Vibration control of floating offshore wind turbines using liquid column dampers

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Abstract. This paper investigates the use of two different liquid column dampers for vibration control of spar-type floating offshore wind turbines (FOWTs). A 16-degree-of-freedom (16-DOF) aero-hydro-servo-elastic model for the FOWT is first established using multi-body based formulation and the Euler-Lagrangian equation, taking into consideration the full coupling of the blade-drivetrain-tower-spar vibrations, a collective pitch controller and a generator controller. It is found from the simulation results that due to the coupling to the spar rigid-body motion, the eigenfrequency of the tower vibration is significantly changed, which needs to be accounted for when tuning the liquid dampers. Tuned liquid column damper (TLCD) is investigated for controlling the lightly damped (due to low aerodynamic damping) tower side-side vibration and blade edgewise vibrations. Further, a newly proposed liquid column damper, the circular liquid column damper (CLCD) is also investigated for blade edgewise vibration control. The large centrifugal acceleration from the rotating blade makes it possible to use liquid column dampers with rather small masses for effectively suppressing edgewise vibrations. By properly tuning the dampers, both types of liquid column dampers are effective in mitigating tower and blade vibrations. Performances of the dampers are compared in terms of the control efficiency and practical considerations.

1. Introduction

Floating offshore wind turbines (FOWTs) must safely withstand harsh environmental impacts including stochastic wind and wave loads. The structural vibrations in the blades, support structure and other components of FOWT are generally larger than those of onshore or fixed-foundation offshore wind turbines. It is well known that for fixed-foundation wind turbines, tower side-side vibration and blade edgewise vibrations are very lightly damped due to the low aerodynamic damping, under normal operational conditions [1,2]. This is also the case for the FOWTs. Vibrations of the structural components not only reduce the fatigue life but also lead to increased maintenance, which is particularly unfavorable for FOWTs installed in deep water far offshore. Hence, there is even greater need for applying vibration control technique on the FOWT, in order to improve the fatigue life, stability and survivability of the FOWT. So far, individual blade pitch controllers [3] have been proposed for damping the problematic motions of the floating platform. This approach suffers from several drawbacks: 1) increased blade pitch actuator usage; 2) almost no effect on the lightly damped tower side-side and blade edgewise vibrations; 3) the robustness of the active control system not guaranteed. Therefore, more robust and cost-effective vibration control techniques/devices need to be developed for FOWTs located at deep waters far offshore. This paper investigated two different types of liquid column dampers for controlling tower side-side and blade edgewise vibrations of FOWTs, namely the tuned liquid column damper (TLCD) and the circular liquid column damper (CLCD). In
order to evaluate the performance of the liquid column dampers, a 16-DOF aero-hydro-servo-elastic model is established for the FOWT system, taking into consideration the full coupling of blade-drivetrain-tower-spar vibrations. Verified by good agreement with FAST, this model can be used as a basis for evaluating different vibration control techniques/devices applied to FOWTs. It is observed from the simulation results of this 16-DOF model that due to the coupling to the spar rigid motion, the eigenfrequency of the tower vibrational mode is changed comparing with that of the fixed-foundation tower, which is important to be accounted for when tuning the liquid dampers. From fully-coupled simulations (by combining the 16-DOF FOWT model with the mathematical models of the dampers), it is seen that by properly tuning, both types of liquid dampers are effective in mitigating structural vibrations of a spar-type FOWT.

2. 16-DOF aero-hydro-servo-elastic model of the FOWT

A 16-DOF aero-hydro-servo-elastic model is established for the FOWT. The model displays the most important characteristics of a floating wind turbine, including time-dependent system matrices, coupled spar-tower-blade-drivetrain vibrations, nonlinear aeroelasticity and nonlinear wave-spar interaction, as well as pitch and generator controllers. Three different coordinate systems have been introduced for formulating the equations of motion of the model. The rigid-body motions of the spar are described in the fixed, global \((X_1, X_2, X_3)\)-coordinate system with its origin fixed at the center of gravity of the undeformed system, the motions of the tower and drivetrain are described in a moving, local \((X'_1, X'_2, X'_3)\)-coordinate system with its origin fixed at the tower bottom, and the motion of each blade is described in the moving \((x_1, x_2, x_3)\)-coordinate system with its origin at the center of the hub. The \(x_3\)-axis is placed along the undeformed blade axis oriented from the hub towards the blade tip. Figure 1 shows a schematic representation of the FOWT model with definition of the coordinate systems and the DOFs.

