Application of two passive strategies on the load mitigation of large offshore wind turbines

Rasoul Shirzadeh and Martin Kühn

ForWind-Center for Wind Energy Research, Institute of Physics, Carl von Ossietzky University of Oldenburg, Küpkersweg 70, 26129 Oldenburg, Germany

E-mail: rasoul.shirzadeh@forwind.de

Abstract. This study presents the numerical results of two passive strategies to reduce the support structure loads of a large offshore wind turbine. In the first approach, an omnidirectional tuned mass damper is designed and implemented in the tower top to alleviate the structural vibrations. In the second approach, a viscous fluid damper model which is diagonally attached to the tower at two points is developed. Aeroelastic simulations are performed for the offshore 10MW INNWIND.EU reference wind turbine mounted on a jacket structure. Lifetime damage equivalent loads are evaluated at the tower base and compared with those for the reference wind turbine. The results show that the integrated design can extend the lifetime of the support structure.

1. Introduction

In recent years, the size of offshore wind turbines is further increased beyond the 5 MW class to reduce the cost of energy. The cost effective design of support structures for such large machines is still a challenging task as the rotor diameter and tower height are exceeding 100 m. The support structure eigenfrequencies are analysed in the early stage of the design procedure to prevent significant resonances between the structural eigenfrequencies and excitations from waves, rotor frequency and its multiples. This can be acquired via the Campbell diagram which plots the eigenfrequency of the entire wind turbine system against the rotor speed including the harmonic excitations 1P, 3P, etc. Figure 1 illustrates qualitatively the trends for the rotor rotational frequency (1P) and blade passing frequency (3P) ranges along with the first eigenfrequency of the support structure as a function of the wind turbine size. In the design procedure of large offshore wind turbines, i.e. 7.5+ MW, with jacket structures, a strong and severe blade passing (3P) resonance is expected at low rotor speeds [1].

Figure 2 shows the Campbell diagram for the 10MW INNWIND.EU reference turbine [2]. It can be seen that at rotor speed of 5.7 rpm, the blade passing frequency (3P) coincides with the first natural frequency of the system. The rotor frequency (1P) is however not problematic as it is found far outside of the operational region. In order to reduce the dynamic excitation, a rotor speed exclusion zone between 5.2 rpm and 6.3 rpm is considered. Compared to the initial exclusion zone chosen in the preliminary stage [3], the exclusion zone is shifted downward by a reduction of the cut-in speed from 5 rpm to 4.5 rpm.

The larger the blades and the support structure of the wind turbine, the higher the bending loads. Consequently, strongly increased fatigue loads are experienced by critical components

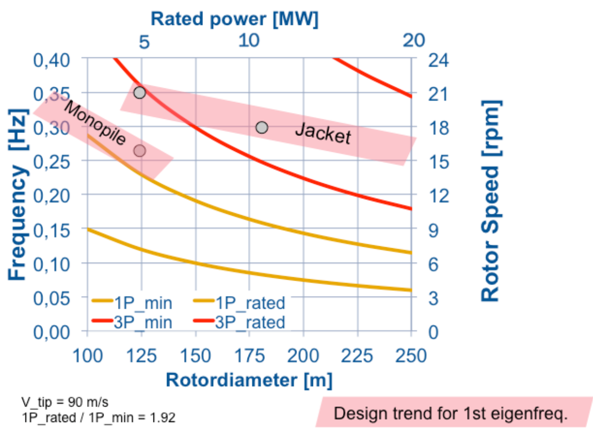


Figure 1. Design trends on the first design eigenfrequency and both the rotor and the blade resonance ranges for three-bladed offshore turbines in the 5 to 20MW class [1].

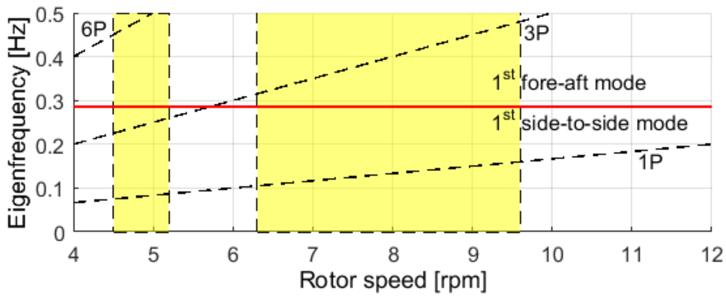


Figure 2. Campbell diagram of the INNWIND.EU 10MW reference turbine. The yellow marked areas represent the operational ranges of the turbine. The cut-in speed is lowered to avoid overlapping with the exclusion zone. The first fore-aft and side-side modes are closely spaced.

