STRUCTURAL CONTROL OF AN OFFSHORE WIND TURBINE USING A SEMI-ACTIVE DEVICE

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Abstract. *Wind energy presents itself, nowadays, as one of energy sources in fast development and implementation all over the world. The project, building and maintenance of wind farms still present lot of challenges for engineers and researches. In this context, offshore wind turbines are found installed on ocean next to the coast which, amongst other advantages, benefit from more intense and consistent wind with less turbulence in these regions. One of the conceptions from this kind of wind turbine is the floating one. This kind of structural system can be analyzed as a discrete model of an inverted pendulum. This type of structure can be vulnerable to excessive vibration caused by wind loads. One of the most used vibration control devices is the Tuned Mass Damper (TMD). It basically consists of a mass-spring-damper system attached to the main structure, the frequency of the damper is tuned to a particular frequency aiming to make the TMD vibrate out of phase with the main system, thus transferring the energy system to the damper. However, passive devices only work properly for the designed frequency range, and wind forces are a random type of excitations. In this sense, a better approach would be designing a semi-active device. Semi-active control joins confidence and simplicity typical of passive systems with active control adaptability. It is characterized by not adding mechanical energy to the structure and by having properties that can change dynamically. These devices can be viewed as controllable passive devices because despite of its changing properties of damping and/or stiffness its action on the structure is passive. In this work, the Bang-Bang control is applied to change TMD's properties as an ON/OFF switch, changing them from one set of properties to another. Considering an inverted pendulum discrete model for TMD, the aim is to change TMD's properties accordingly to the response's frequency in real time, achieving better efficiency than the passive control.*

Keywords: *wind turbine, structural control, pendulum, tuned mass damper, semi-active damper, linear optimal control*

1 INTRODUCTION

Offhsore wind turbines are those installed in the maritime coast, they present a series of advantages like not havind land space limitations, nor an audible or visual impact; on the sea are subjected to less turbulent winds (Pao et al, 2009). On the other hand, unlike onshore systems, stability has to be analyzed due to its shallower foundations. Some offshore wind turbines are floating structures with a mobile base that are subjected to excitations caused by the movement of the sea waves at its bases.

Wind turbines are fixed on the top of high slender towers, these structures are flexible and can present excessive vibration caused by the machine functioning and also by wind forces. A detailed analysis of the tower strucutural behaviour reveals its great importance due to economic reasons since it represents about 30% of the system total cost (Quilligan et al, 2012).

A technology that proposes to reduce excessive vibration of the structures is called structural control. It consists in the addition of external devices such as dampers or application of external forces that change properties of stiffness and/or damping. It can be classified as: passive control, active control, hybrid control and semi-active control. (Housner et al, 1997)

However, passive control strategies have disadvantages: they only are efficient within the very limited frequency range for which they were designed. They are of limited robustness. In the case of structures subjected to wind loads this fact becomes critical, since this type of loading has a random characteristic.

One alternative to passive control is the so called semi-active control. Semi-active devices do not add energy to the structure and their properties can be varied dynamically (Spencer et al, 1997). The semi-active systems are more reliable and more robust than active systems. Damping and stiffness properties can be controlled in an active way by a control signal. These are controllable passive devices since they don't apply any additional force to the structure. Several studies on semi-active devices can be found on literature (Chey et al, 2010; Zemp et al, 2011; Rafieipour et al, 2014).

Despite previous studies and reasonable number of practical applications of structural control in bridges, high towers and buildings, the application of structural control techniques in wind turbines is still a new topic (Murtagh et al, 2008; Lackner et al, 2011; Stewart, 2012). However, despite previous studies, obstacles remain to be overcome. These include: reducing cost/maintenance and increase reliability, efficiency and robustness.

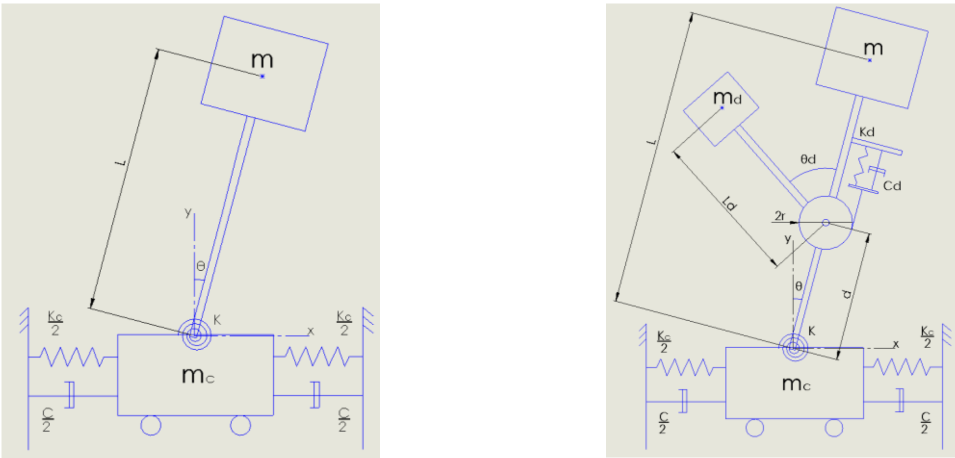
On previous studies (Guimarães et al, 2013; Guimarães et al, 2014) an inverted pendulum model was proposed to study the dynamic behavior and stability of a floating offshore wind turbine. Two passive control systems, a simple pendulum Tuned Mass Damper (TMD) and an inverted pendulum TMD were proposed aiming to reduce the angular amplitude of the main structure. The results obtained showed that inverted pendulum model had a better performance.

