Wind and Structures, *Vol. 22, No. 1 (2016) 107-131*
DOI: http://dx.doi.org/10.12989/was.2016.22.1.107

Vibration control in wind turbines for performance enhancement: A comparative study

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(Received October 19, 2014, Revised November 16, 2015, Accepted November 20, 2015)

Abstract. The need for a more affordable, reliable, clean and secure energy has led to explorations in non-traditional sources, particularly renewable energies. Wind is one of the cleanest energy sources that plays a significant role in augmenting sustainability. Wind turbines, as energy convertors, are usually tall and slender structures, and depending on their location (inland or offshore), they can be subject to high wind and/or strong wave loadings. These loads can cause severe vibrations with detrimental effects on energy production, structural lifecycle and initial cost. A dissipativity analysis study was carried out to know whether wind turbine towers require damping enhancement or rigidity modifications for vibration suppression. The results suggest that wind turbines are lightly damped structures and damping enhancement is a potential solution for vibration lessening. Accordingly, the paper investigates different damping enhancement techniques for vibration mitigation. The efficacy of tuned mass damper (TMD), tuned liquid column damper (TLCD), tuned sloshing damper (TSD), and viscous damper (VD) to reduce vibrations is investigated. A comparison among these devices, in terms of robustness and effectiveness, is conducted. The VD can reduce both displacement and acceleration responses of the tower, better than other types of dampers, for the same control effort, followed by TMD, TSD, and finally TLCD. Nevertheless, the use of VDs raises concerns about where they should be located in the structure, and their application may require additional design considerations.

Keywords: dissipativity analysis; wind turbine; viscous dampers; vibration control; tuned mass dampers; liquid dampers; high-rise structures

1. Introduction

Everyday witnesses increased demand in energy, which requires further investigations on available sources of energy, especially renewable energy that was highlighted in the last decades as per environmental and sustainability demand. Wind energy is one of the cleanest energies that received the attention of researchers and investors because of its availability with low running cost. According to the World Wind Energy Association (WWEA 2015), wind energy is currently the fastest-growing source of electricity in the world, and wind power investment worldwide is expected to expand three-fold, from about $18 billion in 2006 to $60 billion in 2016.

Although the use of wind energy backs to more than hundreds of years (the wind wheel of Heron of Alexandria in the first century AD is the earliest known instance of using a wind-driven

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http://www.techno-press.org/?journal=was&subpage=8 ISSN: 1226-6116 (Print), 1598-6225 (Online)

wheel to power a machine (Drachmann 1961, Lohrmann 1995)), it is yet an immature and growing technology. In recent years, coastal and offshore wind turbines are growing fast, as an attractive means for clean energy production. Due to the nature of the wind in offshore regions, that is powerful and more uniform than inland winds, offshore wind turbines can produce energy effectively (Musial and Butterfield 2006). It should be noted that the extracted energy is proportional to the cubic of the wind speed which means an increase in wind speed has a significant influence on the amount of produced energy. In addition, it is more cost effective to place power generators on a short distance to the market. Hence, considering the fact that the population density is much higher in the coastal area (according to the National Oceanic and Atmospheric Administration (NOAA 2015), 53% of the nation's population of the U.S. lives in coastal areas), it is more economical to use offshore wind turbines instead of inland ones. In the U.S., where wind currently only provides about 1% of the nation's electricity needs, wind has the potential to provide up to 20% of these needs without major changes to the distribution system (Laks *et al.* 2009). The first offshore wind turbine with a fixed foundation was installed off the coast of Denmark in 1991 (Arapogianni and Genach 2013). The first large scale floating offshore wind turbine installed in Norway in 2009 and started to produce energy in 2012 and based on the information presented in Arapogianni and Genach (2013), 40 deep water offshore turbines were installed throughout the world. It is forecasted by the European Wind Energy Association (EWEA) that by 2030, 150 GW of installed offshore wind capacity will be available, enough to power 145 million households (Arapogianni and Genach 2013). However, because of the wind turbine structures' operational conditions and the wind-induced loads (winds do not always blow as the ships wish), there are still many unsolved challenges in expanding wind power and specific attention should be paid during design, construction, installation and maintenance phases of these structures.

One of the important challenges regarding wind turbines' design is to control the structural vibration. Generally, wind turbines are slender structures that are exposed to moderate to high wind loads. For offshore wind turbines, the environmental condition is harsher while they are under the action of a combination of wind and wave. The coupled aero-hydroservo-elastic time domain analysis of these structures is time consuming, particularly in connection with the aerodynamics, controller and the elastic formulation of the structure (Karimirad and Moan 2012). The combination of wind and wave forces may produce extreme vibrations and fatigue loads in the blades, support structure, and other components which can have detrimental effects both on the extracted energy efficiency of the generator and the structural components of the tower such as its fatigue life. For instance, fatigue loads can lead to increased maintenance, reduced availability, more expensive components, and failures (Bossanyi 2003, Veers 2003).

By introducing wind-induced acceleration fragility curves, the study presented by Dueñas-Osorio and Basu (2008) shows that wind turbines are prone to failure from acceleration-sensitive components. They concluded that high levels of accelerations can induce malfunctioning of acceleration-sensitive components, such as, generators, inverters, yaw systems, gearboxes, hydraulic systems, and mechanical brake systems, resulting in reduced annual wind turbine availability. According to their study, the functionality of generators can be affected in the range of 5 to 10 m/s^2, inverters in the range 10 to 15 m/s^2, and electrical controls in the range 15 to 20 m/s^2. They predicted high acceleration levels for the nacelle at the onset of a shutdown wind speed criteria (wind speed = 25 m/s) and showed that for acceleration thresholds of 5, 10, 15, and 20 m/s2, the probabilities of being exceeded are 86%, 64%, 48% and 37%, respectively.

In addition, Ronold and Larsen (2000), Debbarma *et al*. (2010), and Chakraborty and Roy

(2011) are some of other studies that addressed the reliability design of wind turbines. Hence, suppressing unwanted vibrations in wind turbines may lead to an increase in inland and offshore wind turbines' reliability.

2. Dissipativity analysis of a 5 MW NREL wind turbine

As a reference, the 5 MW NREL wind turbine characteristics have been used in the current study (Table 1) (Jonkman *et al.* 2009). According to Jonkman *et al.* (2009) the first fore-aft and side-side natural frequencies of the full-scale wind turbine are 0.322 Hz and 0.314 Hz respectively. The overall generalized mass is assumed placed at a height of 38.2 m. The base diameter of the tower is 6 m and the top diameter is 3.87 m and the structural damping ratio is 1%. The load is considered as external white-noise acting on the primary structure, with a 1Hz cutoff frequency, in order to compare the effectiveness of different vibration suppression methods

The term 'dissipativity analysis' is relatively new to the structural control community. The literature has very little work that can help classify the nature of a control force as being dissipative or non-dissipative. Inaudi (2000) proposed a stochastic index that estimates the probability of the primary control force being dissipative (this index will be defined later). This index was justified by numerical simulations later in Christenson (2003); and the term 'dissipativity analysis' was used (for instance, in Erkus 2011). The dissipativity analysis of an active control force is very important as it can permit the understanding of the performance of both passive and semi-active control systems. As an active control force can be usually optimal (e.g., using a Linear Quadratic Regulator (LQR)), a highly dissipative control force imply that a passive damper can be used to provide the control force in a cheaper way.

The purpose of this dissipative analysis study is to permit the understanding of whether the wind turbine requires damping enhancement or modifying rigidities to reduce a certain response, say the acceleration of the rotor under wind loads. To do so, one can imagine the system modeled as a single-degree-of-freedom (SDOF) spring-mass-dashpot (Fig. 1(a)). The governing equation of this system with a general control device providing a force F can be written as

$$M\ddot{U} + C\dot{U} + KU = P - F \tag{1}$$

where M is the mass, U is the displacement response, C is the damping coefficient, K is the stiffness, P is the excitation force, and F is the control force. Now imagine the system is including an actuator with the purpose to provide a control force (F) that is proportional to the displacement and the velocity response of the primary structure, i.e.

$$F = k_\mathrm{d}U + k_\mathrm{v}\dot{U}. \tag{2}$$

The active controller will provide two gains say k_d (displacement gain) and k_v (velocity gain). This said, the actuator will actually tend to modify both the rigidity (K) and the damping (C) of the primary system by the values k_d and k_v, respectively. From a classical control feedback theory (Soong 1990), and considering the optimization objective to be the acceleration of the system, the control force gains k_d and k_v can be obtained. Now the single-degree-of-freedom (SDOF) system can be tested with a white-noise excitation (P) to permit the understanding of whether damping and/or rigidity modifications has the most significant influence on the reduction of the acceleration response.

