XXXVI Ibero-Latin American Congress on Computational Methods in Engineering
CILAMCE
Rio de Janeiro, 22-25 Nov 2015

GENETIC OPTIMIZATION OF TOWER VIBRATIONS WITH PENDULUM TMD

Gino Bertollucci Colherinhas

Marcus Vinicius Girão de Morais

ginobertollucci@hotmail.com

mvmorais@unb.br

Universidade de Brasília - Faculdade de Tecnologia

Campus Universitário Darcy Ribeiro, Asa Norte - 70.910-900, Brasília, DF, BRAZIL.

Maura Angelica Milfont Shzu

Suzana Moreira Ávila

maura@unb.br

avilas@unb.br

Universidade de Brasília - Faculdade UnB-Gama

Campus UnB Gama, Área Especial Indústria e Projeção A - 72.444-240, Gama, DF, BRAZIL.

Abstract. *The wind turbine is supported by towers, which are slender and flexible due to their geometry and great height. As a consequence, they may experience excessive vibration levels caused by both the operation of the turbine and the wind loads. One common passive control device to solve this problem is the Tuned Mass Damper (TMD). Briefly, it is a pendulum-damper which transfers the energy of the vibration from the main structure to itself, working as a passive device. These passive devices need to be finely tuned to work as dampers; otherwise, they could amplify structural vibration levels. In this paper we propose to optimize the vibration performance of a pendulum TMD applying a Genetic Algorithm (GA) MATLAB toolbox (Colherinhas et al., 2015) in the modeling of the main structure and the TMD via finite element method calling the resources and capabilities of the software ANSYS (Ávila et al., 2015). We restrict the mass, and length of the pendulum to minimize high peaks of the vibration frequencies. Then we compared the finite element optimization using ANSYS with an analytic 2-DOF optimization.*

Keywords: *Wind Turbine, Tuned Mass Damper, Algorithm Optimization, Structural Control.*

1 INTRODUCTION

The current energy crises coupled to the depletion of fossil fuel stocks and the need to reduce emissions of carbonic gas, in order to preserve the environment makes wind power generation a viable and attractive mean for producing electricity (Ávila et al., 2015).

The wind turbine is supported by a tower that can experience excessive vibrations caused by both the wind turbine and the wind forces because of its geometry and great height. A detailed analysis of the structural behavior of the tower is of great importance due to its cost, which can represent approximately 20% of the total cost of the system (Morais et al., 2009).

Given the great progress in analyses and structural dimensioning, with the advances in the material field and construction techniques, higher and slender structures have become more interesting to study. These structures, however, are more vulnerable to excessive vibrations due to dynamic loads such as earthquakes, winds, storms, waves, etc.

An alternative option, widely studied in the last few years, is the structural control. Fundamentally, the structure stiffness and damping properties is changed adding external device or applying external forces. It is classified as passive, active, hybrid, or semi-active control. Several researchers have been studying the use of structural control to help suppress the wind-induced vibrations experienced by wind turbine towers (Stewart & Lackner, 2014).

To minimize these vibrations, we implemented a structural control with a Tuned Mass Damper (TMD) Pendulum type. A TMD is a passive control device composed of a mass-spring-dashpot attached to the structure, aiming to reduce structural vibration response (Soong & Dargush, 1997), since it exhibits a pendulum absorber geometry. After that, a genetic optimization methodology was loaded with multiple objectives to be compared to an existing mathematical and numerical optimization.

We optimized two cases using a GA MATLAB toolbox developed in other research (Colherinhas et al., 2015): the response function of a reduced two-DOF obtained by a Zuluaga's (Zuluaga, 2007) TMD mathematical formulation (Figure 1-b) and the response obtained using an existent finite element package ANSYS (Figure 1-c). Then we compared both cases with a tower without a pendulum TMD (Figure 1-a).