Avila and Guimarães (2015) proposed a semi-active TMD pendulum to control excessive vibration of an offshore floating wind turbine. The structure was modeled as an inverted pendulum discrete model. A bang bang control strategy was considered and simulated, TMD stiffness and damping values were calculated trough optimal control theory.

This work is a continuation of this previous study where more analysis and simulations where performed including stability analysis; controllability and observability analysis; proposal of the ON-OFF strategy criteria and frequency response analysis. Satisfactory results were find when compared to those of passive TMD pendulum.

2 MATHEMATICAL FORMULATION

In this work the behavior of a floating offshore wind turbine is studied through a simplified model of an inverted pendulum (Guimaraes et al, 2014) shown on Fig 1(a), this model assumes some simplifying hypotheses: (a) angular amplitude is kept within boundaries for a linear behavior; (b) a two dimension vibration system is considered; (c) wave loading is disregarded (d) a top mass represents nacelle+blades (Murtagh et al, 2008). It is a system with a spring and a damper at the base that simulates fluid resistance of the sea. The excitation of the system is a dynamic force applied to the mass m, simulating the wind force on the turbine blades. An inverted pendulum working as a tuned mass damper (TMD) for vibration reduction can be seen on Fig. 1(b).



(a) (b)

Figure 1. Model of an inverted pendulum over a mobile base (a) with inverted pendulum TMD installed (b).

Where m is the tip mass at the end of the bar, θ is bar angular amplitude, l is bar's length, K is torsional stiffness, m_c is base's mass, k_c is base's stiffness, c is base's damping coefficient, m_d is TMD pendulum's mass, l_d is TMD pendulum's length, d is the distance from pendulum to bar's base, r is TMD disc's radius, c_d is TMD's damping coefficient, k_d is TMD's stiffness and θ_d is TMD pendulum's angular amplitude related to the bar. The motion equation of the system with control (Figure 1(b)) is given in matrix form:

$$\begin{bmatrix} M_{1,1} & M_{1,2} & M_{1,3} \\ M_{2,1} & M_{2,2} & M_{2,3} \\ M_{3,1} & M_{3,2} & M_{3,3} \end{bmatrix} \begin{bmatrix} \ddot{\theta} \\ \ddot{\theta}_d \\ \ddot{u} \end{bmatrix} + \begin{bmatrix} 0 & 0 & 0 \\ 0 & c_d.r^2 & 0 \\ 0 & 0 & c \end{bmatrix} \begin{bmatrix} \dot{\theta} \\ \dot{\theta}_d \\ \dot{u} \end{bmatrix} +$$

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

$$\begin{bmatrix} F(s).l \\ 0 \\ 0 \end{bmatrix} \tag{1}$$

Where:

$$M_{1,1} = \frac{\rho.l^3}{3} + m.l^2 + m_d(d+l_d)^2 + \rho_d.d.l_d(d+l_d) + \rho_d.\frac{l_d^3}{3}$$

$$M_{1,2} = -m_d.d.l_d - m_d.l_d^2 - \rho_d.d.\frac{l_d^2}{2} - \rho_d.\frac{l_d^3}{3}$$

$$M_{1,3} = m.l + \frac{\rho.l^2}{2} + m_d(d+l_d) + \rho_d.d.l_d + \rho_d.\frac{l_d^2}{2}$$

$$M_{2,1} = -m_d.d.l_d - m_d.l_d^2 - \rho_d.d.\frac{l_d^2}{2} - \rho_d.\frac{l_d^3}{3}$$

$$M_{2,2} = m_d.l_d^2 + \rho_d.\frac{l_d^3}{3}$$

$$M_{2,3} = -m_d.l_d - \rho_d.\frac{l_d^2}{2}$$

$$M_{3,1} = m.l + \frac{\rho.l^2}{2} + m_d(d+l_d) + \rho_d.d.l_d + \rho_d.\frac{l_d^2}{2}$$

$$M_{3,2} = -m_d.l_d - \rho_d.\frac{l_d^2}{2}$$

$$M_{3,3} = m_c + m + m_d + \rho.l + \rho_d.l_d$$

The motion equations of a structural system of n degrees of freedom with active control, in matrix form, are given by:

$$\boldsymbol{M}\ddot{\boldsymbol{y}}(t) + \boldsymbol{C}\dot{\boldsymbol{y}}(t) + \boldsymbol{K}\boldsymbol{y}(t) = \boldsymbol{D}\boldsymbol{u}(t) + \boldsymbol{E}\boldsymbol{f}(t) \tag{2}$$

Where ***M***, ***C*** and ***K*** are, respectively, matrices of order n x n of mass, damping and stiffness: $\boldsymbol{y}(t)$ is a n x 1 vector of displacements; $\boldsymbol{f}(t)$ is a p x 1 vector of applied external forces; and $\boldsymbol{u}(t)$ is a q x 1 vector of control forces. The matrices ***D***(n x q) and ***E***(n x p) define the location of control forces and excitation, respectively.

