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USE OF METAMATERIALS TO REDUCE VIBRATION OF A WIND TOWER SUBJECTED TO AN ARBITRARY STOCHASTIC WIND

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Abstract. *To ensure that wind turbine towers remain economically viable, they must be able to operate for prolonged periods. Therefore, analyzing the dynamic behavior of the structure is crucial to extend its lifespan and optimize its efficiency. In this study, we investigate the attenuation of vibrations in a wind turbine tower using metamaterials. For this analysis, a discrete model of the tower is considered. The finite element method (FEM) is employed in the numerical simulation using a routine developed in Python. The wind effect is described by force signals obtained through a statistical method, using Weibull Probability Distribution as random distribution and Kaimal's power spectral density to simulate the speed wind fluctuations caused by turbulence. The action of the wind is represented by a concentrated force at the upper end of the tower, specifically concentrated at the nacelle. To reduce the tower vibration, identical resonant metamaterials are considered.*

Keywords: *Wind turbine tower, metamaterials, dynamic response, frequency domain, discrete models*

1. INTRODUCTION

One of the great debates of our time is certainly the pursuit of renewable energy sources, as they not only provide fewer environmental impacts and can be naturally replenished, but also contribute to reducing the world's dependence on non-renewable sources such as Petroleum and Coal. Since 2019, fossil fuel consumption accounted for over 81% of global energy production (IEA, 2021). In this regard, wind turbines play a vital role in harnessing clean and renewable energy.

Onshore and offshore wind turbines are subjected to dynamic loads that may be induced by wind, wave and seismic excitation, which potentially threaten their structural integrity and performance. To ensure the safe and efficient operation of wind towers, effective vibration control strategies are essential. In this context, different vibration control methods were proposed by various researchers (Jahani *et al.*, 2022). Considering passive control strategies, the most studied devices are the tuned mass damper (TMD) (Sun and Jahangiri, 2018; Yang *et al.*, 2019; G.B. Colherinhas and M.R.Machado, 2022), followed by the tuned liquid damper (Hemmati *et al.*, 2019). As these devices would be positioned in the turbine nacelle, the practical implementation may be unfeasible due to space limitations.

One passive control strategy that has gained widespread application in recent times for reducing vibrations amplitude is the use of metamaterials. In summary, metamaterials are defined as materials or structures that exhibit some form of spatial periodicity, which are designed in such a way that the material possesses enhanced properties, in other words, in the study of metamaterials, the aim is to develop materials whose physical properties are not found in nature, like negative mass density and negative Poisson's ratio.

According to Hussein *et al.* (2014), the origin of the study of these structures' dates to Newton in his attempt to describe the propagation of sound in air (1686) and more recently with the work of Kock and Harvey (2005), who, based on the study of electromagnetic waves, developed structures capable of refracting and focusing sound waves using an array of small structures compared to the incident wavelength. Regarding periodic structures, Cremer and Leilich (1953) are among the first studies present in the literature. In their work, they sought to investigate harmonic flexural wave motion along a 1D periodic beam.

In general, it is possible to say that the development of metamaterials began in the field of electromagnetism and extended to various other engineering areas, and it can be classified as: Electromagnetic, optical, acoustic, and mechanical metamaterials, figure 1 illustrates this division based on the specific properties of each one.

In this context the use of mechanical metamaterials to control vibrations has been vastly applied to beams (Gao *et al.*, 2019; Sugino *et al.*, 2016; Pai, 2010). This concept of placing multiple resonators at periodic intervals along the structure may be interesting for wind turbine tower control. Compared to traditional TMDs, these resonators have a smaller mass and can be conveniently placed along the tower, possibly making them a more feasible option.

This paper presents an analysis of vibration reduction of a wind tower through the use of metamaterials. To determine

