This paper proposed a novel amplifying damping transfer system (ADTS) as a new damping enhancement solution for high-rise structures like wind turbines. The proposed ADTS can transfer the upper rotation of turbine tower to its bottom with damping amplification mechanism. Hence, viscous damper can be installed on wind turbines in a
very convenient and efficient way. The dynamic characteristics of wind turbines equipped with ADTS were parametrically investigated concerning the influence of the damping, stiffness, and position of the ADTS based on complex frequency analysis. It was found that each mode has a maximum damping ratio, which is affected by the ADTS stiffness and position. The optimal ADTS position of the first mode is about 0.7H (turbine height), and the optimal positions of the second mode are at 0.3H and 0.86H. The proposed ADTS considerably attenuated both drift and acceleration responses of wind turbines caused by winds and earthquakes. For example, as compared to the optimized tuned mass damper, ADTS further decreases the displacement (acceleration) of wind turbine tower by about 22% (38%).

**Keywords:** Amplifying damping transfer system; wind turbine; dynamic characteristics; tuned mass damper.

1. Introduction

As technologies of producing clean energy, wind turbine is in rapid development to reduce the anxieties on the energy shortage. Modern wind turbine essentially comprises a high-rise tower with large concentrated mass at its top (blades, rotor, nacelle, gearbox, etc.). Hence, the turbine structures are very slender and lightly damped, which will be prone to dynamic loadings like winds and earthquakes, making the vibration control problem very critical.

Tuned mass damper (TMD) has been a well-developed structural control strategies, which can be simply realized by a secondary mass, a connecting spring, and a dashpot. Due to the simplicity and effectiveness of TMD, researchers and engineers showed great interest in applying TMD in wind turbines. Subsequently, many updated-TMDs were also developed especially for the application in wind turbines. Despite the fact that TMD has been extensively studied in the recent decade, the shortcomings of applying TMD in wind turbines are also recognized, for examples, (i) the limited space in wind turbines is competing with the large stroke required by the motion of the secondary mass in TMD; (ii) the perfect location of TMD is at the tower top, which may contribute to increase the concentrated mass at turbine top (blades, rotor, nacelle, and gearbox, etc.); (iii) TMD is known to be effective for target modes but may lose its effectiveness for other modes.

Besides TMD, there are many other damping enhancement solutions using novel dampers and magnification strategies which have been successfully applied at other infrastructures. In terms of wind turbines, braced-friction damper system, braced-viscous damper system, scissor-jack braced viscous damper, and magnetorheological dampers, etc., were typically applied to control the vibration of wind turbines. In fact, the above strategies of damping enhancement are all aimed at one problem of how to install typical dampers on wind turbines. Different from other high-rise structures like tall buildings which have their own structural component as the natural support to connect damper, wind turbine is assumed as solely bare cantilever without external support, making it is difficult to install damper like other high-rise structure does.
In terms of the high-rise structures with large aspect ratio, their vibration are known to be dominated by the flexural deformation including both horizontal displacement and rotation, and generally the amplitudes of such displacement and rotation will be larger at the upper part of the high-rise structures. However, on one hand, such large flexural deformation is difficult to utilize because as a two-terminal device, viscous damper needs supplemental support to connect the other terminal, and such support cannot be always available at the upper part of wind turbines. In that case, only the one-terminal device like TMD can be used by just attaching it to the wind turbines. On the other hand, most of the above-mentioned control strategies, including the TMD type and the non-TMD type, only focused on the utilization of the transversal motion rather than the rotation. As a matter of fact, the importance of using rotation has been recognized recently for many infrastructures like tall buildings and metamaterial beams. Hence, novel damping enhancement strategies of using rotation for wind turbines are in urgent need.

This paper proposed a novel amplifying damping transfer system (ADTS) for wind turbines to improve the damping effect. The proposed ADTS can transfer the upper turbine rotation to its bottom with damping amplification. Hence viscous damper can be installed very conveniently and is of high efficiency. In the following sections, the configuration and working mechanism of the proposed ADTS are introduced first in Sec. 2. Then, in Sec. 3 based on the complex frequency analysis, dynamic characteristics of the NREL 5 MW wind turbine example with the proposed ADTS are numerically studied to point out the influences of ADTS parameters. Subsequently, Sec. 4 evaluates the performance of wind turbines equipped with the proposed ADTS in resisting earthquakes and winds, as compared to the TMD technique. Finally, Sec. 5 concludes the new findings.

