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Passive structural control techniques for a 3 MW wind turbine prototype

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Abstract. The dynamic behaviour of the wind turbine tower and blade structure is crucial for structural fatigue propagation of large wind turbines at multi megawatt scale. Mitigation the structural response to windinduced vibrations takes therefore a central role in modern wind turbine design. This study investigates the impact of passive tuned mass damper (TMD) on the dynamic behaviour of the W2E 120/3.0fc wind turbine designed by W2E Wind to Energy. Two different TMD concepts are investigated. A TMD is implemented at the tower top to mitigate fore-aft vibrations of the turbine. Further a TMD is mounted on each turbine blade to mitigate flapwise vibrations of the blades. The performance of the TMDs is evaluated by means of a detailed validated multibody model of the turbine prototype.

1. Introduction

The increasing demand on renewable energy leads to strict requirements on the performance of wind turbines in order to compete on the energy market. This results in the development of wind turbines with increasing rotor diameters and hub heights to achieve a power output at multi megawatt scale. However, this leads to large, slender wind turbine structures that are prone to windinduced vibrations. The increase in structural vibration and the associated increase of fatigue damage experienced by the blade and tower structure can significantly shorten the fatigue lifetime of the system resulting in higher operating and maintenance costs. Limiting the dynamic response and fatigue damage of the turbine structure is thus a key objective for the design of modern multi megawatt wind turbines.

The mitigation of windinduced turbine vibration has been the focus of research during the past decades, and different approaches of wind turbine vibration control emerged [1]. In general vibration control can be categorized into two types. The first approach focuses on the development of advanced control algorithms for the existing turbine control system. Promising results are achieved by individual pitch control [2, 3] and advanced generator torque control [4, 5]. However, a possible drawback of this approach lies in the increased wear on actuators due to higher pitch rates and in the need of additional sensory data as controller inputs. The second type of vibration control aims at mitigating structural vibration by means of structural control techniques. A common approach of structural control is the utilization of an auxiliary spring-

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Figure 1. Overall view of the wind turbine prototype erected in Kankel, Mecklenburg-Western Pomerania, Germany.

mass-dashpot-system further referred to as tuned mass damper (TMD). The use of TMD in the context of wind turbine design primarily concentrates on improving the dynamic behaviour of the blade and tower structure. The mitigation of the dynamic response of the turbine tower using passive TMD is investigated in [6, 7]. The utilization of active TMD for tower vibration control is described in [8, 9]. Passive and active TMD concepts to reduce the vibration of wind turbine blades are investigated by the extensive work in [10] and [11] as well as in [12, 13, 14]. The aforementioned researches in [6, 7, 8, 9, 12, 14] evaluate the performance of the respective structural method based on the 5 MW reference wind turbine defined in [15].

The aim of this study is to investigate the performance of passive TMD to mitigate windinduced vibrations of an industrial wind turbine W2E 120/3.0fc designed by W2E Wind to Energy. A prototype of the wind turbine is erected in Kankel, Mecklenburg-Western Pomerania, Germany, Figure 1. The turbine is a horizontal design and variable speed/variable pitch controlled. The nominal hub height of this prototype is at 100 m. Equipped with a rotor of 120 m diameter, the rated power is 3 MW. The basis to investigate the performance of TMD on the prototype is a detailed multibody model of the turbine prototype to ensure a realistic representation of dynamic behaviour of the turbine model [16]. Within this study two different passive TMD concepts are considered. First a TMD is mounted at the tower top and tuned to the turbine fore-aft eigenmode. In a second step a TMD is mounted at each turbine blade and tuned to the one per revolution frequency (1p) to reduce blade vibration in flapwise direction. The use of the validated multibody model of the real turbine prototype enables a comprehensive and realistic evaluation of TMD performance covering the whole operating range of the wind turbine.

The paper is organised as follows. In Section 2 the multibody model of the W2E 120/3.0fc wind turbine and its interaction with the aerodynamic code and the plant controller is briefly described. Section 3 describes the implementation of the respective TMD concept to the multibody model. Subsequently, the performance of the two TMD concepts to lower the dynamic response of the turbine blades and tower is investigated by utilizing the multibody model of the wind turbine.

