Definition of a passive damping system (TMD) for an industrial steel chimney

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Abstract

The objective of this thesis is to study the implementation effects of a TMD in a steel chimney subjected to aerodynamic effects of the wind, namely the vortex shedding. This effect induces cyclic loading that deteriorates the material, creating fatigue cracking.

To accomplish this objective the performance of a steel chimney with 60 m is evaluated with and without the implementation of the damping system.

The quantification of the dynamic response of the steel chimney caused by the action of the vortex shedding, as well as the acceptable elastic tension amplitude, is done by using the Eurocode [1]. In regard of the of the TMD system, Den Hartog's formulas are used to determine the optimal parameters of the tuned mass damper solution. Finally, various solutions with different masses are studied in order to account for the effects of the mistuning.

The results show that tall steel chimneys, similar to the one studied, need the implementation of a damping system of any kind. The results also revealed that a TMD is a viable damping system solution to steel chimneys.

Key words: TMD, vortex shedding, fatigue, Structural Eurocodes, steel chimneys, damping

1. Introduction

The chimney is a structural element that is used to transport the gases and undesirable fumes resulting from activities such as combustion and other chemical reactions to the outside atmosphere [2].

The height of the chimney influences its ability to send gases into the atmosphere through the " stack " effect. This effect is responsible for the air circulation, due to the difference in density of the fluid, inside and outside the chimney. In addition to the improvement of the capacity to lead pollutants to the outside, the height increase of the chimney also reduces the impact of its dispersion in the surrounding areas [2].

Because of their height and slenderness, industrial chimneys are structures susceptible to the cyclical loads imposed by the wind.[3]. This cyclic action is caused by an aerodynamic phenomenon called vortex shedding.

When the flow of a fluid (in this case atmospheric air) travels through a cylindrical surface, for a certain speed of the flow, it can be observed the formation of a sheet of vortex that "form" and "detach". This phenomenon causes regular fluctuations of the aerodynamic forces, and especially of the transverse component to the flow. If the wind conditions cause oscillation with a frequency close to a natural frequency of the structure, the dynamic response of the chimney is amplified and the phenomenon causes vibrations that can have devastating consequences [4].

Figure 1.1 - Vortex shedding on a cylinder [5].

The frequency of the oscillations is related to a parameter that characterizes the flow, the Strouhal number. This parameter depends on the characteristics of the immersed body section, flow velocity, surface roughness and Reynolds number, it presents the following expression [4]:

$$S_t = \frac{D.f}{V} \tag{1.1}$$

Where D is the diameter of the chimney f is the frequency of the oscillations and V the velocity of the wind. If the frequency of a natural vibration mode is equal to the frequency of the vortex shedding oscillations, it occurs a phenomenon denominated by lock-in and the vibrations are amplified. The velocity for which this phenomenon occurs is defined as the critical velocity.

$$v_{cr,i} = \frac{f_i D}{S_t} \tag{1.2}$$

This cyclic loading causes the deterioration of the material due to the fatigue phenomenon. A load cycle is defined as the duration between maximum stress and minimum stress. In elastic materials, a cyclic load causes a response in terms of periodic-cyclic tension. For such cases, the load cycle is easily defined by the figure which is presented as follows [6].

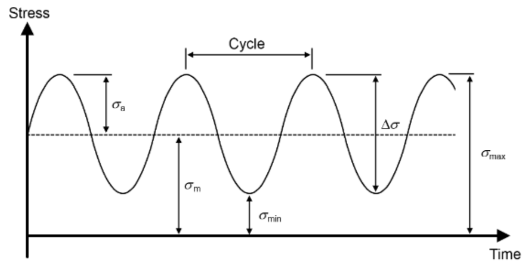

Figure 1.2 - Cyclic response of tension due to a cyclic loading [6].

Every cycle increases the plastic deformation thus decreases the section of the material and consequently reduces the stress resistance. The processed data of the lab testing on different structural details usually is presented as the stress amplitude $\Delta\sigma$ in function of the number of cycles, N, on logarithmic scale. The results originate the $S-N$ curves, as the one given by Eurocode [7] for different structural details categories presented as follows.