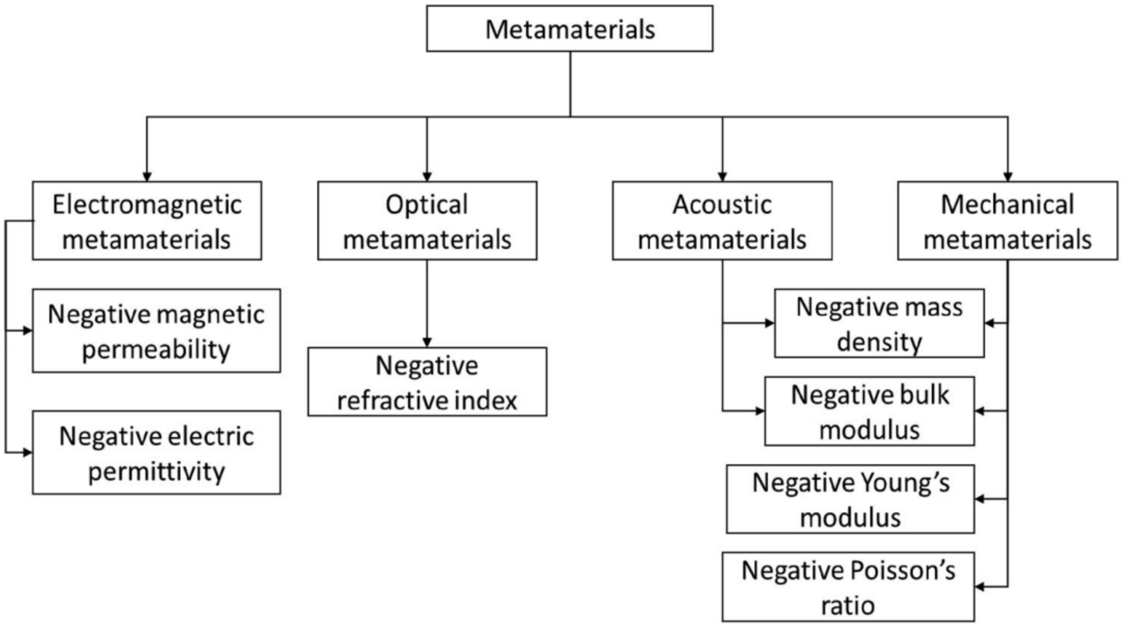


Figure 1: Classification of metamaterials (Balaji *et al.*, 2021)

the force due to wind, a Weibull distribution was used as a random distribution to determine the average wind speeds, and the Kaimal Power Spectral Density (PSD) was applied to simulate wind speed fluctuations caused by turbulence, following the study conducted by Alkmim (2017). The tower's structural representation involves a cantilever beam with an endpoint mass, and its discretized model is derived using the Finite Element Method (FEM). The study evaluates the performance of the metamaterials for different positions of the resonators along the tower and considering attenuation of the first and the fifth modes.

2. MATHEMATICAL MODEL

This section is dedicated to presenting the mathematical model used to assess the dynamic behavior of the turbine, encompassing both the tower-nacelle assembly and the wind-induced forces.

2.1 Tower-nacelle assembly model

The turbine's structural model, used to evaluate its dynamical behavior, is derived using the Finite Element Method (FEM), where beam elements are considered. The simplified model of the tower and nacelle considers a cantilever uniform beam with endpoint mass at the top, as shown in Fig. 2.

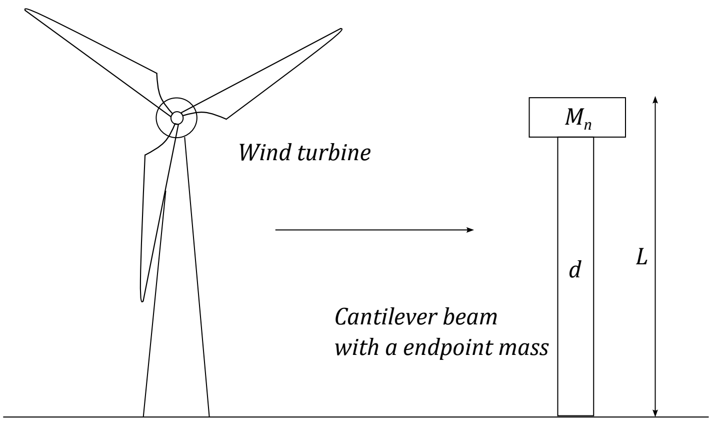


Figure 2: Wind turbine simplified model.