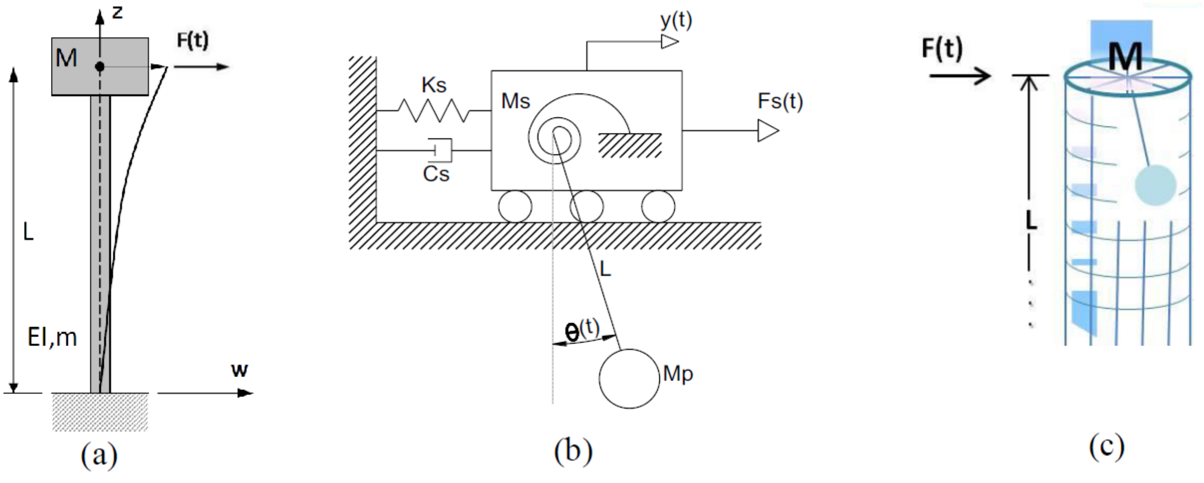

Figure 1: (a) Schematic description of a tower without a TMD; (b) Schematic description of a pendulum TMD attached to a main system; (c) Schematic description of a shell with a tip mass M.

The aim of this article is create a methodology to assist wind turbine design projects, enabling the selection of a configuration that targets a minimization of the frequency peaks. The variable parameters surveyed are: length and mass of the pendulum.

The modeling of the tower for a finite element method was considered using shell element distributed over the nodes of the top (Figure 1-c). The natural frequencies and the mode shapes of the models were obtained as well as transient and harmonic analysis results to determine the dynamic response of the tower in time domain, considering wind loads.

Although structural control of slender and great height towers using pendulum TMDs is widely studied in literature, studies modeling the pendulum TMD with ANSYS it wasn't found yet. This is a potential tool to study the performance of this system attached to excessive vibration vulnerable structures like wind turbines.

Depending on the result obtained for the evaluation of this modeling, fitness is ascribed to each configuration and inputted in a Genetic Algorithm (GA) to optimize the initial values. This optimization method was chosen for its advantages at discontinuous functions with multiple variables and complex formulations.

The following sections describe the modal analysis of a wind turbine tower, the TMD mathematical formulation for the pendulum settings, the methodology, the optimization method chosen (definitions of the chromosome, fitness function, and convergence criteria), and the results.

2 MATHEMATICAL FORMULATION

In this section we describe the modal analysis of the cantilever beam with a tip mass (Figure 1-a) and the 2-DOF mathematical formulation of a pendulum TMD attached to a main system (Figure 1-b).

2.1 Modal analysis

The goal of the modal analysis is to reduce the complex system of partial differential equation which describes the dynamic behavior of a continuous structure. The motion of a cantilever Bernoulli-Euler beam (Figure 1-a) is described in Equation 1 using the theoretical treatment of the modal analysis (Meirovicth, 1967).

$$\frac{\partial^2}{\partial z^2}\left(EI\frac{\partial^2 w(z,t)}{\partial z^2}\right)+\rho\frac{\partial^2 w(z,t)}{\partial t^2}=F(z,t) \tag{1}$$

where $w(z,t)$: normal displacement of the beam axis in some directions; z: distance along the beam axis; ρ: mass per unit length of the beam; F: external force per unit length applied normal to the beam axis in the direction of w; E: Young's modulus; I: area inertia moment for bending.