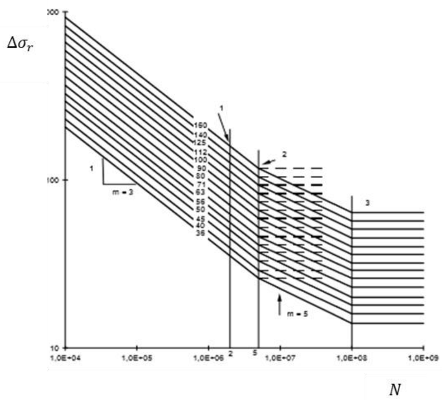

Figure 1.3 - Eurocode's $S-N$ curves [7].

The susceptibility of a chimney to the vortex shedding phenomenon can be measured by a parameter denominated Scruton number.

$$S_C = \frac{2 \times \delta_s \times m_{i,e}}{\rho \times D} \tag{1.3}$$

Where δ_s represents the structural damping, ρ is the air density, $m_{i,e}$ is the equivalent distributed mass and, D is the diameter of the chimney.

The steel chimneys have structural damping in the range of $0{,}02 - 0{,}04$. In order to increase this low value, it can be implemented a damping system to improve the structure's damping capacity. It can be implemented mechanical absorbers, cables with built-in damping and other systems. A solution that can be applied for self-supported steel chimneys is the TMD. A tuned mass damper (TMD) is a device, which is composed of a mass m_t, a spring k_t, and a mechanical damper c_t. It is usually coupled to the top of a structure to reduce its dynamic response [8].

When the structure begins to vibrate the TMD is excited in such a way that it vibrates at the same frequency, however, at a different phase. The kinetic energy of the structure is transmitted to the TMD and dissipated by its mechanical damper. The dynamic behavior of the structure with the TMD can be studied with a mechanical model of two-degrees of freedom.

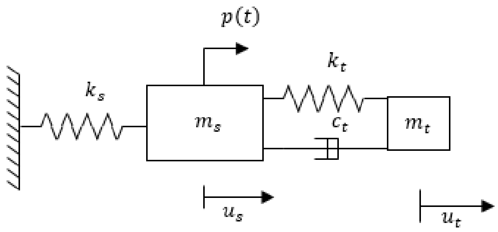

Figure 1.4 - Model of the two-degree of freedom system [9].

Den Hartog studied this model for harmonic loads not taking in account the structural damping. Considering the equilibrium equation for a harmonic load equal to, $p(t) = P_0 \operatorname{sen}(\omega t)$.

$$\begin{cases} m_s\ddot{u}_s + k_s u_s + c_t(\dot{u}_s - \dot{u}_t) + k_T(u_s - u_t) = P_0 \operatorname{sen}(\omega t) \\ \\ m_t\ddot{u}_T + c_t(\dot{u}_t - \dot{u}_s) + k_t(u_t - u_s) = 0 \end{cases} \tag{1.4}$$

If it's considered that, $q = \frac{w_s}{w_t}$, $\mu = \frac{m_t}{ms}$ and $\xi_T = \frac{c_t}{2\omega_t m_t}$, the solution for the equilibrium equation in terms of the structure displacement is equal to[10]:

$$\frac{u_S}{u_{S\,st}} = \sqrt{\frac{(2\xi_T q)^2 + (r_S^2 - q^2)^2}{(2\xi_T r_S q)^2[1 - r_S^2(1+\mu)]^2 + \{r_S^4 - [1 + (1+\mu)q^2]r_S^2 + q^2\}^2}} \tag{1.5}$$

Where, $u_{S\,st} = \frac{P_0}{k_S}$, represents the amplification of the static displacement. In order to optimize the TMD system. Den Hartog decided to analyze the parameters of equation (1.5). He verified that the curves, for different damping values, intersected at two points (P and Q) and could alternate their amplitude, by varying the value of q. By doing this, the amplitude of one point reduced and the other increased. It was also found that the slope of the curve was a function of the TMD damping (ξ_T).