Table 1 NREL reference wind turbine properties (Jonkman *et al*. 2009)

Parameter	**Value**
Rating	5 MW
Rotor orientation, configuration	Upwind, 3 blades
Control	Variable speed, multiple-stage gearbox
Rotor, hub diameter	126 m, 3 m
Hub height	90 m
Cut-in, rated, cut-out wind speed	3 m/s, 11.4 m/s, 25 m/s
Cut-in, rated rotor speed	6.9 rpm, 12.1 rpm
Rated tip speed	80 m/s
Overhang, Shaft Tilt, Precone	5 m, 5°, 2.5°
Rotor mass	110,000 kg
Nacelle mass	240,000 kg
Tower mass	347,460 kg

The modifications in the damping and/or rigidities of the primary structure can be actually obtained by providing an active control force as mentioned above. The displacement gain will be responsible for the rigidity modification by providing non-dissipative control force and the velocity gain will provide dissipative control force all time. For more information on dissipative and non-dissipative control forces, the reader may refer to Inaudi (2000). The probability that the active control force F is dissipative $Pr(F\dot{U} < 0)$ can be predicted based on the values of the displacement and velocity gains (k_d and k_v), provided by the classical active control system (Inaudi 2000). These values will lead to a closed-loop system for which the probability that the control force is dissipative can be expressed as

$$\Pr(F\dot{U} < 0) = \frac{\cos^{-1}(\rho_{\dot{U}F})}{\pi} \tag{3}$$

the correlation coefficient $\rho_{\dot{U}F}$, between the control force F and the velocity $\dot{U}$, on the right hand side of Eq. (3), can be determined from the covariance matrix of the output, using the covariance matrix of the states as a solution of the continuous time Lyapunov equation (e.g., in MATLAB (Attaway 2013) using the command *lyap.m*). Details on obtaining the correlation coefficient $\rho_{\dot{U}F}$ are provided in Aly and Christenson (2008).

One may consider the normalized acceleration response of the SDOF system, which is the ratio of the controlled response to the uncontrolled one, as an optimization objective. Fig. 1 shows this optimization objective versus a varying weight on the velocity gain (viscous damper). This viscous control force can be transformed to a corresponding change in the damping factor ζ of the primary structure as

$$\xi = k_v / 2\omega M \tag{4}$$

in which ω is the natural frequency of the primary structure and M is the generalized mass. The figure shows that viscous dampers can significantly reduce the acceleration response by increasing the damping up to say 20%. After this value there is no significant reduction in the acceleration response, and extra damping can lead to increase in the response as this tend to block the vibratory motion of the primary structure, by providing aggressive damping forces.

Now let's go back to the active controller and consider providing the same force by the above mentioned viscous damper by both damping and displacement gains (i.e., an active control force that tend to modify both rigidity and damping of the primary structure, with the objective to reduce the acceleration response). By increasing the values of the control gains, the active controller will provide a larger portion of the force as non-dissipative, which is witnessed by a lower dissipativity (probability of having dissipative control force) (Aly and Christenson, 2008; Erkus and Johnson, 2011). The response in Fig. 1 (active) shows that the performance of the active controller (i.e. modifying both stiffness and damping of the primary structure) can be better than just modifying damping (viscous) at higher total control forces. The figure also shows that if this control force was provided by a semi-active (on-off) damping device, the performance can be better than the viscous one but less than that of the active controller at relatively high control force. The performance of the semi-active control system using a clipped-optimal controller was predicted in this study using an equivalent semi-active system presented in Aly and Christenson (2008). The equivalent semi-active control system is a linearization of the on-off active control system and has two equivalent energy-based semi-active control gains ($k_{d,s}$ and $k_{v,s}$). The semi-active control gains as a function of the probability that the control force is dissipative Pr (Eq. (3)) and the active control gains (k_d and k_v) can be written as

$$k_{d,s} = \frac{1}{2} k_d \;\; 1 + \sin^2[\pi \;\; \mathrm{Pr} - 0.5] \tag{5}$$

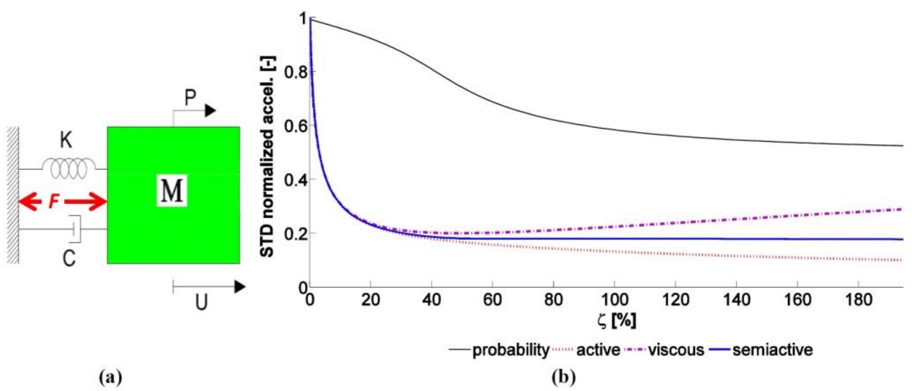

Fig. 1 Dissipativity analysis study: (a) single-degree-of-freedom (SDOF) system and (b) performance of different control systems (active, viscous and semiactive) in reducing the normalized acceleration response with the probability that the active control force is dissipative, versus the corresponding structural damping increase (ζ). The legend 'probability' refers to a stochastic index that estimates the probability of the primary control force (active) being dissipative

$$k_{v,s} = k_{v,s,1} + k_{v,s,2} \tag{6}$$

$$k_{v,s,1} = \frac{|k_d|}{\pi\omega_s} \left(1 = \sin^2\left(\pi(\mathrm{Pr} - 0.5)\right)\right) \tag{7}$$

$$k_{v,s,2} = \frac{k_y\omega_a}{\pi\omega_a}\left\{\frac{\pi}{2} + \left[\sin\left(\pi\left[\left(\mathrm{Pr} - |0.5|\right)\right]\right)\sqrt{1-\sin^2\left(\pi\left[\mathrm{Pr}-0.5\right]\right)} + \tan^{-1}\left(\frac{\sin\left(\pi\left[\mathrm{Pr}-0.5\right]\right)}{\sqrt{1-\sin^2\left(\pi\left[\mathrm{Pr}-0.5\right]\right)}}\right)\right]\right\} \tag{8}$$

$$\omega = \sqrt{\frac{K + k_{d,s}}{M}};\quad \omega_a\sqrt{\frac{K + k_d}{M}} \tag{9}$$

In any case, the performance of the active, the semi-active and the viscous systems at lower control forces (say less than 20% corresponding damping factor) are very similar. In fact, in civil engineering applications, aggressive control forces are not common and the control system may provide a total damping enhancement in the structure of say 2-10%. This control force can be conveniently provided by viscous dampers or a system that can result into a total increase in the damping rather than trying to modify the rigidity and damping by external active/semi-active forces, when the objective is to reduce the acceleration. Based on this conclusion, in the current study, different control systems that attempt to increase damping in the wind turbine primary structure will be considered. Also, reducing the acceleration response will reduce the inertia forces, and most importantly will lead to an improved wind turbine operation.

3. Damping enhancement

In order to reduce the wind turbines' wind-induced displacements, the structure may be as rigid as possible; however, this approach is not the best option as per both structural operations (accelerations can be higher for rigid structures) and commercial concerns (expensive solution). On the other hand, an attractive alternative to reduce vibrations in tall structures is to use external dampers. Dampers are systems that produce dissipating forces on the structure at particular frequencies, which results in a significant reduction in the structure's vibration and motion-induced loads.