Applying boundary conditions and solving the partial differential equation (Ávila et al., 2009) the generalized stiffness and mass of the tower are computed, respectively in Equations 2 and 3.

$$K_s=\frac{\pi^4}{32L^3}EI \tag{2}$$

$$M_s = \frac{mL}{2\pi}\left[\pi\Big(3 + 2\frac{L_e}{L}\Big) - 8\right] \tag{3}$$

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

2.2 Tuned mass damper formulation

The Figure 1-b shows a schematic description of a pendulum TMD attached to a main system composed of a two-DOF. The main system is reduced to a one-DOF model, which corresponds to the mode to be controlled (Soong & Dargush, 1997). The motion considering small displacements to the system is given by the matrix of the Equation 4 (Zuluaga, 2007).

$$\begin{bmatrix} (M_s + M_p) & M_pL \\ M_pL & M_pL^2 \end{bmatrix} \begin{bmatrix} \ddot{y} \\ \ddot{\theta} \end{bmatrix} + \begin{bmatrix} C_s & 0 \\ 0 & C_p \end{bmatrix} \begin{bmatrix} \dot{y} \\ \dot{\theta} \end{bmatrix} + \begin{bmatrix} K_s & 0 \\ 0 & (K_p + M_pgL) \end{bmatrix} \begin{bmatrix} y \\ \theta \end{bmatrix} = \begin{bmatrix} F_s(t) \\ 0 \end{bmatrix} \tag{4}$$

where M_s: main system modal mass; C_s: main system modal damping; K_s: main system modal stiffness; M_p: pendulum mass; C_p: pendulum damping; K_p: pendulum stiffness; L: cable length; g: gravity acceleration; $F_s(t) = F_{s0}e^{i\omega t}$: excitation modal force; $y(t)$: main system displacement; $\theta(t)$: pendulum angular displacement.

Considering $F_s(t) = e^{i\omega t}$, then was obtained $y(t) = H_y(\omega)e^{i\omega t}$ $\theta(t) = H_\theta(\omega)e^{i\omega t}$. Substituting in Equation 4, we obtain the linear equation system given by Equation 5.

$$\begin{bmatrix} -(M_s + M_p)\omega^2 + C_si\omega + K_s & M_pL\omega^2 \\ -M_pL\omega^2 & -M_pL^2\omega^2 + C_pi\omega + (K_p + M_pgL) \end{bmatrix} \begin{bmatrix} H_y(\omega) \\ H_\theta(\omega) \end{bmatrix} = \begin{bmatrix} 1 \\ 0 \end{bmatrix} \tag{5}$$

where $H_y(\omega)$: structure response function in the frequency domain; $H_\theta(\omega)$: pendulum response function in the frequency domain.

Solving this linear equation system (Equation 5), Zuluaga (2007) obtained response functions $H_y(\omega)$ and $H_\theta(\omega)$ in the frequency domain. These responses are present in Equations 6 and 7.

$$H_y(\omega) = \frac{-\omega^2B_2 + i\omega B_1 + B_0}{\omega^4A_4 - i\omega^3A_3 - \omega^2A_2 + i\omega A_1 + A_0} \tag{6}$$

where: $B_0 = K_p + M_pgL$; $B_1 = C_p$; $B_2 = M_pL^2$; $A_0 = M_pK_sgL + K_sK_p$; $A_1 = M_pC_sgL + C_sK_p + C_pK_s$; $A_2 = M_sK_p + M_pK_p + M_sM_pgL + M_p^2gL + C_sC_p + M_pK_sL^2$; $A_3 = M_sC_p + M_pC_p + M_pC_sL^2$; $A_4 = M_sM_pL^2$.

$$H_\theta(\omega) = \frac{-\omega^2B_2 + i\omega B_1 + B_0}{\omega^4A_4 - i\omega^3A_3 - \omega^2A_2 + i\omega A_1 + A_0} \tag{7}$$

where: $B_0 = 0$; $B_1 = 0$; $B_2 = M_pL^2$; $A_0 = M_pK_sgL+K_sK_p$; $A_1 = M_pC_sgL+C_sK_p+C_pK_s$; $A_2 = M_sK_p+M_pK_p+M_sM_pgL+M_p^2gL+C_sC_p+M_pK_sL^2$; $A_3 = M_sC_p+M_pC_p+M_pC_sL^2$; $A_4 = M_sM_pL^2$.