2. Detailed multibody model of the wind turbine

The overall dynamic behaviour of a wind turbine during operation is strongly nonlinear and depends on the complex interaction between turbine structure, turbine controller and



Figure 2. Interaction scheme of the wind turbine simulation model.

environmental conditions. Using multibody simulation allows for a thorough investigation of the turbine dynamics during operation, resulting in a realistic representation of overall structural loads for different operation conditions. A highly coupled, detailed multibody model of the turbine prototype is built up in the general purpose multibody program SIMPACK [17]. An overall view of the turbine model as well as a schematic interaction scheme between aerodynamic code, multibody model and plant controller is given in Figure 2. The blades as well as the tower of the turbine prototype are modelled as flexible body parts based on a modal representation. The two-stage planetary gear drive train model is built up by connecting rigid body parts via user-defined force elements. Overall the multibody model comprises 110 degrees of freedom. The same controller software that operates the physical turbine prototype is used for the multibody model. In doing so a realistic representation of the turbine operational behaviour is ensured. The aerodynamic forces and torques acting on the turbine blades are implemented using the software packages Aerodyn and TurbSim, both provided by the National Renewable Energy Laboratories (NREL) [18]. The multibody model is validated by extensive measurements taken on the turbine prototype. The measurements were carried out according to the International Electrotechnical Commission (IEC) standard 6140013 [19]. A detailed description of the multibody model and its validation is given in [16].

3. Wind turbine with TMD

In order to improve the overall dynamic response of the wind turbine structure to aerodynamic loads two TMD concepts are incorporated into the turbine structure. By tuning the natural frequency of the TMD to the vibration frequency of the primary structure (wind turbine) vibration energy will be transferred to the TMD causing damping forces on the primary structure and thus mitigating the vibration of the primary structure. The mass m of the TMD is coupled to the primary structure by a spring element c and dashpot element d. Together m, c and d are the tuning parameters of the TMD. Typically the mass m of a TMD is low (in the range of a few percent) compared to the mass of the primary structure. A common design theory to calculate optimal values of the tuning parameters is given in [20]. The optimal frequency ratio between the natural frequency of the TMD f_{TMD} and the vibration frequency f_1 of the primary



Figure 3. Side view of the wind turbine multibody model equipped with a TMD at the tower top (left) and the corresponding idealised sketch (right).

structure is given by

$$f_{\rm TMD} = \frac{1}{1+\lambda} f_1 \,. \tag{1}$$

The parameter λ in (1) is the ratio between the mass m of the TMD and the mass m_1 of the primary structure. The optimal damping ratio δ_{TMD} follows from

$$\delta_{\rm TMD} = \sqrt{\frac{3\lambda}{8(1+\lambda)^3}} \,. \tag{2}$$

In a first step a TMD is implemented to the tower top of the multibody model, see Figure 3. The objective is to mitigate the windinduced vibration of the wind turbine tower along wind direction by tuning the TMD to the first eigenmode of the turbine in fore aft direction (along wind direction) according to (1) and (2). The TMD is mounted to the tower top by a prismatic joint that allows for a translational displacement of the TMD in fore aft direction relative to the tower top. The mass m of the TMD is set to 10% of the modal mass of the first eigenmode in fore aft direction.

In a second step a TMD is attached to each blade of the wind turbine rotor. Here the objective is to mitigate the flapwise (out of rotor plane, along wind direction) vibration of the blades in the 1p frequency range which is caused by the rotational sampling of turbulence, wind shear and tower shadow effects. Although best results would be expected by mounting the TMD at the tip of each blade the TMD is mounted at approximately $\frac{2}{3}$ of the total blade length as practical trade off between potential TMD performance and available space inside the turbine blade [11]. Figures 4 and 5 show the blade TMDs attached to the turbine model. Similar to the TMD at the tower top the TMDs are attached to the blades by a prismatic joint that allows for a relative translational displacement of the TMD in fore aft direction. The blade TMD mass m is set to 10% of the modal mass of the first blade eigenmode in for aft direction. It is important to note that the 1p flapwise vibration of the blades depends on the rotational speed of the rotor and has thus a nonlinear behaviour over the operating range of the wind turbine. As a first investigation within the scope of this contribution the blade TMDs are tuned to 1p frequency at rated speed of the wind turbine while the nonlinear behaviour below rated speed is not accounted for.

It should be emphasized that the objective of these investigations is to show the fundamental effects of idealised TMDs on the dynamic behaviour of the 3 MW wind turbine prototype. The various problems that arise with a complete mechanical design of TMD is not considered within this contribution.





Figure 4. Overall view of the wind turbine model with Blade TMDs

Figure 5. Side view of the wind turbine model equipped with a TMD at the rotor blade (left) and the corresponding idealised sketch (right).

