# VIBRATION CONTROL OF THE SET TOWER AND WIND TURBINE UNDER THE WIND INFLUENCE

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***Abstract.*** *Wind power generation is based on the atmospheric air flow, and it is a renewable energy source, clean and of low environmental impact. The wind kinetic energy is converted into electrical power by a set of blades and a rotor. The rotor is connected though a transmission system to a power generator. The whole chain blade/rotor/transmission/generator is named wind turbine. The conventional wind turbines, horizontal type, are usually mounted on the top of a tower, and due to its geometry and great hight the tower structure is flexible and exposed to excessive vibration caused by the turbine working cycle and the blade-wind interaction. The detailed analysis of the tower structure behavior is of great importance because its cost, which is about 20% of the total system cost. In this work a study of the dynamic stability of the system tower-turbine is done, where the fluid-structure coupling analysis employs a simplified model, a turbine without blades. For this study mainly the first vibration modes is considered, and the structure is excited by a turbulent wind profile up to a hight of 50m from the base of the tower. It is also designed a passive control device to reduce excessive vibrations of the tower structure. A TMD with pendulum absorber geometry is considered.*

***Keywords:*** *Structural Control, Tuned mass damper, Wind Engineering; Reduced Order; Dynamic System; Wind Stochastic Modelling.*

## 1. INTRODUCTION

The current energy crisis coupled to the depletion of fossil fuel stocks and the need to reduce emissions of carbonic gas, preserving the environment makes wind power generation a viable and attractive mean of producing electricity.

The wind power generation has its origin in the movement of atmospheric air masses. The energy is extracted from the wind by the conversion of the kinetic energy into electrical energy by a set of blades attached to a rotor. The rotor moves a transmission system linked to an electrical converter. The mechanical set blade / rotor / transmission / converter, used for electricity generation, is named wind turbine. The wind power generation is a source of renewable energy that is clean, with low environmental impact and available in many places around the world. The use of this energy source for electricity generation in commercial scale was made possible, mainly, by the world wide oil crises in the 70s. At this time, Europe and the USA developed this technology to decrease their dependence in oil and coal (Mukund, 1999).

The set wind turbine / blade is fixed and sustained by tower that due to its geometry and great height can present excessive vibrations caused by the wind turbine working and also by wind forces. A detailed analysis of the structural behavior of the tower is of great importance because of its cost that can represent approximately 20% of the total system cost.

A solution to the problem of excessive vibrations studied by many researchers in the last years is the structural control. It provides structural properties changes, enhancing damping, stiffnes and strength, by installing external devices or applying external forces. Some researchers have been studying the use of structural control to help suppress the wind-induced vibrations experienced by wind turbine towers (Murtagh et al, 2008; Colwell and Basu, 2009).

One of the earliest structural control devices, already extensively studied (Den Hartog, 1956; Soong & Dargush, 1997) is the Tuned Mass Damper (TMD). The TMD is designed as a mass-spring-dashpot device which is tuned to a specific structure natural frequency in order to transfer the vibration energy from the main system to the auxiliary mass that vibrates out of phase.

An alternative geometry of the TMD is the pendulum vibration absorber, this device has its natural period depending on the length of its cable, and only can be considered as a linear oscillator when the vibration amplitudes are small (Orlando & Gonçalves, 2005; Avila et al, 2006, Gerges & Vickery, 2005, Zuluaga et al, 2008). Pendulum vibration absorbers have been installed in high-rise buildings, bridges and other civil structures to attenuate wind-excited vibrations (Fischer, 2007; Korenev & Reznicov, 1993). Compared to a conventional vibration absorber made of movable mass and flexible member, a simple pendulum is more rugged, easily constructed, and suitable for heavy-duty jobs.

In this work, a study of the dynamic behavior of the system tower-turbine is done, where the fluid-structure coupling analysis employs a simplified model, a turbine without blades. For this study only the first vibration mode is considered, and the structure is excited by a turbulent wind profile up to a height of 60m from the base of the tower. The wind speed is modeled by the combination of the Van der Hoven's model, which is considered as one of the best known references in large-band wind speed modeling, and the von Karman's model, also a well known reference for the turbulence component (Morais et al, 2009). The equations of motion are obtained for a structure modeled as a single degree of freedom and coupled to a pendulum tuned mass damper (TMD) when is subjected to random dynamic excitations as white noise and wind forces. A pendulum parametric study is carried out to improve the control system performance reducing satisfactorily the *rms* structural response.