3 METHODOLOGY

Numerical simulations of the wind turbine with a pendulum TMD attached were performed in ANSYS. The dynamic behavior of this structure is studied through harmonic analysis. Below, we describe the methodology used to model this system with ANSYS.

Firstly, it was created a function in MATLAB that calls the software ANSYS and interacts every loop with it, sending information created by the GA MATLAB toolbox.

Then, using the software ANSYS, a mass-spring system was modeled with a pendulum TMD with basically MASS21, BEAM4 and COMBIN14 elements. A COMBIN14 element represents the system structural stiffness. Another COMBIN14 element models a three-dimensional spring to describe the torsional stiffness that connects the tower to the pendulum via CP command. The pendulum system was modeled as a beam element with Euler-Bernoulli coupled BEAM4 and with a mass element MASS21 at the end.

The tower used in this work was modeled with shell elements (Figure 1-c), using 712 nodes. A constant force was applied only in the x axis, and the the analysis was made only for this axis. A MASS21 element defines the nacelle + blade mass at the tower top. The pendulum is connected to the top of the structure via COMBIN14. The tower is modelled with finite elements of quadratic rectangular shell SHELL93. To simulate the stiffening conferred by the set nacelle + blade to the tower, the CERIG command is used. This command is responsible for setting a rigid region without adding mass to the system.

4 OPTIMIZATION

In this section we discuss, briefly, how works the GA MATLAB toolbox constructed in other research (Colherinhas et al., 2015) and define restrictions of the pendulum length and mass, to optimize the 2-DOF formulation and the finite element proposition.

A Genetic Algorithm (GA) is a stochastic algorithm that mimics natural phenomena as operators, proposed to find the best pendulum setting. GAs are search techniques based on the processes of natural selection for survival through population genetics (Holland, 1992). For the GAs to start evolving, we can use the steps selection, recombination (crossover), mutation, and replacement, where the survival-of-the-fittest mechanism can be applied to the candidate solutions (Goldberg, 1989) (Haupt & Haupt, 1998).

In the initialization, the algorithm replies the evolutionary genetics and generates a random population by uniform distribution. The chromosomes represent the elements in evolutionary algorithm space, where their features, called genes (quantified by values called alleles), are the problem inputs to be rated by the fitness function (Colherinhas et al., 2014).

The selection allocates more copies of those solutions with higher fitness values and the roulette-wheel select is the procedure chosen to accomplish this idea. After that, recombination combines parts of two parental solutions to create new, possibly better, solutions. The mutation, on the other hand, randomly modifies the solutions' allele. At the end, a percentage of the best elements is simply copied to the next generation using an elitism probability. This is a mechanism to guarantee the best solutions are transmitted to the current generation (Goldberg, 1989).

The algorithm feeds the system back and evaluates each element from the population. Using the aforementioned evolutionary strategies, the fittest elements will have more probability to pass their features to the next generations.

They are depicted by real base, where each gene matches to the hereditary characteristics to be combined and evaluated. The chromosome is described by the following function:

$$C = [L_p; m_p] \tag{8}$$

where L_p: pendulum length; m_p: pendulum mass.

The real base is useful when the parameters to be optimized are continuous variables (Rahmat-Samii & Michielssen, 1999). Working with any decimal digits of precision the computer uses flow point numbers to represent the chromosome and its size is equal to the vector that represents the problem solution; thereby each gene represents one problem variable (Rodrigues, 2007).

Variable restrictions were inputted to control this optimization faster. The lanes for pendulum length L_p (m) and pendulum mass m_p (kg) are:

$$1.00 \leq L_p \leq 5.00 \tag{9}$$

$$7000 \leq m_p \leq 12000 \tag{10}$$

The performance of each element of the population is measured by the fitness function. It rates the capability of the element (given a pendulum setting) to minimize high vibration level.