According to Petyt (2010), the element matrices of mass and stiffness are given by:

$$[M_e] = \frac{n_e \rho A l_e}{420} \begin{bmatrix} 156 & 22l_e & 54 & -13l_e \\ 22l_e & 4l_e^2 & 13l_e & -3l_e^2 \\ 54 & 13l_e & 156 & -22l_e \\ -13l_e & -3l_e^2 & -22l_e & 4l_e \end{bmatrix}, \qquad [K_e] = \frac{EI}{l_e^3} \begin{bmatrix} 12 & 6l_e & -12 & 6l_e \\ 6l_e & 4l_e^2 & -6l_e & 2l_e^2 \\ -12 & -6l_e & 12 & -6l_e \\ 6l_e & 2l_e^2 & -6l_e & 4l_e^2 \end{bmatrix} \tag{1}$$

where n_e is the number of elements, ρ is the density of the tower, A is the cross-section area of the tower, l_e is the length of the element, E is the Elasticity modulus of the tower and I is the cross-section moment of inertia of the tower. Global stiffness and inertia matrices are obtained from the element matrices presented in Eq. (1), incorporating the nacelle mass, M_n, and the linear resonators (modeled as mass-spring-damper oscillators). The equation of motion of the tower-nacelle-metamaterials assembly is:

$$[M]\{\ddot{u}\} + [C]\{\dot{u}\} + [K]\{u\} = \{F\}, \tag{2}$$

where $[M]$ is the global mass matrix, $[C]$ is the global damping matrix, $[K]$ is the global stiffness matrix, and $\{F\}$ is the global load vector.

2.2 Wind model

To model the wind behavior, a random distribution function is employed to represent the probability distribution of wind speed. The Weibull distribution, widely recognized for this purpose, is adopted in the current study. The function is given by:

$$P(\bar{U}) = \frac{k}{c}\left(\frac{\bar{U}}{c}\right)^{k-1} \exp(\frac{\bar{U}}{c}) \tag{3}$$

Based on the data obtained by Ozawa (2017) through the least squares method for the city of Ventania (PR), it is used $k = 2.81$ and $c = 3.94$. The obtained Weibull distribution is shown Figure 3. Once the parameters of the Weibull distribution are known, it is possible to generate mean velocity values, $\bar{U}$, that follow this distribution using the Inverse Cumulative Distribution Function (ICDF) as follows:

$$\bar{U}(P) = -c[ln(1-P)]^{1/k}. \tag{4}$$

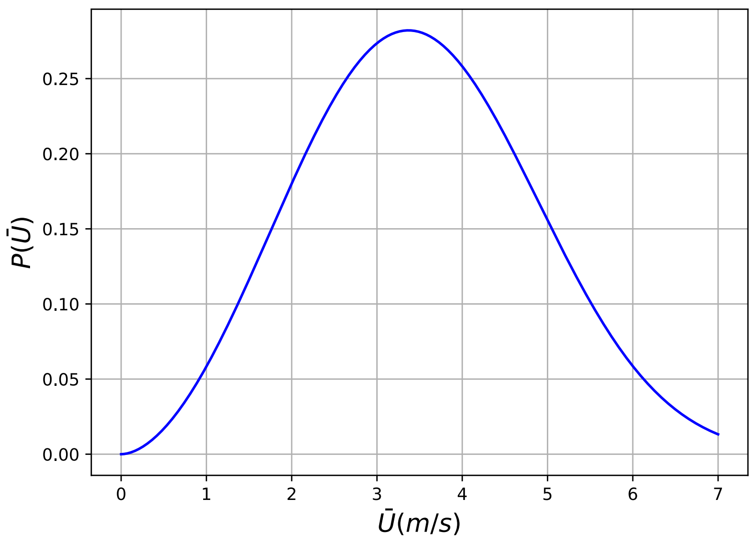


Figure 3: Weibull Distribution

The wind behavior data obtained by Ozawa (2017) were measured at a height of 10m. Therefore, due to the impracticality of empirically measuring these values for all heights, Custódio (2007) proposed a mathematical model to determine wind velocities at any desired position:

$$\frac{v_1}{v_2} = \frac{ln\left(\frac{h_1}{z_0}\right)}{ln\left(\frac{h_2}{z_0}\right)} \tag{5}$$

where v is the velocity and h is the height at points 1 and 2, and z_0 is the roughness length at the location.

In the present study, the height of the wind tower is $44.075\ m$ and the roughness length used is 0.05, as described by Ferreira *et al.* (2010). Hence, utilizing Eq. (4) for a probability of 0.9, we obtain a wind velocity of Ozawa's tower. Substituting this value into Eq. (5), we derive a velocity of $6.78\ m/s$, as illustrated in Figure 4.