Usually, the formulation and solution of control problems present equations of motion Equation (1) as state equations:

$$\dot{\boldsymbol{z}}(t) = \boldsymbol{A}\boldsymbol{z}(t) + \boldsymbol{B}\boldsymbol{u}(t) + \boldsymbol{E}\boldsymbol{f}(t) \tag{3}$$

where ***z(t)*** is the state vector $4n + 2q$; ***A*** corresponds to the matrix system state $4n + 2q$ x $4n + 2q$; ***B*** and ***E*** are the control and disturbance input matrices; Finally, the vector $\dot{\boldsymbol{z}}(t)$ represents the state of the structural system which contains the relative velocity and responses to acceleration of the structure with respect to the ground. The details of each vector and matrix are listed as follows:

$$\boldsymbol{z}(t)=\begin{bmatrix}\boldsymbol{x}(t)\\ \dot{\boldsymbol{x}}(t)\end{bmatrix},\quad \boldsymbol{A}=\begin{bmatrix}0 & I\\ -\boldsymbol{M}^{-1}\boldsymbol{K} & -\boldsymbol{M}^{-1}\boldsymbol{C}\end{bmatrix},\quad \boldsymbol{B}=\begin{bmatrix}0\\ \boldsymbol{M}^{-1}\boldsymbol{D}\end{bmatrix},\quad \boldsymbol{E}=\begin{bmatrix}0\\ \boldsymbol{M}^{-1}\boldsymbol{H}\end{bmatrix} \tag{4}$$

Where $\boldsymbol{x}$(t) and $\dot{\boldsymbol{x}}(t)$ are system displacement and velocities vectors, respectively. Matrices $\boldsymbol{D}$ and $\boldsymbol{H}$ define the location of control forces and of external loading, respectively.

3 SEMI-ACTIVE CONTROL STRATEGY

The Bang Bang control, also called control ON / OFF control, is a feedback controller that suddenly changes between two limit values. This device compares the input with a target value, so that if the output exceeds the input, the actuator is switched off, otherwise, the actuator is now on. Figure (2) shows an example of a bang bang controller block diagram. This is a low cost controller, further its simplicity and convenience.

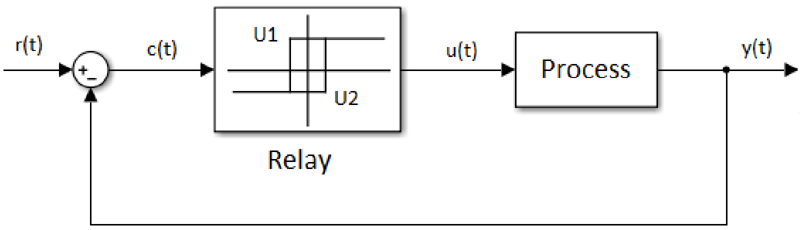


Figure 2. Bang bang control (Moore, 2014).

The control strategy is to control structural response using bang bang control, varying pendulum TMD stiffness k_d and damping c_d, switching from one extreme set of values to the other. The optimal parameter values (k_d and c_d) are obtained based on linear optimal control algorithm (linear quadratic regulator – LQR) (Meirovitch, 1990).

As decision criteria to the bang bang controller to turn on or off, analyzing the time response, the period between two maximum peaks of angular displacement amplitude is calculated and then it can be calculated the response oscillation frequency. Depending on these frequency values it switches the controller ON or OFF.

To define the optimal parameter values (k_d and c_d), first a LQR controller is designed assuming an active pendulum TMD system and neglecting the actuator dynamics. The optimal actuator force $u(t)$ is defined by the gain matrix $\boldsymbol{G}$. The actuator force is not really applied at the TMD, this force is applied through a semi-active damper.

The linear optimal control problem consist in finding the control vector $\boldsymbol{u}(t)$ that minimizes the performance index J subject to the constraint (4). In structural control, the performance index is usually chosen as a quadratic function in $\boldsymbol{z}(t)$ and $\boldsymbol{u}(t)$, as follows

$$J=\int_{t_0}^{t_f}\left[\mathbf{z}^{\mathrm{T}}(t)\mathbf{Q}\mathbf{z}(t)+\mathbf{u}^{\mathrm{T}}(t)\mathbf{R}\mathbf{u}(t)\right]dt \tag{5}$$