There are different types of dampers that generally can be classified into three major categories: passive, active and semi-active dampers. An example of passive dampers is the tuned mass damper (TMD), which is usually tuned to a specific frequency, the dominant natural frequency of the structure in most of the cases, to protect the primary structure and prevent the resonance phenomenon. In this type of damping, the damper is tuned once and no adjustments will be done during its operation. In contrast, active tuned mass dampers (ATMDs) involve the use of different types of actuators (Soong 1990, Chen *et al.* 2011). The active force allows realizing both the precise frequency and damper tuning at the same time. However, in order to control a primary structure subjected to excitation, external forces are needed, therefore the power consumption can be a significant concern in this approach (Choi *et al.* 2008, Weber 2014). To overcome this drawback, hybrid schemes were considered with different combinations and configurations to improve the efficacy of the control system (Park and Ok, 2015). In addition, many semi-active tuned mass damper (STMD) concepts have been developed. In a semi-active damper, the actuator's target is to adjust the frequency of the damper to the dominant frequency of the structure. Controllable friction dampers have also been implemented in STMDs to control their energy dissipation or to increase the relative motion of STMDs for higher efficiency (Lin *et al.* 2010a, b). The semi-active damper cannot inject mechanical energy into the structural system, i.e., requires minimal operating power, but it can adjust damping to reduce unwanted vibration of the systems with lower performance than the counterpart active control system (Choi *et al.* 2008).

During the last decades, different damping enhancement technologies were explored for vibration suppression, however, up to now, most of the damper systems that have been suggested for these structures are based on passive systems. In what follows some of these systems are addressed.

3.1 Tuned mass dampers (TMDs)

The tuned mass damper (TMD) is a mass spring system with oil damper that is installed on the vibrating primary structure at anti-node position to reduce vibrations (Den Hertzog 1934). The principle is tuning the mass (which is usually about 2% of the primary structure) to a particular structural frequency, so that when that frequency is excited, the damper will suppress the vibration by it's out of phase motion. The concept of TMD was first applied by Frahm in 1909 in ships (Connor 2003). The applications of TMDs are considered widely by researchers and have received attention more than other types of dampers due to their simplicity and efficiency, especially at the target response frequency. A SDOF system and its schematic location at the nacelle of a wind turbine can be idealized as in Fig. 2. The governing equation of motion for this system can be written as follows

$$(1+\bar{m})\ddot{U} + 2\xi\omega\dot{U} + \omega^2 U = \frac{P}{M} - \bar{m}\ddot{u} \tag{10}$$

$$\ddot{u} + 2\zeta_d\omega_d\dot{u} + \omega_d{}^2 u = -\ddot{U} \tag{11}$$

where $\bar{m} = m/M$. Here M and m are the masses of the primary structure and the damper, respectively. Also

$$\omega^2 = \frac{K}{M} \tag{12}$$

$$C = 2\zeta\omega M \tag{13}$$

in which K is the stiffness, ξ is the damping ratio, U is the displacement of the primary structure, and u is the displacement of the mass of the TMD.

The common design parameters for TMDs are the mass, the stiffness, and the damping ratio of the auxiliary system. According to Warburton and Ayorinde (1980), the primary structure can be treated as an equivalent single degree-of-freedom (DOF) system if its natural frequencies are well separated.

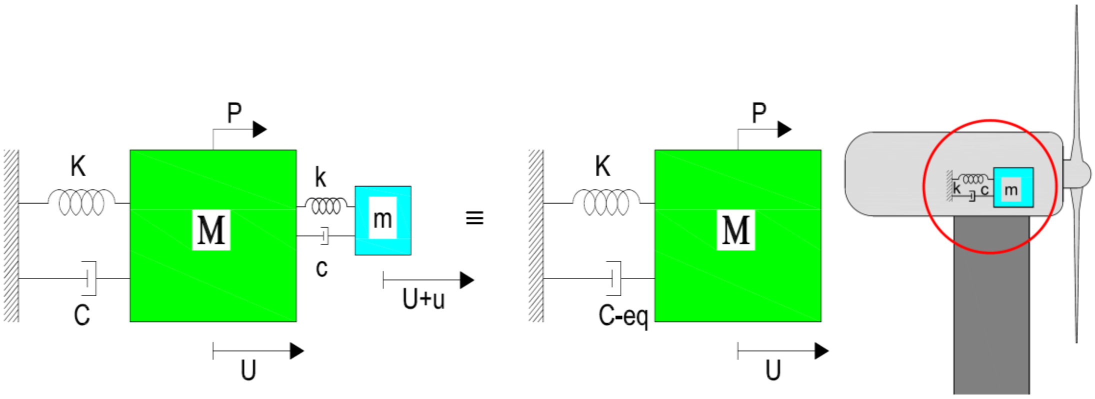

Fig. 2 A simplified model of a SDOF system with a TMD and the equivalent model and its schematic location at the nacelle of a horizontal axis wind turbine

Then the effect of the TMD can be viewed as being equivalent to changing the damping ratio of the original system from the value ξ=C/2(KM)$^{0.5}$ to a larger value ξ_e (Simiu and Scanlan, 1996). The equivalent system has the same mass and stiffness as the primary structure, but with damping C_e=2 ξ_e(KM)$^{0.5}$. Then it can be shown that (Si *et al*. 2014)

$$\xi_e = \frac{1}{2}\frac{\alpha_1(\alpha_2\alpha_3-\alpha_1)-\alpha_0\alpha_3^2}{\alpha_0(\alpha_2\alpha_3-\alpha_1)+\alpha_3(\beta_1^2-2\alpha_0)+\alpha_1} \tag{14}$$

here

$$\left.\begin{aligned} &\alpha_0 = f^2, \quad \alpha_1 = 2f\left(\xi f+\xi_d\right), \quad \alpha_2 = 1+f^2\left(1+\mu\right)+4f\xi\xi_d \\ &\alpha_3 = 2\xi+2\xi_d f\left(1+\mu\right), \quad \beta_1 = 2\xi_d f \end{aligned}\right\} \tag{15}$$

where $\mu = m/M, \quad f = \omega_d/\omega$.

For a certain tuning frequency ratio, *f*, the optimal value of the damping coefficient of the TMD that will minimize the displacement response of the primary structure under white-noise excitation is given by

$$\xi_d^{opt} = \frac{\sqrt{\mu}}{2} \tag{16}$$

and the corresponding equivalent damping is

$$\xi_e^{opt} = \frac{\sqrt{\mu}}{4} + 0.8\xi > \xi \tag{17}$$

Feng and Mita (1995), proposed formulae for estimating the optimum parameters of the TMD by minimizing the mean square response of the primary structure to a white-noise force excitation for wind analysis. For wind loading, they proposed the following absorber parameters

$$f = \frac{\sqrt{1+(\mu/2)}}{1+\mu}, \quad \xi_d^{opt} = \frac{1}{2}\sqrt{(1+\mu)^4 - \left(\frac{2+\mu}{1+\mu}\right) f^2 + \frac{1}{1-\mu}} \tag{18}$$

Murtagh *et al.* (2008) investigated the use of TMDs for the mitigation of the along-wind vibrations of a wind turbine. In their model, the mass ratio of the TMD was 1% and the optimal tuning frequency ratio was 0.99. They concluded that, if the system parameters do not change dramatically with time, the use of passive TMD is justifiable due to its ease of use, low cost and no need for external power.

Si *et al.* (2014) investigated the effect of passive TMD to control the vibration of offshore floating spar type wind turbines. In this numerical study, they tried to optimize effective parameters of the TMD for this type of wind turbines. They suggested placing the damper inside the upper part of the spar segment. They also found that, under the condition that the turbine works above rated in resonant motion, minimizing the spring coefficient would achieve much load reduction and power quality improvement.

Stewart and Lackner (2014) investigate the misalignment effect of wind and wave on offshore wind turbines. They showed that this misalignment causes large loads on the tower in the side-side direction, which are less than, but of the same order at the fore-aft loads. However, they showed that the primary impact of the TMDs is on the side-side loads, which were reduced by over 40%, as a result of the minimal structural damping in this direction, compared to the aerodynamic damping provided by the rotor in the fore-aft direction.

Regardless of the benefits of tuned spring mass dampers, it should be taken into consideration that they have some limitations. It is worthy to mention that the wind turbine may vibrate at different frequencies due to different reasons, such as:

(a) Forced vibrations: the disturbing force excites the primary structure at a frequency that differs from the target resonance frequency.