e.g. the tower base. Innovations are required in lowering the loads experienced by the support structure. This goal can be met through a variety of load mitigation strategies e.g. implementation of passive or (semi)-active damping devices and different control and regulation concepts. Nowadays, the application of a tuned mass damper (TMD) is becoming increasingly practical for the load mitigation of offshore wind turbines. Previous works have shown the potential for the integration of TMDs in the tower top location [4]. In addition, different solutions have been proposed for the application of semi-active or active dampers, e.g. toggle-brace viscous fluid damper (VFD), tuned liquid column damper (TLCD), magnetorheological (MR) damper and hybrid mass dampers, see [5–7].

To control the dynamic excitation of the support structure, two innovative concepts are designed and integrated in the INNWIND.EU 10MW reference wind turbine (RWT). In the first study, the results of the integration of a passive structural damper to reduce the support structure loads are presented. A passive tuned mass damper (TMD) is designed to realize the best configuration according to the calculated tower base lifetime and damage equivalent loads (DELs) for the reference number of cycles of $N_{ref} = 10^7$ and the S-N curve slope of $m = 4$. During the second analysis, the numerical model of a toggle-brace VFD is developed. Aeroelastic simulations are performed in DNV GL Bladed software [8] for the offshore 10MW INNWIND.EU

RWT mounted on a jacket structure. The loads obtained from two concepts are then compared with those for the reference turbine. We show that the applied loads can be effectively mitigated by the use of damping strategies which would result in an extension in the lifetime of the support structure.

2. Implementation of Wind Turbine Innovations

2.1. Description of the performed studies

The performed studies are based on the offshore 10MW INNWIND.EU RWT [9]. A schematic picture of the jacket and the reference wind turbine (RWT) structures as well as the considered coordinate system are displayed in Figure 3. The design water depth is applied according to the North Sea site with a water depth of 50 m. The support structure consists of a 4-leged, x-braced jacket structure. The 3D turbulent wind field with the Kaimal spectrum is generated for six random seeds while the wave kinematics is modelled based on the irregular wave model with the JONSWAP spectrum. No yaw misalignment angle is assumed in this analysis. The design load case (DLC) 1.2 based on wind class IA according to IEC61400-1 standard [10] is considered and fatigue loads are calculated during the full operating wind speed range. The wind and wave characteristics are listed in Table 1.

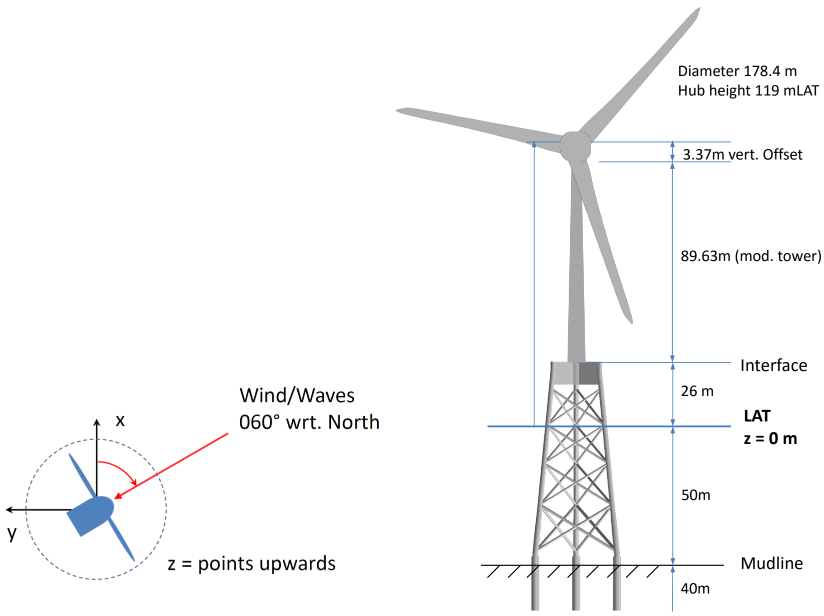


Figure 3. The global coordinate system (left) and a schematic drawing of the RWT and jacket structures (right).

