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VIBRATION CONTROL OF AN OFFSHORE WIND TURBINE MODELED AS AN INVERTED PENDULUM

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Abstract. *Wind energy presents itself, nowadays, as one of energy sources in rapid development and implementation all over the world. The project, building and maintenance of called wind farm still present lots of challenges for engineers and researches. In this context, offshore wind turbine are found: wind turbines installed on ocean next to coast which, between other advantages, gets benefit of more intense and consistent wind with less turbulence in these regions. One of conceptions from this kind of wind turbine is the floating. This kind of structural system can be analyzed like a discrete model of inverted pendulum. This article presents one modeling of inverted pendulum analyzing its dynamic behavior and stability. The floating wind turbine structure because of its geometry and great height, can experience excessive vibrations caused by its own functioning and also by wind forces. A solution to the problem of excessive vibrations that has been studied by many researchers in the last few years is structural control. One of the earliest structural control devices, already extensively studied is the Tuned Mass Damper (TMD). The TMD is designed as a mass-spring-dashpot device that is tuned to a specific structural natural frequency to transfer the vibrational energy from the main system to the auxiliary mass that vibrates out of phase. An alternative geometry for the TMD is the pendulum vibration absorber. The natural period of this device depends on the length of its cable, and can be considered as a linear oscillator only when the vibrational amplitudes are small. This work studies the dynamic behavior of offshore wind turbines modeled as a stable inverted pendulum. A mass-spring pendulum absorber is attached to the main system in order to reduce the excessive vibrations. Numerical simulations are performed to define TMD parameters that improve the control device performance. However, passive devices only work properly for the design frequency range, and wind forces are random type of excitations. Better results would be achieved if a robust control is developed. Thus study will serve as a basis for the proposition of a semi-active device.*

1 INTRODUCTION

Wind energy presents itself, nowadays, as one of energy sources in fast development and implementation all over the world. The design, construction and maintenance of the so called wind farms still present lots of challenges for engineers and researchers. The turbine is fixed for support in towers, which due to its geometry and high level are slender, flexible and might present excessive vibration caused by turbine´s own operation, as also by wind action. Detailed analysis of support tower structural behavior reveals itself of big relevance due to cost factor, since it represents about 30% of system total cost [1].

The reduction of vibrations in structures under the action of the wind has increased its need and importance in view of the tendency of construction of buildings more high and flexible. This kind of structure is more vulnerable to occurrence of excessive vibration coming from loading caused by wind, which may affect human comfort and structure safety. The wind turbine support towers follow that tendency, becoming higher and slender and might reach up to 120 meters in some cases. One of the highest towers nowadays is Enercon E-126 which produces 7.5 MW and is 135 meters tall [1]. High vibration amplitudes compromises safety of towers and, also, wind turbine performance.

Offshore wind turbines are installed on ocean next to the coast, they present some advantages like getting benefits of more intense and consistent wind with less turbulence in these places [2]. Examples of offshore and onshore wind farms are shown on Figure 1(a) and (b).

(a) (b)

Figure 1: (a) Offshore wind park at Cumbria (http://www.windenergyplanning.com/category/offshore-wind-turbines/, visited on 18/06/2013); (b) Onshore wind park at Britain (http://www.dailymail.co.uk/news/article-2013233/The-wind-turbine-backlash-Growing-public-opposition-thwarts-green-energy-drive.html , visted on 18/06/2013).

One conception of offshore wind turbines is the floating device, which is installed more distant from coast. This structural system present an additional dynamic excitation source compared to onshore wind turbine: the loading action from ocean wave’s movement in its base. The dynamic behavior of a floating wind turbine can be studied modeling the structural system as an inverted pendulum discrete model [3].

Structural control is a technology which aims to reduce excessive vibration levels by installing external devices or applying external forces which promotes changes on properties of system’s stiffness and damping [4].

