

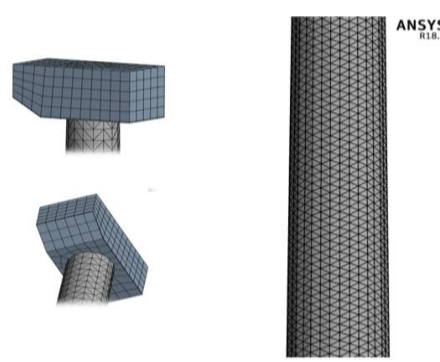
The structure was initially designed in the Autodesk Inventor software, not taking into account the ladder and structural defects, and the introduction of the wind turbine on the tower was made in a simplified way, where this was represented by a massive box fitted on the top of the structure, having weight equivalent to the total weight of the rotor.

Simplifications of nacelle consist of a satisfactory simulation practice, which is also used in other works [8, 9, 10, 11, 12]. The wind towers are subjected to different dynamic stresses due to, for example, the rotor rotation, the action of the wind, and the blade pass through the tower. However, in this paper, these dynamic effects will not be included, since the focus here is on observing the natural frequencies of the structure, analyzing the natural vibration of the tower when subjected only to free vibration.

The mesh was then created in the structure using the ANSYS software, which can be seen in Figure 3. In this step, the convergence of the results in relation to the first natural frequency of the system was considered, where in Figure 4 it is possible to see the amount of elements generated, as well as the first natural frequency equivalent to 1.56 Hz.



(a)



(b)

DOI: 10.25286/repa.v6i2.1465

Figure 3: Meshes in - (a) proximity to the hole and base, (b) tower and upper part.
Source: Author.

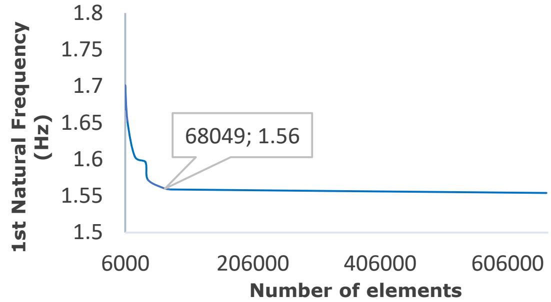


Figure 4: Mesh convergence.
Source: Author.

As a parameter for analyzing the frequencies obtained numerically, the results obtained experimentally by Dias, L. **[13]** will be considered. In the test, two accelerometers were used at a height of 1.41 m in two different positions (x and y). The tower was excited using a PCB 086b20 impact hammer, and the first average frequencies obtained can be seen in Table 1. For this test, intermediate vibrations were not taken into account, such as vibrations in the wind turbine, and the frequency spectrum obtained in the test can be seen in Figure 5.

Table 1: Frequencies obtained experimentally.

	Frequencies (Hz)
First mode of vibration	1.61
Second mode of vibration	5.99

Source: Dias, L. **[13]**.

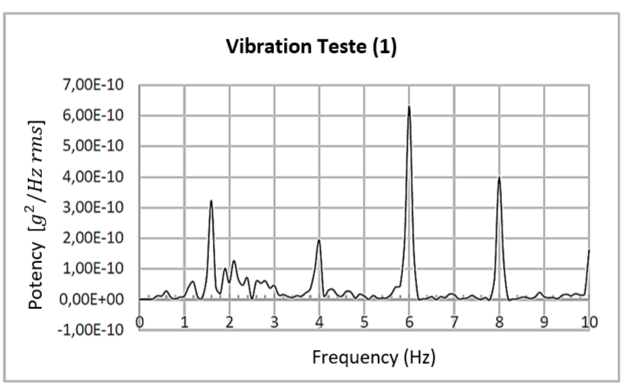
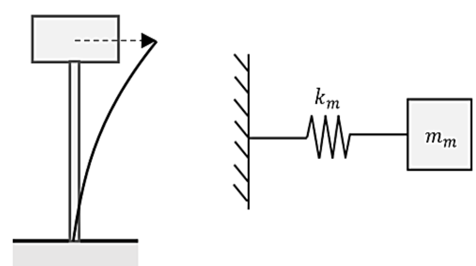


Figure 5: Spectrum of frequencies obtained experimentally. [adapted].
Source: Dias, L. **[13]**.