2. Problem Formulation

2.1. NREL 5 MW wind turbine

The modern NREL 5 MW wind turbine reported by Jonkman et al. is selected as the example in this study (see Fig. 1). The turbine tower has a total height of 87.6 m, and its outer diameter and thickness decrease from 6 m to 3.87 m and from 0.027 m to 0.019 m, respectively, with the increase of height. The tower steel has Young’s modulus of 210 GPa, a Poisson’s ratio of 0.3, and an effective density of 8500 kg/m$^3$, respectively. A concentrated mass of 350 ton is located at the top of turbine tower to represent the mass of nacelle, blades, and hub. Readers are suggested to refer to Jonkman et al. for more details of the parameters of the NREL 5 MW wind turbine. The first four frequencies of the wind turbine model are calculated as $f_1 = 0.2988$ Hz, $f_2 = 3.0247$ Hz, $f_3 = 9.0850$ Hz, and $f_4 = 18.5958$ Hz. The first four modal shapes of the NREL 5 MW wind turbine are shown in Fig. 2. In addition, a 2% Rayleigh damping is adopted for the uncontrolled wind turbine.
2.2. Amplifying damping transfer system

Figure 3 presents the sketch diagram of the proposed ADTS, which can be used to control either the fore-aft vibration or side-to-side vibration of wind turbines. The proposed ADTS is mainly made up of outrigger trusses, cables, rocker, and dampers. In particular, a pair of outriggers is cantilevered at the upper part of the turbine tower. A rocker is pinned at turbine bottom and can rotate freely.
Pre-stressed cables are used to connect the outrigger ends with the rocker. Two viscous dampers are applied to connect the flange ends of the rocker and ground. When wind turbines develop flexural deformation caused by lateral loading like winds and earthquakes, the tower rotation will trigger the outrigger to rotate and produce considerable vertical displacement at the outrigger ends. Then, this vertical displacement will drive the rocker through pre-stressed cables, and the rocker will rotate like the traditional Chinese Diabolo game. In this way, the viscous dampers will be either compressed or tensioned to provide supplemental energy dissipation to the whole turbine structures (see Fig. 3).

From the above illustration, it is known that the proposed ADTS provides a new solution for wind turbines to install damper, which not only utilizes the large rotation of the upper part but is also very convenient since viscous dampers can be placed at the turbine bottom. Moreover, the outrigger-cable-rocker system will further amplify the motion, resulting in significant vertical displacement at the rocker end to drive the viscous damper. Hence, the proposed ADTS can use a small damper to achieve a large damping effect for wind turbines. Last but not least, the proposed ADTS avoids the implementation of large extra mass at the top of wind turbine as traditional TMD does.

2.2.1. Deformation analysis
As shown in Figs. 3 and 4, when wind turbines are subjected to lateral loading, the flexural deformation of the turbine tower will produce rotation $\theta$ at the outrigger position, resulting in vertical displacement at outrigger ends ($u_O$) when a small deformation assumption is adopted

$$u_O = \theta R,$$  \hspace{1cm} (1)
where $R$ is the outrigger length. Then, this vertical displacement will drive the wheel of rocker to rotate through cables. The vertical displacement at point B of the rocker is

$$u_B = u_O - u_c,$$

(2)

where $u_c$ is the deformation of cables. Then the rotation of the rocker is approximately obtained as follows based on the small deformation assumption

$$\theta_2 = u_B / L_1.$$  

(3)

The resultant damper deformation at rocker end will be calculated as

$$u_D = \theta_2 L_2 = (\theta R - u_c) L_2 / L_1.$$  

(4)

In this way, the rotation at the upper position of the turbine tower is transferred to its bottom.