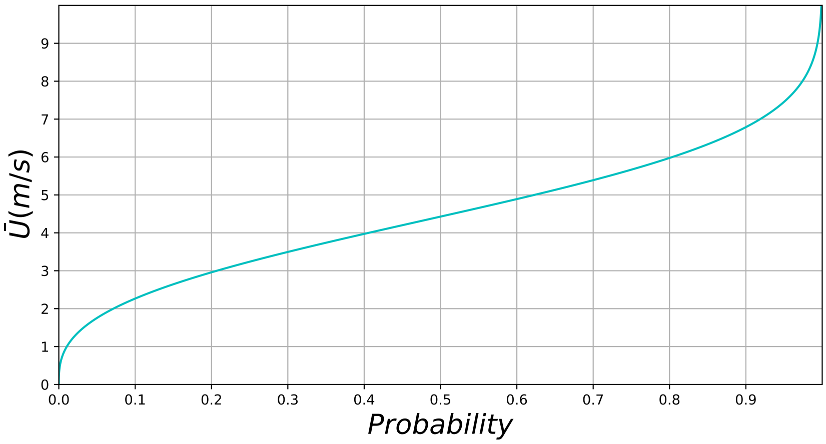


Figure 4: Probability density for the studied tower

Finally, it is possible to use the wind velocity to generate the wind force based on the Kaimal power spectral density (PSD), as described by equation 6:

$$S_{xx}(\omega) = \frac{4\sigma_u^2 L_{1u}/\bar{U}}{[1 + 6\omega(L_{1u}/\bar{U})]^{5/3}} \tag{6}$$

where $L_{1u} = 171.7$, as specified by the International Electrotechnical Commission (Burton *et al.* (2011)). And the generated PSD is shown in fig. 5

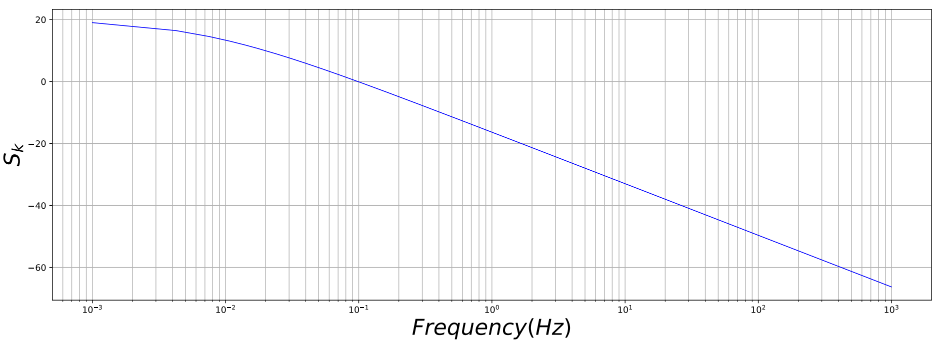


Figure 5: Kaimal PSD

3. RESULTS AND DISCUSSION

In this section the dynamical response of the wind turbine tower under wind forces excitation is evaluated with and without control. The tower is discretized with 30 nodes, where each node has one degree of rotational freedom and one degree of translational freedom, resulting in 60x60 stiffness and inertia matrices. The parameters used in numerical simulations are presented in Table 1:

Table 1: Tower parameters.

Parameter	Value
Elasticity modulus of the tower, E	2.1×10^{11} Pa
Tower diameter, d	2.715 m
Tower height, L	44.075 m
Nacelle's mass, M_n	95000 kg
Density of the tower, ρ	7850 kg/m^3

The damping matrix [C] is defined proportionally to the stiffness matrix [K]. In this study, a negligible damping was considered as follows: $[C] = 10^{-12}[K]$.

As the wind model has a stochastic nature, the dynamical response of the system is evaluated by its spectral density function, S_{yy}, that related with the spectral density function of the excitation, S_{xx}, as follows:

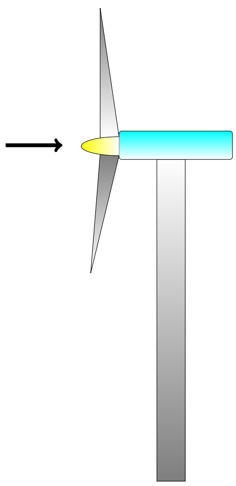
$$S_{yy}(\omega) = |H(\omega)|^2 S_{xx}(\omega), \tag{7}$$