One of typical mechanisms of structural control is the so called Tuned Mass Damper (TMD), which control the structure by energy transfer between the main structure and an auxiliary mass. As its name suggests, this device is tuned in a structure natural frequency, being basically projected to control structures that vibrates predominantly in a given vibration mode, in general the first, which is the case of high towers [5].

One of alternative geometry of TMD is the pendulum shape [6], it is attached to the main structure and its movement excites the TMD device. This damper has its vibration period depending of its cable length. Another relevant aspect is that the pendulum only can be considered as a linear oscillator when the vibration amplitudes are very small.

This work presents an inverted pendulum model to study the dynamic behavior and stability of a floating wind turbine. A passive control system, a pendulum TMD is connected to the structure aiming to reduce the angular amplitude of the main structure. To improve control system performance, a parametric study is carried out changing TMD damping and stiffness.

2 INVERTED PENDULUM MODEL

For modeling the dynamic behavior of an offshore wind turbine, this work studies the behavior of an inverted pendulum discrete model [7] like the one shown in Figure 2(a). A tip mass is placed at the the bar end. The system excitation is a dynamic force applied at the tip mass, simulating the wind force applied at blade/nacelle set.

The system's equations of motion are calculated as follows:

$$x_m = l.\sin\theta = l.\theta \; ; \; y_m = l.\cos(\theta) = l \tag{1}$$

$$x_b = \frac{l}{2}.\sin(\theta) = \frac{l}{2}.\theta \; ; \; y_b = \frac{l}{2}.\cos(\theta) = \frac{l}{2} \tag{2}$$

$$K_E = \frac{1}{2}\left(m.l^2.\dot{\theta}^2 + \frac{1}{3}\rho.l^3.\dot{\theta}^2 + m.\dot{u}^2 + 2.m.l.\dot{\theta}.\dot{u} + l.\rho.\dot{u}^2 + m_c.\dot{u}^2\right) \tag{4}$$

$$P_E = \frac{1}{2}(K.\theta^2 + (2.m.l + \rho.l^2)g.\cos\theta + k_c.u^2) \tag{5}$$

$$D_E = \frac{1}{2}(c.\dot{u}^2) \tag{6}$$

$$\frac{d}{dt}\left(\frac{\partial K_E}{\partial \dot{\theta}}\right) - \left(\frac{\partial K_E}{\partial \theta} - \frac{\partial P_E}{\partial \theta} - \frac{\partial D_E}{\partial \dot{\theta}}\right) = F(t).l$$

$$\frac{d}{dt}\left(\frac{\partial K_E}{\partial \dot{u}}\right) - \left(\frac{\partial K_E}{\partial u} - \frac{\partial P_E}{\partial u} - \frac{\partial D_E}{\partial \dot{u}}\right) = 0 \tag{7}$$

$$\begin{bmatrix} \frac{\rho.l^3}{3} + m.l^2 & m.l + \frac{\rho.l^2}{2} \\ m.l + \frac{\rho.l^2}{2} & m_c + m + \rho.l \end{bmatrix} \begin{bmatrix} \ddot{\theta} \\ \ddot{u} \end{bmatrix} + \begin{bmatrix} 0 & 0 \\ 0 & c \end{bmatrix} \begin{bmatrix} \dot{\theta} \\ \dot{u} \end{bmatrix} + \begin{bmatrix} K - m.g.l - \frac{\rho.g.l^2}{2} & 0 \\ 0 & k_c \end{bmatrix} \begin{bmatrix} \theta \\ u \end{bmatrix} = \begin{bmatrix} F(t).l \\ 0 \end{bmatrix} \tag{8}$$

Where m is the tip mass at the end of the bar, x_m and y_m are m coordinates, ρ is bar linear density, x_b and y_b are gravitational centre bar coordinates, θ is bar angular amplitude, l is bar length, K is torsional stiffness, g is gravity acceleration, m_c is car mass, u is car horizontal displacement, k_c is car stiffness, c is car damping coefficient, K_E is total system kinetics energy, P_E is total system potential energy, D_E is total system dissipated energy and $F(t)$ is external force applied at the system.