## 2. MODAL ANALYSIS

The objective of the modal analysis is to reduce the complex system of partial differential equations that describe the dynamical behaviour of a continuous structure. This approach is simpler and is described by a system of ordinary differential equations that considers the motion of an equivalent one-dimensional structure. A theoretical treatment of modal analysis is given in Meirovitch (1967). Examples of modal analysis are given for vortex induced vibration in slender beam and others examples, (Blevins, 1986).

The motion of a cantilever Bernoulli-Euler beam is described as follows:

$$\frac{\partial^2}{\partial z^2}\left(EI\frac{\partial^2 w(z,t)}{\partial z^2}\right)+m\frac{\partial^2 w(z,t)}{\partial t^2}=F(z,t)\text{, with } I=\int_S \xi^2\mu\, dS \qquad (1)$$

where $w(z,t)$ is the displacement normal to the beam axis in some direction, $z$ is the distance along the beam axis; $m$ is the mass per unit length of the beam; $F$ is the external force per unit length applied normal to the beam axis in the direction of $w$; $I$ is the area inertia moment for bending; $S$ is the cross-sectional area; $\xi$ is the distance from the shear centre of the beam in the $w$ direction and $\mu$ is the mass per unit volume of the section. In general, the modulus of the inertia moment and the mass per unit length vary along the span of the beam. However, in this example they are considered to be uniform over the span. For a cantilever beam with a tip mass, the appropriate boundary conditions are:

$$w(0,t)=\partial_z w(0,t)=0 \quad \text{and} \quad EI\partial_z^2 w(0,t)=\omega_n^2\Phi'(L)J_M \quad EI\partial_z^2 w(L,t)=\omega_n^2\Phi(L)M \qquad (2)$$

where $M$ and $J_M$ are respectively the mass and corresponding rotatory inertia mass at the free end.

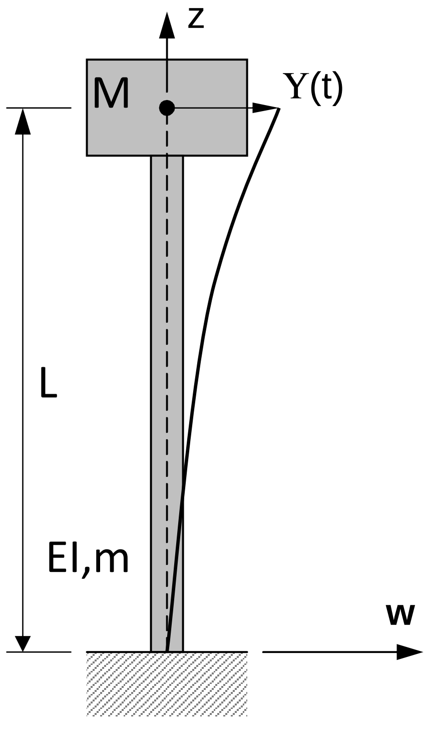


Figure 1 – Schematic description of a cantilever beam with a tip mass.

Applying the Fourier method into the associated equation for free vibration ($EI\partial_z^2 w + m\partial_t^2 w = 0$), we carry out the solution of Eq. (1) in terms of a separated space-time function:

$$w(z,t) = \Phi(z)\Upsilon(t) \tag{3}$$

Finally, it is obtained:

$$\frac{1}{\Phi}\frac{\partial^4\Phi}{\partial z^4} = -\frac{m}{EI}\frac{1}{\Upsilon}\frac{\partial^2\Upsilon}{\partial t^2} = const. \tag{4}$$

The solution of Eq. (4) is the following,

$$\begin{aligned} \Phi(z) &= C_1\sin(\alpha z) + C_3\cos(\alpha z) + C_3\sinh(\alpha z) + C_4\cosh(\alpha z) \\ \Upsilon(t) &= A\sin(\omega_n t) + B\cos(\omega_n t) \quad n = 1,2,3,\dots \end{aligned} \tag{5}$$

where $\omega_n = \left(n^2\pi^2/L^2\right)\sqrt{EI/m}$ is the natural frequency of vibration, and $\alpha_n = \omega_n^2 m/EI$ . The boundary conditions are used to determine the constants $A$ , $B$ and $C_i$ , $i = 1,2,3,\dots$ . Thus the spatial end-conditions of the cantilever beam with tip mass, Eq. (2), are obtained by expressing one of the fixed end constants as a function of the others. Substituting into Eq. (5) (Murtagh et al, 2004):

$$\frac{\Phi_n(z)}{C_1} = \sin(\alpha_n z) - \sinh(\alpha_n z) + \left[\frac{\sin(\alpha_n z) + \sinh(\alpha_n z)}{\cos(\alpha_n z) + \cosh(\alpha_n z)}\right]\left[\cosh(\alpha_n z) - \cos(\alpha_n z)\right] \tag{6}$$