where $S_{xx}(\omega) = F(\omega)$ and the transfer function is

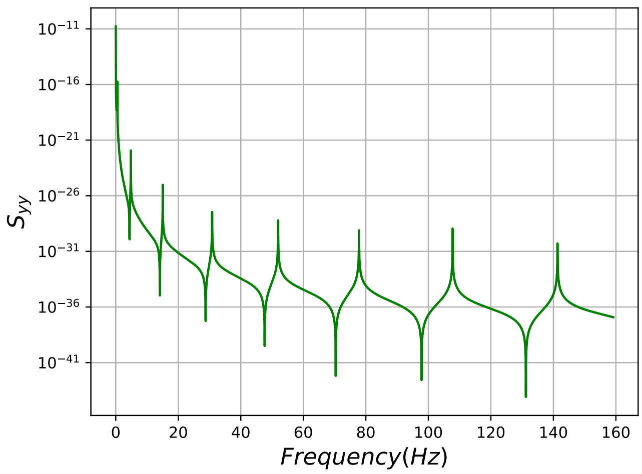
$$H(\omega) = [-\omega^2 M + j\omega C + K]^{-1}. \tag{8}$$

3.1 Response to wind excitation without control

The wind force is assumed to be concentrated at the nacelle, thus, located at the free end of the tower, as represented by Figure 6a. The frequency response is presented in Figure 6b. The response correspond to the translational vibration of the top node, which is also the location where the excitation is applied. The first eight natural frequencies are observed, with the first one being very low at $\omega_1 = 0.463$ Hz.



(a)



(b)

Figure 6: a) Schematic representation of the turbine with concentrated force action due to wind (Araújo, 2021); b) PSD response to the concentrated force applied at the top of the tower with an average wind velocity of 6.78 m/s

3.2 Vibration attenuation

To evaluate the performance of the metamaterials in reducing the vibration amplitudes, two frequencies are considered as target frequencies, one at a time, being the first and fifth natural frequencies of the turbine:

- $\omega_{n1} = 0.463\ Hz$
- $\omega_{n5} = 51.93\ Hz$

Initially, only one dynamic vibration absorber (DVA), also called TMD, is taken into account, followed by the consideration of metamaterials. For both cases, a mass equal to 5% of the total mass of the tower is defined for the 5th mode, and 15% for the 1st mode.

3.2.1 System response with dynamic vibration absorber

The dynamic vibration absorber (DVA) is modeled as a mass-spring oscillator. To target the first natural frequency, the DVA has a natural frequency of 0.463 Hz and a mass equivalent to 15% of the total tower mass, amounting to 14,250 kg. Conversely, when targeting the fifth mode, the DVA's natural frequency is set to 51.93 Hz, with a mass equal to 5% of the total tower mass, equivalent to 4,750 kg. This higher mass used at the frequency of 0.493 Hz was employed to highlight the effects of the absorber, which are less significant at low frequencies.

The stiffness of the dynamic vibration absorber is then defined as follows:

$$K_{ab} = \omega_{ab}^2 M_{ab} \tag{9}$$

where ω_{ab} corresponds to each one of the target frequencies, i.e., 1st and 5th natural frequencies.

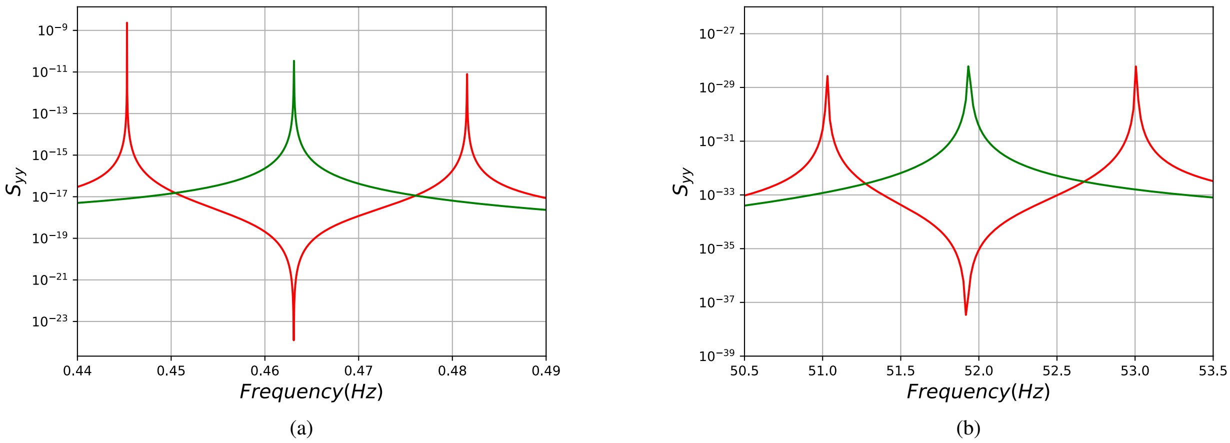


Figure 7: Response with (in red) and without (in green) the presence of the concentrated mass absorber: a) 1st Mode, and b) 5th Mode