The blade is modelled as an Euler-Bernoulli beam in the \((x_1, x_2, x_3)\)-coordinate system, with variable mass per unit length and variable bending stiffness in the flap-wise and edgewise directions. Each blade is related with two DOFs. \(q_1(t), q_2(t), q_3(t)\) denote the flap-wise tip displacement of each blade in the positive \(x_1\)-direction. \(q_4(t), q_5(t), q_6(t)\) denote the edgewise tip displacement of each blade in the negative \(x_2\)-direction. The related mode shapes are taken as the undamped fundamental eigenmodes \(\Phi_f(x_3)\) and \(\Phi_e(x_3)\) in the flap-wise and edgewise directions, when the blade is fixed at the hub. Further, the tower is also modelled as an Euler-Bernoulli beam in the \((X'_1, X'_2, X'_3)\)-
coordinate system, with its motions defined by the translational DOFs $q_7(t)$ and $q_9(t)$, indicating the fore-aft and side-sid e-vibrations, respectively.

The drivetrain is modelled by DOFs $q_9(t)$ and $q_{10}(t)$ using St. Venant torsional theory. The sign definition in Figure 1 applies to a gearbox with odd number of stages. $q_9(t)$ and $q_{10}(t)$ indicate the deviations of the rotational angles at the hub and at the generator from the nominal rotational angles $\Omega t$ and $N\Omega t$, respectively, where $\Omega$ is the rated rotational speed of the rotor and $N$ is the gear ratio. Correspondingly, $q_9(t)$ and $q_{10}(t)$ are deviations of the rotational speeds at the hub and the generator from the nominal values.

The spar is modeled by 6 rigid DOFs, $q_{11}(t), q_{12}(t), q_{13}(t), q_{14}(t), q_{15}(t)$ and $q_{16}(t)$, specifying surge, sway, heave, roll, pitch and yaw motions of the spar, respectively. The rigid body motions of the spar will induce velocity contributions to the tower and the blades, leading to highly coupled system matrices. For example, the tower segment velocity in the local $(X'_1, X'_2, X'_3)$- coordinate system is written as:

$$v_T(X'_3, t) = (\Phi_{TF} q_7 + q_{11} + (h_T + X'_3)q_{15})t' + (\Phi_{TS} q_8 + q_{12} - (h_T + X'_3)q_{14})t' + +q_{13}k'$$

where $\Phi_{TF}$ and $\Phi_{TS}$ are the fundamental mode shapes of the tower fore-aft and side-sid e-vibrations, respectively, $h_T$ is the distance from the tower bottom to the origin of the global coordinate system. $t'$, $k'$ are the unit base vectors of the $(X'_1, X'_2, X'_3)$- coordinate system. Clearly, the motions of the spar have significant impact on the motion of the tower. Similarly, the motions of both the spar and the tower will have an impact on the motion of the blades.

Multi-body based formulation has been carried out for the total kinetic energy $T$ and total potential energy $U$ of the system, and structural dynamics of the FOWT is derived using the Euler-Lagrange equation:

$$\frac{d}{dt} \left( \frac{\partial T}{\partial q} \right) - \frac{\partial T}{\partial q} + \frac{\partial U}{\partial q} = f_a + f_h + f_g + f_m + f_{gen} - C_s q$$

where $q(t)$ is the DOF vector, $f_a, f_h, f_g, f_m$ and $f_{gen}$ are the load vectors due to the aerodynamic loads, hydrodynamic loads, gravity loads, mooring system loads and generator torque, respectively. $C_s$ is the structural damping matrix. $f_a$ is calculated using the modified BEM with due consideration of the fluid-structure interaction, and $f_h$ is calculated using the Morrison’s equation including the fluid-structure interaction. For calculating $f_m$, we have established different mooring line models, including linear spring, quasi-static, lumped mass and FE models. In the present paper for the numerical efficiency, quasi-static mooring line model is coupled to the 16-DOF model for evaluating the performance of the liquid dampers. Furthermore, a generator controller and a collective PI pitch controller are also included in the model.