3.2 Proposal of a TMD for the structure

Tuning the TMD means choosing the appropriate mass, stiffness and damping parameters for a given actuation frequency, so that the movement of the main structure be minimized. Therefore, first the structure is reduced in an equivalent mass-spring system of 1 DOF, where its structural parameters are replaced by its modal (or generalized) parameters that correspond to the mode of vibration in which the TMD will act. Figure 6 shows schematically this system of 1 DOF, representing the first mode of flexion of the structure, where k_m and m_m correspond to the modal stiffness and mass. The modal mass is obtained from ANSYS itself, and with that, the modal stiffness can be calculated, since the values of m_m and natural frequency ω_m are known. Considering that steel towers have, in particular, a lower level of structural damping **[14, 15]**, it was considered for this first approach that the structural damping is approximately zero. In addition, for cases where this value is reduced ($\xi_s \leq 1$), the design of the TMD can be performed considering the damping of the structure as zero **[7]**.



(a) (b)

Figure 6: (a) Flexion mode (b) simplified to a modal mass-spring system.

When introducing the mass-spring-damper TMD into the structure, the system then changes from 1 DOF to 2 DOF, as shown in Figure 7, where k_t, c_t, and m_t correspond to the stiffness, damping coefficient and mass of the TMD, respectively, and the dynamic equilibrium equations of this system can be seen in Equation (4). These parameters are then dimensioned from the equations of optimal parameters, as shown in Equations (2) and (3), where adapting them to the system in Figure 7, they become as follows, as shown in Equations (5) and (6).

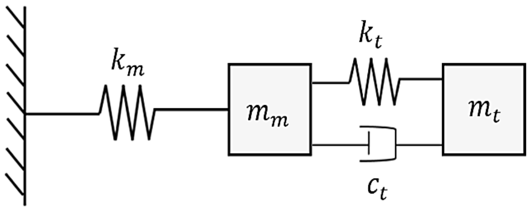


Figure 7: Mass-spring system with coupled TMD.

$$\begin{cases} m_m\ddot{x}_m + c_t(\dot{x}_m - \dot{x}_t) + k_m x_m + k_t(x_m - x_t) = 0 \\ m_t\ddot{x}_t + c_t(\dot{x}_t - \dot{x}_m) + k_t(x_t - x_m) = 0 \end{cases} \quad \textbf{(4)}$$

$$q_{optimal} = \left(\frac{1}{1+\mu}\right) \quad \textbf{(5)}$$

$$\xi_{t,optimal} = \frac{\sqrt{\mu}\sqrt{1+\mu}}{1+\mu} \quad \textbf{(6)}$$

The mass of the TMD m_t should be chosen initially, where it is suggested that its value be about 0.5% to 1% of the total mass of the structure, and in towers of larger size it is also observed the use of values around 3% **[10, 16]**. The total mass of the structure can also be obtained using ANSYS. From the pre-determination of m_t and the knowledge of the modal mass m_m, it is then possible to find the mass ratio μ, and calculate $q_{optimal}$ and $\xi_{t,optimal}$, from where it will be possible to obtain the stiffness and damping parameters of TMD, as well as its frequency ω_t.

After obtaining the TMD's parameters, the system response is simulated using the MATLAB software, where an algorithm was developed based on the previous equations. The tool used to solve the system of differential equations was ODE45 through the application of the fourth-order Runge-Kutta Method. The FFT (Fast Fourier Transform) function was also used to obtain the system's response in the frequency domain.

4 Results and Discussion

4.1 Dynamic Analysis

From the dynamic analysis performed in ANSYS it was possible to obtain the natural frequencies and modes of vibration of the structure. It was observed that for each pair of frequencies the values were very close. The first two pairs of frequencies obtained numerically are shown in Table 2.

Table 2: Natural frequencies obtained numerically.

Frequencies (Hz)	
First mode of vibration	1.56
Second mode of vibration	1.59
Third mode of vibration	6.79
Fourth mode of vibration	6.89

Source: Author.

It was also possible to observe that for each pair of frequencies the modes of vibration are similar, acting however in opposite directions. Figure 7 shows the first and second modes of vibration of the structure, corresponding to flexion modes, where what differentiates them is the direction of action, one in the X direction and the other in the Z direction. Since the first modes of vibration are the most important and likely to occur, then the other modes will not be placed here.