So, figure 7 shows the response of the top node without (in green) and with (in red) DVA, focusing solely the region of the target frequency. The figure 7a represents the case of the first target frequency, while Figure 7b correspond to the fifth. Results show the classical performance of DVAs, exhibiting a reduction in vibration amplitudes within a narrow range around the target frequency.

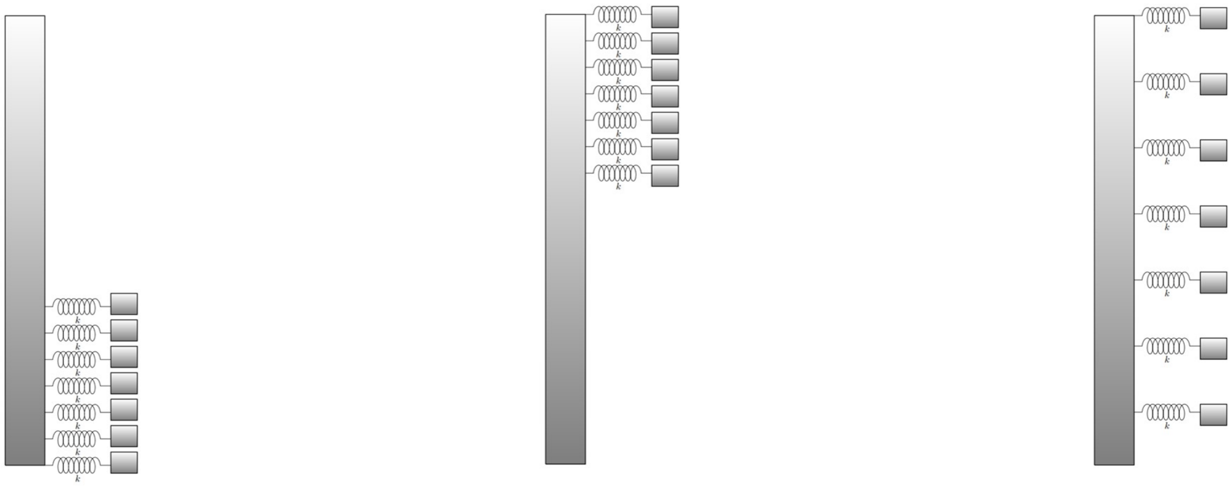
3.2.2 System response with metamaterials

At this point, metamaterials are employed to reduce the vibrations of the tower-nacelle assembly. Three different configuration of the resonators are considered: at the base, distributed along the tower and at the top. In the first case, the first resonator is positioned at the bottom of the structure, followed by the subsequent resonators in the subsequent nodes. The distributed case involves a periodic arrangement of the resonators along the tower. In the top case, the first resonator is placed at the top of the structure, followed by the subsequent resonators in the subsequent nodes. The distribution of the resonators is represented in figure 8.

In all analyzed cases, the presented responses correspond to the top node, corresponding to the nacelle vibration.