Where $\boldsymbol{Q}$ is a $2n$ x $2n$ positive semi-definite matrix and $\boldsymbol{R}$ is a q x q positive definite matrix. The matrices $\boldsymbol{Q}$ and $\boldsymbol{R}$ are referred to as weighting matrices, whose magnitudes are assigned according to the relative importance attached to the state variables and to the control forces in the minimization procedure. The minimization problem leads to a Riccati differential equation system:

$$\dot{\mathbf{P}}(t) + \mathbf{P}(t)\mathbf{A} - \frac{1}{2}\mathbf{P}(t)\mathbf{B}\mathbf{R}^{-1}\mathbf{B}^{\mathbf{T}}\mathbf{P}(t) + \mathbf{A}^{\mathbf{T}}\mathbf{P}(t) + 2\mathbf{Q} = \mathbf{0}, \qquad (6)$$
$$\mathbf{P}(t_f) = \mathbf{0}$$

where $\boldsymbol{P}(t)$ is the Ricatti matrix. The control vector $\boldsymbol{u}(t)$ is linear in $\boldsymbol{z}(t)$. In this case, the linear optimal control law is

$$\mathbf{u}(t) = \mathbf{G}(t)\mathbf{z}(t) = -\frac{1}{2}\mathbf{R}^{-1}\mathbf{B}^{\mathbf{T}}\mathbf{P}(t)\mathbf{z}(t) \qquad (7)$$

where $\mathbf{G}(t) = -\frac{1}{2}\mathbf{R}^{-1}\mathbf{B}^{\mathbf{T}}\mathbf{P}(t)$ is the control gain. In most structural applications, numerical analysis show that Riccati matrix $\boldsymbol{P}(t)$ stays constant during the control range, converging fast to zero on the neighborhood of t_f (Meirovitch, 1990). Thus $\boldsymbol{P}(t)$, in most cases, can be approximated by a constant matrix $\boldsymbol{P}$ and Riccati equation (6) is reduced to

$$\mathbf{PA} - \frac{1}{2}\mathbf{PBR}^{-1}\mathbf{B}^{\mathbf{T}}\mathbf{P} + \mathbf{A}^{\mathbf{T}}\mathbf{P} + 2\mathbf{Q} = \mathbf{0}, \qquad (8)$$

and the constant control gain is given by

$$\mathbf{G} = -\frac{1}{2}\mathbf{R}^{-1}\mathbf{B}^{\mathbf{T}}\mathbf{P} \qquad (9)$$

As it was stated above the control vector $\boldsymbol{u}(t)$ is proportional to state vector $\boldsymbol{z}(t)$ which contains system displacements and velocities. The elements of control gain matrix $\boldsymbol{G}$ that multiplies θ_d and $\dot{\theta}_d$ are considered to set optimum values for pendulum TMD parameters k_d and c_d.

4 NUMERICAL RESULTS

The present work numerical study considered properties of the turbine NREL 5 MW (Stewart et al, 2011), they are exhibited in Table 1.

Table 1: Wind turbine's properties (Stewart et al, 2011)

Rating	5 MW
Rotor, hub diameter	126.3 m

Hub Height	90 m
Rotor mass	110,000 kg
Tower Mass	347,460 kg

Also the following values were considered for system parameters (Guimarães et al, 2014): l = 90 m, m_c = 240,000 Kg; g = 9.81 m/s^2, K = 8.94 x 10^8 N/rad, m = 110,000 kg, m_d = 49,267 kg, d = 79.6 m. The wind dynamic loading was modeled as an applied load at the inverted pendulum mass. Two loading cases were considered: harmonic force and white noise force.

In previous studies (Guimarães et al, 2014), parameter values for pendulum TMD were found out through a parametric study: k_d = 5.9 x 10^6 N/m, c_d = 4.4 x 10^5 Ns/m and l_d = 7.4 m. It results on an efficient vibration control of the wind turbine tower when subjected to a harmonic force approximation of wind load and also for a white noise wind load. In order to improve the performance of this structural control device it is proposed in the present work a semi-active pendulum TMD that varies k_d and c_d properties through a bang bang (ON/OFF) control.

Aiming to design an efficient semi-active controller, the values of parameters k_d and c_d were optimized via classical optimal control algorithm (LQR), assuming an active pendulum TMD system. The optimal actuator force $u(t)$ is defined by the gain matrix $\boldsymbol{G}$, obtained through solution of Riccati equation (Equation 8). It is not really applied at the TMD, this force is applied through the semi-active damper.

Optimal control effectiveness is directly related to a proper choice of the weighting matrices $\boldsymbol{Q}$ and $\boldsymbol{R}$. The flexibleness on this matrices choice allows the generation of a family of multiple different controllers; this represents the great advantage, as well as, the great disadvantage of this method. It is extremely important to carry out a detailed parametric study of weighting matrices in order to ensure control robustness (Avila et al, 2010).