![Figure 2: Comparison of steady state responses between FAST and the 16-DOF model](image-url)
The NREL 5 MW baseline wind turbine [4] has been used to formulate the turbine structural systems, and the spar type floating foundation together with the mooring line parameters are taken from [5]. The rotational sampled turbulence is simulated using an AR model following the procedure in [2], and the JONSWAP spectrum has been used for generating irregular sea waves. Using the 4th order Runge-Kutta method with time step of 0.02 s, simulations of the 16DOF FOWT model have been carried out in the time domain. The steady state responses of the developed 16-DOF FOWT model are compared with that from FAST [4], for wind speeds ranging from cut-in (3 m/s) to cut-out (25 m/s). Good agreement has been obtained for all responses (including structural responses and controller responses), and the results of some DOFs are shown in Figure 2.

Table 1 shows the comparison of the system natural frequencies of the land-based (fixed-foundation) turbine between FAST and the 16-DOF model, by using eigenvalue analysis. In this case, the rigid body DOF’s of the floating foundation of the 16-DOF model is deactivated, resulting in a 10-DOF fixed-foundation model.

Table 1: Comparison of system natural frequencies in [rad/s] for the land-based wind turbine.

|                  | Blade flapwise | Blade edgewise | Tower fore-aft | Tower side-side |
|------------------|----------------|----------------|----------------|----------------|
| 10-DOF           | 4.178          | 6.846          | 2.124          | 2.100          |
| FAST             | 4.394          | 6.847          | 2.036          | 1.960          |
| %                | -4.9           | 0.0            | 4.4            | 7.1            |

Table 2: Comparison of component and coupled system natural frequencies in [rad/s] for the FOWT.

|                  | Tower fore-aft | Tower side-side | Spar roll | Spar pitch |
|------------------|----------------|-----------------|-----------|------------|
| 16-DOF - component | 2,496          | 2,484           | 0,2070    | 0,2069     |
| 16-DOF - coupled  | 3,106          | 3,111           | 0,2070    | 0,2069     |
| %                | -19.62         | -20.18          | 0.00      | 0.00       |

For the FOWT, the tower motion is strongly coupled with the spar rotations. This is seen in Table 2 by comparing the natural frequencies based on the component and coupled frequencies. Further, Figure 3 shows the tower side-side top displacement in both the time domain and frequency domain, for mean wind speed $V_0 = 15$ m/s, turbulence intensity $I = 0.1$, the significant wave height $H_s = 6$ m and the wave peak period $T_p = 10$ s. From Figure 3(b), it is seen that due to the coupling to the spar roll motion, the eigenfrequency of the tower side-side vibration is shifted from 2.48 rad/s for the fixed foundation to 3.14 rad/s for the present case of floating foundation. This can be verified by evaluating the off-diagonal terms in the system mass matrix between the 8th DOF and the 14th DOF.

![Figure 3](image-url)  

Figure 3: Tower side-side top displacement. (a) Time series. (b) Fourier amplitude spectrum

3. Two types of liquid column dampers for the FOWT
The main advantages of liquid dampers are the ease of fabrication and installation, and minimal maintenance after installation, which make the devices very cost-effective and especially suitable for FOWTs placed far offshore. For blade-mounted liquid dampers, the large centrifugal acceleration from the rotating blade makes it possible to use dampers with rather small liquid masses for effectively suppressing blade edgewise vibrations. For the design and analysis of tower-mounted dampers, it’s very important to take into consideration the shifted tower frequency due to coupling with the spar.
3.1 Tuned liquid column damper (TLCD) for blade

As shown in Figure 4, the TLCD is composed of a U-shaped tube (one horizontal column and two vertical columns) partly filled with liquid, with an orifice installed at the center of the horizontal column for energy dissipation. When this U-shaped liquid container is mounted inside the blade with a changing azimuthal angle, it should be fabricated in a closed configuration so as to prevent the liquid from running out of the tube. In this case, an extra slim tube is included connecting the two vertical columns, in order to balance the pressure above the liquid column during oscillation.