(b) Higher resonance frequencies: another resonance frequency of the primary structure is excited than the resonance frequency that was used for the design of the TMD.

(c) Time-varying target resonance frequency due to environmental impacts: the resonance frequency of the target mode differs from the resonance frequency that was used for the design of the TMD due to temperature effects and/or life loads on the structure (Weber 2014, Aly 2014a, b).

Also, the passive energy-absorbing devices are usually installed to suppress vibrations in a single axis while change of wind direction may require vibration absorbers located in two orthogonal directions. In this case it may be necessary to install two separate dampers in both orthogonal directions to control the two responses of a high-rise structure (Aly 2014a, b). But, this requires considerable cost and space (Min *et al*. 2014a).

Dinh and Basu (2015) applied a mathematical model to investigate the performance of single and multiple tuned mass dampers for passive control of spar-type floating wind turbines. In their study, they considered aerodynamic, hydrodynamic and buoyancy effects on a 5 MW NREL spar-type offshore wind turbine. Different TMD configurations were examined and they concluded that a horizontal TMD can effectively control the sway motion of the nacelle and the roll of the spar. Also, a spar TMD is more effective than a nacelle TMD to control nacelle the sway, and the control effectiveness is not much improved by using more than two TMDs.

3.2 Tuned sloshing dampers (TSDs)

Tuned sloshing dampers are a type of passive energy-absorbing devices that are proposed to reduce the wind-induced vibration problems of tall buildings (Fig. 3). TSDs, that are a type of tuned liquid dampers (TLDs), use the sloshing motion of liquid in containers to control the excited vibration. Coupled dynamic equations of the TSD and the structure may be written as (Chang and Qu 1998)

$$\begin{bmatrix} 1+\mu_2 & \mu_3 \\ \mu_3 & \mu_1 \end{bmatrix} \begin{Bmatrix} \ddot{U}(t) \\ \ddot{u}(t) \end{Bmatrix} + \begin{bmatrix} 2\xi\omega & 0 \\ 0 & 2\mu_1\xi_{TSD}\omega_{TSD} \end{bmatrix} \begin{Bmatrix} \dot{U}(t) \\ \dot{u}(t) \end{Bmatrix} + \begin{bmatrix} \omega^2 & 0 \\ 0 & \mu_1\omega^2{}_{TSD} \end{bmatrix} \begin{Bmatrix} U(t) \\ u(t) \end{Bmatrix} = \begin{Bmatrix} \dfrac{F(t)}{M} \\ 0 \end{Bmatrix} \tag{19}$$

in which, μ_1 and μ_3 are identical to the mass ratio of the total mass to the first modal mass of the structure (i.e., $\beta\rho L_w hT/M$ where T is the water tank width); μ_2 is the mass ratio of total liquid mass over the first modal mass of the system (i.e., $\rho L_w h_t/M$); β is the value of $8L_w \tanh(\pi h/L_w)/\pi^3 h$. The fundamental frequency of the sloshing liquid is calculated based on the shallow wave theory as (Min *et al.* 2014a)

$$\omega_{TSD} = \sqrt{\frac{\pi g}{L_w}\tanh\left(\frac{\pi h}{L_w}\right)} \tag{20}$$

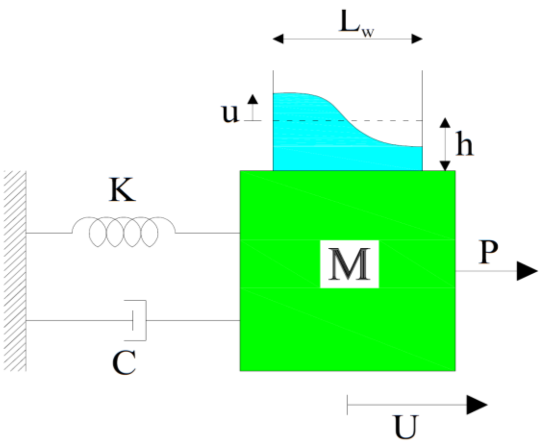

Fig. 3 Simplified model of a tuned sloshing damper: U is the horizontal displacement of the primary structure; u is the height of sloshing motion; M, C and K, respectively represent the first modal mass, damping and stiffness coefficients of the primary structure; P is the excitation force acting on the primary structure

Different studies have been accomplished to investigate the effectiveness of applying liquid dampers for suppressing excitation forces on tall structures. Pirner and Urushadze (2007) introduced effectiveness parameter as follows, to check the performance of the TSDs

$$\varepsilon = STD\left(\frac{F_{H_2O}}{F_0}\right) \tag{21}$$

where *STD* refers to the standard deviation, F_{H_2O} is the harmonic excitation force needed for the excitation of the required amplitude of the horizontal motion of the appliance with water and F_0 is the harmonic excitation force needed for the excitation of the required amplitude of horizontal motion of the appliance without water. In their experimental study, they have tested various perforated partitioning in the tank (Fig. 4) and compared with no partitioning tank in order to find the effectiveness of the tank in different cases under the horizontal motion. Their study could not show that the partitioning has any kind of preferred ability over no-obstacle tanks. They also checked the influence of dynamic viscosity and showed that using a liquid with less dynamic viscosity is more effective.

The damping force of tuned liquid dampers is controlled by the fraction of liquid contribute to control the load which is controlled by the three volume dimensions and liquid density. Moreover, the damping force is controlled by the liquid level velocity related to the loading intensity (Chang 1999, Lee *et al.* 2012). Hence, the optimal frequencies and damping ratios for liquid dampers should be determined to a given loading intensity. By assuming a constant damping coefficient and Gaussian white noise excitation of winds, unified optimal frequency ratios of λ_{opt} and damping ratios of ξ_{opt} are obtained as (Min *et al.* 2014a)

$$\lambda_{opt} = \frac{(1+\mu_2-(1/2)\gamma)^{1/2}}{1+\mu_2} \tag{22}$$

$$\xi_{opt} = \frac{1}{2}\left[\frac{\gamma(1+\mu_2-(1/4)\gamma)}{(1+\mu_2)(1+\mu_2-(1/2)\gamma)}\right]^{1/2} \tag{23}$$

where γ is defined as μ_2^2/μ_1.

3.3 Tuned liquid column dampers (TLCDs)

Since in TSDs just the liquid near the free surface participates in the sloshing motion, tuned liquid column dampers (TLCDs) are proposed to increase the participation of liquid. Min *et al.* (2014b) performed design and testing of a tuned liquid mass damper for attenuation of the wind responses of a full-scale building. The device shows significant performance in vibration suppression of real full-scale buildings. To further enhance the performance of liquid column dampers, shape memory alloys devices were used recently by Gur *et al.* (2014). A TLCD is a passive type control device which absorbs the motion of the primary structure by the motion of liquid in a column container. TLCDs are U-shaped tubes of uniform rectangular or circular cross-section, partially filled with a volume of liquid (Fig. 5). Vibration energy is transferred from the structure to the TLCD liquid through the motion of the rigid container exciting the TLCD liquid (Huo *et al.* 2013). TLCD suppress the structure's vibration through the gravitational restoring force acting on the displaced TLCD liquid. The viscous interaction of the liquid and the container will dissipate the vibration energy. In order to increase the efficiency during the vibration, the liquid has to pass through orifices installed inside the TLCD container which cause liquid head loss and subsequently more energy dissipation. Relatively low maintenance requirements and the availability of the liquid to be used for emergency purposes are some of the potential advantages of liquid vibration absorbers (Hitchcock *et al.* 1997a, b). The equation of motion for TLCD's is as follows

$$m\ddot{u}+c\dot{u}+ku=-\alpha_x m\ddot{U}_n \tag{24}$$

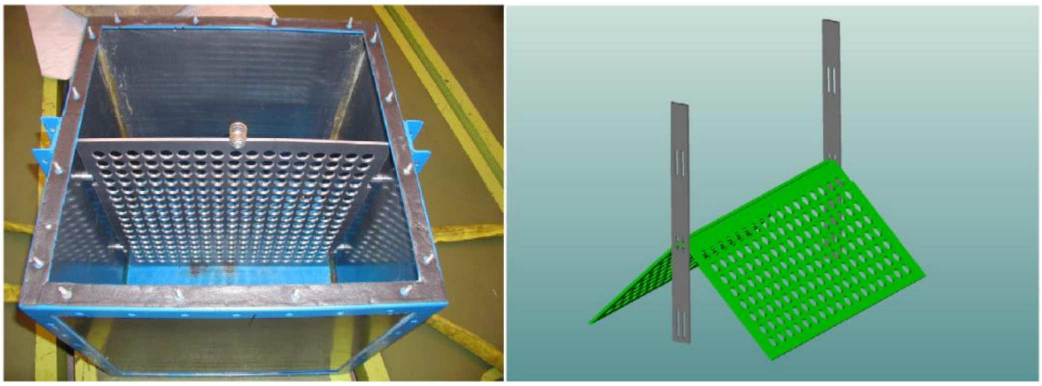

Fig. 4 Various perforated partitioning used in a sloshing tank (Pirner and Urushadze 2007)

