Passive Control of Spar Type Floating Wind Turbine using Effective Economic Optimal Design Values

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Abstract

This paper studies the performance of a passive linear tuned mass damper on controlling motion of a floating wind turbine. Controlled and uncontrolled analytical models of the spar FWT is established using Newton's second law and conservation of angular momentum theory. The aerodynamic, hydrodynamic, mooring and buoyancy forces are determined and coupled with the system. For the controlled model, the TMDs are located in different locations and are tuned to different frequency ratios to reduce the motion of the FWT in different directions. The economic optimal design values are considered for the passive controller. The control performance is evaluated by the reduction of root mean square response in each degree of freedom. The results reveals that the linear TMD tuned to the frequency of pitch degree of freedom reduces the translational motion and rotational motion by 5% and 12% respectively. However, tuning the linear tuned mass damper to the frequency of surge degree of freedom provides 8% and 6% motion reduction in translational and rotational degrees of freedom. Also, it has been shown that installing the linear TMD inside the spar is more effective than installing the TMD inside the nacelle.

Keywords: Spar floating wind turbine; conservation of angular momentum; linear tuned mass damper; passive control

1. Introduction

Due to the advantages of renewable energies over the fossil fuels, renewable energies have been attractive in different engineering fields [1]-[12]. Offshore wind turbines offer larger renewable power generation in comparison with the onshore wind turbines due to steadier and stronger wind speed in marine areas. In many countries, areas with allowable water depth to install offshore fixed wind turbines are highly limited and hence, floating wind turbines are more attractive than the fixed offshore wind turbines. In addition, due to higher wind speeds and possibility of generating higher electrical power in locations farther from the shore, the wind turbine size has increased which increases the flexibility of the wind turbine structures. Therefore, structural control methods have been investigated to reduce the motion and vibrations of the floating wind turbine.

Various methods have been introduced to reduce vibrations of fixed and floating offshore wind turbines. These controlling methods can be generally divided into three groups: 1- passive, 2- active and 3- semi-active. Passive controlling methods are the simplest form of controlling methods among the other general categories. Passive controllers do not require energy input to reduce the vibrations of the structure. Passive linear TMDs were introduced to reduce the vibrations of offshore wind turbines [13]–[17]. Another type of a passive tuned mass damper, known as tuned liquid column damper was introduced and shown to be effective in reducing the vibrations of offshore wind turbines [18]–[21]. Passive multiple tuned mass dampers have also been investigated by researchers to show the effectiveness of multiple TMDs in reducing vibrations of offshore wind turbines [22], [23]. Recorded met ocean data reveals that real wind and wave loadings are not in one direction. Therefore researchers introduced passive controlling methods which are able to reduce the vibrations in different directions [24]–[28].

In addition to passive controlling methods, active and semi-active controlling methods have also been investigated to reduce the vibrations of the offshore wind turbines. Active tendons were installed inside the blades to reduce the edgewise and flapwise vibrations of the blades [29], [30]. Semi-active controlling methods require much smaller energy input in comparison with the active controlling methods. It was shown that semi-active controlling methods are more effective than passive controlling methods in reducing the vibrations of offshore wind turbines [31]–[34].

The above literature review shows that most mentioned researches have focused on reducing vibrations of fixed offshore wind turbines. Only some studies have studied the vibration reduction of floating wind turbines which mostly have simplified the structural modeling of the floating wind turbine either to in-plane directions or by linear simplified methods. Therefore, this paper studies the effectiveness of a passive linear TMD in reducing vibrations of a spar floating wind turbine. The floating wind turbine is modeled using conservation of momentum method which models the non-linarites of the structure. The aerodynamic, hydrodynamic, hydrostatic and mooring loads are determined via accurate methods. The TMD frequency is tuned to different degrees of freedom frequencies. Also, the linear TMD is placed in different locations to determine the most optimum location in reducing the root mean square response of the floating wind turbine. It is shown that the linear TMD tuned to the frequency of pitch degree of freedom reduces the translational motion and rotational motion by 5% and 12% respectively. However, tuning the linear tuned mass damper to the frequency of surge degree of freedom provides 8% and 6% motion reduction in translational and rotational degrees of freedom. Also, it has been shown that installing the linear TMD inside the spar is more effective than installing the TMD inside the nacelle.