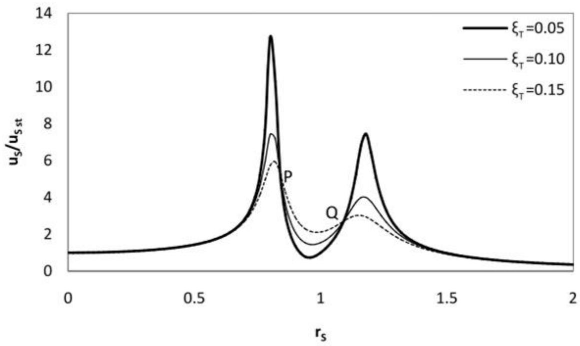

Figure 1.5 - Amplification curves for different values of ξ_T [10]

So, to optimize the system Den Hartog found the optimum values for, q, and. ξ_T, to achieve the TMD's maximum efficiency. The formulas of the optimal parameters are presented as follows:

$$q_{opt} = \frac{1}{1+\mu} \tag{1.6}$$

$$\xi_{opt} = \frac{1}{2}\sqrt{\frac{3\mu/2}{1+3\mu}} \tag{1.7}$$

A TMD tuned with these optimal parameters provide the following additional damping to the structure:

$$\Delta\xi_{eq} = \frac{1}{2}\sqrt{\frac{\mu/2}{1+\mu/2}} \tag{1.8}$$

According to [11], the increase of structural damping expressed in logarithmic decrement can also be calculated using the maximum amplification curve of the displacements, $H_{dmáx} = \text{máx}\left(\frac{u_T}{u_{S\,st}}\right)$

$$H_{dmáx} = \frac{2\pi}{2\delta_{eq}} \tag{1.9}$$

2. Method

2.1. Description of the case of study

For the purpose of studying the application of the TMD, a practical example of a self-supporting steel industrial chimney was analyzed. To evaluate the effects of vortex shedding, a chimney with the dimensions equal to an example examined by Dyrbye and Hansen [37], subjected to a wind of zone A of Portugal, was considered.

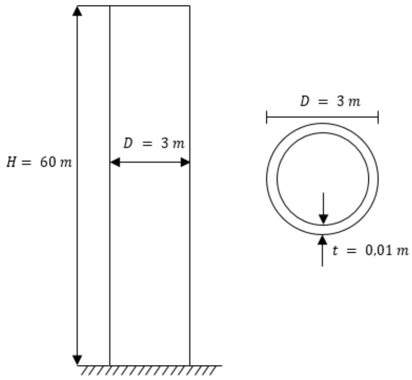

Figure 2.1 - Self-supported steel chimney with 60 meters of height (image not scaled)

The chimney has an inside layer to protect from internal corrosion coupled to the shaft structure with a mass equivalent to 260 kg/m, distributed along the height and vibrating with the structural mass.

2.2. The effects of the wind

In this study the static and aerodynamic effects of the wind were compared.

The static effects of the wind occur due to the pressure exerted by the velocity on the chimney's surface. The equivalent static force of the wind is equal to [1]:

$$F_W = c_s . c_d . c_f . q_p . A_{ref} \tag{2.1}$$

Where $c_s . c_d$ is a structural coefficient c_f is a strength coefficient A_{ref} is a reference area (that for steel chimneys is considered to be its diameter) and q_p represents the pressure due to the mean velocity of the wind considering the intensity of turbulence. This pressure has the following distribution [1].

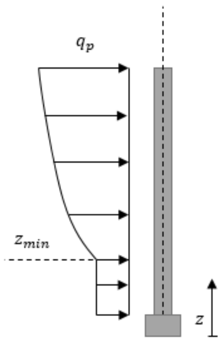

Figure 2.2 - Pressure of the wind static effect [4]

This pressure is equal to [1]:

$$q_p = \left[1 + 7 . I_{v(z)}\right] . \frac{1}{2} . \rho_{ar} . V_m(z)^2 \tag{2.2}$$

Where $I_{v(z)}$ is the intensity of turbulence and $V_m(z)$ represents the mean wind speed. The effects of the vortex shedding are a cyclic transversal inertial force that is mobilized by the displacement due to the vibrations [1],

$$F_{vs,i} = m(z).(2\pi f_i)^2.\emptyset_i(z).w_{i,máx} \tag{2.3}$$

Where, $m(z)$, represents the distributed mass, f_i is the natural frequency of the vibration mode, $\emptyset_i(z)$ is the vibration mode shape and $w_{i,máx}$ is the maximum displacement at the top of the chimney and it's equal to [1]:

$$\omega_{i,max} = \frac{1}{S_t^2}.\frac{1}{S_c}.K_\varphi.K_w.D \tag{2.4}$$

Where K_φ and K_w represent the modal shape and the correlation length coefficients respectively.