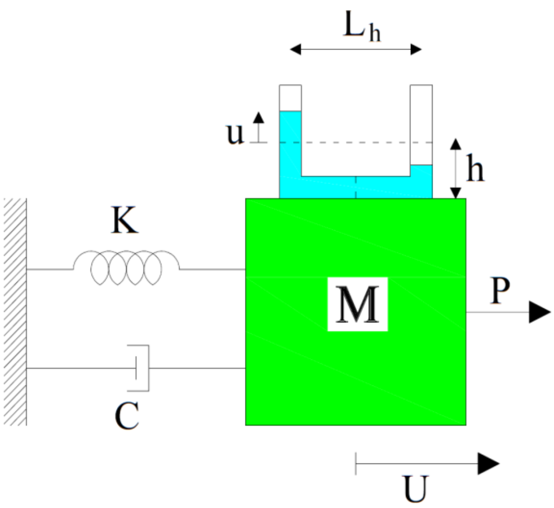

Fig. 5 Simplified model of a tuned liquid column damper

where $m=\rho A(2h+L_h)$ is the mass of the liquid in the TLCD. Here, ρ is the liquid density, A is the cross sectional area, h is the static height of the liquid in the columns and L_h is the length of the horizontal part of the TLCD; $c=(\rho A\xi_{TLCD}|\dot{u}|)/2$ is the equivalent damping of the TLCD in which ξ_{TLCD} is the coefficient of head loss due to the orifice and $\dot{u}$ is the relative velocity of the liquid in the TLCD; $k= 2\rho Ag$ and is the stiffness of the liquid in vibration. α_x is defined as the shape function of the TLCD and is $L_h /(2h+ L_h)$. Also, in a structure with n degree of freedom (DOF), $\ddot{U}_n$ is the acceleration of the nth DOF. Therefore, the natural circular frequency of the motion can be written as $\omega_{\mathrm{TLCD}} = \sqrt{k/m} = \sqrt{2g/(2h+L_h)}$.

Colwell and Basu (2009) modeled an offshore wind turbine and examined the excitation of multi degree of freedom system under the action of wind and wave. They showed that for the over rated wind velocity of 30m/s, the peak displacement of the structure could be diminished up to 35.5% by installing the TLCD without wave attack. Also in a harsh case of wind and wave attack, employing TLCDs leads to decrease of about 38% in the peak response. In operational condition, with rotating blades, TLCDs can reduce the responses by approximately 60%. Mensah and Dueñas-Osorio (2014) also conducted a study to explore the reliability of inland wind turbines that are equipped with TLCDs. They showed that the response of a baseline TLCD with 1% of the mass of the primary structure minimizes peak responses by as much as 47%.

Huo *et al.* (2013) explored the optimal design of liquid dampers to control the structures using Genetic Algorithms (GAs). They proposed that the best control performance for the TLCDs can be achieved with the damping ratio of 0.1. They also showed that the mass ratio of the TLCD for this damping coefficient is about 3%.

However, the most detrimental problem of using TLCD’s in slender structures such as wind turbines is the total length of liquid column. The larger the value of the horizontal section of the TLCD, the more effective the TLCD will be, but the horizontal space for TLCD is limited, which means that a relatively low value of the length ration α (horizontal length/total length) should be used.

3.4 Smart and viscous dampers

Since passive TMDs are normally tuned to a certain vibration frequency, they cannot completely control the system under broad band of excitation frequencies. Hence different types of controllable TMDS are developed that active and semi-active dampers are some of the examples of them. Experiments have shown that active tuned mass dampers are effective solutions for vibration reduction in high-rise structures (Lu *et al.* 2003, Soong 1990, Wu and Pan 2002); however, they are large and heavy systems. Moreover, active dampers consume considerable power that makes them inappropriate solution for wind turbine towers as the main purpose is to produce renewable energy, rather than consuming it the production place. Other types of dampers that can be used as alternatives to TMDs and ATMDs are viscous and semi-active dampers. These devices do not require frequency tuning (Smith and Willford 2007).

Smart damping technology is a type of semi-active control that employs variable dampers (e.g., variable orifice, magneto-rheological (MR) fluid and electro-rheological fluid dampers) (Aly *et al.* 2011, *Metwally et al.* 2006). Some of the recent examples of these systems are MR semi-active TMD (MR-STMD), shape memory alloy TMD (SMA-TMD) and nonlinear TMD (NTMD). The idea of a MR-STMD is to replace the passive oil damper, in a classical TMD, by a MR damper which is used to adjust the natural frequency and damping of the MR-STMD to the actual frequency of the primary structure (Weber and Maślanka 2012). The SMA-TMD dissipates the input energy through hysteretic phase transformation of its microstructure under cyclic loading. That is, due to its force-deformation characteristic, a shape memory alloy (SMA) spring can reduce the displacement of the TMD. Also, it is shown that the mass ratio can be less than that in a classical linear TMD (Mishra *et al.* 2013). A NTMD is composed of four springs: two acting in a geometric way to harden the system, and two acting in a linear way. Therefore, in comparison with a linear TMD, a NTMD has more effectiveness over a broader range of frequencies, and it does not suffer from the problem of amplification just outside the target bandwidth and requires relatively lower mass ratio (Alexander and Schilder 2009).

In smart and viscous dampers (VDs), the viscous force generated in the damper can be used to control a wide band of excitation frequencies. Some of the advantages of these types of dampers are their continuous damping controllability, quiet operation, simple configuration, low to zero power consumption, and high control stability (Choi 2008). Fig. 6 illustrates a configuration model of these types of dampers with an outer bracing system. It is worthy to mention that bracing systems have been used for the control of tall buildings under wind loads by several researchers, for instance, Kim *et al*. (2014) presents a wind-induced vibration control of tall buildings using hybrid buckling-restrained braces. The system showed significant effectiveness in vibration suppression. Drift magnification for outer bracing systems for tall buildings with a lever mechanism connection was used by Aly *et al*. (2011) for improving the effectiveness of the dampers (see also Rezaee and Aly 2015). Accordingly, in the current study, a lever mechanism is used to magnify the displacement, for better performance of the VD dampers in reducing the vibrations of the wind turbine. In addition, to the fact that a designer can bring damping in wind turbines to higher levels, by employing different control systems, depending on their location, wind turbines under fluid structure interaction may develop significant hydrodynamic and/or aerodynamic damping.