![Figure 4: Tuned liquid column damper (TLCD) for edgewise blade vibration control](image)

Unlike the conventional uniform cross-section, in the present study the cross-sectional areas of TLCD are proposed to be non-uniform, where the vertical and horizontal column cross-sectional areas are denoted as \( A \) and \( A_0 \) respectively, and \( \alpha = A / A_0 \) is the area ratio. This makes the design of the damper more flexible. The horizontal width, the vertical height, and the total length of the liquid inside TLCD are denoted \( R \), \( B \) and \( L \), respectively, where \( L = 2H + B \). Thus, the overall mass of the liquid inside the TLCD is:

\[
m = (2HA + BA_0) \rho \tag{3}
\]

where \( \rho \) is the mass density of the liquid. \( \gamma = B / L \) is defined as the horizontal length ratio.

Energy dissipation of the liquid can be characterized and adjusted by the size of the orifice opening. The damping force of the liquid motion when it passes through the orifice is expressed as:

\[
F_d = c_d |v_0| \dot{v}_0 \tag{4}
\]

where \( \dot{v}_0 \) is the velocity of the liquid in the horizontal column, which fulfills the continuity condition \( \dot{v}_0 A_0 = \dot{v} A \), with \( \dot{v} \) being the velocity of the liquid in the vertical column. \( c_d \) is the damping coefficient indicating the energy dissipation, as specified in the following form:

\[
c_d = \frac{1}{2} \xi \rho A_0 \tag{5}
\]

where \( \xi \) is the head loss coefficient determined by the opening ratio of the orifice.

The motion of the liquid inside TLCD can be described by the degree of freedom \( \psi(t) \), indicating the displacement of the liquid in the vertical column. When coupled to the edgewise vibration of the rotating blade, the equations of motion of the coupled blade-TLCD system become highly nonlinear, and detailed equations are given in [6]. The angular eigenfrequency \( \omega_d \) of TLCD are obtained as [6]:

\[
\omega_d = \frac{\sqrt{2x_0 - L(1 - \gamma)}}{L + (\alpha - 1) \gamma L} \Omega \tag{6}
\]

where \( x_0 \) is the coordinate at the blade where the TLCD is mounted, \( \Omega \) is the rotational speed of the rotor. The design parameters of TLCD tune out to be the frequency ratio (the ratio between \( \omega_d \) and the eigenfrequency of the structure), overall mass \( m \), area ratio \( \alpha \), horizontal length ratio \( \gamma \), and head loss coefficient \( \xi \). The optimal performance of TLCD is obtained by carefully choosing all these parameters.
3.2 Circular liquid column damper (CLCD) for blade

Inspired by the pendulum, the newly proposed liquid damper is a circular tube/column partly filled with certain amount of liquid, and hence is termed as a circular liquid column damper (CLCD), as shown in Figure 5. The cross-section of the circular tube is circular as well. Due to the axisymmetric nature of the damper geometry, it allows for consistent definition of local dynamic behavior of the liquid irrespective of the azimuthal angle of the rotating blade, and is therefore especially suitable for blade vibration control.

![Figure 5: Circular liquid column damper (CLCD) for edgewise blade vibration control](image)

The radius of the circular tube (distance from the center point \(O\) to the central axis of the tube) is denoted \(R\), and the radius of the cross-section is denoted \(r\). Hence, the total dimension of the damper can be calculated as \(H = 2(R + r)\), and \(\beta = r/R\) is defined as the radius ratio.

Inside the tube, the liquid is assumed to be connected, filling a segment with a central angle of \(2\Theta_0\) of the completer circle. The motion of the liquid is specified by the degree of freedom \(\theta(t)\), indicating the clockwise rotation of the gravitational center of the liquid \([7]\). From dynamics point of view, the liquid inside the circular tube virtually acts as a pendulum during its oscillation. Therefore, the CLCD can be represented as an equivalent mathematical pendulum with the equivalent mass \(m_e\) and the equivalent length \(R_e\), which are proved to be \([7]\):

\[
m_e = m \frac{1 + \frac{1}{4} \beta^2}{(1 + \frac{1}{4} \beta^2)^2}, \quad R_e = R \left(1 + \frac{1}{4} \beta^2\right)
\]

where \(m\) is the overall mass of the liquid.