2.2. Description of the load mitigation concepts

Two load mitigation concepts namely the tuned mass damper and the viscous fluid damper which have already been introduced in [11] are considered. These concepts are integrated into the final design of the support structure to ensure an optimised and affordable design. Figure 4 illustrates schematic models of these two concepts. Time series of combined wind and hydrodynamic loads are extracted for both concepts to calculate the Fatigue Limit State (FLS) of the offshore wind

Table 1. Environmental conditions for the wind and wave parameters.

Wind speed [m/s]	Long. Turb., I_u [%]	Lat. Turb., I_v [%]	Vert. Turb., I_w [%]	H_s [m]	T_p [s]	Occurrence [hours/year]
4	20.4	16.30	10.20	1.10	5.88	874.7
6	17.5	14.00	8.75	1.18	5.67	992.8
8	16.0	12.80	8.00	1.31	5.67	1181.8
10	15.2	12.16	7.60	1.48	5.74	1076.3
12	14.6	11.68	7.30	1.70	5.88	1137.2
14	14.2	11.36	7.10	1.91	6.07	875.6
16	13.9	11.12	6.95	2.19	6.37	764.7
18	13.6	10.88	6.80	2.47	6.71	501.3
20	13.4	10.72	6.70	2.76	6.99	336.0
22	13.3	10.64	6.65	3.09	7.40	289.4
24	13.1	10.48	6.55	3.42	7.80	130.4

turbine in operational conditions. In the current study, no wind-wave misalignment is considered. Fatigue loads are evaluated at the tower base and compared with the reference design.

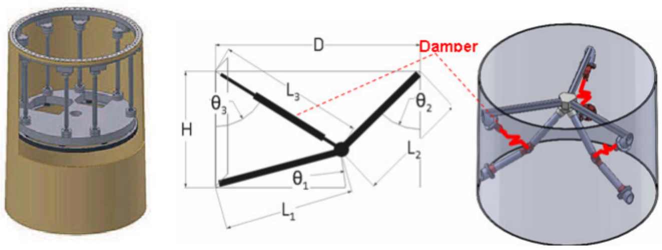


Figure 4. Two passive strategies integrated in the tower top of the 10MW INWWIND.EU RWT, the TMD [12] (left) and the toggle brace viscous fluid damper [13] (right).

During the first concept, the numerical modeling of a passive tuned mass damper mounted on the tower top is performed (see Figure 4). This consists of a mass-spring-damper system which is characterised by its mass, the resonance frequency and the damping factor. The damper is omnidirectional meaning that it can vibrate at all direction. Tuned mass dampers are widely implemented and tested in recent commercial wind turbines, e.g. Vestas V90-3MW. However, the initial INNWIND.EU 10MW reference jacket is designed without considering any damper in the tower. As a consequence, the initial jacket design is not optimised and needs further improvements. The integration of a TMD has the potential to improve the jacket design by reducing interface loads. The interface spot, as shown in Figure 3, is the connection point between the tower and the jacket foundation, i.e. the tower base.

According to [4], the TMD mass is approximately either 8% of the nacelle mass or 6% of the tower top mass. Other studies propose to use a value between 2-4% of the modal mass associated with the fundamental lowest eigenfrequency which is commonly used for civil engineering structures [4]. In this analysis, simulations are performed with a TMD with mass ratios of 1% and 2% with respect to the modal mass. The resonance frequency of the TMD is tuned to the first fundamental mode of the support structure (the first fore-aft mode) and is able to reduce the amplitude of vibrations around this frequency.

The second concept describes the mathematical representation of a VFD. VFDs have already been used in civil engineering structures to dissipate seismic energy. Up to now, the application of VFDs in wind turbines did not come into practice and further investigations are required to accommodate a damper network system of this type in the tower. The optimal location of the damper is a challenging task. In this study, we assume that the VFD is installed at the tower top, whereas maximum vibrations occur and hence a higher damping efficiency is desired.

The design formulas for a VFD system are based on the study described in [14]. For a system represented in Figure 4, the following relation exists:

$$u_D = fu \tag{1}$$

where u_D is the relative displacement along the axis of the damper, u is the relative displacement of the attachment points and f is the magnification factor. The magnification factor depends on the configuration of the bracing system and is usually larger than 1. For a diagonal toggle-brace-damper system, it equals to $\cos\theta$ where θ is the inclined angle of the damper [14]. The horizontal component of the force exerted by the damper on the structure, F, is obtained from:

$$F = fF_D \tag{2}$$

with F_D, the damper force. For a VFD, the damper force along its axis is proportional to the velocity and can be written as below:

$$F_D = C\dot{u}_D^{\alpha} \tag{3}$$

where C is the damping coefficient of the damper, $\dot{u}_D$ represents the relative velocity between the attachment points of the damper and α is the damper nonlinearity. In this paper, we assume $\alpha = 1$ which models a linear damper. Combining Eqs. 1 and 3 and knowing that any real number can be expressed as the product of its absolute value and its sign function, the magnitude of the damping force can be expressed as:

$$F_D = Cf^{\alpha}|\dot{u}|^{\alpha}sgn(\dot{u}) = C_0|\dot{u}|^{\alpha}sgn(\dot{u}) \tag{4}$$