As described by Méthod and Ilinca (2007), because of the non-homogeneity of the boundary conditions, the principle of orthogonality ($\int_0^L \Phi_n \Phi_n \, dz = 0, i \neq j$) between mode shapes is not respected. However, It can be overcome by Gram-Schmidt orthogonality process using the fundamental mode shape $\Phi_1(z)$, as first vector, to orthogonalize the other mode shapes. Paz (1997) provides an approximated solution using the principle of virtual works and the Rayleigh's method. To determine the equivalent one-degree-of-freedom for a dynamic system with distributed mass and stiffness, Figure 1, it can be assumed that the fundamental mode shape is described as follow:

$$w(z,t) = \Upsilon(t)\Phi(z) = \Upsilon(t)\left[1 - \cos\left(\frac{\pi z}{2L}\right)\right] \tag{7}$$

Then, if this expression is substituted into Eq. (1),

$$M^*\ddot{\Upsilon} + K^*\Upsilon = F \tag{8}$$

The generalized mass and the generalized stiffness of the tower are computed, respectively as:

$$K^* = \int_0^L EI\left[\Phi''(z)\right]^2 dz = \int_0^L EI\frac{\pi^4}{16L^4}\cos^2\left(\frac{\pi z}{2L}\right)dz \quad \therefore \quad K^* = \frac{\pi^4}{32L^3}EI \tag{9}$$

and

$$M^* = M + \int_0^L \left[1 - \cos\left(\frac{\pi z}{2L}\right)\right]^2 dz = M + \frac{mL}{2\pi}(3\pi - 8) \quad \therefore \quad M^* = \frac{mL}{2\pi}\left[\pi\left(3 + 2\frac{L_e}{L}\right) - 8\right] \tag{10}$$

where the tip mass $M = mL_e$ is defined proportional as an equivalent length $L_e$ .

The equivalent force for the distributed mass forced by the atmospheric wind force is computed as:

$$
\begin{aligned}
\frac{F^*}{C_D A} &= \int_0^L F(t,z)\Phi(z)dz = \int_0^{60}\left[\tfrac{\rho_a}{2}V_{60}^2(t)\cdot S_1^2 S_2^2\left(\tfrac{z}{z_o}\right)\right]\cdot\left(1-\cos\frac{\pi z}{2L}\right)dz \\
&= \frac{\rho_a}{2}\int_0^{60} V_{60}^2(t)\cdot S_1^2\cdot\left(\frac{z}{z_o}\right)^{0.34}\cdot\left(1-\cos\frac{\pi z}{2L}\right)dz \qquad (11)\\
\therefore &= 19.185\frac{\rho_a}{2}V_{60}^2(t)
\end{aligned}
$$

## 3. WIND MODEL

The wind speed is modeled by combining two power spectra models, Van der Hoven's and von Karman's models. These models accurately simulate the whole wind speed band considering short, medium and long-term frequency components. Van der Hoven (1957) collected a large amount of experimental data measuring the wind speed over different time periods at a height about 100m. Van der Hoven power spectrum contains data that approximates all frequency band, ranging from 0.0007 to 900 cycles/hour (Van der Hoven, 1957; Nichita et al, 2002; Morais et al, 2009). However, Van der Hoven's model treated the turbulence component as a stationary random process, and consequently the model can only be applied to describe the wind speed over a time scale of hours, days, etc. Therefore, it is necessary to use von Karman's model to simulate the turbulence component as a non-stationary process, (Nichita et al, 2002; Morais et al, 2009). By using Van der Hoven's power spectrum ( $S_{vv}^{VH}$ ) the wind speed signal is reconstructed with the following relations.