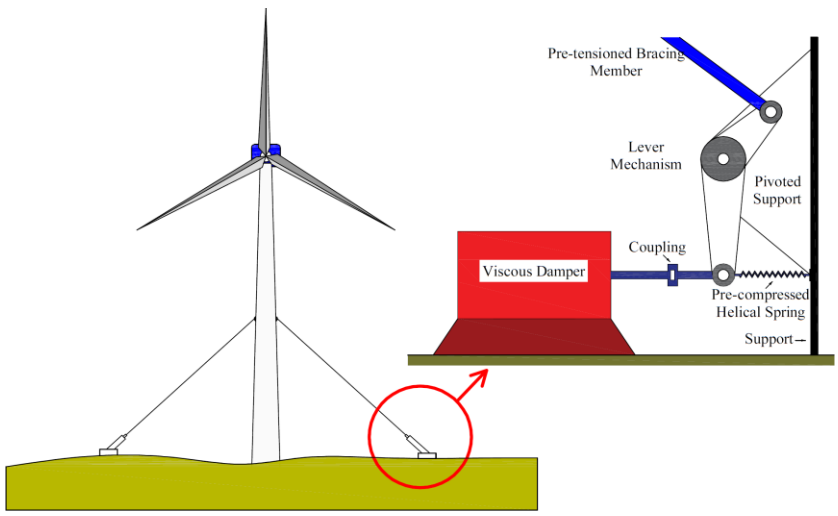

Fig. 6 A schematic representation of the viscous damper (VD) installation: configuration with a lever mechanism connection for displacement amplification to enhance the performance of the dampers

4. Hydrodynamic/aerodynamic damping

The damping of the structure mainly comprises of aerodynamic and hydrodynamic damping (for offshore turbines). The hydrodynamic forces per unit length on the floating part of the floating wind turbine can be estimated by the Morison's formula (Faltinsen 1995)

$$dF = \rho \frac{\pi D^2}{4}\left(C_m \dot{U}_r + \dot{U}_w\right) + \frac{\rho}{2} C_d D \left|U_r\right| U_r \tag{25}$$

This equation comprises the term of added mass and Froud-Krylov inertia forces (first term) and the quadratic viscous drag (second term). Here, D is the cylinder diameter, $\dot{U}$ is horizontal relative acceleration and $U_{r=}$ U_w- U_B is the relative velocity (U_w is the water particle velocity and U_B is the velocity of the body). C_m and C_d are added mass and quadratic drag coefficients respectively. The hydrodynamic damping can be written as

$$dF_{drag} = \frac{\rho}{2} C_d D \left|U_r\right| U_r \tag{26}$$

If $U_r \geq 0$

$$dF_{drag} = \frac{\rho}{2} C_d D\left(U^2{}_w + U^2{}_B - 2U_W U_B\right) \tag{27}$$

else ($U_r < 0$)

$$dF_{drag} = -\frac{\rho}{2} C_d D\left(U^2{}_W + U^2{}_B - 2U_W U_B\right) \tag{28}$$

The term proportional to body velocity in this equation represents the damping force and so it can be added to the left hand side of the equation of motion. Both damping force and excitation forces are dependent on the water particle velocity, sea water density, diameter of the strip and the drag coefficient. Therefore, by increasing the quadratic drag coefficient both damping and the excitation forces will be increased. However, Karimirad and Moan (2012) showed that, for spar type floating offshore wind turbines, by increasing the drag coefficient the surge resonant response of low frequency rigid body motion will be reduced (Kaimal *et al.* 1972).

Moreover, both the blades and the tower of the wind turbine can be considered as slender structures. Hence, the snap wise velocity component is much lower than the stream wise component, and therefore it is assumed in many aerodynamic models that the flow at a given point is two dimensional (Hansen 2008). The wind force acting on the blade can be estimated as follows (Kaimal *et al.* 1972)

$$f_t \cong (Vk)^2 - 2VkL\dot{U}_5 \cos U_5 - 2Vk\dot{U}_1 \tag{29}$$

Here V is the upstream wind velocity, $\dot{U}_1$ is the surge motion velocity of the structure, U_5 is the pitch motion response, and L is the airfoil section distance from the mean water level surface. For more information about k please refer to the Kaimal *et al.* (1972). The terms in negative sign are the aerodynamic damping which can be added to the left hand side of the equation of the motion.

5. Comparison among different types of dampers

In this section, in order to recommend the use of a certain damping enhancement technique for wind turbines, it is important to first study the performance of using different dampers to suppress the vibration of the primary structure, in a comparative study. Accordingly, comparisons are presented among TMD, TSD, TLCD and VDs for the 5MW wind turbine. Using MATLAB (Attaway 2013), codes were developed that permit the simulation of the primary structure (wind turbine) with and without the control devices (TMD, TSD, TLCD, and VDs). For VDs, the equivalent damping factor for the controlled structure is equal to that of a classical TMD, in order to provide a fair condition for comparison.

Fig. 7 shows a dimensionless parameter L^2/hT for the TLCD and the TSD, for different mass ratios: L (L_h or L_w, depending on the type of damper: TLCD or TSD) and h are the dimensions of the tank/column (as shown in Figs. 3 and 5), T is the width of the tank/column (in the perpendicular plane). It should be noted that the parameter that control the tuning of the frequency of the liquid damper is the length dimension (L) of the liquid container. The figure depicts that the ratio of the square value of the length (L) to the other two dimensions (hT) does not change significantly with the change in the mass ratio. While for the TSD, by increasing the mass ratio, the tank shape is going toward a cubic shape with a square cross-section (i.e., the width T is increased significantly leading to shape modification of the tank and not a scale up of the same shape, which is the case of the TLCD).

By considering different mass ratios, in the numerical optimization, optimal response criteria can be obtained for different optimization objectives, for instance, normalized displacement and normalized acceleration. The term normalized means the relative values between the controlled STD (standard deviation) responses and the uncontrolled STD responses (without the control devices), i.e.

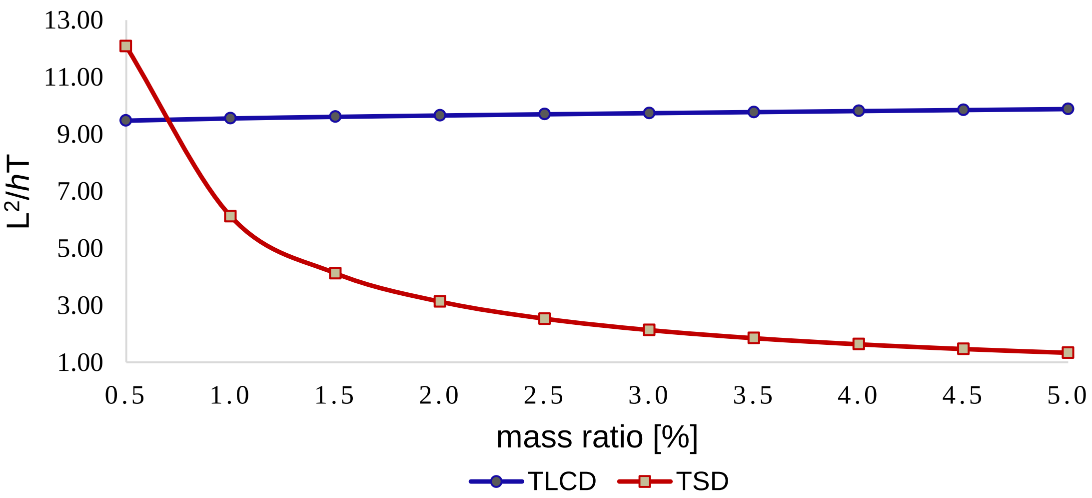

Fig. 7 Dimensionless values of L^2/hT versus the mass ratio parameter for the TLCD and the TSD

$$normalized\ response = \frac{STD\ of\ the\ controlled\ response}{STD\ of\ the\ uncontrolled\ response}. \tag{29}$$

Fig. 8 shows the normalized displacement response of the primary structure controlled by a TMD for with a 2% mass ratio. The figure shows that for this TMD the optimum values of the tuning frequency ratio and the damping factor for the TMD are 0.925 and 0.09, respectively. The TMD is able to reduce the STD of the displacement response by 38%.

The results presented in Aly (2014a, b) show that the mass ratio of the TMD has a significant impact on the tuning parameters, the response of the primary structure, and the stroke of TMD. As shown in Fig. 9, by increasing the mass ratio, the optimum tuning frequency ratio is reduced. This reduction is significantly dependent on the optimization parameter (minimizing displacement or acceleration of the primary structure). The figure also shows that a TMD with a relatively high mass ratio will require increase in the damping ratio of the device, which is favorable in limiting the stroke of the TMD as shown in Fig. 10. The effect of the mass ratio on the normalized (ratio of the response with TMD to the response of the primary structure without TMD) displacement responses of the primary structure and the normalized stroke of the TMD is shown in Fig. 10. By increasing the mass ratio of the TMD, significant reduction in the response of the primary structure can be achieved. Also the mass ratio has a significant influence on limiting the stroke of the TMD. Typically at higher mass ratios, additional increase in the mass of the TMD will not bring significant reduction to the responses. In any case, the current paper focuses on a comparative study among TMD, TLCD, TSD, for which the mass ratio was assumed to be 2%.