The optimal control algorithm does not lead to a control force truly optimal in certain cases, since the excitation is ignored in obtaining the Riccati matrix (Avila, 2002). Because of this, two loading cases were considered to obtain the optimum control force $u(t)$: harmonic and white noise. Comparing their angular displacement on time history, it was concluded that the parameters for harmonic loading has better results (Avila et al, 2015). The parametric study for harmonic loading case provided the following weighting matrices and the corresponding gain matrix:

$$\boldsymbol{Q} = 5\,x\,10^6 \begin{bmatrix} 1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 \end{bmatrix} \boldsymbol{R} = 10^{-4}$$

$$\boldsymbol{G} = [2.897 * 10^6 \quad 2.924 * 10^4 \quad 1.677 \quad -1.150 * 10^8 \quad 1.725 * 10^6 \quad -2.633 * 10^6]$$

Gain Matrix $\boldsymbol{G}$ coefficients are applied to displacements and velocities $\theta, \theta_d, u, \dot{\theta}, \dot{\theta}_d$ and $\dot{u}$ respectively, however only the coefficients associated to θ_d and $\dot{\theta}_d$, are considered on the semi-active case. The optimum values of k_d and c_d are obtained adding $\boldsymbol{G}$[2] and $\boldsymbol{G}$[5] to passive k_d and c_d values, respectively, obtaining: k_d = 5.9292 x 10^6 N/m and c_d = 2.1654 x 10^6 Ns/m.

Comparing those optimum parameter values to passive parameter values (k_d = 5.9 x 10^6 N/m and c_d = 4.4 x 10^5 Ns/m to the optimum ones it can be observed that a significant increase occurs only for c_d, showing that a semi-active device varying only damping parameters would be satisfactory.

Verifying stability through Routh criterion it was verified that the system is stable. To keep that guaranteed the response should be on small displacement boundaries, keeping the system with linear behavior.

Evaluating system controllability it is verified that all analysed systems are controllable. Controllability is the s the ability of a non-restricted input vector $\boldsymbol{u}$ (t) modify the state vector $\boldsymbol{x}$(t) from an initial state x_0 to any final state x_f during a finite time (Ogata, 2008). In the considered systems, the input vector $\boldsymbol{u}$ (t) is related to the force applied by the TMD.

Evaluating observability property, it can be observed that systems are fully observable. The observability is the ability to determine an initial state vector $\boldsymbol{x}_0$ from observing $\boldsymbol{y}(t)$ over a finite time (Ogata, 2008). In the systems considered, the output vector $\boldsymbol{y}(t)$ is related to the state vector $\boldsymbol{x}(t)$.

Table 2 presents natural frequencies and associated mode vibration to each one of the three systems considered: without contol; OFF position and ON position. The uncontrolled system does not have mode vibration components associated to θ_d, the TMD angular displacement.

Table 2: Natural frequencies and vibration modes

	Natural frequencies (rad/s)	Vibration modes		
		θ	θ_d	u
Uncontrolled	0.8259	1.0000	-	0.0000
	0.0001	-0.9996	-	1.0000
OFF	1.1107	-1.0000	-0.8389	0.0000
	0.6699	-1.0000	0.3104	-0.0000
	0.0001	0.6964	0.9747	1.0000
ON	1.1156	-1.0000	-0.8434	0.0000
	0.6713	-1.0000	0.3035	-0.0000
	0.0001	0.6918	0.9755	1.0000

Figure 3 presents the frequency response of the uncotrolled system (black curve), ON position (red curve) and OFF position (blue curve). It can be noticed that the response on the frequency range of uncontrolled resonance are reduced on both controlled cases. Comparing the ON and OFF curves, depending on the frequency a lower amplitude can be obtained switching from ON to OFF position and vice-versa.

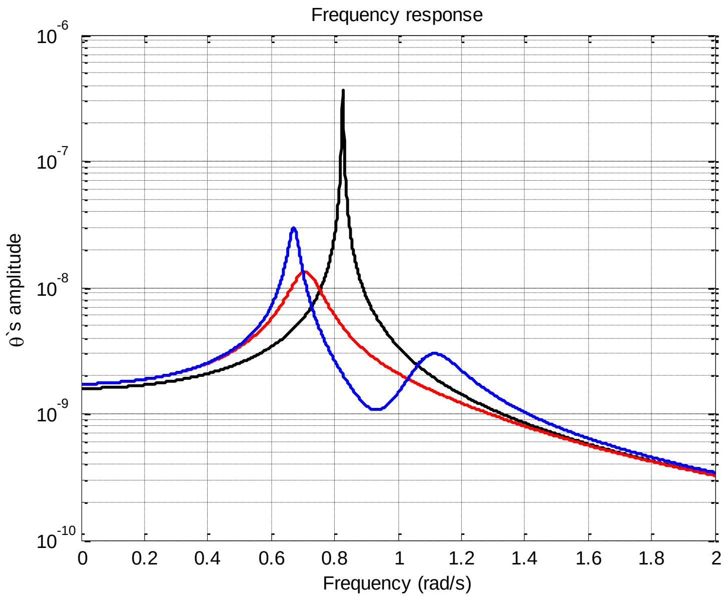


Figure 3. Frequency response of the uncotrolled system (black curve), ON position (red curve) and OFF position (blue curve)

A time domain analysis was performed, three situations were considered for analysis: (1) structure without control, (2) system with passive TMD considered as the semi-active turned OFF and (3) system with semi-active ON, with optimum parameter for harmonic loading. Time domain simulations for harmonic loading were performed varying forcing frequency in a range from 0 to 2 rad/s.