$$A_i = \frac{2}{\pi}\sqrt{\frac{1}{2}\left[S_{vv}^{VH}(\omega_i) + S_{vv}^{VH}(\omega_{i+1})\right]\left[\omega_{i+1} - \omega_i\right]} \qquad (12)$$

and

$$v(t) = \sum_{i=0}^{n} A_i \cos(\omega_i t + \varphi_i) \qquad (13)$$

where $A_i$ is the amplitude at the harmonic frequency $\omega_i$ and $S_{vv}^{VH}(\omega_i)$ is the corresponding value of the power spectrum density. The phase angle is randomly generated with a uniform distribution in the domain $[-\pi, \pi]$. At $i = 0$, $\omega_0 = 0$, $\varphi_0 = 0$ and $A_0 = \bar{v}$, where $\bar{v}$ is the mean speed, computed at a time interval greater than the largest period in Van der Hoven's power spectrum density ( $T = 2\pi/\omega_1$ ). For accurately capturing the medium- and long-term wind speed components, the discrete frequency values $f_0 = 0,\ f_1 = 0{,}001\ldots,\ f_{30} = 3$ cycles/hour are extracted from Van der Hoven's power spectrum. Thus, the wind speed is computed as:

$$v(t) = v_{ml}(t) + v_t(t) \qquad (14)$$

where $v_{ml}(t)$ is the medium- and long-term component calculated by

$$v_{ml}(t) = \sum_{i=0}^{30} A_i \cos(\omega_i t + \varphi_i) \qquad (15)$$

with $n = 30$ and $v_t(t)$ is the turbulence component computed by von Karman's power spectrum ( $S_{vv}^{VK}$ ).

$$S_{vv}^{VK}(\omega) = \frac{0.475\sigma^2 \frac{L}{\bar{v}}}{\left[1 + \left(\frac{\omega L}{\bar{v}}\right)^2\right]^{5/6}} \qquad (16)$$

where $\sigma$ and $L$ are the turbulence intensity and length respectively, also $S_{vv}^{VK}$ depends on the value of the mean speed $\bar{v}$. The implementation of the non-stationary turbulence component is done by applying a shape filter with a white noise input, whose transfer function is:

$$H_F(j\omega) = \frac{K_F}{(1 + j\omega T_F)^{5/6}} \tag{17}$$

The static gain is computed having the variance of the colored ($y_c(t)$) and white ($y_w(t)$) noises equal to 1, (Welfonder, 1997), the following expression results from this condition:

$$K_F \approx \sqrt{\frac{2\pi T_F}{B\left(\frac{1}{2}, \frac{1}{3}\right) T_S}} \tag{18}$$

where

$$T_F = \frac{L}{\bar{v}} \tag{19}$$

and $B$ designates the beta function and $T_S$ is the sampling period for the turbulence component computation:

$$v_t(t) = \hat{\sigma}_v \, y_c(t) \tag{20}$$

with $\hat{\sigma}_v$ being the estimated value of the standard deviation:

$$\hat{\sigma}_v = k_{\sigma,v} \bar{v} \tag{21}$$

where $k_{\sigma,v}$ is determined experimentally as the slope of the curve which approximates the relation between $\bar{v}$ and $\hat{\sigma}_v$, (Welfonder at al, 1997). The wind speed model can still be adapted to different sites by modifying the parameters $k_{\sigma,v}$ and $L$. For simplification, the whole process of the wind speed modelling, previously described, is summarized in a computation procedure, (Morais, 2009).

## 4. PENDULUM ABSORBER

It is possible to absorb the vibrations of a structure by the motion of a pendulum, suspending a mass with the required damping. The absorber mass may have the form of a massive body of small dimensions, of a plate or a ring, according to the space available, (Fischer, 2007).

In Figure 2 is shown a schematic diagram of a pendulum tuned mass damper attach to a main system constituting a mixed rotational and translational two-degree of freedom model, being the main system idealized as a single degree of freedom for the vibration mode to be controlled, (Soong & Dargush, 1997).