One of the most used passive devices on structural control is the tuned-mass damper (TMD) [5]. One of the alternatives geometries of TMD is in pendulum shape [8]. Figure 2(b) shows the discrete model of an inverted pendulum with a pendulum TMD installed.

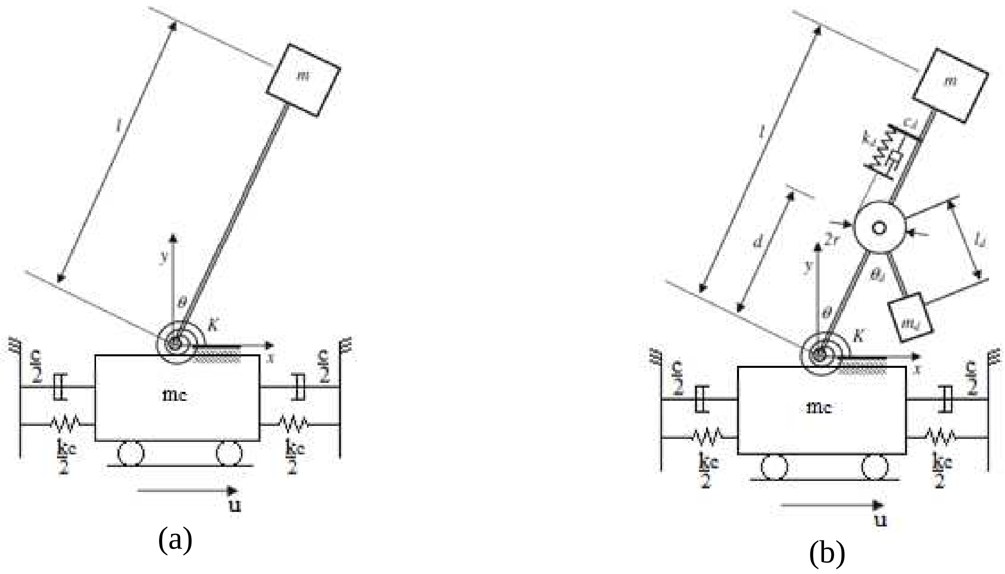


Figure 2: (a) System without TMD; (b) System with TMD.

The dynamic equilibrium equations of the controlled system are obtained similarly as the ones above:

$$x_m = l.\sin\theta = l.\theta\ ;\ y_m = l.\cos(\theta) = l \tag{9}$$

$$x_b = \frac{l}{2}.\sin(\theta) = \frac{l}{2}.\theta\ ;\ y_b = \frac{l}{2}.\cos(\theta) = \frac{l}{2} \tag{10}$$

$$x_d = d.\sin(\theta) + l_d(\theta_d - \theta) = d.\theta + l_d(\theta_d - \theta)\ ;\ y_d = d.\cos(\theta) - l_d.\cos(\theta_d) = (d - l_d) \tag{11}$$

$$K_E = \frac{1}{2}\Big(m.l^2.\dot{\theta}^2 + m_d.d^2.\dot{\theta}^2 + m_d.l_d^2\big(\dot{\theta} - \dot{\theta}_d\big)^2 + 2.m_d.d.l_d.\dot{\theta}\big(\dot{\theta}_d - \dot{\theta}\big) + \frac{1}{3}\rho.l^3.\dot{\theta}^2 + m.\dot{u}^2 + 2.m.l.\dot{\theta}.\dot{u} + m_d.2.l_d\big(\dot{\theta}_d - \dot{\theta}\big).\dot{u} + 2.m_d.d.\dot{\theta}.\dot{u} + l.\rho.\dot{u}^2 + m_c.\dot{u}^2\Big) \tag{12}$$