5 RESULTS

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. Although in practice tower stiffness and mass vary along its height, this work considers these properties constant, since the following example properties were taken from a previous work (Murtagh et al, 2005) where the same simplifications were undertaken. 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 $210109N/m^2$ and $7850kg/m^3$, respectively. The tower carries a nacelle and a rotor system mass of $19876kg$. Damping was considered negligible (Ávila et al, 2013). The TMD properties were obtained following the design criteria suggested by Oliveira et al (2014). The torsional stiffness and damping were respectively $k_p = 1247900N/m$ and $c_p = 9024.9Ns/m$.

Using Equations 2 and 3 and considering wind force as a harmonic excitation, we obtained the following generalized rigidity, mass, and force parameters:

$$M_s\ddot{Y} + K_sY = F_s(t), \text{ where } M_s \cong 34.899kg \text{ and } K_s \cong 46.3671N/m \tag{11}$$

Inputting these values in the algorithm, many tests were made to determine the best probabilities of crossover, mutation, and elitism, as well as the population and generation sizes. The parameters were adjusted with: 7% probability of mutation; 70% crossover; 10% elitism; 150 chromosomes per generation. After 50 generations, the figures were plotted containing the curves of the best general fitness for each generation with their mean values.

A simple case aiming to reduce the two high system response peaks was loaded and was compared with an analytic optimization (Colherinhas et al., 2015). The low peak of the response was not considered for the optimization. The fitness function was ruled by Equation 12.

$$f_{fitness_1} = \frac{1}{\max H_y(\omega)_i}, \text{ where } i = 1, 2, \ldots, N_{ind} \tag{12}$$

where $\max H_y(\omega)_i$: maximum response of each generation; N_{ind}: size of the population.

The frequency response of the GA optimization of the structure using finite element and the 2-DOF were plotted, as well as the structure without a pendulum TMD (Figure 2).