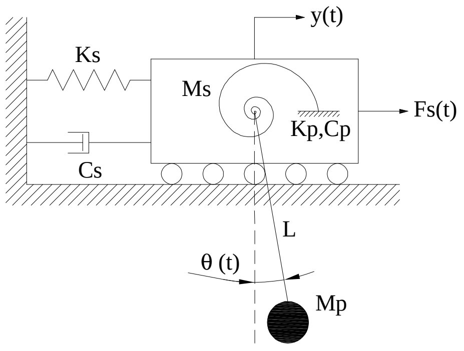


Figure 2 – Mixed two-degree of freedom system subjected to force excitation $\mathbf{F_s(t)}$

The governing equations if the pendulum rotation is small, are given by:

$$\left(M_s + M_p\right)\ddot{y} + M_p L\ddot{\theta} + C_s\dot{y} + K_s y = F_s(t) \tag{22}$$

$$M_p L\ddot{y} + M_p L^2\ddot{\theta} + C_p\dot{\theta} + \left(K_p + M_p g L\right)\theta = 0 \tag{23}$$

where *y(t)* is the displacement of the main system and θ(t) is the rotation of the pendulum. The main system is modeled as a single degree of freedom system with $M_S$, $C_S$, $K_S$ and $F_S(t)$ being the generalized mass, damping, stiffness and force, respectively. The pendulum is represented for a mass $M_P$, with damping $C_P$, stiffness $K_P$ and cable length $L$.

Zuluaga (2007) conducted an optimization study considering a structure subjected to a white noise random excitation given by power spectral density functions. The following expressions were obtained for the optimal length $L_{opt}$ and the optimal damping ratio $\xi_{opt}$ of the pendulum.

$$L_{opt} = \frac{\left(2g(\mu+1) + 2\sqrt{(\mu g + g)^2 + 2\omega_a \omega_S{}^2(\mu+2)}\right)(\mu+1)}{2\omega_S^2(\mu+2)} \tag{24}$$

$$\xi_{opt} = \frac{\sqrt{\mu(\mu+2)(3\mu+4)(\mu+1)\left((\mu g + g)^2 + \omega_a \omega_S{}^2(\mu+2) + g(\mu+1)\sqrt{(\mu g + g)^2 + 2\omega_a \omega_S^2(\mu+2)}\right)}}{2\omega_S^2(\mu+2)^2} \tag{25}$$

where μ is the mass ratio; $\omega_S$ is the structure natural frequency; $\omega_P$ is the pendulum natural frequency and $\omega_a$ is the ratio between pendulum mass and stiffness $\omega_a = K_P / M_P$ .

## 5. NUMERICAL EXAMPLE

A simple model is considered for tower representation with realistic dimensions to illustrate this work. The proposed model consists of a wind turbine tower connected to a nacelle, as presented in Figure 1. The system is modelled as a uniform cantilever beam of circular hollow cross-section with a tip mass (the nacelle) at its free end. The tower is constructed from steel with a hub height of 60m, a width of 3m and a shell thickness of 0.015m. The elastic modulus and density of the steel are assumed to be 2.1×1011N/m$^2$ and 7850 kg/m3, respectively. The tower carries a nacelle and a rotor system mass of 19.876kg. The drag coefficient used for the tower was 1.2, with the density of air of 1.25kg/m3. We assume none modal damping ratios. Equations (9), (10) and (11), using the precedent data, produces the following generalized rigidity, mass and forces parameters:

$$M^*\ddot{Y} + K^*Y = F^*(t) \text{, where } M^* \cong 34{,}899kg \text{ and } K^* \cong 46{,}3671 N/m \tag{26}$$

The estimated frequency is $\omega_{reduced} \cong 0.58$ Hz, close to the literature result $\omega_{reduced} \cong 0.567$ Hz, Murtagh (2004). The relative error, inferior to 2.5%, justifies the simplified Rayleigh model.

The wind speed modelling is a composition of the Van der Hoven's and the von Karman's power spectra (Morais et al, 2009). The wind is simulated as a coherent temporal signal. For every test case performed, the wind speed model uses the following parameters $L = 180m$ , $k_{\sigma,v} = 0.16$ , $T_{s1} = 0.5$ s, $T_s = 0.5$ s, $\Delta\omega = 0.002$ rad/s, $M = 5000$ and $N = 100$. Figure 3 presents the wind temporal evolution speed along five hours (18,000s). The superior cut-off frequency is 1Hz according to Nyquist criteria. Figure 3 shows medium- and long-term wind speed components $v_{ml}(t)$ (thick red line), and the total wind speed $v(t)$ (thin blue line) as represented in Eq. (14). The turbulence component $v_t(t)$ is also important with respect to the medium- and long-term components $v_{ml}(t)$ .