2.3. The definition of the TMD

The critical wind velocity of the second mode of vibration revealed high enough to be considered as not relevant. So, to decrease the amplification of the displacement due to the vortex shedding a TMD tuned to the frequency of the first vibration mode was considered. To study the dynamic behavior of the chimney with a TMD, a mechanical model with two-degrees-of-freedom is assumed. This model is characterized by the stiffness k_t and the modal mass relative to the first vibration mode M_0.

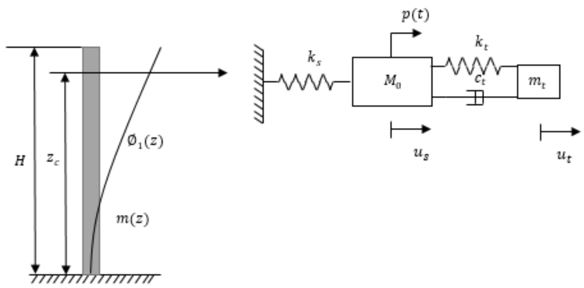

Figure 2.3 - Model representing the chimney with the TMD adapted from [11]

For a cantilever beam with a constant cross section and a uniform distributed oscillating mass, the modal mass corresponding to the fundamental mode of flexion is 62% of the total mass [4],

$$M_0 = 0,62mH \tag{2.5}$$

For the TMD definition, was adopted a solution in Hirsch system. As shown in the following figure, the mass of the TMD is materialized by a ring, with a quadrangular section, composed of steel, supported by a pendulum structure.

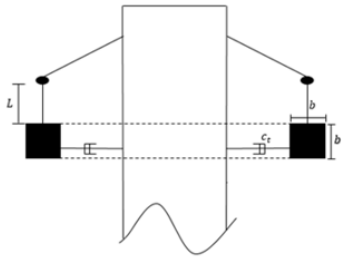

Figure 2.4 - The Hirch system.

The frequency of the TMD (assuming that the TMD is a perfect pendulum) is then adjusted, considering the oscillatory movement of a pendulum. This movement is defined with the length of the metal bars, suspending the mass ring, by the following formula:

$$f_T = \frac{1}{2\pi}\sqrt{\frac{g}{L}} \tag{2.6}$$

Where g represents the gravitational acceleration and L represents the suspension length. It should be noted that the springs coupled to the shaft also give stiffness to the movement of the TMD, which means that this value of length of the bars is only an approximation of the real length, therefore, the rigorous adjustment of the frequency must be made, take into account, the length of the bars and this stiffness simultaneously.

2.4. Taking in account the mistuning

The modal characteristics of the chimney are known only to a degree of accuracy, since the modal response is affected, for example, by the effect of temperature or corrosion which modifies its properties. Furthermore, in the construction, the frequency and damping specified in the design do not exactly match. This can cause a mistuning that deteriorates the efficiency of the TMD. In this regard, Ricciardelli [12] proposes a minimum mass to the TMD, to take in account the effect of the mistuning.

$$\mu_{min} = \left[\frac{\rho d^2}{m}\left(\frac{(2\gamma+1)C_L^2 2L\rho d^2}{256\pi^{7/3}{S_t}^4(3+\lambda)Bm\left({}^{y}/_{d}\right)^2} - \frac{K_a}{c}\right)\right]^{1/\varepsilon} \tag{2.7}$$

B - Bandwidth of the vortex shedding spectrum
C_L - Lift Coefficient vortex shedding
K_a - Negative aerodynamic damping
λ - Slenderness of the chimney
d - Diameter of the chimney
L - Shedding correlation length
γ - Coefficient related to the vibration mode shape
y - Maximum displacement on the top of the chimney

The values c and ε are parametric values related to the mistuning (the error between the optimal tuning q_{opt} and actual tuning), that can be consulted in [12].