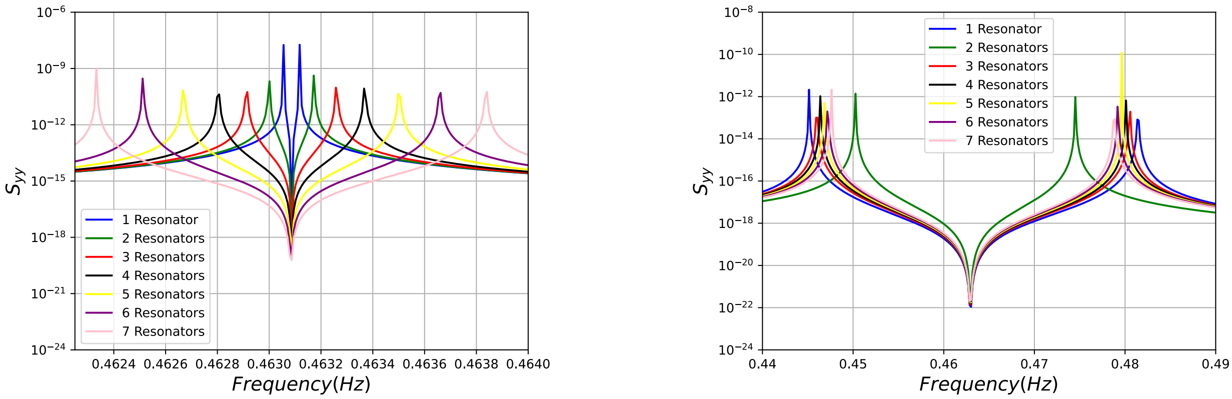
Initially, the first natural frequency of the structure is considered as the target frequency. Different numbers of resonators are taken into account, from 1 (that correspond to an ADV) to 7. That way, figure 9 shows system response solely in the region of the target frequency. It is important to emphasize that in each case, the combined mass of all resonators correspond to 15% of the total structure mass. The case with resonators at the base, as shown in Figure 9a, correspond to the worst performance of the metamaterials, which is expected due to the structure's smaller vibration amplitudes at the bottom. In the distributed and the top cases, it is evident that the configuration with only one resonator exhibits the best performance, and there is no significant difference observed when using 6 or 7 resonators. Further increasing the number of resonators beyond 7 does not yield any noticeable difference compared to the 7-resonator case.

Now, the fifth natural frequency of the structure is the target one. The same analysis conducted for the first frequency is carried out again, as shown in Figure 10. Notably, for a higher frequency, the use of metamaterials proves to be more effective than the ADV. The results showcase the classical outcomes typically obtained in literature when metamaterials are employed for vibration reduction, with a broader attenuation band observed when the number of resonators is increased.

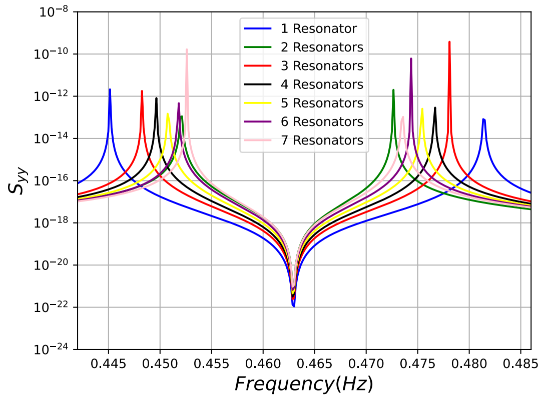


(a) (b) (c)

Figure 8: Simplified model of resonators distribution: a) At the base, b) at the top, and c) distributed along the tower.



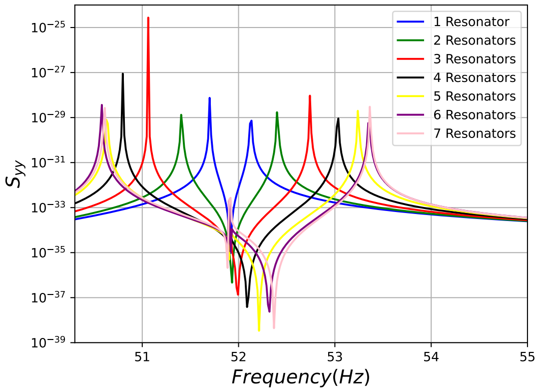
(a) (b)



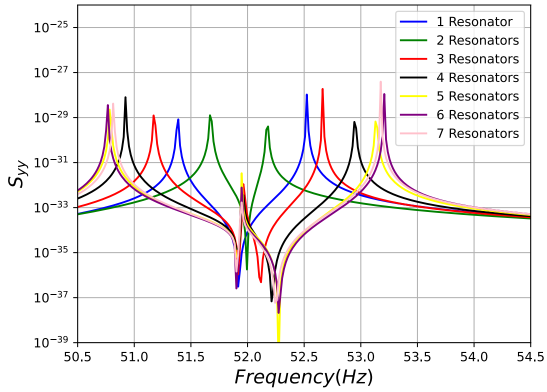
(c)

Figure 9: PSD response of the structure when resonators are tuned to the first frequency and positioned: a) At the base, b) at the top, and c) distributed along the tower.