2. Analytical model

In this section, the equations of motion for the floating wind turbine are derived by using conservation of angular momentum and Newton's second law. It is essential to introduce different coordinate systems and to determine the system's equations of motion with respect to the mentioned coordinates. Also, the external forces and moments, which are resulted from buoyancy, hydrodynamic, wind, and mooring cables are determined.

The floating wind turbine is considered as two rigid bodies, the tower and the rotor and nacelle assembly. (X, Y, Z) is earth-fixed global coordinate system located at system's center of mass. (x_1, y_1, z_1) and (x_2, y_2, z_2) are body fixed and are located at first and second body's center of mass respectively. (x_s, y_s, z_s) is the coordinate system for the entire system, which is assumed to be parallel to (x_1, y_1, z_1) and is located on the system's center of mass [35]–[37].

Six unknown degrees of freedom is related to the first body which includes three translational, and 3 rotational degrees of freedom. Additionally, two known degrees of freedom for rotor spinning, φ ,and nacelle yaw, β which describes the relative motion of the second body to the first body.

The schematic image of the floating wind turbine and the coordinates are illustrated in Fig. 1. The misalignment angle between wind and wave loading is denoted by θ.

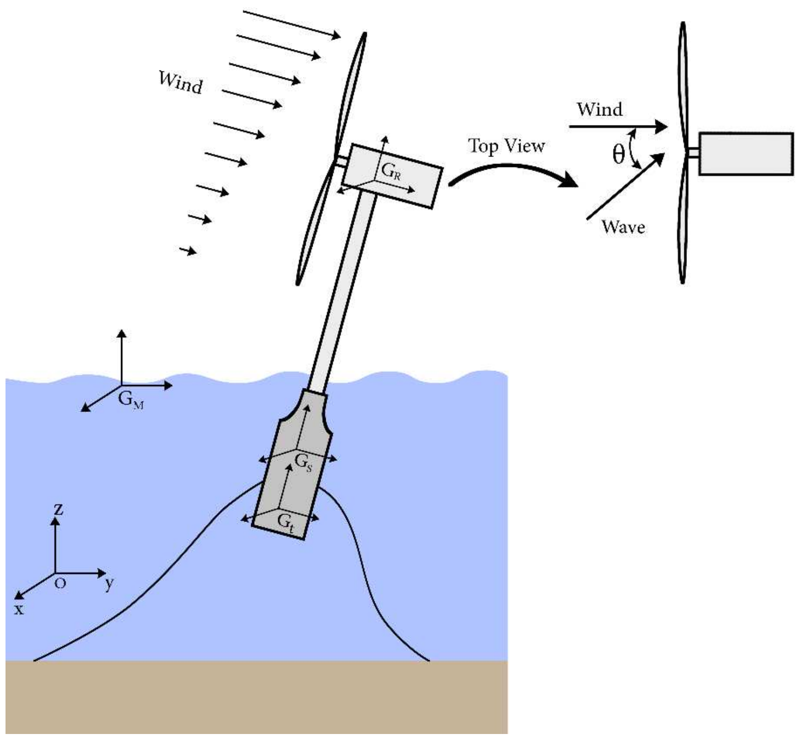

Fig. 1. Schematic image of the spar type FWT

Newton's second law is applied to the system to determine the translational equations:

$$\sum F^{C_I} = (\mathrm{m}_1 + \mathrm{m}_2)\,\mathrm{a}_{G_s}^{C_I} \tag{1}$$

where $\sum F^{C_I}$ is the summation of all external forces. Also, m_1 and m_2 denote the mass of first and second body respectively. Additionally, the linear acceleration of the system which is shown as $\mathrm{a}_{G_s}^{C_I}$ is equal to $\left[\ddot{X}_1, \ddot{X}_2, \ddot{X}_3\right]$.