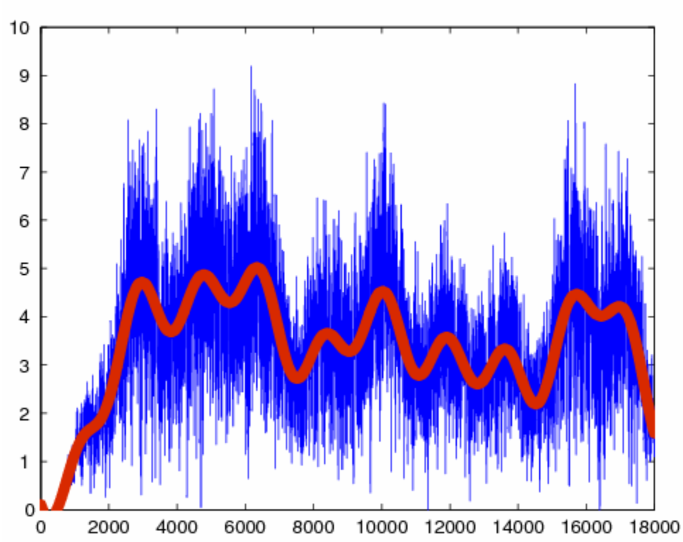


Figure 3 – Characteristic wind velocity acting on tower, $t$ [s] x $v$ [m/s].

First of all, it is simulated the behaviour of the wind tower under a white noise wind effect. It was considered a time history of one hour of duration and amplitude pic-to-pic of 1000 N (*F = [ t = 1:1:3600 ; 1000*rand(size(t)) ]*). It was used the optimum values of the pendulum length L, mass ratio μ and stiffness ratio $K_P/K_S$ proposed by Zuluaga (2007): $\mu$ = 0.08, L =1.75m, $K_P/K_S$ = 10%. Two simulations were conducted with this random excitation, with and without TMD pendulum control, Table 1. The simulation with control presents a reduction of almost 75% of pic-to-pic and effective *rms* tip displacement comparing to the case without control.

Table 1. Effective *rms* tip displacement (cm) response to a white noise temporal excitation.

| Tip displacement (cm) | $y_{pic\text{-}to\text{-}pic}$ | $y_{RMS}$ |
|---|---|---|
| without control | 6.0 | 1.9 |
| with control | 1.5 | 0.5 |

However, using Zuluaga's values to control the dynamic response of the tower subjected to wind loading, worse results were obtained. Therefore, a parametric study was carried out to find the better results considering this wind excitation. Figure 4 presents the effective *rms* tip displacement varying frequency ratio $\alpha$ for three different values of mass ratio $\mu$ = [0.06, 0.08, 0.10]. The results are summarized in Table 2. As highlighted by Zuluaga (2007), we choose low values of mass ratio to avoid non-linear effects, e.g., divergent instability. The most efficient parameters are $\alpha$ = 1.15, $\mu$ = 0.06, reducing 32,5% *rms* tip displacement without control. Figure 5 presents displacement time evolution with and without control, considering these parameters.

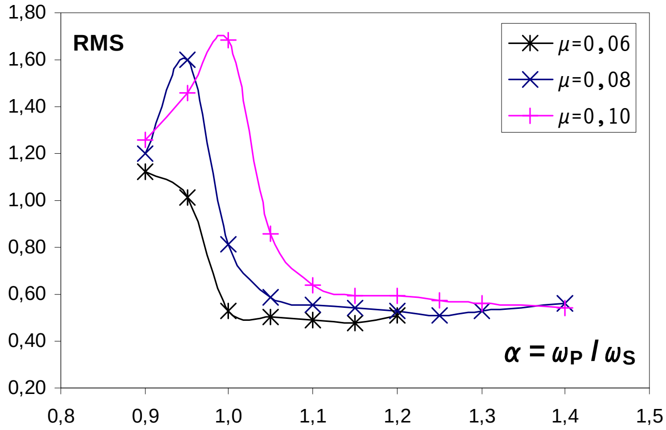


Figure 4. Effective RMS tip displacement (cm) response to a wind action presented at Figure 3, as function of the ratio frequency $\alpha$ ($\omega_p$ / $\omega_S$).