2.2.2. Force analysis

Assuming a linear damper is adopted, its damping force is

$$F_D = c_d u_D,$$

(5)

where $c_d$ is the damping coefficient. Then, the moment equilibrium at point A is

$$2 F_D L_2 + (F_B - F_B') L_1 = 0.$$  

(6)
The moment provided by ADTS to wind turbine is through the cables, which is

\[ M_O = (F_B - F_B')R. \]  

(7)

Substituting Eq. (6) into Eq. (7) yields

\[ M_O = \frac{2L_2R}{L_1} F_D = -\frac{2L_2R}{L_1} c_d \dot{u}_D = -2 \left( \frac{L_2}{L_1} \right)^2 R c_d (\dot{\theta} R - \dot{u}_c). \]  

(8)

If the deformation of the cable is neglected, Eq. (8) is degraded to

\[ M_O = -2 \left( \frac{L_2}{L_1} \right)^2 c_d \dot{\theta} = -2 R^2 C_d \dot{\theta}, \]  

(9)

where the damping coefficient of actual viscous damper \( c_d \) will be significantly amplified by \( (L_2/L_1)^2 \) times to \( C_d \).

2.2.3. Equivalent model

Based on the above illustration, an equivalent simplified model of wind turbines with ADTS can be established as shown in Fig. 5, where the effect of ADTS is approximately equivalent to a rotational spring attached at the intermediate of the turbine tower. The resisting moment provided by this rotational spring is further equivalent to the cantilever outrigger attached with viscous damper. The dashpot model with damping coefficient \( C_d \) is used here to represent the viscous damper in ADTS, while the connecting spring with stiffness \( k_C \) is adopted to represent the
actual flexibility of ADTS. In this regard, the resisting moment provided by the ADTS is described as

\[ M_O = -2F_d R = -2Rk_C (u_O - u_d) = -2RC_d \dot{u}_d, \]

(10)

where \( R \) is the cantilever length of the outrigger; \( F_d \) is the force at outrigger end; it is noteworthy that the damping coefficient \( C_d \) used in Eq. (10) is the amplified damping coefficient of viscous damper.

The resisting moment shown in Eq. (10) can be rewritten as follows by eliminating \( u_d \)

\[ \dot{M}_O = A_1 \dot{u}_O + A_2 M_O, \]

(11)

where \( A_1 = -2Rk_C; A_2 = -k_C/C_d. \)

2.3. Dynamic equation

Taking the seismic excitation case as an example, the control equation of the uncontrolled wind turbine subjected to the earthquake is expressed as

\[ M \ddot{X} + C \dot{X} + KX = -M \delta x_g, \]

(12)

where \( M, C, \) and \( K \) are the mass, damping, and stiffness matrices of the turbine tower; \( X = [x_1, \theta_1, \ldots, x_n, \theta_n]^T \) is the displacement vector with transversal displacement \( x_j \) and rotation \( \theta_j \) at each node; \( x_g \) is the ground motion acceleration; \( \delta \) is the loading vector; dot represents derivative with respect to time.

When ADTS is adopted, the dynamic equation is rewritten as

\[ M \ddot{X} + C \dot{X} + KX + \Psi M_O = -M \delta x_g, \]

(13)

where \( M_O \) is the resisting moment of ADTS determined by Eq. (11); \( \Psi \) represents the position vector.

Recognizing that the rotational displacement and velocity at the outrigger position are obtained as \( \theta_j = \Psi^T X \) and \( \dot{\theta}_j = \Psi^T \dot{X} \), respectively, the control equation of wind turbine with ADTS is combined as

\[
\begin{bmatrix}
M & 0 \\
0 & 0
\end{bmatrix}
\begin{bmatrix}
\dot{X} \\
\dot{M}_O
\end{bmatrix} +
\begin{bmatrix}
C & 0 \\
-A_1 \Psi^T & 1
\end{bmatrix}
\begin{bmatrix}
\dot{X} \\
\dot{M}_O
\end{bmatrix} +
\begin{bmatrix}
K & \Psi \\
0 & -A_2
\end{bmatrix}
\begin{bmatrix}
X \\
M_O
\end{bmatrix} =
\begin{bmatrix}
-M \delta \\
0
\end{bmatrix}
\dot{x}_g.
\]

(14)

Rewriting Eq. (14) into state-space expression yields

\[ \dot{X}_S = A_S X_S + B_S \dot{x}_g, \]