Highly nonlinear equations of motion of the blade-CLCD system have been derived using Lagrange’s equation, and the angular eigenfrequency \(\omega_d\) of CLCD are obtained as \([7]\):

\[
\omega_d = \sqrt{\frac{x_0}{R_e}} \Omega
\]

(8)

Similar to TLCD, the energy dissipation of the liquid can be characterized and adjusted by the size of the orifice opening, and the damping force of the liquid motion due to passage of liquid through the orifice is:

\[
F_d = c_d |\dot{\theta}| \dot{\theta}
\]

where \(\dot{\theta}\) is the rotational velocity of the gravitational center of the liquid. The damping coefficient \(c_d\) due to the orifice is written as \(c_d = \frac{1}{2} \xi \rho r^2 R^3\) \([7]\), with \(\xi\) being the head loss coefficient. The design parameters of CLCD tune out to be the frequency ratio (the ratio between \(\omega_d\) and the eigenfrequency of the structure), overall mass \(m\), radius ratio \(\beta\), and head loss coefficient \(\xi\). The optimal performance of CLCD is obtained by carefully choosing all these parameters.

It is clear that CLCD combines the property of TLCD where the energy dissipation can be adjusted by the orifice opening, and the property of a pendulum which has a larger effective mass than the
TLCD. As will be shown in the following simulation results, CLCD with the same amount of liquid mass will have better damping effect on the blade edgewise vibration than TLCD.

### 3.3. TLCD for tower vibration control of FOWT

Only TLCD is considered for tower vibration control, because its flat bottom facilitates its installation to the nacelle (tower top). Comparing with the bladed-mounted TLCD, the analysis and design of the tower-top mounted TLCD is simpler, the theory (as well as the equation of motion) of which has been well developed for the building-mounted TLCD in civil engineering field [8]. This is exactly the case for the fixed-foundation wind turbines. The angular eigenfrequency of the TLCD is expressed as:

$$\omega_d = \frac{2g}{\sqrt{L+L'(1-\xi)\gamma L}} \tag{10}$$

where $g$ is the gravitational acceleration. Comparing with Eq. (6), it is seen that the motion of the tower-mounted TLCD is dominated by gravitational acceleration, while the motion of the blade-mounted TLCD is dominated by the centrifugal acceleration $x_0\omega^2$, which is much larger than $g$. Therefore, large amount of liquid mass is needed for the tower-mounted TLCD, while it is possible to use the TLCD with rather small masses for effectively suppressing blade edgewise vibrations, as will be shown in the simulation results.

However for FOWTs, special attention needs to be paid when designing tower-mounted TLCD using Eq. (10). As shown in Figure 3, due to the coupling to the rigid-body roll motion of the floating spar, the eigenfrequency of the tower side-side vibration is shifted from 2.48 rad/s for the fixed foundation to 3.14 rad/s for the present case. The TLCD thus needs to be designed so that its angular eigenfrequency $\omega_d$ is tuned to the shifted eigenfrequency of the tower side-side mode (3.14 rad/s). It should also be noted that since the total length of the liquid column is inverse proportional to $\omega_d$, it actually becomes more easier to tune the TLCD for FOWT than for the fixed foundation wind turbines, because the required total length of the liquid column is reduced.

### 4. Performance of the liquid column dampers on the FOWT

Fully-coupled simulations have been carried out by combining the 16-DOF FOWT model (with quasi-static mooring line model) with the mathematical models of the dampers, in order to evaluate the performance of different liquid column dampers.