The effective damping coefficient, $C_0 = Cf^{\alpha}$, has an important impact on the damper force. The damper force is applied to the structure and mitigates the external forces exerted on the structure. Several strategies can be used to calculate the viscous damper force. One of the most practically available parameters is the acceleration which can be easily measured using accelerometers.

Figure 5 demonstrates the mechanism to calculate the viscous damper force using the tower top simulated accelerations. A Simulink model of a VFD is developed and connected to Bladed via an external Dynamic-Link Library (DLL). The simulated time series of accelerations at the ending points of the damper where it is attached to the tower are recorded. Velocity signals can be attained by integrating the simulated accelerations. The damper force, F_D, is then obtained in Simulink using both velocities and the effective damping coefficient. This force is divided in two and returned to the Bladed whereas it will be applied as reaction forces at the damper ending points (see Figure 5). For multiple dampers, the damper force is split up into several components based on the orientation of the viscous dampers. The damper model performs as a passive device if a constant damping coefficient is assumed. Alternatively, for a semi-active device, the effective damping coefficient can be provided as a lookup table which is calculated and scheduled based on external or operating conditions. For such a damper, it acts semi-actively as the damper force is adapted based on the environmental conditions. In this study, however, a constant effective damping coefficient is chosen at all wind speeds.

3. Design Load Verification

In this section, results of the load verification with the integration of two concepts are discussed. Figure 6 demonstrates the DELs of the tower base moment in the fore-aft (M_y) and side-to-side

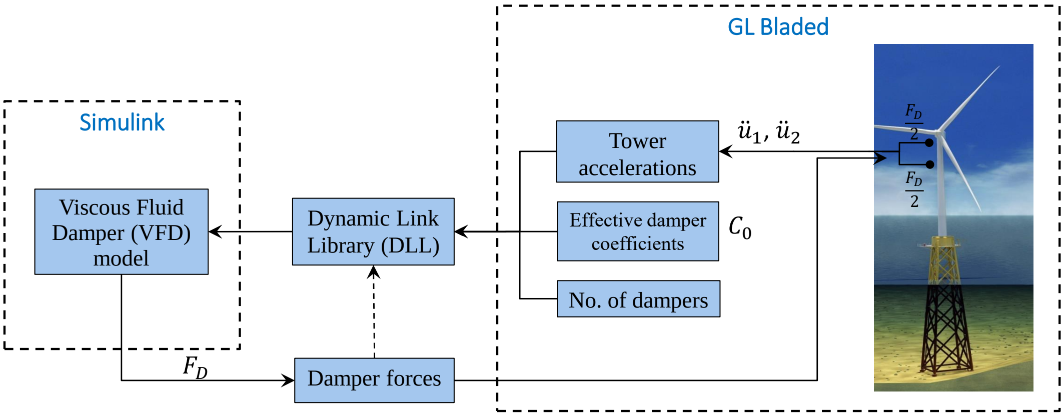
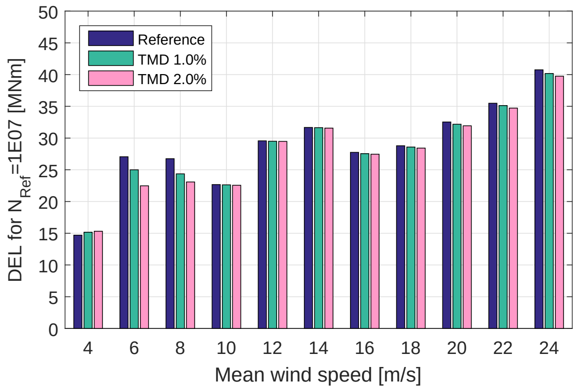


Figure 5. Numerical modeling of the VFD shows the mechanism to calculate the damper forces using the tower accelerations.