To determine the rotational equations of motion, the conservation of angular momentum theory is used as follows:

$$\sum \vec{M} = \dot{\vec{\mathrm{H}}}_{G_s}^{s} + \vec{\omega}_{B_1}^{C_1} \times \vec{H}_{G_s}^{s} \tag{2}$$

where $\sum M$ is the external moment resulted from external loadings. Also, $\vec{H}_{G_s}^{s}$ denotes the angular momentum of entire system in C_s coordinate system. $\dot{\vec{\mathrm{H}}}_{G_s}^{s}$ is the change of momentum amplitude for the entire system and $\vec{\omega}_{B_1}^{C_1}$ describes the absolute angular velocity. The total angular momentum is equal to the summation of angular momentum of each body.

The angular momentum of the system can be determined as follows:

$$\vec{H}_{G_s}^{s} = I_s \vec{\omega}_s + \vec{H}' \tag{3}$$

where I_s is the inertia tensor and $\vec{H}' = (-I_B \dot{\psi} \sin\beta, I_B \dot{\psi} \cos\beta, I_C \omega_{yaw})$. The absolute time derivative of angular momentum of the system can be determined as:

$$\dot{\vec{H}}_{G_s}^{s} = \dot{I}_s \vec{\omega}_s + I_s \dot{\vec{\omega}}_s + \dot{\vec{H}}' \tag{4}$$

By substituting the angular momentum of the system and its derivative into the newton's second law and conservation of angular momentum theory, the equations of the motion can be written in a matrix form as:

$$\tilde{M}\ddot{\tilde{X}} + \tilde{C}\dot{\tilde{X}} + \tilde{K}\tilde{X} = \tilde{Q}_{wave} + \tilde{Q}_{wind} + \tilde{Q}_{buoy} + \tilde{Q}_{mooring} + \tilde{Q}_{gravity} \tag{5}$$

It should be noted that, $\tilde{M}$, $\tilde{C}$ and $\tilde{K}$ are the mass, damping and stiffness matrixes which are time variant. Variables $\tilde{Q}_{wave}, \tilde{Q}_{wind}, \tilde{Q}_{buoy}, \tilde{Q}_{mooring}, \tilde{Q}_{gravity}$ are the force and moment vectors resulted from the external loadings.

3. Loading

The mooring forces, buoyancy force, hydrodynamic and wind loads are derived in this section

3.1. Wind Load

The most dominant load on offshore wind turbine is the wind loading. Therefore, it is important to model this force accurately. The mean velocity is calculated using logarithmic wind profile:

$$\overline{v}(z) = \frac{V_{hub} \log(z / z_0)}{\log(H_{ref} / z_0)} \tag{6}$$

where V_{hub} is the mean velocity at the hub height. z_0 is the roughness length. The Kaimal spectrum model is used to model the turbulent component of the wind.

After determining the wind velocity on the wind turbine, the blade element momentum method is used to determine the lift and drag forces acting on the wind turbine blade. The lift and rag forces can be expressed as follows:

$$F_L = \frac{1}{2}\rho_a V_{rel}^2 c C_l,\ \mathrm{F}_D = \frac{1}{2}\rho_a V_{rel}^2 c C_d \tag{7}$$

where ρ_a is the density of air and c is the chord length and the C_N and C_T are the normal and the tangential coefficients.

By developing MATLAB codes and using the principle of virtual work, the virtual work done by the wind loading and generalized wind loading can be determined.