(15)

where \( X_S = \begin{bmatrix} \dot{X} \\ \dot{M}_O \end{bmatrix} \); \( B_S = \begin{bmatrix} -\delta \\ 0 \end{bmatrix} \); \( A_S = \begin{bmatrix} -M^{-1}C & -M^{-1}K & -M^{-1}\Psi \\ \Psi^T & 0 & 0 \\ A_1 \Psi^T & 0 & A_2 \end{bmatrix} \) is the system matrix.
3. Dynamic Characteristics of Wind Turbines with ADTS

3.1. Complex frequency analysis

The damping amplification effect of the proposed ADTS has been qualitatively illustrated in Sec. 2.2. To quantitatively evaluate the damping effect provided by ADTS to wind turbines, the dynamic characteristics especially the damping ratio of each mode are in the spotlight. By studying the eigenvalues of the system matrix $A_S$, the influence of ADTS on wind turbines can be quantified. The eigenvalues $\lambda_k$ of the system matrix $A_S$ are determined in complex conjugate pairs as

$$\lambda_k = \mu_k \pm i\nu_k,$$

(16)

where $i = \sqrt{-1}$ is the imaginary unit. Then, the modal frequency and modal damping ratio for the $k$th mode can be obtained, respectively, as

$$\omega_k = |\lambda_k|, \quad \xi_k = -\mu_k/|\lambda_k|.$$

(17)

3.2. Parametric study

There are mainly three parameters of ADTS that influence the performance of wind turbines—ADTS’s damping coefficient ($C_d$), position ($H_{ADTS}$), and connecting stiffness ($k_C$). Without loss of generality, the following normalized parameters are defined:

$$c = \frac{C_d R^2}{H \sqrt{mEI}}; \quad \alpha = \frac{H_{ADTS}}{H}; \quad \beta = \frac{EI}{2R^2\alpha Hk_C},$$

(18)

where $EI$ is the bottom bending stiffness of the turbine tower; $m$ is the distributed mass of turbine tower at the bottom. In fact, $c$ is the normalized ADTS damping; $\alpha \in (0, 1)$ is the normalized height of the ADTS position; $\beta$ is the normalized stiffness ratio of the bending stiffness of turbine tower to the ADTS stiffness. The defaulted values of these parameters are: $c = 0.02$, $\alpha = 1.0$, and $\beta = 0.5$ for the following study.

3.2.1. ADTS damping

The influence of ADTS damping on the dynamic characteristics of wind turbines with ADTS is presented in Fig. 6 where $\omega_{UC}$ represents the natural frequencies of the uncontrolled wind turbines. It is observed that by increasing the ADTS damping, the damping ratios of wind turbines are all first increased and then decreased for each mode; the modal frequencies all gradually increase and then remain constant. Therefore, it can be inferred that there exists an optimal ADTS damping for each mode, which renders the system capable of achieving a maximum modal damping ratio. As shown in Fig. 6(a), the maximum modal damping ratio is monotonically decreased for higher modes. Moreover, it is found that the value of the optimal
ADTS damping varies with the mode, which means that no one pair of ADTS has the ability to simultaneously reduce vibrations of all of the modes to their minimum. The possible reason for this effect is owing to the flexibility of ADTS, which may absorb the majority deformation when ADTS stiffness is inadequate, in this way, the motion of viscous damper will be limited. Similar conclusions have been drawn for the applications of viscous damper with support systems.

3.2.2. ADTS’s connecting stiffness

The influence of ADTS stiffness on the dynamic characteristics of wind turbines with ADTS is depicted in Fig. 7. It is obvious that the ADTS stiffness plays an essential role in affecting wind turbines’ dynamic characteristics. The phenomenon that modal damping ratios first increase and then decrease is still observed in Fig. 7. More importantly, the maximum achievable damping ratio for each mode is considerably influenced by the ADTS stiffness. From the definition in Eq. (18), it is known that the smaller $\beta$ value indicates the larger ADTS connecting stiffness. The maximum achievable modal damping ratios all gradually increase as ADTS has larger stiffness. In the meantime, the corresponding modal frequencies and required ADTS damping coefficients to achieve the maximum modal damping ratio also increase. In addition, it is noteworthy that the maximum damping ratio cannot be improved arbitrarily. As shown in Fig. 7, an upper limit in modal damping ratio/frequency is observed, where the increase of ADTS stiffness rarely enhances the maximum damping ratio.