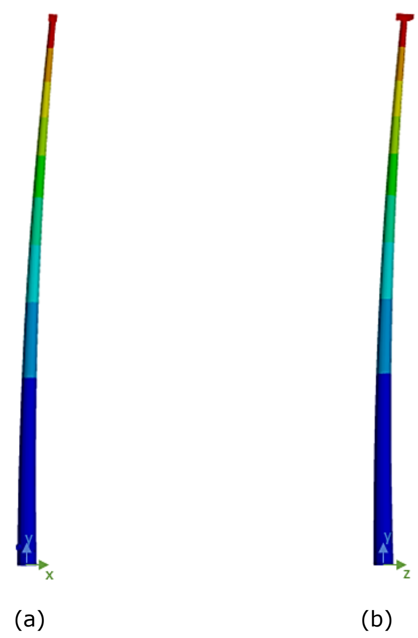


Figure 7: (a) 1st and (b) 2nd mode of vibration.

Thus, given the similarity to each pair of frequencies, the average of these pairs will now be considered, that is, the first frequency corresponds to the average of the first pair of similar frequencies, and

DOI: 10.25286/repa.v6i2.1465

the second frequency to the average of the second pair of similar frequencies.

4.2 Comparison of Numerical and Experimental Results

From the values of frequency obtained experimentally, presented in Table 1, and the values of frequency obtained numerically (considering the average of the pairs of frequency), it is possible to observe a small error related to the first natural frequency, equivalent to 1.86%, and a 14.19% error related to the second frequency, as shown in Table 3.

Table 3: Experimental and numerical frequency results and respective errors.

Natural Frequency	Experimental	Numerical	***Error (%)***
First mode of vibration **(Hz)**	1.61	1.58	***1.86 %***
Second mode of vibration **(Hz)**	5.99	6.84	***14.19 %***

Source: Author.

Then, considering that the first frequency was the best observed numerically and experimentally, and because it is considered the main one it was then considered as a parameter of TMD tuning. Therefore, TMD will act when this frequency is reached, vibrating out of phase in relation to the movement of the structure.

Some factors may have contributed to the reported differences such as, for example, the simplification of the numerical model of the tower, where the ladder present in the structure, structural defects, and the real shape of the wind turbine were not considered. The wind turbine was simplified as a slightly decentralized solid box at the top of the tower, aiming to simulate the effect of the rotor's center of gravity on the response of the structure, but being an approximation since the real value is not known.

Another reason for such differences may have been due to the experimental test, where vibrations from the wind turbine may have generated interference in the results. The test performed was an environmental vibration test, that is, the response of the structure was obtained being subject to its working and operating conditions. Environmental vibration tests like this provide large amounts of data that need to be handled carefully, as the captured signal receives contribution from various frequencies, such as wind, rotor and blade rotation, and also from the generator. Therefore, being this type of signal considered white noise, the natural frequencies of the structure can be recognized through the identification of frequencies with higher energy content. Therefore, a small flaw in this identification can also be considered as the cause of the difference in the results. In addition, the position of the sensors may have been another cause of the observed differences, where it is suggested that they be used at different points of the structure for a more satisfactory test.

4.3 Optimal TMD's parameters and response of the structure

Initially, the mass of the TMD was considered to be 1% of the total mass of the structure (equivalent to approximately 752.37 kg), which corresponds to m_t = 7.52 kg. The modal mass referred to the first mode of vibration (f =1.58 Hz, equivalent to ω_m= 9.93 rad/s) is obtained from numerical analysis, and corresponds to m_m = 353.28 kg. With this, the modal stiffness is

$$k_m = {\omega_m}^2 . m_m = 34\,814.1\ N/m. \tag{7}$$

The mass ratio μ is equivalent to

$$\mu = \frac{m_t}{m_m} = 2.13\%. \tag{8}$$

With the value of μ, the frequency value of TMD is found, in order to obey the optimal ratio of frequencies q. Therefore,

$$q_{optimal} = \left(\frac{1}{1+\mu}\right) = 0.98 \tag{9}$$

$$\omega_t = 0.98\,.\,\omega_m \cong \ 9.73\ rad/s. \tag{10}$$

From Equation (6) it is also possible to find the optimal value of the damping factor for TMD, from where the damping coefficient c_t will be obtained.

$$\xi_{t,optimal} = \frac{\sqrt{\mu}\,.\sqrt{1+\mu}}{1+\mu} = 0.14 \tag{11}$$

$$c_t = \xi_{t,optimal}\,.2.m_t.\omega_t = 20.48\ N.s/m. \tag{12}$$

With these values of m_t and ω_t, the stiffness required for TMD will be

(13)

$$k_t = {\omega_t}^2 . m_t = 711.65 \ N/m.$$

Table 4 presents, briefly, the values of the parameters obtained for the TMD.