3.2. Wave load

The wave force acting on circular cylindrical structural members can be estimated using Morison's equation as follows:

$$\vec{f}_M^I = \frac{1}{2}\rho C_d D\vec{V}_{rt}\left|\vec{V}_{rt}\right| - C_a\rho\frac{\pi}{4}D^2\dot{\vec{V}}_t + C_m\rho\frac{\pi}{4}D^2\dot{\vec{V}}_n \tag{9}$$

where C_d , C_a and C_m are the drag, added mass and inertia coefficient respectively. Also, ρ is the sea water density and D is the diameter of the hull. $\dot{\vec{V}}_t$, $\dot{\vec{V}}_n$ and $\vec{V}_{rt}$ are the normal component of wave acceleration, structural acceleration and velocity of the water particle relative to the tower. The normal component of water particle acceleration is determined by: $\dot{\vec{V}}_n = \vec{u}_z^I \times (\dot{\vec{V}} \times \vec{u}_z^I)$. Parameter $\dot{\vec{V}}$ represents the wave acceleration vector in the C_I coordinate system and $\vec{u}_z^I$ is the unit vector along the central axis of the tower in the C_I coordinate system, which is equal to: $\vec{u}_z^I = T_{s\to I}(0,0,1)$.

The structural velocity and acceleration are calculated by the following equations:

$$\vec{V}_t = (X_1, X_2, X_3) + T_{s\to I}(\vec{\omega}_{B_1}^{C_1} \times \vec{r}_{i/G_s}) \tag{10}$$

$$\dot{\vec{V}}_t = (\dot{X}_1, \dot{X}_2, \dot{X}_3) + T_{s\to I}(\dot{\vec{\omega}}_{B_1}^{C_1} \times \vec{r}_{i/G_s} + \vec{\omega}_{B_1}^{C_1} \times (\vec{\omega}_{B_1}^{C_1} \times \vec{r}_{i/G_s})) \tag{11}$$

Where $\vec{r}_{i/G_s}$ is the vector radius from CM of the system to the segment with unit length. The normal velocity of the water particle relative to the tower is expressed as $\vec{V}_{rt} = \vec{u}_z^I \times (\vec{V}_{rwt} \times \vec{u}_z^I)$. The relative velocity of the wave to the segment of the submerged tower is shown as $\vec{V}_{rwt} = \vec{V} - \vec{V}_t$ in which $\vec{V}$ is the wave kinematic velocity.

To define the velocity and acceleration of the water particles, it is considered that the sea waves have a regular sinusoidal shape. According to this theory the surface elevation, velocity and acceleration of particles are computed:

$$\eta(y,t) = \frac{H}{2}\cos\left(ky - \omega' t\right) \tag{12}$$

$$V = (0,u,v)$$
$$\begin{cases} u = \dfrac{\pi H}{T} \times \dfrac{\cosh k(x+d)}{\sinh kd}\cos(ky-\omega' t) \\ v = \dfrac{\pi H}{T} \times \dfrac{\cosh k(x+d)}{\sinh kd}\sin(ky-\omega' t) \end{cases} \tag{13}$$

$$\dot{V} = (0,\dot{u},\dot{v})$$
$$\begin{cases} \dot{u} = \dfrac{2\pi^2 H}{T^2} \times \dfrac{\cosh k(x+d)}{\sinh kd}\sin(ky-\omega' t) \\ \dot{v} = -\dfrac{2\pi^2 H}{T^2} \times \dfrac{\cosh k(x+d)}{\sinh kd}\cos(ky-\omega' t) \end{cases} \tag{14}$$

where H , T , k and ω' are wave height, time period of wave, wave number and angular wave frequency respectively. The wave force on the platform of the system and the wave moment in the C_s system is computed by numerically integrating over the submerged length of the tower as follows:

$$\begin{aligned}\vec{F}_{wave} &= \int \vec{f}_M^I \\ \vec{M}_{wave} &= \int (\vec{r}_{i/G_s} \times (T_{I\to s}\vec{f}_M^I)dz\end{aligned} \tag{15}$$

The wave loading along surge and sway by considering the wind-wave misalignment angle, β , can be expressed as:

$$\begin{aligned}F_{wave,surge} &= F_{wave}\cos\beta \\ F_{wave,sway} &= F_{wave}\sin\beta\end{aligned} \tag{16}$$

3.3.Mooring forces

To determine the mooring load, a dynamic linking library, Moordyn [38], is coupled to the MATLAB codes. This model uses a lumped mass approach to discretize the cable dynamics over the length of the mooring line.