As shown in Fig. 8 for a prescribed ADTS stiffness value ($\beta$), the maximum damping ratio, corresponding frequency, and the required ADTS damping are all decreased for higher modes. The above results also highlight the importance of ADTS stiffness. If it is not adequate (i.e. large $\beta$ value), ADTS will not provide...
Fig. 7. Influences of ADTS's connecting stiffness.
sufficient damping ratio for wind turbines no matter how much ADTS damping is adopted. Hence, the stiffness of ADTS should be guaranteed when applied to wind turbines.

3.2.3. ADTS position

ADTS position also significantly influences the maximum achievable damping ratio. The influence of ADTS position on the dynamic characteristics of wind turbines with ADTS is summarized in Fig. 8. It is observed that there are one, two, three, and four wave peaks in the damping ratio curves and corresponding frequency curves for the first, second, third, and fourth mode, respectively. Hence, there are one to four optimal ADTS positions for the 1st to the 4th mode. The optimal ADTS positions are near 0.7H for the first mode; 0.3H and 0.86H for the second mode; 0.18H, 0.56H and 0.93H for the third mode; and 0.13H, 0.43H, 0.70H, and 0.95H for the fourth mode, respectively. When the ADTS is located at the “valley” of one specific mode, it indicates that the ADTS becomes invalid for this mode. Apparently, there
Fig. 9. Influences of ADTS’s position on the maximum modal damping ratios, corresponding modal frequencies, and required damping coefficient.

exists no perfect position that is always effective for all modes. For instance, the optimal position for the first mode is about 0.7 H, but this position contributes little to the second mode.

The maximum modal damping ratios decrease as β increases, and this trend is generally in agreement with the corresponding modal frequencies. Hence, it is highlighted again that the stiffness of ADTS should be guaranteed to achieve satisfactory additional damping for wind turbines. For example, to achieve 10% damping
ratio for the first mode, the normalized ADTS stiffness ratio $\beta$ is not suggested to be larger than 4.0.

The required ADTS damping to achieve the maximum modal damping ratio generally decrease by increasing the height of ADTS position, i.e. the turbine bottom requires larger damping while the top part needs smaller damping. However, the damping curves are not monotonically decreasing for higher modes. For example, the 4th mode may amplify the required ADTS damping at some specific position when $\beta$ is low.

### 3.3. Brief summary

From the above parametric study, the influences of ADTS parameters can be briefly summarized as: (i) there exists an optimal ADTS damping for each mode, which renders the system capable of achieving the maximum modal damping ratio; (ii) the maximum modal damping ratio is influenced by ADTS stiffness and position; (iii) the ADTS stiffness should be guaranteed to provide satisfactory modal damping ratio; (iv) the optimal ADTS position for the first mode is about 0.7H, but this position contributes little to the second mode.

### 4. Dynamic Responses of Wind Turbines with ADTS

To evaluate the vibration attenuation of ADTS for wind turbines when subjected to winds and earthquakes, three cases—uncontrolled wind turbines (UC), wind turbines with ADTS, and wind turbines with TMD—are adopted for comparison hereafter.

#### 4.1. Numerical example

TMD is widely used for vibration control of wind turbines, and its optimal design is the most critical issue. This paper adopted an algebraic solution (typically PatternSearch algorithm) to minimize the amplitude of transfer function for wind turbines with TMD. The TMD mass was assumed as 36096 kg by referring. Then, the TMD damping and stiffness were optimized by minimizing the amplitude of transfer function. Detail parameters are listed in Table 1. The transfer function curves of wind turbines with/without optimized TMD are shown in Fig. 10.

<table>
<thead>
<tr>
<th>TMD</th>
<th>ADTS</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass (kg)</td>
<td>Damping (N/m)</td>
</tr>
<tr>
<td>36096</td>
<td>2.19e4</td>
</tr>
</tbody>
</table>
a typical flattened curve was observed for TMD, verifying the effectiveness of the optimized parameters.

Detailed parameters of the ADTS are presented in Table 1. It is noteworthy that the ADTS damping coefficient here is the equivalent amplified damping coefficient $C_d$; the damping coefficient $c_d$ used in the actual viscous damper shown in Fig. 3 is much smaller due to the damping amplification effect of the proposed ADTS (see Eq. (9)). For the sake of a fair comparison, the first mode damping ratios provided by TMD and ADTS are set to be equal to about 10.3%.