Table 4: TMD's parameters for vibration control of the first mode of vibration of the structure.

Parameter	Value	Unit
Mass	7.52	kg
Stiffness	711.65	N/m
Damping	20.48	N.s/m

Once the optimal TMD parameters were determined, it was then possible to obtain the system response graphs shown in Figure 8, through MATLAB.

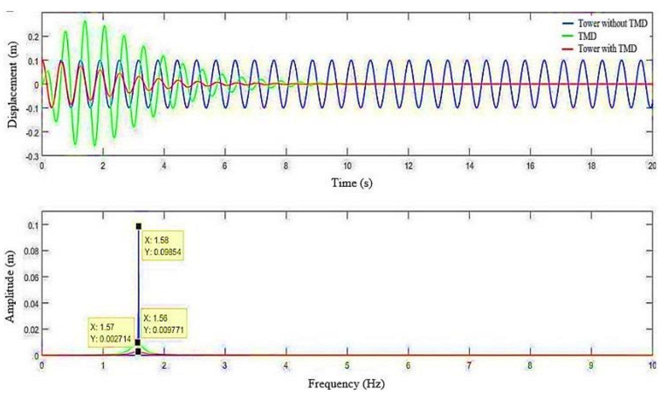


Figure 8: Response of the spring-mass system with and without coupled TMD, in the time and frequency domain: m_t=7.52kg, k_t=711.65 N/m, c_t=20.48 N.s/m.

As can be observed, the presence of TMD resulted in the rapid attenuation of the tower vibration. The two new frequencies obtained for the present case were f_1 =1.56 Hz and f_2 =1.57 Hz as can be seen in the frequency domain response graph. In this same graph it is possible to visualize the detachable difference that exists in the amplitude for the frequencies of the system without TMD and with TMD, where these amplitudes are related to the amount of energy in the vibration movement for the simulation time considered. Therefore, the system response without TMD presents the highest peak because throughout the simulation time its vibratory energy is constant, while the system response with TMD presents much smaller amplitudes, for the mass of the TMD and for the mass of the tower, because during the simulation time both masses present a decrease of their vibration energy.

Increasing the mass m_t, it was observed that the vibration attenuation occurs faster and that the amplitude of the displacement suffered by the TMD becomes smaller. In addition, from the frequency domain graph it was noticed that the peaks of the curve of the Tower-TMD system become smaller and closer to each other in height **[17]**. This can be observed in Figure 9, where m_t was increased by 10% of the total mass of the structure and the other parameters were updated within the optimal conditions. However, although the use of a larger mass is apparently a good choice, it is advisable to evaluate the design limitations such as, for example, the space available in the tower for TMD installation.

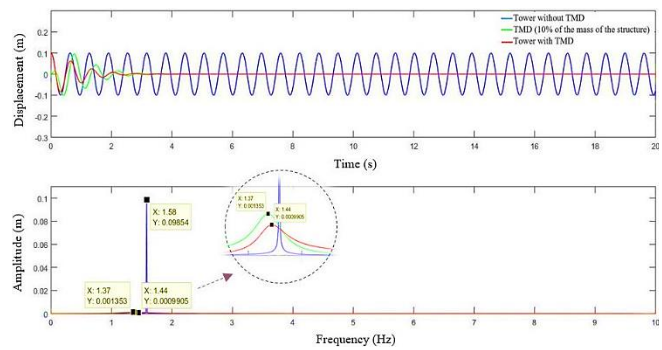


Figure 9: Response of the spring-mass system with and without coupled TMD, to m_t equal to 10% of the total mass of the structure: m_t=75.24 kg, k_t=5108.62 N/m, c_t=520.78 N.s/m.

4.4 Influence of the change of TMD's parameters on the structure response, without the correct adequacy to optimal conditions

Now, through the variation of TMD parameters so that they disperse from their optimal values, it will be possible to observe how the vibration attenuation of the structure occurs and, consequently, to evaluate what is the need to use the equations of optimal parameters correctly.