The effectiveness value relative to semi-active device in ON and OFF position is measured by *rms* values as follows:

$$EFF = \frac{\theta_{OFF} - \theta_{ON}}{\theta_{OFF}} \text{ x } 100\ \% \tag{10}$$

Figure 4 shows effectiveness value as a function of forcing frequency. On the frequency range where effectiveness has positive values ON is effective and turns to be ineffective for negative values of efficiency.

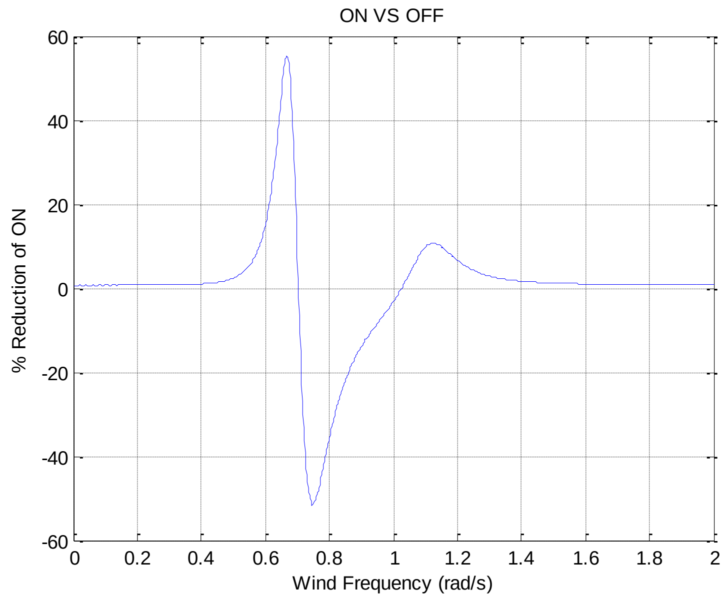
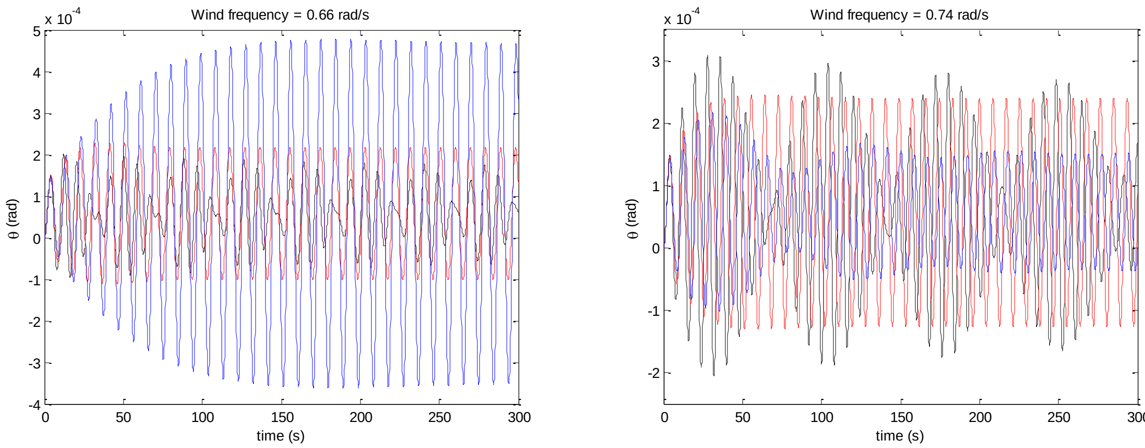


Figure 4. Effectiveness of HSA semi-active control

Figures 5(a) and 5(b) shows angular displacement θ time history for system without control (black curve), OFF case (red curve) and ON case (blue curve), considering forcing frequencies of 0.66 rad/s and 0.74 rad/s, respectively. These frequencies values match to maximum and minimum effectiveness curve peaks of Figure (4).



(a)

(b)

Figure 5. Angular displacement time history with control ON (red curve) and OFF (blue curve) and without control (black curve)

Table 3 presents values of rms percent over uncontrolled system (RMS%) for time evolution displacement response for ON and OFF position considering two different excitation frequencies.

Table 3: RMS percentage displacements.

	RMS% ON	RMS% OFF
$\omega_f = 0.66$	145.84%	313.67%
$\omega_f = 0.74$	106.76%	71.10%

Note that, as expected from Figure 4, to $\omega_f = 0.66\ (rad/s)$ the ON position is more effective, while to $\omega_f = 0.74\ (rad/s)$ the OFF position is more effective on reducing the response. Note also that, as expected from Figure 3, the uncontrolled system presents lower vibration amplitudes for $\omega_f = 0.66\ (rad/s)$, because for this frequency the system without control has a small amplitude response

Figure 6 shows angular displacement time history when the structure is subjected to a white noise wind loading.