$$P_E = \frac{1}{2}\big(K.\theta^2 + k_d.r^2.\theta_d^2 + (2.m.l + 2.m_d.d + \rho.l^2)g.\cos\theta - 2.m_d.l_d.g.\cos(\theta - \theta_d) + k_c.u^2\big) \tag{13}$$

$$D_E = \frac{1}{2}\Big(c_d.\big(r.\dot{\theta}_d\big)^2 + c.\dot{u}^2\Big) \tag{14}$$

$$\begin{gathered}
\frac{d}{dt}\left(\frac{\partial K_E}{\partial \dot{\theta}}\right) - \left(\frac{\partial K_E}{\partial \theta} - \frac{\partial P_E}{\partial \theta} - \frac{\partial D_E}{\partial \dot{\theta}}\right) = F(t).l \\
\frac{d}{dt}\left(\frac{\partial K_E}{\partial \dot{\theta}_d}\right) - \left(\frac{\partial K_E}{\partial \theta_d} - \frac{\partial P_E}{\partial \theta_d} - \frac{\partial D_E}{\partial \dot{\theta}_d}\right) = 0 \\
\frac{d}{dt}\left(\frac{\partial K_E}{\partial \dot{u}}\right) - \left(\frac{\partial K_E}{\partial u} - \frac{\partial P_E}{\partial u} - \frac{\partial D_E}{\partial \dot{u}}\right) = 0
\end{gathered} \tag{15}$$

$$\begin{bmatrix} \frac{\rho.l^3}{3} + m.l^2 + m_d(d - l_d)^2 & m_d.d.l_d - m_d.l_d^2 & m.l + \frac{\rho.l^2}{2} + m_d(d - l_d) \\ m_d.d.l_d - m_d.l_d^2 & m_d.l_d^2 & m_d.l_d \\ m.l + \frac{\rho.l^2}{2} + m_d(d - l_d) & m_d.l_d & m_c + m + m_d + \rho.l \end{bmatrix} \begin{bmatrix} \ddot{\theta} \\ \ddot{\theta}_d \\ \ddot{u} \end{bmatrix} + \begin{bmatrix} 0 & 0 & 0 \\ 0 & c_d.r^2 & 0 \\ 0 & 0 & c \end{bmatrix} \begin{bmatrix} \dot{\theta} \\ \dot{\theta}_d \\ \dot{u} \end{bmatrix} +$$

$$\begin{bmatrix} K - g\left(m.l - \frac{\rho.l^2}{2} - m_d(d - l_d)\right) & -m_d.g.l_d & 0 \\ -m_d.g.l_d & k_d.r^2 + m_d.g.l_d & 0 \\ 0 & 0 & k_c \end{bmatrix} \begin{bmatrix} \theta \\ \theta_d \\ u \end{bmatrix} = \begin{bmatrix} F(t).l \\ 0 \\ 0 \end{bmatrix} \tag{16}$$

Where m_d is TMD pendulum mass, l_d is TMD pendulum length, d is distance of pendulum to bar base, r is TMD disc radius, c_d is TMD damping coefficient, k_d is TMD's stiffness and θ_d is TMD pendulum's angular amplitude related to the bar.