(M_x) directions, respectively. M_x and M_y are represented in the global coordinate system with X and Y axes pointing towards the north and west, respectively, when nacelle is faced to the wind inflow. The results are only for one set of load setup where the wind-wave misalignment angle is zero. The simulations are carried out with six random seeds for the turbulent wind and DELs are averaged at each wind speed. For all results shown here, the exclusion zone is activated. The influence of the TMD is not significant in the fore-aft direction, especially near the rated wind speed, while the TMD can effectively mitigate the fatigue loads in the sideways direction. It should be noted that despite the rotor speed exclusion zone is active, a resonance can be seen for the reference design which is further mitigated by the TMD even in the fore-aft direction.



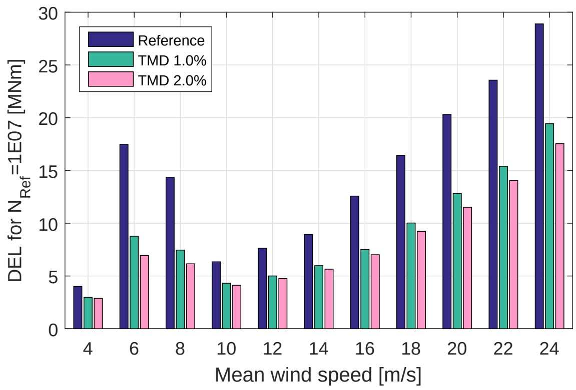


Figure 6. DELs of the tower base moment in the fore-aft, M_y, (left) and sideways, M_x, (right) directions for different mass ratios of the TMD.

This is in agreement with findings of Kuhnle [11] where it has been shown that the DELs can be improved by the application of a tuned mass damper at the tower top location. In addition, the TMD is more effective for the sideways direction which is due to the low aerodynamic damping in this direction. The aerodynamic damping is the dominant phenomenon in the fore-aft direction and therefore the performance of the TMD is marginal in this direction. Except for DELs in the fore-aft direction in a region around the rated wind speed, the TMD effectively dissipates side-to-side loads in the whole operational range.

The maximum reduction of DELs occurs in the sideways direction with a TMD with the mass ratio of 2% at the wind speed of 6 m/s which corresponds to a 60% reduction with respect to the reference turbine configuration. For a TMD with the mass ratio of 2%, the lifetime weighted DELs are reduced by 4% and 24.3% in the fore-aft and side-to-side directions respectively.

As explained in the previous section, the second concept is a viscous fluid damper installed at the tower top location. Three toggle brace viscous fluid dampers with 120 phase shifting i.e. 0° towards north, 120° and 240°, are considered. The VFD model is created in Simulink and called through a DLL at each time step. Since the damper force is proportional to the effective damping coefficient, C_0, a larger damper force is achieved with increasing the damping coefficient. Therefore, the effective damping coefficient is increased step by step up to a value for which the simulations became unstable. This corresponds the damping coefficient which causes maximum load reduction.

A typical time interval of the nacelle displacements in the fore-aft and side-to-side directions are demonstrated in Figure 7. It is obvious that the tower top vibration, particularly in the sideways direction, is considerably dissipated compared to the reference configuration.

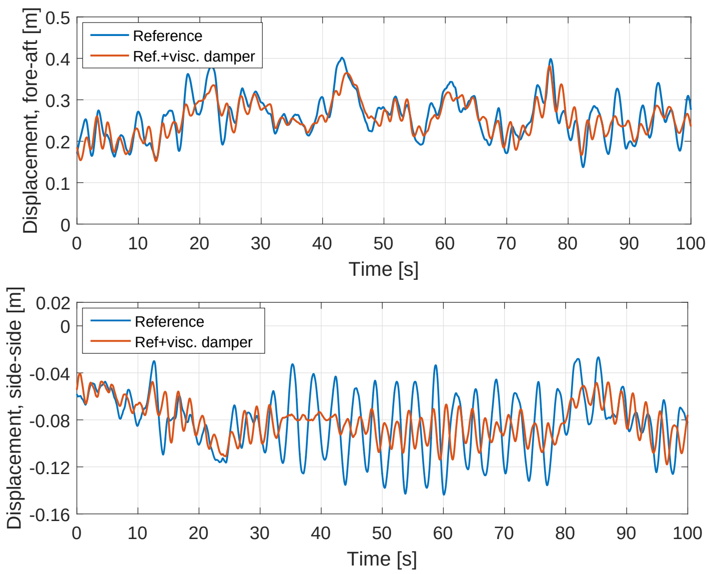


Figure 7. Nacelle displacement with and without VFD in the fore-aft (top) and side-to-side (bottom) directions at wind speed of 12 m/s.