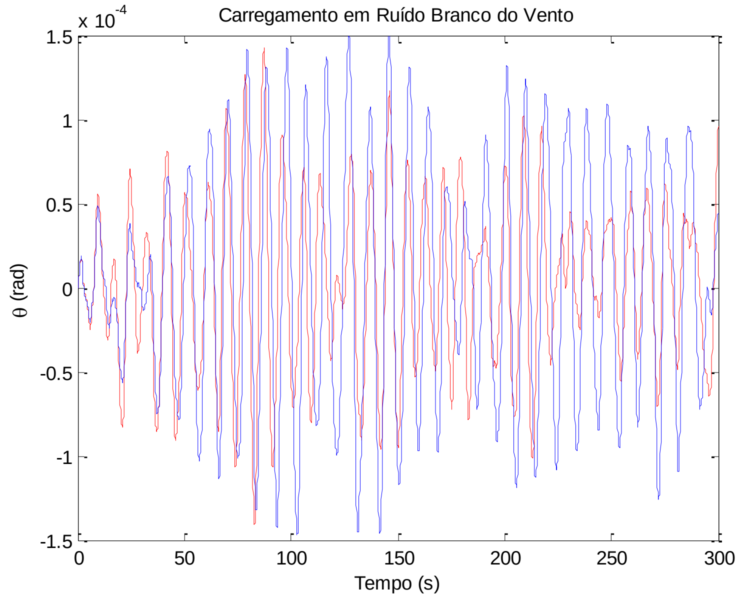


Figure 6. Angular displacement time history with control ON (red curve) and OFF (blue curve): white noise loading.

For this excitation, the RMS% response for ON position was of 18.43%, while for OFF position was of 23.51%. In this case the ON position achieves a better performance.

In a way to evaluate the bang bang control strategy, a harmonic wind force varying its frequency along time was considered. The frequency varies from 0,66 rad/s to 0,74 rad/s after 170 seconds, and then from 0,74 rad/s to 1,5 rad/s after 340 seconds as it can be noticed in Figure 7. The numerical simulation is performed comparing the bang bang results with the results from the system always in OFF position and always in ON position.

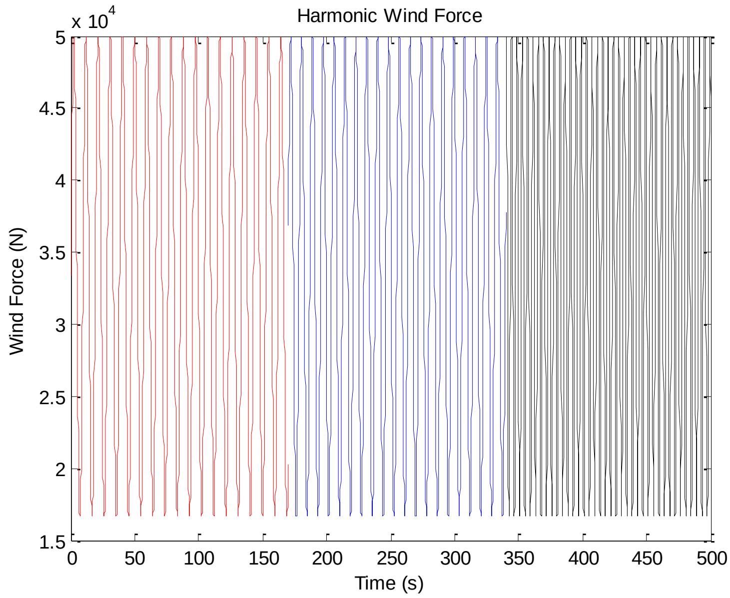


Figure 7. Wind harmonic force time evolution.

As decision criterion to the bang bang controller to turn ON or OFF, analyzing the time response, the period between two maximum peaks of angular displacement amplitude is calculated and then it can be calculated the response oscillation frequency. If the calculated frequency is greater than 0,709 rad/s and less than 1,043 rad/s that are the zeros of the effectiveness curves, bang bang switches to OFF, on the other hand it switches to ON position.

Figure 8 shows angular displacement time evolution to the varying frequency harmonic load.