Fig. 11 shows the normalized displacement response of the primary structure controlled by a TSD with a 2% mass ratio. The figure shows that for this TSD, the optimum values of the tuning frequency ratio and the damping factor for the TSD are 0.96 and 0.06, respectively. The TMD is able to reduce the STD of the displacement response by 35.5%.

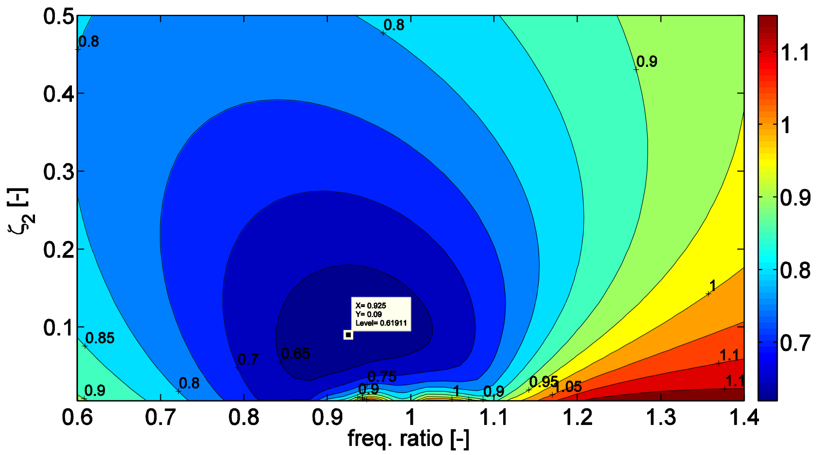

Fig. 8 Effect of changing the frequency and damping ratios of the TMD on the normalized displacement response of the primary structure

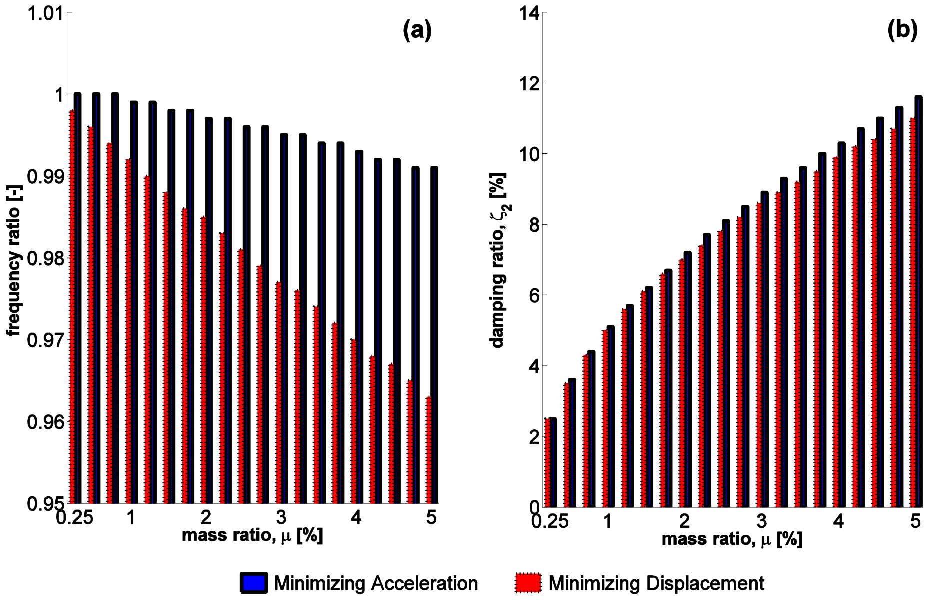

Fig. 9 Effect of changing the mass ratio on the optimal parameters of the TMD for two optimization objectives (minimizing acceleration and minimizing displacement of the primary structure): (a) tuning frequency ratio and (b) damping ratio

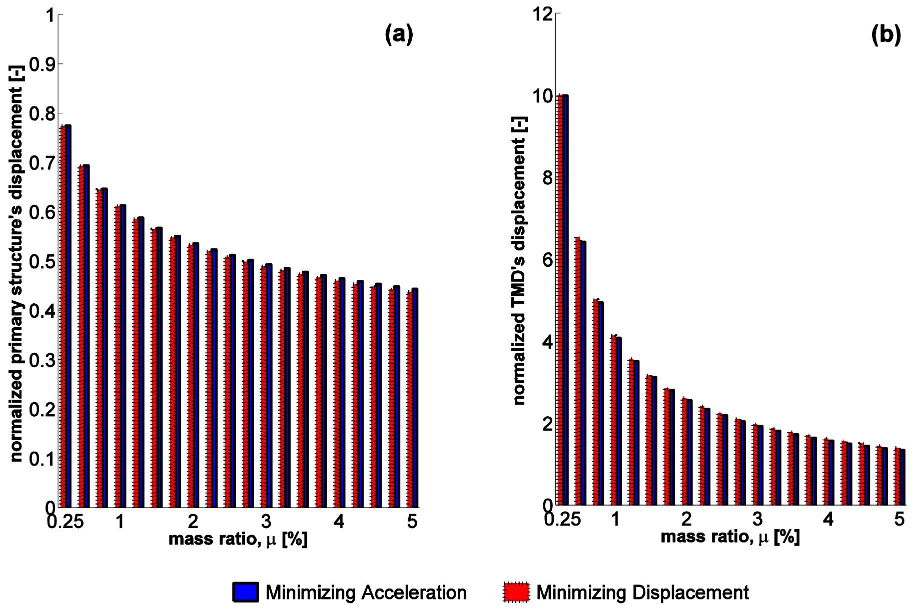

Fig. 10 Effect of changing the mass ratio on the normalized displacement response for two optimization objectives (minimizing acceleration and minimizing displacement of the primary structure): (a) normalized displacement of the primary structure and (b) normalized stroke of the TMD

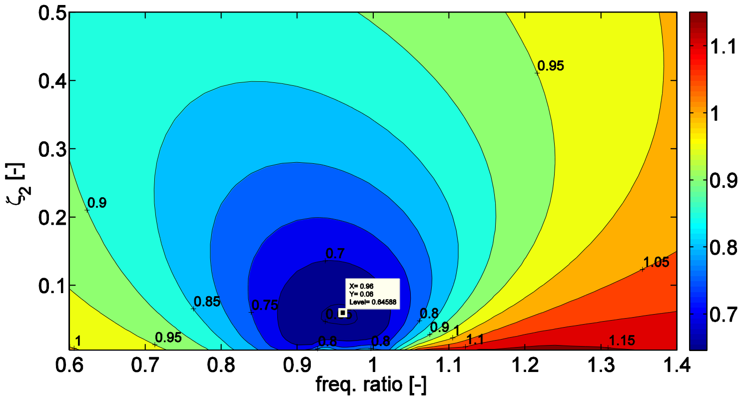

Fig. 11 Effect of changing the frequency and damping ratios of the TSD on the normalized displacement response of the primary structure

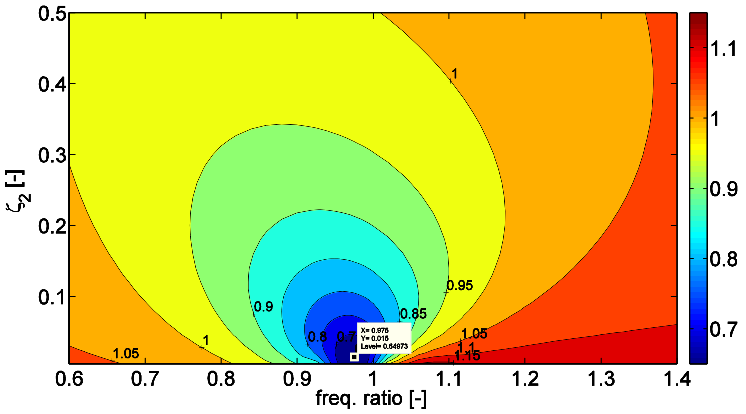

Fig. 12 Effect of changing the frequency and damping ratios of the TLCD on the normalized displacement response of the primary structure

Fig. 12 shows the normalized displacement response of the primary structure controlled by a TLCD with a 2% mass ratio. The figure shows that for this TLCD, the optimum values of the tuning frequency ratio and the damping factor for the TSD are 0.975 and 0.015, respectively. The TMD is able to reduce the STD of the displacement response by 35%. A comparison among Figs. 8-10 shows that the TMD can bring the best response reduction to the primary structure with better robustness than the TSD and the TLCD. The TLCD performance is very sensitive to the tuning frequency.