Table 2. Pendulum length and effective *rms* tip displacement $y_{RMS}$ (cm) as function of frequency ratio $\alpha$ ($\omega_p$ / $\omega_S$) for response a wind action, Figure 3.

| $\mu$ = 0.06 | | | $\mu$ = 0.08 | | | $\mu$ = 0.10 | | |
|---|---|---|---|---|---|---|---|---|
| $\alpha$ | L (m) | $y_{RMS}$ | $\alpha$ | L (m) | $y_{RMS}$ | $\alpha$ | L (m) | $y_{RMS}$ |
| 0.90 | 1.96 | 1.1220 | 0.90 | 1.78 | 1.2030 | 0.90 | 1.66 | 1.2566 |
| 0.95 | 1.83 | 1.0112 | 0.95 | 1.66 | 1.6031 | 0.95 | 1.54 | 1.4557 |
| 1.00 | 1.71 | 0.5291 | 1.00 | 1.55 | 0.8131 | 1.00 | 1.44 | 1.6847 |
| 1.05 | 1.61 | 0.5060 | 1.05 | 1.45 | 0.5898 | 1.05 | 1.34 | 0.8586 |
| 1.10 | 1.52 | 0.4889 | 1.10 | 1.37 | 0.5521 | 1.10 | 1.26 | 0.6418 |
| 1.15 | 1.44 | 0.4764 | 1.15 | 1.29 | 0.5404 | 1.15 | 1.19 | 0.5910 |
| 1.20 | 1.36 | 0.5128 | 1.20 | 1.22 | 0.5286 | 1.20 | 1.13 | 0.5922 |
| 1.25 | | | 1.25 | 1.16 | 0.5093 | 1.25 | 1.07 | 0.5769 |
| 1.30 | | | 1.30 | 1.11 | 0.5284 | 1.30 | 1.02 | 0.5638 |
| 1.40 | | | 1.40 | 1.01 | 0.5587 | 1.40 | 0.93 | 0.5412 |

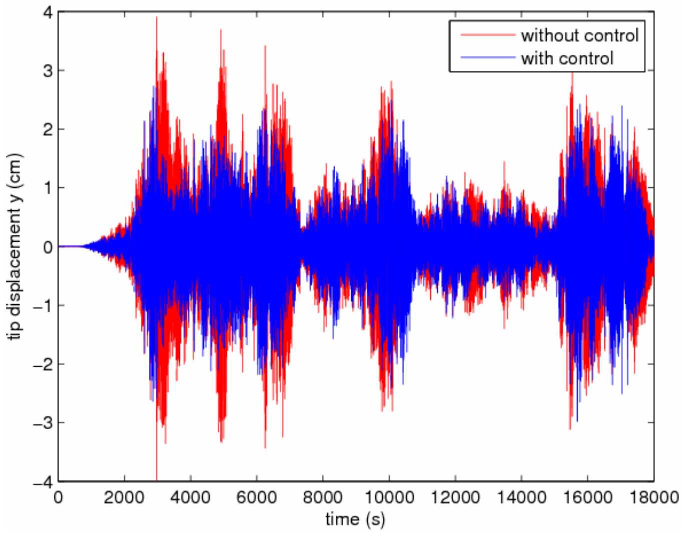


Figure 5. Displacement time evolution y (cm) with and without control to parameter values $\alpha$ = 1.15 and $\mu$ = 0.06.

The performance of the pendulum using Zuluaga's optimum parameters, considering white noise excitation, was more efficient than the one using parametric study results for the wind excitation. If a detailed optimization study is conducted, considering the Van der Hoven's and von Karman's wind excitation, the performance of the passive pendulum absorber can be improved. However, it is more indicated a future research considering semi-active devices that are more robust than passive ones, when the structure is excited by random forces.

## 6. CONCLUSIONS

The wind turbine tower was modelled as a cantilever beam with a tip mass. This continuous model is reduced to a one-degree-of-freedom mass-spring model. The atmospheric wind velocity uses a composition of the Van der Hoven's and von Karman's power spectra to calculate a stochastic wind speed temporal history. A passive structural control device, the tuned mass damper (TMD) is proposed to reduce excessive vibrations. The TMD is designed having pendulum geometry; it is attached to the structure. First it is analyzed the structural response when subjected to a white noise random excitation, the optimum parameters of Zuluaga (2007) for the pendulum are adopted producing good rms response reductions. Subsequently the dynamical analysis considers the Van der Hoven's and von Karman's wind excitation; in this case Zuluaga's parameters do not have good results. So, it is performed, in this case, a parametric study to improve the pendulum efficiency. Passive energy devices performance is very sensitive to changes in the dynamic excitation, because of that it is recommended more research on semi-active control devices, which are more robust, particularly when the structure is subjected to random excitation, like the problem studied here: wind turbine tower subjected to wind forces.

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