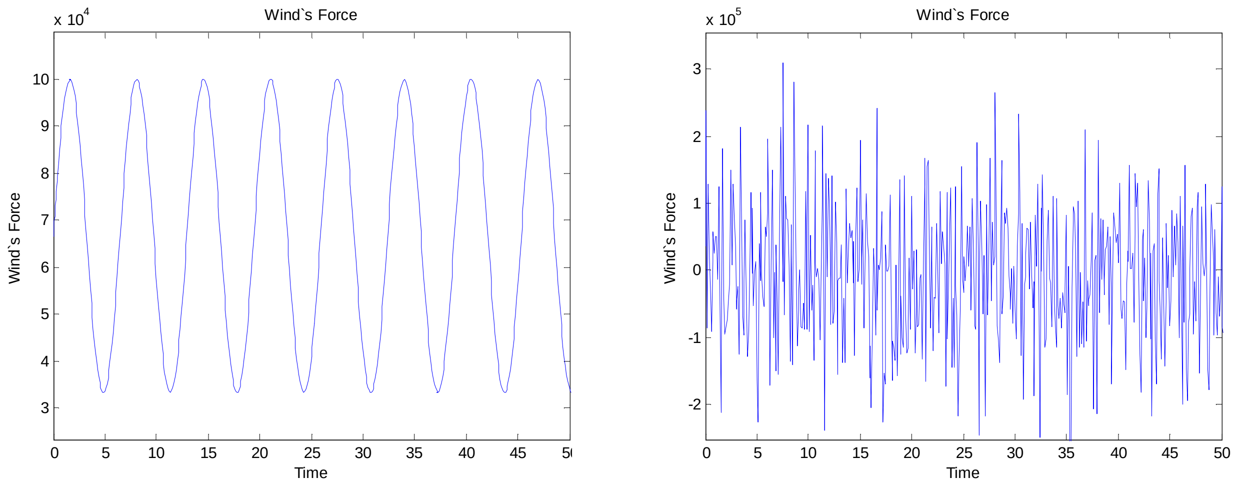
3 NUMERICAL RESULTS

The present work numerical study considered properties of the turbine NREL 5 MW [9], they are reproduced here in Table 1.

Rating	5 MW
Rotor, hub diameter	126, 3 m
Hub Height	90 m
Rotor mass	110,000 kg
Tower Mass	347,460 kg

Table 1: Wind turbine's properties [9].

Also the following values were considered for system parameters: $l = 90[m]$, $m_c = 240000[kg]$, $g = 9.81[m/s^2]$, $K = 8.27 * 10^8[N/rad]$, $m = 110000[kg]$, $m_d = 11000[kg]$, $d = 70[m]$, $l_d = 67.5[m]$, $k_d = 10^6[N/m]$, $c_d = 10^6[N.s/m]$. The wind dynamic loading was modeled as an applied load at the inverted pendulum mass. Two loading cases were considered: harmonic force ($F(t) = 66000 + 34000 * \sin(0.97 * t)\ [N]$) and white noise, both showed in Figure 3.



(a) (b)

Figure 3: Loading: (a) Harmonic force (b) White noise.

From equations of motion (8) and (16) it was obtained the corresponding transfer functions $G(s)$. Analyzing the denominator polynomial roots of $G(s)$, the corresponding system poles are presented on Table 2. With these poles, it could be checked out system stability. Both cases with and without control were considered.

Stability relation is analyzed verifying system pole's real part signal, to the system be stable all pole real parte should be negative. As can be verified on Table 2, all pole real part are negative, so the system is stable without and with control, moreover the TMD pendulum is also stable.

However, even the system being stable it should be also verified its angular amplitude to guarantee that the system behavior stays linear. This angular amplitude must not be greater than 0.26 rad.