4.2. Wind-induced responses

Generally, the wind-induced vibration in the fore-aft direction is more critical than the side-to-side because the thrust force will act on the turbine blades in the fore-aft direction. Thus, the wind-induced responses of wind turbine in the fore-aft direction are investigated hereafter. The wind force acting on the turbine tower at height $z$ is calculated as

$$F_{sh}(z) = 0.5\rho_a C_D D (V + v)^2,$$  \( \text{(19)} \)

where $\rho_a$ is the air density; $C_D$ is the drag coefficient; $D$ is the tower diameter; $V$ is the mean wind velocity, where logarithmic law mean wind speed profile is adopted hereafter; $v$ is the fluctuating wind velocity, whose power spectra density used the Simiu spectrum. \( \text{[23]} \)

The thrust force at the turbine blade is approximately calculated as

$$F_T = 0.5\rho_a \pi R_T^2 V_S^2 (1 + 2v_S/V_S) C_T,$$  \( \text{(20)} \)
where $V_S$ and $v_S$ are the mean wind velocity and fluctuating wind velocity at the hub height; $R_T$ is the rotor radius, $C_T$ is the thrust coefficient calculated by

$$C_T = 4a_0(1 - a_0),$$

where $a_0 \in (0, 1)$ is the induction factor, and a value of 0.5 is adopted in this study. For more details about the wind effect on wind turbines, readers are suggested to refer to Ref. [4]. Fig. 11 compares the simulated wind speed spectrum with the target Simiu spectrum, where a satisfactory agreement is observed, validating the effectiveness of the simulation. Figure 12 presents time-history result of wind force components in the fore-aft direction. It is evident that the thrust force acting on
the blades is much larger than wind force loading on turbine tower, indicating that the thrust force is the main source for the wind force on the fore-aft directions.

Figure 13 compares the maximum wind-induced response envelopes of wind turbines with different control strategies subjected to an operating wind speed of 11.4 m/s at hub. Both wind-induced drift and acceleration responses are reduced by TMD as compared to UC. In the meantime, ADTS is also effective and achieves a very similar performance as compared to the TMD case since these two are designed to have the same first-mode damping ratios. For example, as compared to the TMD cases, ADTS further decreases the wind-induced drift of wind turbine tower by about 5.5% (from 4.329e-3 to 4.091e-3); while, the wind-induced acceleration response at the turbine tower top is further decreased by about 5.4% (from 0.3471 m/s² to 0.3282 m/s²).

The time-history results of wind-induced responses for wind turbines with different control strategies are shown in Fig. 14. In terms of the vibration mitigation, the proposed ADTS is slightly outperformed than TMD for either drift or acceleration responses. Moreover, it is observed that both wind-induced displacement and acceleration responses are long-period, indicating they are dominated by the first-mode vibration of wind turbine. This statement is further validated by the wind-induced
Fig. 14. Time-history results of wind-induced responses for wind turbines with different control strategies: (a) top displacement; (b) top acceleration.

Fig. 15. Wind-induced response spectra of wind turbines with different control strategies: (a) top displacement; (b) top acceleration.
response spectra shown in Fig. 15, where a significant peak is observed at a frequency of 0.3 Hz, whose value is also the fundamental frequency of UC. However, the application of ADTS significantly attenuates this peak.

4.3. Seismic responses

To investigate seismic performance of wind turbines with ADTS under real earthquakes, a set of 56 near-fault ground motions is selected from the database of Pacific Earthquake Engineering Research Centre (PEER) to match the target ASCE code spectrum for a site with characteristic values of $S_{DS} = 1.2 g$ and $S_{D1} = 0.8 g$. The design response spectrum is expressed as

$$S_a = \begin{cases} 
S_{DS} \left(0.4 + 0.6 \frac{T}{T_0}\right), & T < T_0 \\
S_{DS}, & T_0 \leq T \leq T_S \\
\frac{S_{D1}}{T}, & T_S < T \leq T_L \\
\frac{S_{D1}T_L}{T_L^2}, & T_L < T,
\end{cases}$$