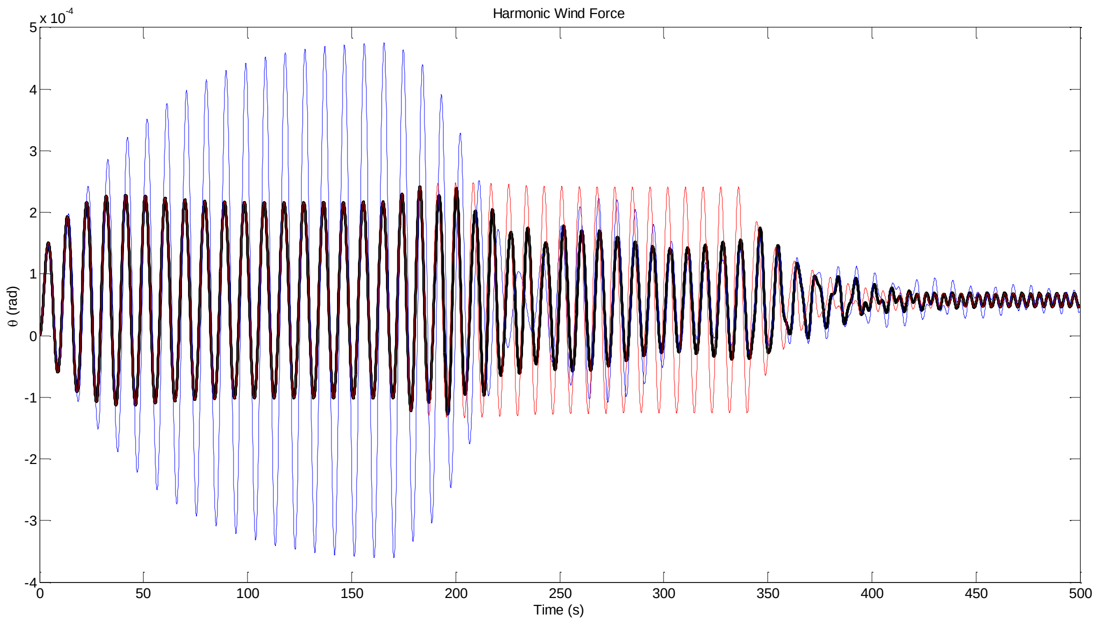


Figura 8. Angular displacement time history with Bang Bang (black curve), control ON (red curve) and OFF (blue curve): harmonic with varying frequency loading.

Table 4 presents RMS% values for ON/OFF positions and Bang Bang over the periods of time: 0s to 170s, 170s to 340s, 340s to 500s and 0s to 500s.

Table 4: RMS% for ON/OFF positions and Bang Bang.

	RMS%			
	0s to 170s	170s to 340s	340s to 500s	0s to 500s
Bang Bang	153.48%	77.24%	22.69%	79.74%
OFF	330,65%	121.90%	25.08%	153.87%
ON	153.48%	114.80%	26.11%	96.30%

It can be observed that the Bang Bang case presented the lower RMS% value in all periods, kept the amplitude vibration next or lower of the other cases during all the time analysed, showing to be efficient relating to the frequency variations of the excitation force, considering the ON/OFF criteria.

Figure 9(a) shows the frequency response for Bang Bang (black curve), control ON (red curve) and OFF (blue curve). It can be observed that the black curve (Bang Bang) has lower amplitudes or is near of the lowest amplitude curve. Figure 9(b) shows a zoom of the frequency response.

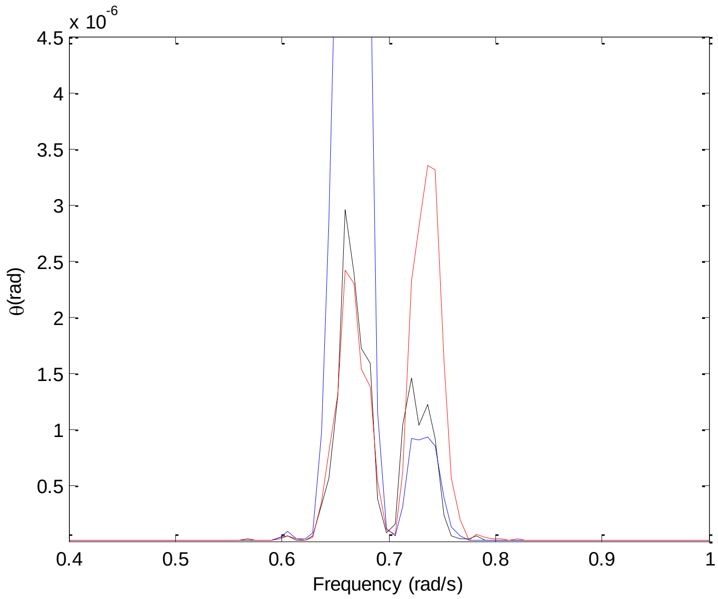


Figura 9: Frequency response for Bang Bang (black curve), control ON (red curve) and OFF (blue curve)

Calculating RMS% displacement response it was obtained the following values: Bang Bang = 81.37%, ON = 93.55% and OFF = 252.05%. It can be noticed that Bang Bang strategy has the better performance compared to the two other cases.

5 CONCLUSIONS

In this work a semi-active TMD pendulum is proposed to control excessive vibration of an offshore floating wind turbine. The structure is modelled as an inverted pendulum discrete model. This model is presented as a preliminary model for structural control analysis, the results serve as a basis for real structure design with a more careful modelling. A bang bang (ON/OFF) control strategy was considered, semi-active TMD stiffness and damping values were calculated trough optimal control theory.

The frequencies values in which ON and OFF positions have better effectiveness were calculated. These frequencies are important to decide when the control keeps ON or OFF position. A decision criterion is proposed by calculating the period between two maximum peaks of angular displacement amplitude and, with it, the response oscillation frequency is calculated. This criterion is evaluated by simulating the system with an harmonic wind force varying its frequency along time, comparing its results to ON and OFF only control. The bang bang strategy had the best performance in all cases considered, which supports the criterion proposed.

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