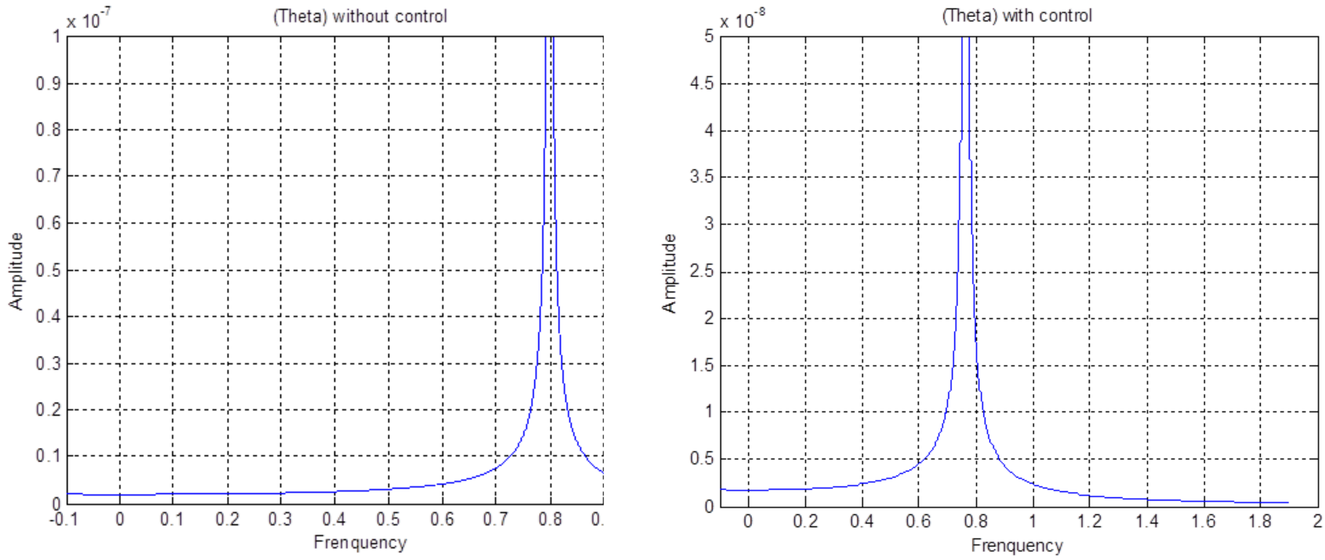
Poles to θ without control	Poles θ With control	Poles to θ_d
	-1.1285 + 1.0776i	-1.1285 + 1.0776i
-0.0018 + 0.7987i	-1.1285 - 1.0776i	-1.1285 - 1.0776i
-0.0018 - 0.7987i	-0.0034 + 0.7634i	-0.0034 + 0.7634i
-0.0034 + 0.0000i	-0.0034 - 0.7634i	-0.0034 - 0.7634i
-0.0000 + 0.0000i	-0.0032 + 0.0000i	-0.0032 + 0.0000i
	-0.0000 + 0.0000i	-0.0000 + 0.0000i

Table 2: System's poles.

The system natural frequencies on both analyzed cases were obtained through a modal analysis. The values of natural frequencies are presented in Table 3. Figure 4 shows θ frequency response plots. In both cases, the modes associated to the second natural frequency have greater influence at the response.

Without control natural frequencies $\left[\frac{rad}{s}\right]$	Controlled natural frequencies $\left[\frac{rad}{s}\right]$
	1.564110571354939
0.798725245061805	0.761604184809907
0.000119124536744	0.000115381013955

Table 3: Natural frequencies without and with control.



(a) (b)

Figure 4: Frequency reply (a) without control and (b) with control.

Figure 5 shows angular displacement θ evolution, with a harmonic force applied, comparing cases without and with TMD control. It is not verified a significant angular amplitude reduction.