First, the mass m_t was varied, where the other parameters k_t and c_t were kept constant, that is, they were not updated according to the equations of optimal parameters. Figure 10 shows the answer for the case of m_t=4.51 kg, that is, reduced by 40% of

DOI: 10.25286/repa.v6i2.1465

its initial value. Figure 11 shows the answer for the case of m_t=10.53 kg, that is, increased by 40%.

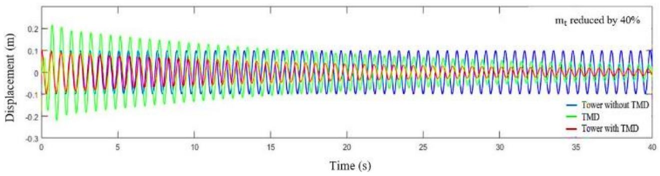


Figure 10: Response of the spring mass system with and without coupled TMD, to m_t reduced by 40% (m_t=4.51 kg), and k_t=711.65 N/m, c_t=20.48 N.s/m.
Source: Author.

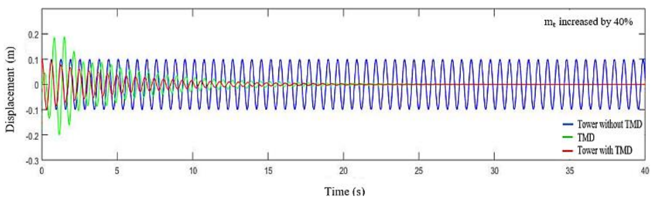


Figure 11: Response of the spring-mass system with and without coupled TMD, to m_t increased by 40% (m_t=10.53 kg), and k_t=711.65 N/m, c_t=20.48 N.s/m.

In both graphs, less efficient results in vibration attenuation are observed, mainly with m_t reduction. In cases where the parameters are even more distant from their optimal values, it is possible to observe situations where attenuation does not occur, as shown in Figure 12.

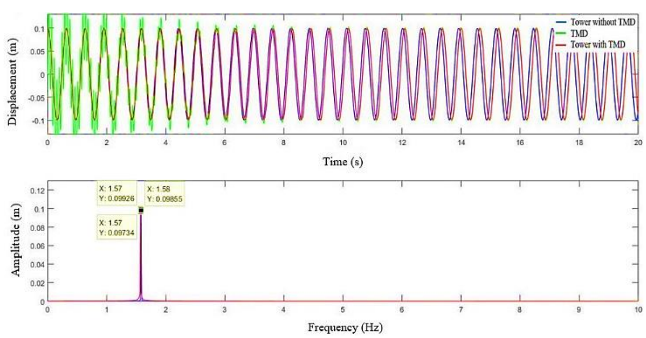


Figure 12: Response of the mass-spring system with and without coupled TMD, to m_t=9 kg, k_t=3 N/m and c_t=20040 N.s/m.

It is also valid to point out that, in cases where the TMD is designed to act when a certain external force is applied, if its parameters are not carefully implemented, it will be possible to observe that there may be an increase in the dynamic amplitude of the tower.

Similarly, the damping c_t was also varied, where the other parameters k_t and m_t were kept constant, that is, they were not updated according to the equations of optimal parameters. Figure 13 shows the answer for the case of c_t= 4.08 N.s/m, that is, reduced by 80% of its initial value. Figure 14 shows the answer for the case of c_t= 36.80 N.s/m, that is, increased by 80%.

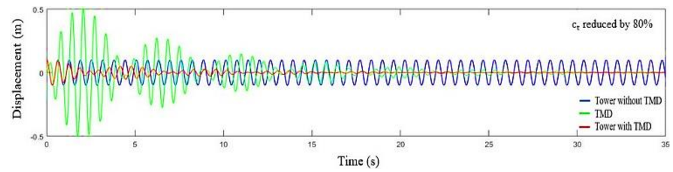


Figure 13: Response of the spring-mass system with and without coupled TMD, to c_t reduced by 80% (c_t=4.08 N.s/m), and k_t=711.65 N/m, m_t=7.52 kg.