#### 4.1. Performance of TLCD on the blade edgewise vibration control

![Figure 6: Performance of the TLCD](image)

Figure 6 shows the performance of TLCD on suppressing blade edgewise vibrations in both time and frequency domains, when $V_0 = 15$ m/s, $I = 0.1$, $H_u = 6$ m and $T_p = 10$ s, and the FOWT is under normal operational conditions. The TLCD is placed at the location of $x_0 = 45$ m of the 63-m long blade, with the following design parameters: total liquid mass 41.7 kg (corresponding to 3% modal
mass of the blade), area ratio \( \alpha = 1 \), \( B = 2.4 \) m, \( H = 0.52 \) m, \( \xi = 5 \), and the frequency ratio 0.98. It is seen that the modal loads from gravity result in a large harmonic motion in edgewise direction, with an angular frequency of 1P (1 per rev, corresponding to the rotational speed of the rotor 1.27 rad/s). On top of this deterministically harmonic-varying motion, oscillations related to the edgewise eigenvibration are also clearly observed. The TLCD has no effect on the gravity induced 1P motion, but effectively add damping into the edgewise eigenvibration. From the Fourier amplitude in Figure 6b), the peak corresponding to edgewise eigenfrequency is significantly reduced by the TLCD, while the peak at 1P cannot be influenced by the damper. As mentioned, due to the large centrifugal acceleration, the TLCD with a mass of 41.7 kg can be very effective in suppressing blade edgewise vibrations. To quantify the reduction in the blade edgewise vibrations, fatigue equivalent loads are calculated and compared. This is done by first calculating the loads based on Euler-Bernoulli beam theory and then performing rainflow counting on the time series of the loads. From the rainflow counting, the range and number of cycles are used to estimate the fatigue equivalent loads with \( N = 10^7 \) cycles and a slope of the SN curve of \( m = 9 \) for the glass fiber blades and \( m = 4 \) for the steel tower. For Figure 6, the fatigue equivalent loads at the blade root are 1253.8 kNm without TLCD and 1124.7 kNm with TLCD, representing a 10.3% reduction with the TLCD.

On the contrast to the gravity induced 1P motion, the edgewise vibration (in the fundamental edgewise mode) is stochastic in nature, and is influenced by both the stochastic external loads and the operational condition of the turbine. Under some conditions, the aerodynamic damping becomes negative, and large amplitude oscillation or even aeroelastic instability may take place. Figure 7 shows the performance of the TLCD (same parameters used as in Figure 6) under aeroelastic instability conditions, where the structural modal damping of the blade is set to be -0.8 to mimic the negative aerodynamic damping. It is seen that the attached TLCD can totally eliminated the instability, implying that significant damping is introduced by the TLCD to the fundamental edgewise mode to overwhelm the negative aerodynamic damping. The fatigue equivalent loads at the blade root are reduced by 54.2% with the TLCD.

4.2. Performance of CLCD on the blade edgewise vibration control

Figure 8 shows the performance of CLCD on suppressing blade edgewise vibrations in both time and frequency domains, when \( V_0 = 15 \) m/s, \( I = 0.1 \), \( H_8 = 6 \) m and \( T_p = 10 \) s, and the FOWT is under normal operational conditions. The CLCD is also placed at the location of \( x_0 = 45 \) m of the 63-m long blade, with the following design parameters: total liquid mass 41.7 kg (corresponding to 3% modal mass of the blade), radius ratio \( \beta = 0.05 \), \( R = 1.74 \) m, \( \xi = 6 \), and the frequency ratio 0.98. The fatigue equivalent loads at the blade root are reduced by 13.0% with the CLCD. Similar results have been obtained comparing with Figure 6, since both CLCD and TLCD are mounted at the same location and the total liquid mass is the same (41.7 kg). However by careful comparing Figure 8 and Figure 6, one can observe that CLCD actually introduces more damping into the edgewise mode, and
the peak corresponding to edgewise eigenfrequency is almost totally eliminated by the CLCD (although the 1P peak is still not influenced). This is due to the fact that CLCD has the property of a pendulum, and thus has larger effective mass than the TLCD (the total liquid mass is the same). Actually, the effective mass of the TLCD is only the liquid inside the horizontal column.

Figure 8: Performance of the CLCD \((x_0 = 45 \text{ m}, \text{liquid mass } m = 41.7 \text{ kg}, R = 1.74 \text{ m}, \beta = 0.05, \xi = 6, \text{frequency ratio}=0.98)\) on blade edgewise vibration control, under normal operational conditions (a) time-series. (b) Fourier amplitude spectrum.