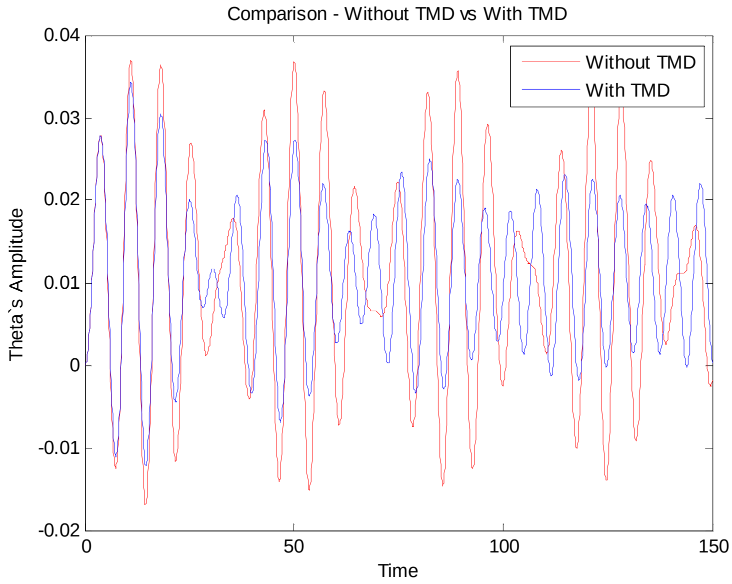


Figure 5: Time evolution of angular displacement with and without control (harmonic force).

However, modifying parameters to $k_d = 10^5[N/m]$, $c_d = 10^5[N.s/m]$ e $l_d = 90[m]$, it's possible to verify at Figure 6 a significant improvement in performance. But pendulum's l_d length, in practice, is limited by constructive restraints, thus this solution would be unfeasible.

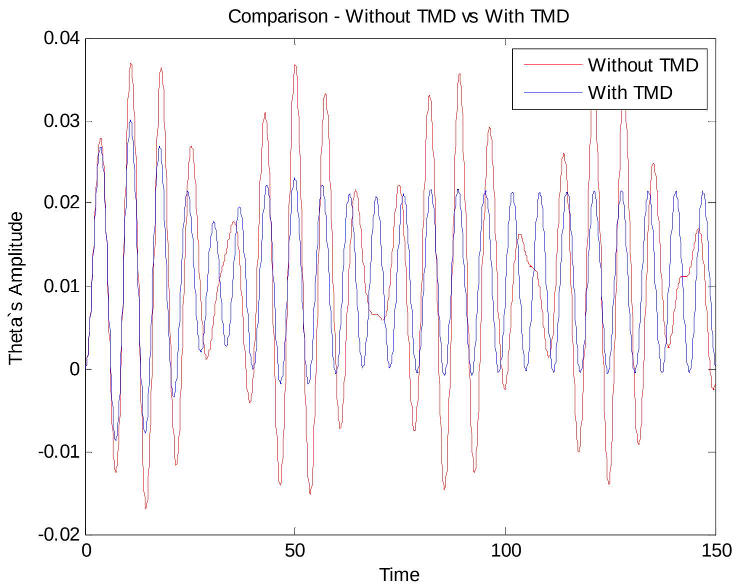


Figure 6: Time evolution of angular displacement with and without control, modifying parameters (harmonic force).

Applying the white noise loading shown on Figure 3(b), an amplitude reduction can be noticed in certain time intervals, but an amplitude increase of these values in others intervals, as can be verified on Figure 7. In this case, TMD is not efficient.