The tower base DELs of the bending moments in the fore-aft and sideways directions are plotted in Figure 8. It should be noted that these plots are for a loading setup where the wind-wave misalignment is zero. It can be seen that the DELs are remarkably decreased in both directions. The maximum load reduction, i.e. around 69%, occurs in the sideways direction at the wind speed of 6 m/s. In addition, the lowest loads reduction occurs near the rated wind speed where the highest variations of aerodynamic forces take place. Although these results are calculated without the wind-wave misalignment angle, the interface loads and moments calculated for the full load setup are likely to be resulted in optimized jacket geometry.

The lifetime weighted equivalent loads of the tower base moments for the reference turbine and both strategies are compared in Figure 9. The values are normalised with respect to the DEL of the reference turbine in the fore-aft direction. It can be concluded that the DELs are

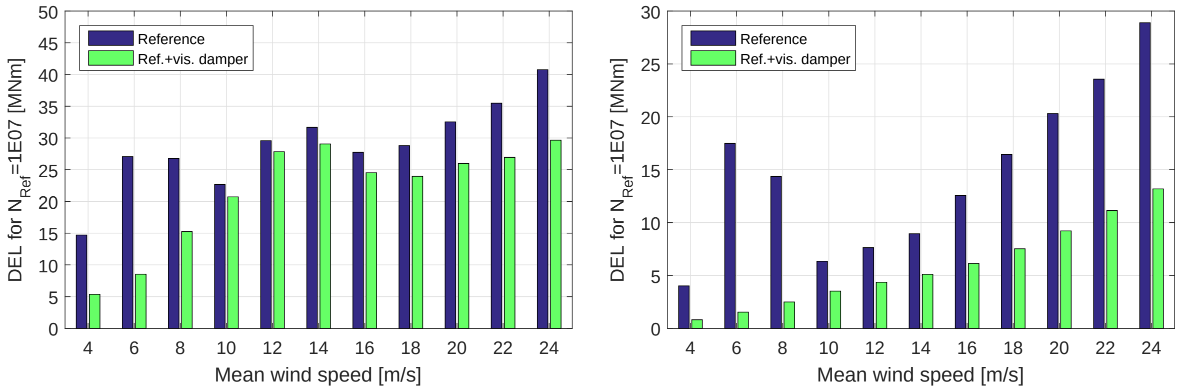


Figure 8. Tower base moment DELs in the fore-aft (left) and side-to-side (right) directions with integration of a VFD.

strictly improved with passive viscous dampers. The application of the VFD is however more effective than TMD.

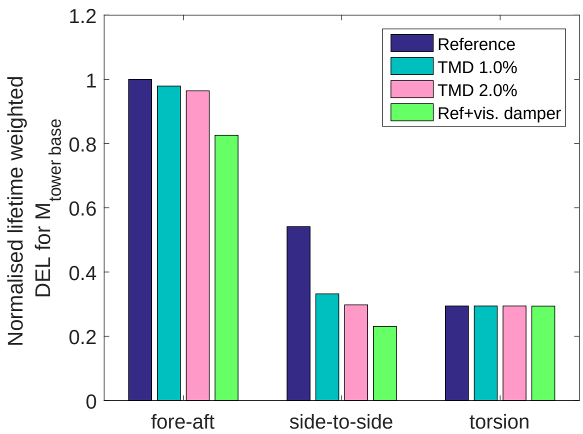


Figure 9. Normalised lifetime weighted DELs of the tower base moments calculated for different damping strategies.

4. Conclusions

This paper presents the results of the integration of two damping concepts in the 10MW INNWIND.EU reference wind turbine. These two concepts are: a passive tuned mass damper (TMD) and a toggle brace viscous fluid damper. The improved tower base loads are obtained and compared with the reference design.

It can be concluded that for the load setup where inflow wind is aligned with the wave direction, the DELs in the sideways direction, where no aerodynamic damping is active, can be lowered up to 29% and 69% compared to the reference case for respectively a TMD and viscous fluid damper. In this condition, the impact of the TMD in the fore-aft direction is however not significant while the viscous damper gives remarkably lowered loads in both directions.

The integration of the damping concepts could have a positive impact on the lifetime of the system. The integration of the developed strategies could be considered for an optimized jacket design with reduced entire mass and applied loads for the INNWIND.EU 10MW RWT.

Acknowledgments
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