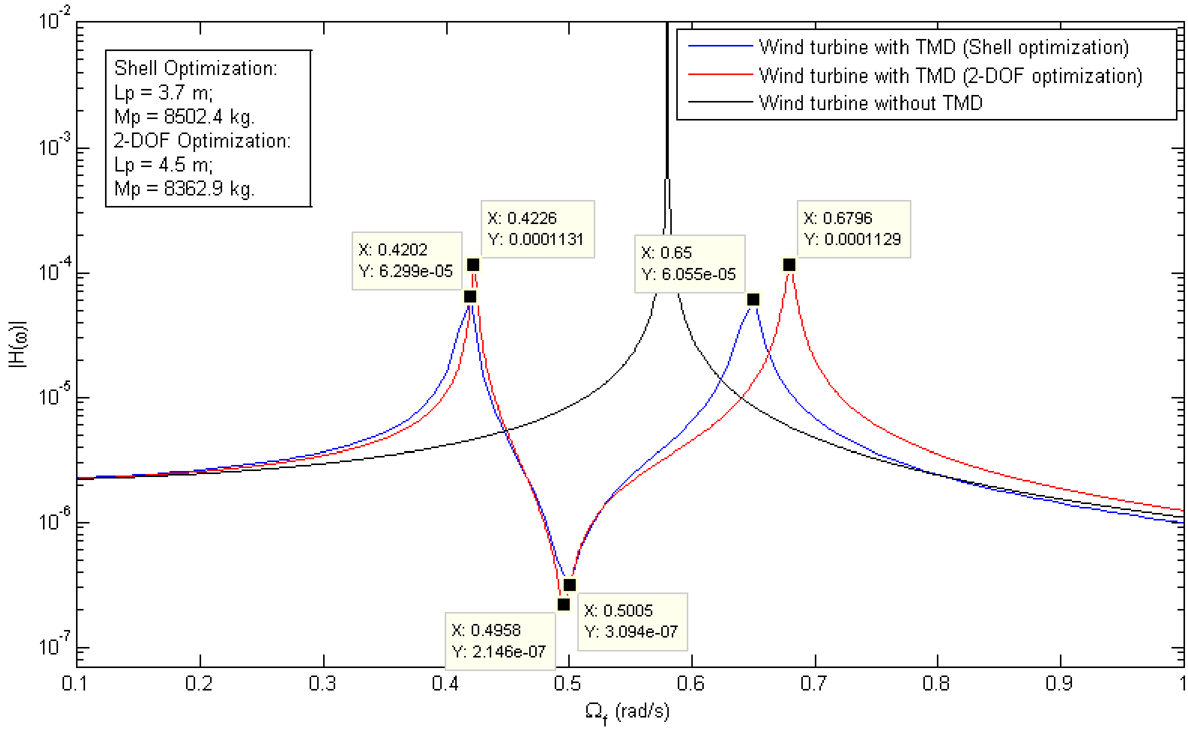

Figure 2: Displacement harmonic response of the wind turbine: with TMD (Shell and 2-DOF optimization) and without TMD.

In the Table 1 we compare the values of the length (L_p) and mass (M_p) of the pendulum for Shell and 2-DOF optimization.

Table 1: Comparison of the length and mass of the pendulum for a Shell and 2-DOF optimization.

	$L_p(m)$	$M_p(kg)$
SHELL optimization	3.75	8502.4
2-DOF optimization	4.50	8362.9
Error	20.0%	1.6%

In Table 1 it was showed that the pendulum length are different for both configurations in 20.0% while the mass of the pendulum presents 1.6% of error. A lot of explanations can be concluded based on these results. Initially, we can say that exists more than one combination for both parameters that the local minimum converges using the GA optimization. A sensibility analysis is necessary in future studies to see the reason why the searched values, that minimize both functions, converge to different configurations.

An interesting fact is although the optimization gives different pendulum combinations (specially in the length), the natural frequencies for the first and second modes are very close. In Table 2 we can see the difference of the natural frequencies (in Hz) for the first and second modes in the optimization of the SHELL and the 2-DOF.

Table 2: Comparison of tower natural frequencies for the Shell and 2-DOF optimization.

Natural Frequencies (Hz)	1st mode	2nd mode
SHELL optimization	0.4202	0.6500
2-DOF optimization	0.4226	0.6796
Error	0.6%	4.0%

6 CONCLUSION

This paper presents a genetic algorithm application to optimize the vibration performance of a pendulum TMD using the finite element package ANSYS called by a GA toolbox created in MATLAB. We used harmonic analysis to obtain the frequency-response function of a tower with a pendulum TMD. The continuous system that describe the tower was reduced to the first mode. Then the structure system was described as a 2-DoF dynamic system.

Using a genetic algorithm with floating point chromosome, we propose minimize high peaks of the vibration frequencies for two cases: the response function of a reduced two-DOF obtained by a Zuluaga's TMD mathematical formulation (Zuluaga, 2007) and the response obtained using an existent finite element package ANSYS (Ávila et al., 2015). Then we compared both cases with a tower without a pendulum TMD.

It was observed that the optimization gives different combinations for the pendulum configurations (the maximum difference was 20% in pendulum length). Although these results diverge, we can see that the natural frequencies values found in ANSYS optimization are close when compared with the 2-DoF optimization, as expected. This can indicate that the GA implementation and the finite element package can obtain satisfactory values, because the algorithm try to do exactly the fitness function objectives.

We can realize more studies about the fitness function that will try align the two curves of the Figure 2, and this can approximate both pendulum configurations. But in a first look over the results we can say that the algorithm works as expected, and the ANSYS finite element package works fine when combined with the GA MATLAB toolbox optimization.

Another interesting possibility that will be made in future studies will be release the settings for pendulum damping ratio and stiffness in the GA optimization, and this can enable the signal filtering of the frequency response aiming a min-max restriction.

ACKNOWLEDGEMENTS

The authors greetings CNPq (Brazilian Scientific Conseil) and MCTI (Industrial, Science and Technology Ministry) by financial support (Process 406895/2013-9) referent to scientific project edital MCTI/CNPq 74/2013 ”Capacitação laboratorial e formação de recursos humanos em Energia Eólica (Linha 1)”.

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