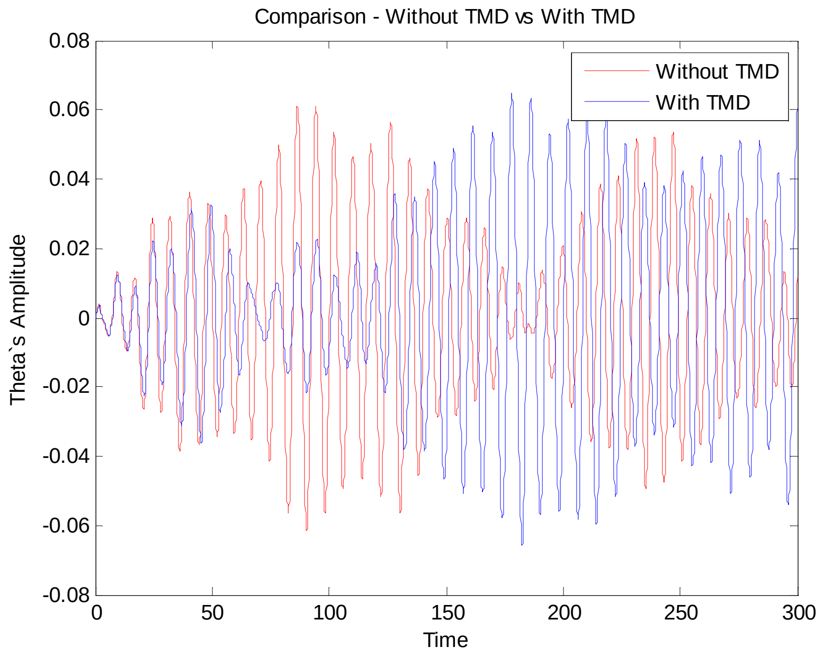


Figura 7: White noise.

Varying, again, the parameter's values to $k_d = 10^5[N/m]$, $c_d = 10^5[N.s/m]$ e $l_d = 90[m]$, an amplitude reduction can be noticed as show non Figure 8. However, as previously mentioned, this pendulum's length is unfeasible.

The conclusion is that pendulum TMD, in both loading cases, only reaches good efficient levels for high l_d length values, and it's not a good solution for this kind of problem because of practical restraints.

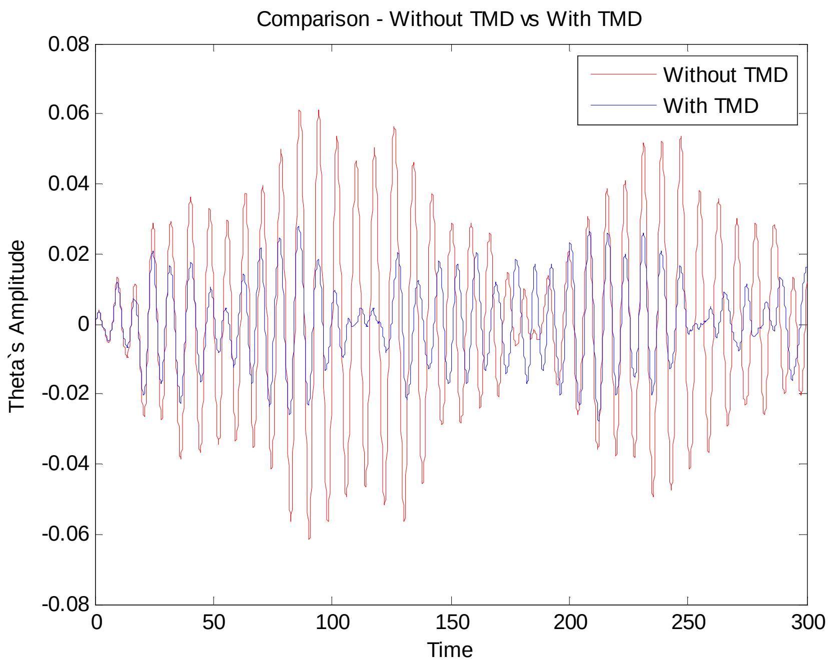


Figura 8: Time evolution of angular displacement with and without control, modifying parameters (white noise).

In all graphics and results presented it was observed that angular amplitudes were all less then of $0.26\,rad$, maintaining linear hypothesis, also to TMD θ_d's angular amplitude it is verified.

In this work, TMD control system only presented a good performance for l_d high values. A dynamic absorber of mass-spring inverted pendulum type may be able to solve this problem [8].

4 CONCLUSIONS

The dynamic behavior and stability of an offshore wind turbine are studied. Different dynamic loading are considered. Structural system is analyzed using an inverted pendulum discrete model. To reduce system angular amplitude displacement, a pendulum TMD is connected to the structure. TMD performance is verified by performing a parametric study. However a good control system efficiency is only reached adopting high values of pendulum, impracticable on constructive point of view. For a better TMD performance, it could be tested an inverted pendulum TMD, another idea is to design a semi-active control mechanism.

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