3.4.Buoyancy Load

The instantaneous buoyancy of the floating system can be expressed as follows:

$$\vec{F}_B^I = (0, 0, \rho g \pi r^2 h_1) \tag{18}$$

where ρ is the water density, r is the radius of the cylinder and h_1 is the instant submerged length of the cylinder,

To determine the moments, the center of buoyancy should be calculated, if the distance between center of buoyancy and CM of the system is assumed as $\vec{r}_{B/G_s} = (\xi_B, \eta_B, \zeta_B)$, the distance can be expressed as:

$$\begin{cases}\xi_B = -\dfrac{tt_{31}r^2}{4tt_{33}h_1} \\ \eta_B = -\dfrac{tt_{32}r^2}{4tt_{33}h_1} \\ \zeta_B = -h_G + \dfrac{h_1}{2} + \dfrac{r^2(tt_{31}^2 + tt_{32}^2)}{8tt_{33}^2 h_1}\end{cases} \tag{19}$$

The resulting external moment is defined as $\vec{M}_B^s = \vec{r}_{B/G_s} \times T_{C_I\to C_s}\vec{F}_B^I$

4. Simulation and Results

The NREL 5MW spar type wind turbine model is used as a case study. The wind speed is considered to be 12 *m/s* with a turbulence intensity of 10%. The significant wave height is 2 *m* with a 10 *s* period. The wind time history for the mentioned case is demonstrated in Fig. 2. It should be noted that the wind time history is generated in three different axes and the wind turbine is facing the wind speed with higher values.

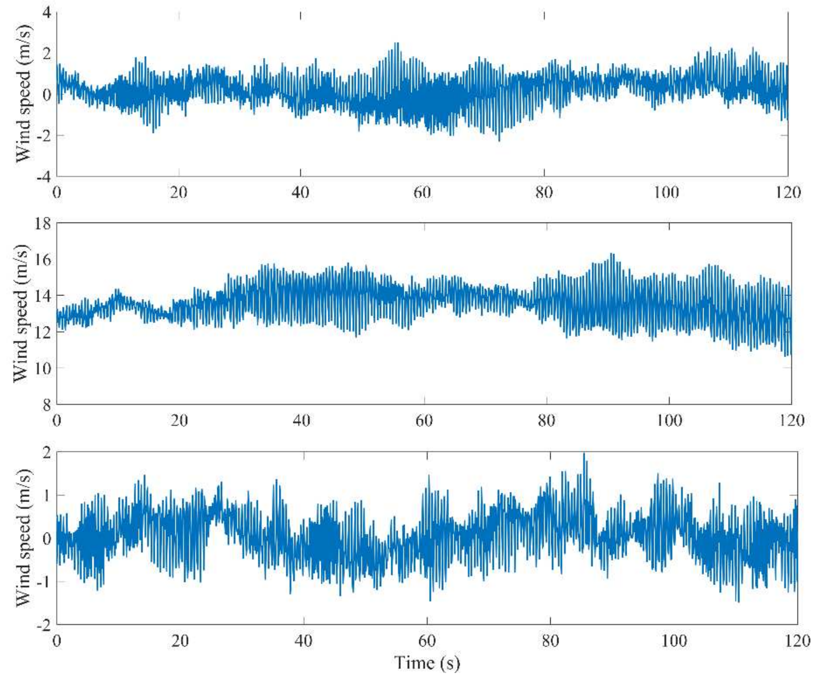

Fig. 2. Three dimensional wind time history ($v = 12m/s$ and turbulence intensity of 10%)

Fig. 3 shows the translational and rotational motion reduction between the baseline wind turbine and controlled case. The passive linear TMD is installed inside the spar and the frequency is tuned to 0.008 Hz which is the natural frequency of the wind turbine system in translational degree of freedom. It can be seen that the motion of the floating wind turbine in surge and pitch degree of freedom has reduced. It should be noted that the motion of the floating wind turbine in other degrees of freedom are not reduced since the linear TMD is only coupled to the in-plane vibrations.