### 4.3. Performance of TLCD on the tower side-side vibration control

Figure 9 shows the performance of TLCD on suppressing tower side-side vibrations in both time and frequency domains, when \(V_0 = 15 \text{ m/s, } l = 0.1, H_s = 6 \text{ m and } T_p = 10 \text{ s, and the FOWT is under normal operational conditions. The design parameters are: total liquid mass } 1.16 \times 10^4 \text{ kg, area ratio } \alpha = 1, B = 1.24 \text{ m, } H = 0.41 \text{ m, } \xi = 10, \text{frequency ratio}=0.99)\). The TLCD is tuned to the shifted tower frequency 3.14 rad/s. It is observed that the TLCD effectively introduce damping to the tower side-side vibrational mode, and the peak corresponding to the shifted tower side-side eigenfrequency (3.14 rad/s) is almost totally eliminated by the damper. The fatigue equivalent loads at the tower base are reduced by 14.5% with the TLCD. Comparing with the blade-mounted TLCD, the liquid mass used for the tower-mounted TLCD is much larger, partly due to the fact that the motion of the liquid is dominated by gravity instead of the centrifugal acceleration, and partly due to the fact that the tower is heavier than the blade. It has also been seen that if the TLCD is tuned to the fixed foundation tower frequency (2.48 rad/s), the effect of the TLCD on tower side-side vibration control becomes negligible (results not shown here).

Further, it should be noted that in all the simulations above, the parameters of the liquid column dampers are just sub-optimal by simple try-and-error procedure. Systematic optimizations of the parameters of the liquid damper have not been carried out in the present study yet, due to the fact that the simulation of the fully-coupled FOWT-damper system is computationally expensive.

### 4.4. Discussion on the practical application
It has been proved that CLCD with the same amount of liquid mass generally has better damping effect than the TLCD because of the larger effective mass. Therefore CLCD is superior to TLCD in this regard. However, there are more design parameters for TLCD, making the design of this device more flexible. Actually, the geometry of the TLCD can be easily adjusted according to the blade configuration, which is especially helpful when the space of the outer part of the blade is limited. CLCD, on the other hand, may suffer from the space constraints inside the blade, since the geometry of the CLCD is fixed as soon as the radius is fixed (by tuning condition). It is observed that the motion of the liquid mass for CLCD only takes place in a small region of the circle, making it possible to devise a damper with an arc tube rather than a complete circular tube. This may facilitate the installation of CLCD inside the blade when the space is limited. Still, there are some practical challenges in the implementation of the dampers inside the blades because of the space constraints and the risk of fluid leakage. Replacement of the dampers might also be a design challenge to overcome for the wind turbine design engineers.

5. Conclusions
This paper investigated the performance of two types of liquid column dampers on vibration control of FOWTs subjected to wind and wave loads. A 16-DOF aero-hydro-servo-elastic model is established for the spar-type FOWT, taking into account the full coupling of the blade-drivetrain-tower-spar vibrations, a collective pitch controller and a generator controller, and nonlinear aeroelasticity and hydrodynamics. The simulation results from this 16-DOF model agree very well with that from FAST, and the model is considered a good basis for evaluating the performance of the liquid column dampers when fully-coupled simulations are carried out.

Two types of liquid column dampers, the TLCD and the CLCD, have been introduced in detail, in terms of the dynamic modelling principle and the design parameters. Both dampers with a very small amount of liquid mass can effectively suppress blade edgewise vibrations, while the CLCD is slightly more efficient than the TLCD. For FOWTs, the eigenfrequency of the tower side-side mode is significantly changed comparing with that of the fixed-foundation case, due to the coupling effect to the rigid-body roll motion of the spar. Therefore when tuning the tower-mounted liquid column dampers for FOWT, special attention needs to be paid and the dampers should be tuned to the shifted tower side-side frequency. Simulation results show that by properly tuning the dampers, both types of liquid column dampers are effective in mitigating structural vibrations of FOWTs.

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