(22)

where $S_{DS}$ is the design spectral response acceleration parameter at short periods; $S_{D1}$ is the design spectral response acceleration parameter at 1-sec period; $T$ is the fundamental period of the structure; $T_0 = 0.2 \frac{S_{DS}}{S_{DS}}$; $T_0 = \frac{S_{D1}}{S_{DS}}$; $T_L$ is the long-period transition period. For more details about the selected records, readers are suggested to refer to FEMA P695. Pseudo acceleration response spectra of all ground motion records are shown in Fig. 16.
Using the selected records, the seismic responses are obtained for the following: (a) UC, (b) wind turbines with ADTS, and (c) wind turbines with TMD. The maximum seismic response envelopes of wind turbines with different control strategies are compared in Fig. 17. From these results, it is observed that TMD is effective for controlling seismic drift response but ineffective in controlling the acceleration responses of wind turbines when subjected to earthquakes. As a comparison, the proposed ADTS significantly reduces both inter-story drift and acceleration responses of wind turbines caused by earthquakes. For example, as compared to the TMD cases, ADTS further decreases the displacement of wind turbine tower by about 22% (from $5.358 \times 10^{-3}$ to $4.183 \times 10^{-3}$), reducing the risk of failure in wind turbine towers. In the meantime, ADTS further reduces the seismic acceleration response of the tower top by about 38% (from $1.866 g$ to $1.16 g$), resulting in lower costs in the possible failure of the expensive equipment at the tower top.

Figure 18 presents the time-history results of seismic responses for wind turbines with different control strategies when subjected to one typical earthquake. It is observed that the proposed ADTS is more effective than TMD in controlling either seismic drift or acceleration responses. Moreover, the acceleration responses contain more components from higher modes. Thus, its time-history result is not as smooth as the drift responses.
Wind Turbines with ADTS

Fig. 18. Time-history results of seismic responses for wind turbines with different control strategies when subjected to RSN2114_DENALI_PS10-317: (a) top drift ratio; (b) top structural acceleration.

Fig. 19. Seismic response spectra of wind turbines with different control strategies when subjected to RSN2114_DENALI_PS10-317: (a) top displacement; (b) top acceleration.
Figure 19 shows the power spectral density (PSD) curves of the seismic response spectra of wind turbines with different control strategies. From this figure, it is known that the displacement response of turbine tower is dominated by its 1st mode vibration; while the higher modes vibrations (especially the second mode in this example) contribute a lot to the seismic acceleration responses. TMD effectively controls the first mode vibration of wind turbine with considerable reduction in the amplitude of PSD curve at a frequency of 0.3 Hz, at which the ADTS also performs similarly for the 1st mode amplitude. In terms of the second mode vibration, TMD rarely contributes to mitigating the second mode vibration since TMD is targeted on the first mode and it is de-tuned for the second mode and also the higher modes. Fortunately, although ADTS already provides very significant reduction in the first-mode vibration, it is still very effective for higher modes.

5. Conclusion

This paper proposed a novel ADTS to enhance the damping effect of wind turbines. Dynamic characteristics of the NREL 5MW wind turbine example with the proposed ADTS were systematically studied; the performance of ADTS in resisting earthquakes and winds was evaluated as compared to TMD. The most important findings of this paper are summarized below:

- The proposed ADTS can transfer the rotation of the turbine tower from the upper position to its bottom with damping amplification, providing a novel solution of enhancing the damping effect of wind turbines.
- There exists an optimal ADTS damping coefficient achieving the maximum modal damping for each turbine mode. Such maximum modal damping ratio is influenced by ADTS stiffness and ADTS position: the larger the ADTS stiffness, the larger the maximum damping ratio; the optimal ADTS position of the first mode is about 0.7 H, and the optimal ADTS positions of the second mode are at 0.3 H and 0.86 H.
- The proposed ADTS significantly reduces both drift and acceleration responses of wind turbines caused by winds and earthquakes. This is because ADTS is effective for both fundamental and higher modes, which is superior to TMD which may lose its effectiveness for higher modes’ vibration. For example, as compared to the TMD cases, ADTS further decreases the seismic displacement (acceleration) of wind turbine tower by about 22% (38%).

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