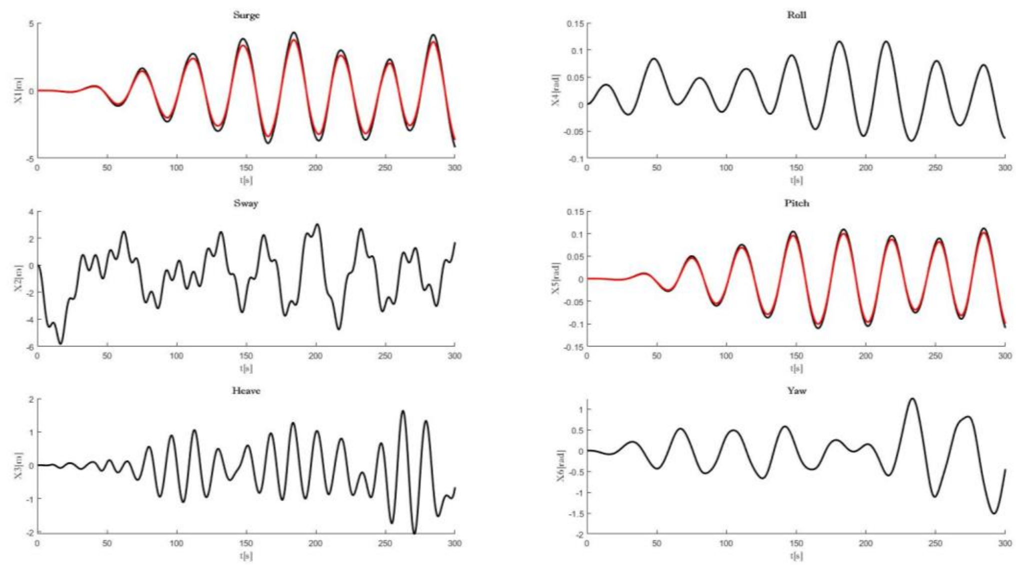

Fig. 3. Vibration mitigation of FWT with linear TMD frequency tuned to 0.008 Hz.

Fig. 4 shows the translational and rotational motion reduction between the baseline wind turbine and controlled case. The passive linear TMD is installed inside the spar and the frequency is tuned to 0.03 Hz which is the natural frequency of the wind turbine system in rotational degree of freedom. It can be seen that the motion of the floating wind turbine in surge and pitch degree of freedom has reduced. But, the reduction in pitch direction is more significant than the reduction in surge direction.