To further investigate the robustness of the three control devices in addition to the VDs, Fig. 13 shows the normalized displacement and acceleration response of the wind turbine's tower for different uncertainties in the stiffness of the primary structure (±10%). The figure shows that the VD is the most robust, followed by the TMD, the TSD and finally the TLCD. The TLCD is very sensitive to uncertainties in the structural stiffness. At the same time, an increase in the nominal stiffness (by 2-3%) may result into displacement and acceleration response reduction, however this solution is not robust and increase in the damping of the system can bring more response reduction.

Fig. 14 shows that not only the damping enhancement techniques can reduce the displacement and acceleration response, but also they can reduce significantly the overall foundation load (35-40%). Here again the VD is the best option to reduce the foundation forces and it is followed by the TMD, the TSDD and the TLCD, respectively. Stiffness enhancement is not a robust solution (uncontrolled response with ±10 stiffness uncertainty).

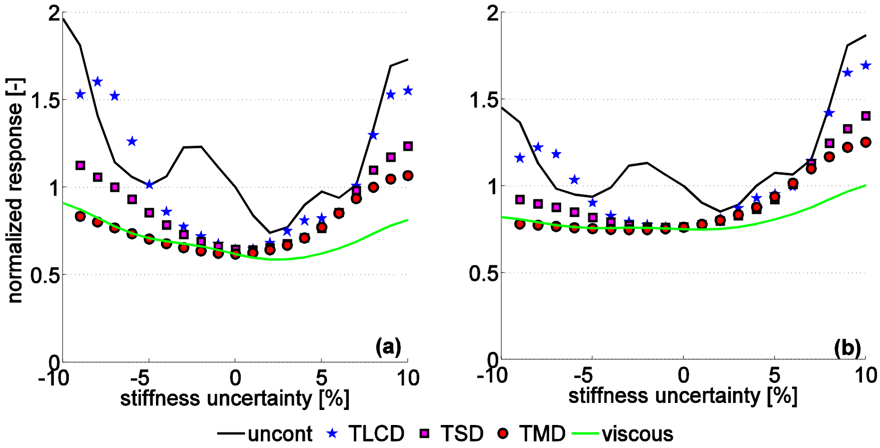

Fig. 13 Normalized responses of the primary structure with different control systems versus the uncertainty in the stiffness: (a) displacement and (b) acceleration

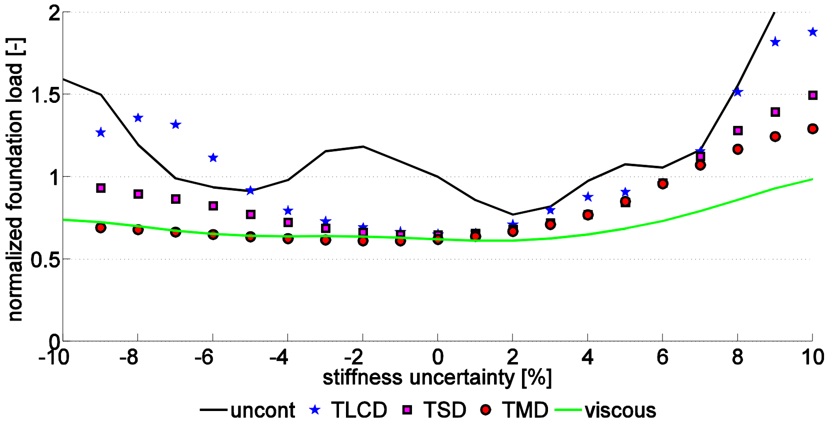

Fig. 14 Normalized foundation loads of the primary structure with different control systems versus the uncertainty in the stiffness

For the use of viscous dampers in slender and cantilever structures, outer bracing and a lever mechanism connection can help magnify the displacement across the dampers and hence improve their performance (Aly *et al*. 2011). However, one of the challenging problems about VDs is their need for maintenance and adjustments. Another challenge is the use of these devices in the marine environment, particularly deep waters, so their use may be limited to inland wind turbines. Also, they may need sufficient space around the structure for the bracing system.

It seems that TSD are attractive devices for offshore wind turbines. One of the advantages of TSDs is their capability to suppress a wide range of frequencies. So the load effect and consequently the vibration direction is no matter for the damper, while this can be one of the limitations of using a classical TMD.

6. Discussion

While the focus of the current investigations is on a comparative study among different damping enhancement techniques, the study did not address the potential coupling between the blades and the tower. Such coupling may affect the response of the wind turbine to wind/wave action. For instance, the study presented by Dinh *et al.* (2013), which investigated the effects of blade-nacelle-spar vibration coupling, suggests that such coupling may have a significant effect on the nacelle and spar responses. Also, in comparison with a fixed foundation wind turbine, a floating wind turbine may experience a reduction in the response amplitude at higher frequencies, as per a positive hydrodynamic damping effect. Additional studies used coupled models to investigate the response of wind turbines (Wayman *et al.* 2006, Lackner and Rotea 2011, Sebastian and Lackner 2012, Dinh and Basu 2015). However, Karimirad and Moan (2012) presented a simplified aero-hydro-dynamic method to calculate the dynamic responses of floating offshore wind turbines (FOWTs). They considered that for floating wind turbines, the general motions and structural responses, such as nacelle surge motion and acceleration as well as the bending moment and shear force at the tower-spar interface, are dominated by rigid body motions rather than elastic deformations. Hence, they used rigid body motion method to simplify the analysis. They showed that the sophisticated coupled aero-hydro-servo-elastic time domain analysis of these structures is time consuming, particularly in connection with the aerodynamics, controller and the elastic formulation of the structure while the simplified method is faster and the results are acceptable. In any case, the focus of the current study was on the performance of different damping enhancement techniques, when applied to the wind turbine modeled as a SDoF system.

7. Conclusions

Wind energy as a reliable alternative to fossil fuels is growing fast. In recent years, wind energy harvesting from coastal and offshore zones has increased, due to wind availability and population concentration along the coast. Accordingly, larger wind turbine structures with taller towers are being built, which brings the challenge of dealing with severe vibrations and excessive wind loads that can be detrimental to the structural lifecycle and energy production. To alleviate these issues, different damping enhancement technologies are available. In the current study, a comparative study among several passive dampers including TMDs, TSDs, TLCDs and VDs was conducted. The important results are summarized as follows:

1. The dissipativity analysis carried out shows that wind turbines are lightly damped structures and the enhancement of damping will suppress excessive vibrations, rather than modifying rigidity. The performance of active, semi-active and viscous systems at relatively low control forces are very similar. The optimal control force can be conveniently provided by viscous dampers or a system that can increase the damping rather than trying to modify the rigidity. This makes a passive control force (damping force), between the ground and the primary structure, competitive with active and semi-active control forces.
2. The comparison between controlled (with damping enhancement devices) and uncontrolled responses shows the effectiveness of the control system to suppress vibrations.
3. The proposed damping enhancement techniques show that not only they can reduce the displacement and acceleration responses, but also they can reduce the overall foundation loads, and hence can improve both the performance and the resilience of the wind turbine.
4. In terms of robustness, VDs are the most robust, followed by TMDs, the TSDs and finally TLCDs.
5. Using VDs with outer bracing and a lever mechanism connection can help magnify the displacement across the dampers and hence improve their performance; however, one of the challenging problems about VDs is their use in the marine environment, particularly deep waters, so their usage may be limited to inland wind turbines. Also, they need enough space around the structure for the bracing system.
6. One of the advantages of TSDs is their capability to suppress a wide range of frequencies. So the load effect, and consequently the vibration direction, is no matter for the damper. While this is one of the limitations of using, for example, a classical TMD.

Acknowledgments

Financial support was received from the National Science Foundation (NSF Award No. 1361908), to attend a Planning Grant meeting (I/UCRC for Windstorm Hazard Mitigation), and to present the results of this study. Also, it would not be possible to present the results of the current study at the 14th International Conference on Wind Engineering without the travel support provided by NSF, through the American Association for Wind Engineering (AAWE).

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