3. Results and discussion

3.1. Comparison of the different TMD solutions

For the considered example, the static effect of the wind, although it caused stresses with a higher value in the base of the chimney due to the consideration of loss of resistance because of fatigue, the vortex shedding proved to be the determinant load of design. For a life period of 50 years the chimney without the TMD revealed to have insufficient damping capacity to resist to the number of cycles induced by the vortex shedding. Firstly, it was considered a solution of TMD with a, $\mu = 0{,}04$, this solution revealed to be sufficient to control the amplitude of the oscillations for a fatigue life durability of 50 years.

Table 3.1 - Dynamic response of the chimney with and without the TMD.

Dynamic response	Without TMD	With TMD ($\mu = 0{,}04$)
S_c	7,11	78,21
$w_{i,máx}(m)$	0,206	0,019
$M_b(kN.m)$	3941,62	359,75
$\sigma_{máx}(MPa)$	55,76	5,19
$\Delta\sigma_{máx}(MPa)$	111,52	10,38
$\Delta\sigma_r(MPa)$	24,98	24,98

Not only due to the mass reduction of the ring considered, that consequently means less costs, but also to minimize the impact of the TMD on the original structural natural frequencies, it was studied the minimum mass of TMD needed to provide the sufficient damping in this situation, using the method of Ricciardelli and the method of Den Hartog, and considering a mistuning of 5%. The results are presented as follows.

Table 3.2 - Minimum mass of the TMD considering Den Hartog's and Ricciardelli's methodology.

Methodology	Mistuning	$\mu(\%)$	$m(kg)$
Den Hartog	$\lvert\beta - 1\rvert = 0$	0,7	260,4
	$\lvert\beta - 1\rvert = 0{,}05$	1,85	688,2
Ricciardelli	$\lvert\beta - 1\rvert = 0$	0,14	52,8
	$\lvert\beta - 1\rvert = 0{,}05$	0,75	279

The verified differences can be explained by the divergences in the two methodologies. In the first methodology, the action of vortex shedding was considered as a harmonic action that amplifies an equivalent mechanical model of 2 degrees of freedom composed of the modal mass of the structure and the TMD. On the other hand, in the methodology conceived by Ricciardelli, it assumes the effect of the vortex shedding as a negative aerodynamic damping. This has an implication in the tuning formulas of the device, as well as in the increase of the damping due to its implementation.

3.2. The variation of the slenderness

Using the same chimney of the example, it was studied the variation of diameter and height in order to see the effect of the variation of slenderness. Contrary to what could be expected, by increasing the height of the chimney, consequently increasing its slenderness, the moment at the base decreases. Although the height of the chimney it's higher, it is found that the frequency of the structure decreases. Making the structure less rigid, the elastic restitution force, for the same displacement, is smaller. This, however, is verified till, $H = 78\,m$, at which point the mobilization of the second vibration mode is considered. In this case, for the, same diameter because the Scruton number maintains the same, the displacement at the top of the chimney maintains the same. The number of cycles increases, but for this case, this isn't relevant as the determinant structural detail of chimney was already in its fatigue limit resistance, limit for which the stress resistance it's constant for an increase of the number of load cycles.

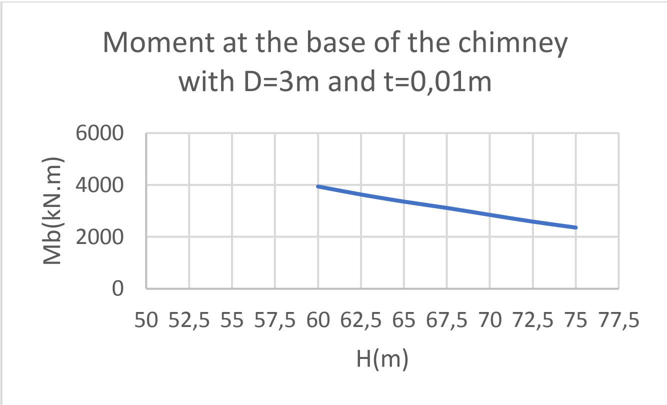

Figure 3.1 - Moment in the base of the Chimney with different heights and $D = 3m$ and $t = 0{,}01m$.