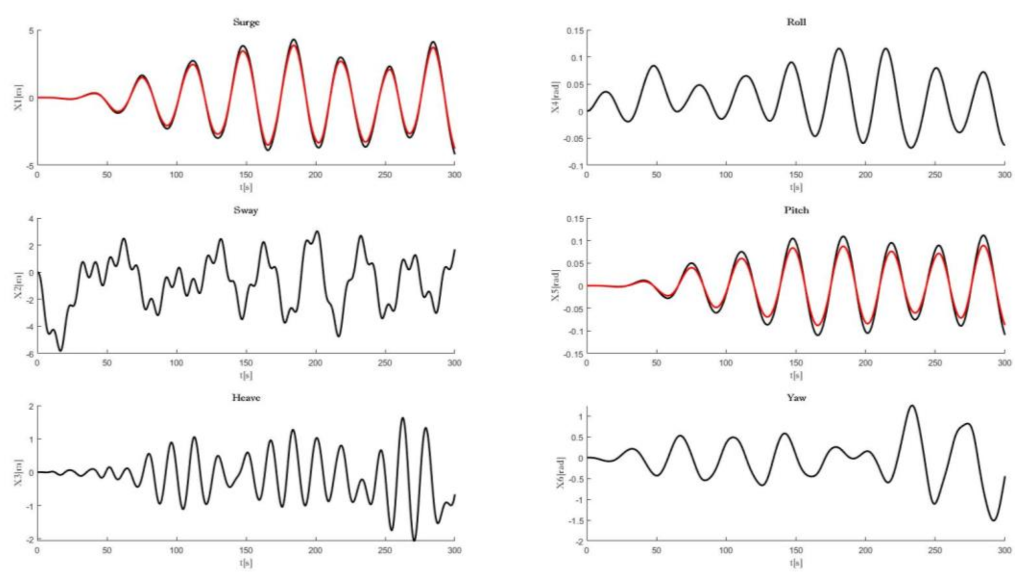

Fig. 4. Vibration mitigation of FWT with linear TMD frequency tuned to 0.03 Hz.

Fig. 5 shows the surge and pitch motion reduction between the baseline wind turbine and controlled case. The passive linear TMD is installed inside the nacelle and the frequency is tuned to 0.03 Hz which is the natural frequency of the wind turbine system in rotational degree of freedom. It can be seen that the motion of the floating wind turbine in surge and pitch degree of freedom has reduced. By comparing the results presented in Fig. 5 and Fig. 4, it can be concluded that installing the TMD inside the spar is more effective than installing it inside the nacelle.

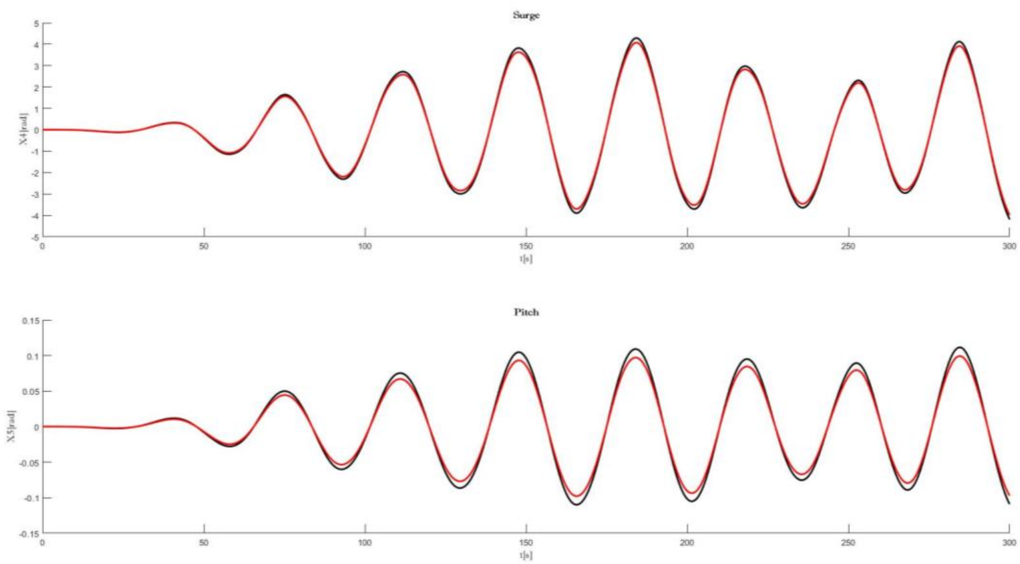

Fig. 5. Vibration mitigation of FWT with linear TMD frequency tuned to 0.03 Hz and installed inside the nacelle.

5. Conclusion

In this paper, the spar floating wind turbine is modeled using conservation of momentum method which considers the nonlinearities of the system in the model. A linear tuned mass damper is installed in nacelle and the platform. The TMD is also tuned to different frequencies to evaluate the effectiveness of the TMD in reducing the vibrations of the floating wind turbine in different degrees of freedom. It is shown that the linear TMD tuned to the frequency of pitch degree of freedom reduces the translational motion and rotational motion by 5% and 12% respectively. However, tuning the linear tuned mass damper to the frequency of surge degree of freedom provides 8% and 6% motion reduction in translational and rotational degrees of freedom. Also, it has been shown that installing the linear TMD inside the spar is more effective than installing the TMD inside the nacelle.

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