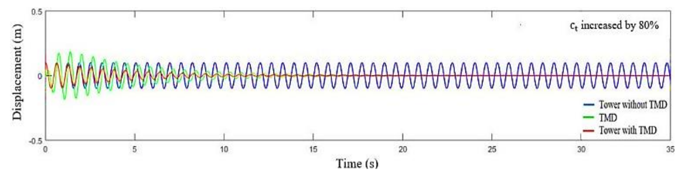


Figure 14: Response of the spring-mass system with and without coupled TMD, to c_t increased by 80% (c_t=36.80 N.s/m), and k_t=711.65 N/m, m_t=7.52 kg.

With the reduction of the damping coefficient, it is observed that the displacement of the TMD starts to present greater amplitudes, and the vibration of the tower remains for a longer time. It is also possible to notice the inconsistency in the vibration attenuation, where in some periods there is a slight increase in the vibration where it immediately reduces again. On the other hand, with the increase of the damping coefficient, the TMD starts to present smaller displacements, but the time of vibration attenuation of the tower continues to be greater than the time observed in the responses obtained through the optimal parameters (Figures 8 and 9).

In all cases, it is notable that the responses tend to be more efficient as the TMD's parameters converge to their values considered optimal.

As a last consideration, it is worth noting that, since free vibration has been treated here, the amplitude of the masses displacements observed in the responses is dependent on the initial displacement of the tower adopted in the simulations (this is a case of a Possible and Indeterminate System). However, in a constructive TMD project, it is important that an experimental study of the tower vibration amplitudes be carried out, thus obtaining adequate initial conditions for the simulation. Depending on these conditions, the diameter of the tower where the TMD would be installed (upper internal diameter) must be large enough to allow the displacement of m_t and to accommodate the other elements of the system, such as spring, damper and guides. However, regardless of such imposed conditions, the application of the optimal parameters of the TMD will imply an optimized attenuation of the tower vibration.

The construction and implementation of the TMD would consist of another stage of the study, where it would be necessary to gather the design restrictions, from the dimensions to the limited commercial availability of elements (as is the case of the dampers in relation to their damping factor). In this work, constructive details are not within the objective. However, Figure 15 schematically shows a possible arrangement of the elements in the system, where the ends would be fixed to the internal walls of the tower (the TMD could also, alternatively, be implemented inside the nacelle, or even outside the tower, which possibly require a different arrangement of the elements). However, this is only a possible implementation, as the elements (spring and damper) could also be placed both on the same side, as shown by LIMA, D. M. **[10].** He proposes a constructive model, where guides and rails are implemented for the translation of the mass in the correct direction. It is also worth mentioning that TMD should act in two different directions (X and Z) regarding the 1st and 2nd modes of vibration of the tower (see Figure 7), therefore two complete TMDs can be applied to the system (one for each direction mentioned), or a special arrangement for just one TMD must be developed taking in consideration acting in two direction.

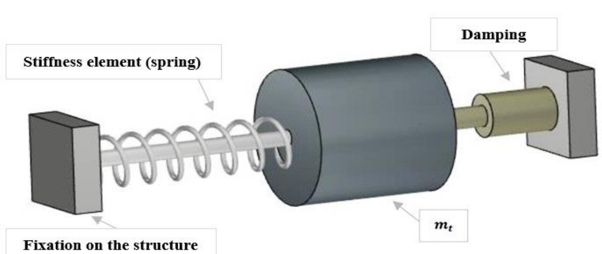


Figure 15: Scheme showing possible arrangement of system elements.

5. Conclusions

Through the dynamic analysis of a wind tower, performed numerically, it was possible to extract information of frequency and modes of vibration of the tower, where when comparing them with experimental results, the first frequency was the one that presented the most satisfactory result, what made this frequency chosen for TMD tuning. These differences may have resulted from simplifications of the numerical model, or from possible errors in the identification of natural frequencies through experimental analysis, since in an environmental vibration test the structure receives the contribution of different excitation frequencies, and then, the natural frequencies must be carefully identified through the energy content of the signals obtained.

In the proposal of a TMD for the structure it was possible to verify that the use of the equations of optimal parameters generated satisfactory results, providing the best result in the vibration attenuation of the tower. This statement could be confirmed by observing that the results were more efficient when the TMD's parameters converged to their values considered optimal.

For the simulated case, of free vibration, the system is not being continuously excited by external force and, therefore, in the simulation it is only observed how fast or slow the vibration attenuation occurs due to the variations in the parameters. However, in case of simulations with forced systems, the inappropriate use of TMD parameters can lead to amplification of the system's vibration, causing the resonance effect, and that would be a proposal for future work.

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