For a comprehensive comparison of the results, Figure 11 illustrates the structural responses under different control scenarios: without control (in green), with ADV (in red), and with 7 resonators at the base (in purple), at the top (in blue), and distributed (in yellow). For the lower target frequency, as shown in Figure 11a, the ADV demonstrates superior performance in vibration attenuation. However, the case with metamaterials at the top proves to be an interesting alternative,



(a)



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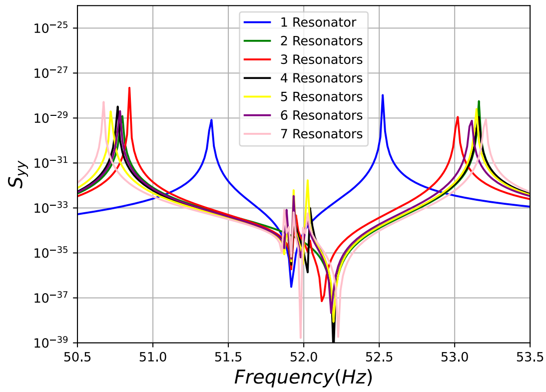
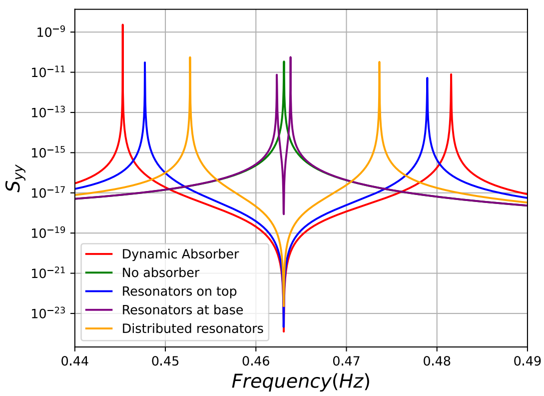


(c)

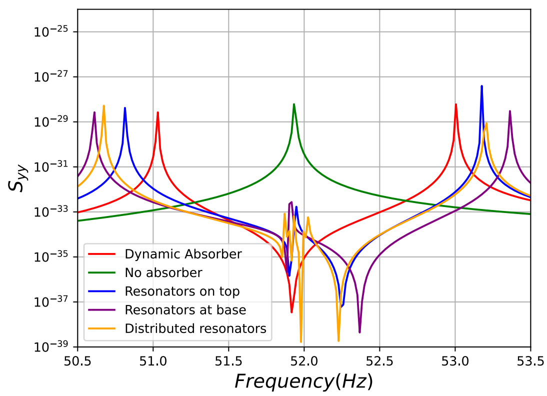
Figure 10: PSD response of the structure when resonators are tuned to the fifth frequency and positioned: a) At the base, b) at the top, and c) distributed along the tower.

as it exhibits a performance close to the ADV and may be more feasible to implement.

For the fifth frequency case, due to it being a higher frequency, the metamaterial demonstrates superior performance when compared to the case with ADV. In this scenario, the resonators at the base exhibited the best performance. The optimal positioning depends on the modal shape of the structure.



(a)



(b)

Figure 11: Comparison between PDS responses without and with control when resonators are tuned to the: a) 1st Mode and b) 5th Mode.

4. CONCLUSION

This paper seeks to add a little more to the theory of these intriguing and innovative materials, metamaterials. Moreover presenting a possible solution for passive vibration control of wind towers.

Based on above, When considering the metamaterials, three different configurations were considered concerning the resonators positioning: at the base, distributed along the tower and at the top. Similar to the ADV case, when the targeting the first natural frequency, the total of the resonators is equivalent to 15% of the total tower mass, while when targeting the fifth mode, the total mass is equal to 5% of the total tower mass.

For the fifth mode, a higher frequency, it is evident that the use of resonators yielded significantly better results than the DVA, especially when distributed along the base of the tower. However, at the lower frequency, the single ressonator (ADV) was slightly more effective. Nevertheless, practical implementation of this DVA might be unfeasible due to its large mass, whereas in the second case, this mass is distributed among several resonators. Thus, for the 1st mode, despite the more modest outcome, the implementation of resonators may be an interesting alternative, as it exhibits a performance close to the ADV and may be more feasible to implement.

Therefore this analysis provides important results for the optimization and selection of vibration attenuation systems in wind towers, considering different vibration modes and practical effectiveness in implementing control devices. Additionally, the study could be further explored for the case of low frequencies to achieve better results. This could be achieved by adding significant damping or exploring different resonator configurations.

5. ACKNOWLEDGEMENTS

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