The same logic was applied to the diameter of the chimney. In this case, it was decided to increase the diameter, in order to decrease the slenderness.

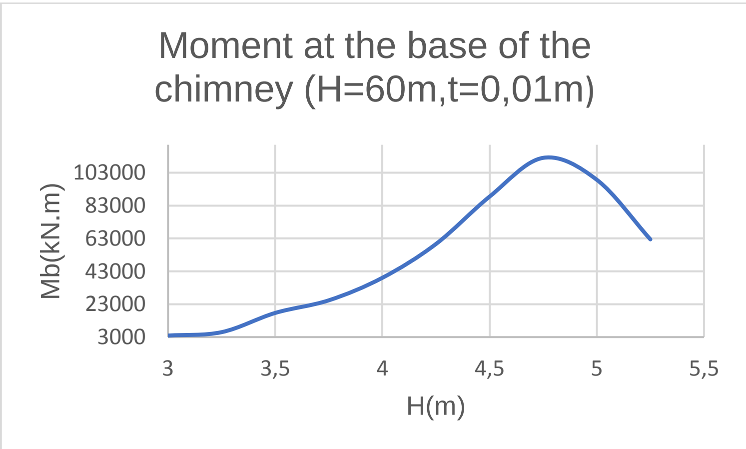

Figure 3.2 – Moment at the base of the chimney (H=60m and t=0,01m).

The displacement at the top increases significantly because there is a reduction in the number of Scruton, i.e. an increase of the susceptibility of the structure to the vortex shedding phenomenon.

Although the mass distributed is higher, which would predict a reduction in the susceptibility to the phenomenon, the fact is that the susceptibility increases, due to the larger diameter. The moment in the base, increases as in the previous case. However, the increase it's considerably higher due to the simultaneously increases of the stiffness, displacement and also because of the decrease of the damping (for $\frac{H}{D} < 18$ the damping is equal to $\delta_s = 0{,}02$ [1]) . For a diameter of, $\approx 4.75\,m$, the lateral force coefficient reduces, because of the increase oh the critical velocity of the first vibration mode (for $\frac{v_{cr,1}}{v(z)_m} > 0{,}83$, c_{lat} decreases linearly) which causes a decrease of the displacements at the top of the chimney and in the moment at the base of the structure. This also causes a reduction in probability of the phenomenon occurring, which means a lower reduction of the resistance. However, the increase of the applied moment, is not proportional to the resistance decrease. These considerations are valid up to a diameter of, $5.5\,m$. For this diameter the critical velocity of the first mode is reduced to the point where the effect of vortex shedding is disregarded for this mode, and completely. The TMD have no impact on the on the static effects of the wind due to this fact, these effects weren't considered in this study.

4. Conclusions

This work aimed to analyze the viability of the implementation of a TMD system as a solution to reduce the vibrations due to the effect of vortex shedding.

The chimney, with high slenderness, ($H/D = 20$), proved to be highly affected by the phenomenon, to the point of being the most conditioning action because of the loss of resistance consequence of the fatigue cracking process.

A definition of different geometries, in order to diminish the slenderness, revealed not to have the intended impact. The reduction of the structure's slenderness, within certain limits, even revealed a more unfavorable dynamic response. This leads to the conclusion that the consideration of a damping system becomes, in certain cases of steel chimneys, indispensable.

It is concluded that TMD is a viable damping solution, presenting the following advantages:

• The formulas, for tuning the optimal parameters, are easy to apply

• It increases greatly the structural damping with relatively small masses

• Although the efficiency of the system is very susceptible to the correct frequency tuning, it is possible to overcome this weakness, with a not very significant mass increase.

• A solution is achieved that has little impact on the vibration modes characteristics

5. References

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