4. Numerical Simulation

For the investigation of the influence of the TMDs on the dynamic behaviour of the industrial 3MW wind turbine the validated multibody model of the turbine described in section 2 is utilized. A total of 84 design load cases (DLC) are defined in accordance to the GL guideline [21] for the simulation of the multibody model. The 84 DLC comprise mean wind speeds ranging from 4 m/s to 25 m/s in steps of 1 m/s, each with 4 different stochastic windseeds and turbulence intensity (TI) of 12%. Each DLC is simulated for a 10 minute time series. The performance of the respective TMD is evaluated based on the time series and corresponding frequency spectra of the fore-aft bending moment at the tower base and the flapwise bending moment at the blade root, respectively. In addition the standard deviation (STD) of the bending moment time series is calculated for each mean wind speed and averaged over the 4 different windseeds. In a last step the fatigue loading resulting form the bending moments is estimated for a 20-year life cycle by extrapolating the calculation results according to the GL guideline [21] and calculating a damage equivalent load range (DELR) defined in [19]. Further load case scenarios such as Pitch Runaway, Emergency Stop or Extreme Operating Gust are not investigated within the scope of this contribution. Due to confidentiality the results are scaled with respect to their mean value (time series data) and with respect to the tuning frequency of the respective TMD (frequency data).

4.1. Tower TMD results

Figure 6 compares selected time series data and corresponding frequency spectra of the fore-aft bending moment at the tower base with and without TMD at the tower top ranging from 4 m/s to 20 m/s in steps of 4 m/s.

At the cut-in wind speed of 4 m/s the wind turbine displays resonant behaviour in fore aft direction which can be seen in the corresponding frequency spectra. Here the major part of vibration energy is concentrated around the eigenfrequency of the first turbine eigenmode in

fore aft direction (tuning frequency f_1 of the TMD), and a clear mitigation of the vibration around the tuning frequency by the TMD is observed. The large peak in the frequency range below $0.2 f/f_1$ results from the very slow change in wind speed which is also be observed in the time data.

The slow changes in wind speed become more dominant for the turbine vibration as shown in the calculation results for the 8 m/s DLC were the major part of vibration energy is concentrated at frequencies below $0.2 f/f_1$. The mitigating effect TMD on the turbine vibration can still be observed around the tuning frequency f_1 , however, only a very limited amount of vibration energy is located around f_1 .

With wind speeds exceeding the rated speed of 10.7 m/s the turbine controller starts to limit the rotor speed to nominal speed by pitching the rotor blades. The pitching action of the controller acts like high pass filter on the windinduced loads on the tower top. This effect is shown in the time data of the 12 m/s DLC were the very slow changes in wind speed are compensated by the pitch control. However, the major part of vibration energy remains below f_1 , and the impact of the TMD on the turbine vibration remains rather limited. Compared to the 12 m/s DLC no significant changes in the fore aft vibration of the turbine occur for the 16 m/s DLC.

At high wind speeds the distribution of vibration energy becomes broadband as shown by the frequency spectra corresponding to the tower bending moment at 20 m/s mean wind speed. A significant part of vibration energy is again concentrated around the tuning frequency f_1 resulting in a higher impact of the TMD on the dynamic response of the turbine in fore aft direction.

The STD of the time series results is calculated for each mean wind speed and averaged over the 4 different windseeds. The STD quantifies the amount of variation in the time data and therefore serves as a measure of performance of the TMD since STD in the bending mode vibration can be interpreted as an indicator for fatigue damage. The bar diagram in Figure 7 shows the reduction of STD due to the implementation of the TMD for each mean wind speed. The diagram confirms the already in Figure 6 observed influence of the TMD on the turbine fore aft vibration. At cut-in the TMD shows overall best performance, whereas with increasing wind speed the STD reduction drops below 1% since vibration energy is concentrated outside of the tuning frequency. With wind speed above 15 m/s the energy distribution widens and the STD reduction increases to approximately 6%. A possible cause for the negative impact of the TMD at 11 and 12 lies in the switching between control strategies as the wind speed is close to rated speed.

In order to estimate the fatigue loading of the fore aft bending moment at the tower base over a full cycle of 20 years according to the GL guideline [21], the 10 minute time series are accumulated by means of a site-specific Weibull distribution of the mean wind speed. The Weibull distribution of the turbine prototype site is indicated by the red graph in Figure 7. It is an indicator for the share of each mean wind speed time series in the accumulated 20 years fatigue loading. Thus the impact of the TMD on the fatigue damage is further reduced by the site-specific environmental conditions, since the turbine prototype operates mainly at wind speeds were the TMD performance is the lowest. The TMD at the tower top of the wind turbine reduces the fatigue DELR of the fore aft bending moment at the tower base by 1.72%.

4.2. Blade TMD results

Figure 8 shows selected time series data and corresponding frequency spectra of the flapwise bending moment at the blade root with and without tower TMD ranging from 4 m/s to 20 m/s in steps of 4 m/s.

Compared to the results from the TMD mounted to the tower top, the results from the blade TMD simulation show little to no difference in dynamic behaviour between the turbine with and without equipped TMDs. At mean wind speed below the 10.7 m/s rated wind speed a significant

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Figure 6. Selected time series (left) and corresponding frequency spectra (right) of the fore-aft bending moment at the tower base with and without Tower TMD (tuning frequency f_1) for the DLC (top down) 4 m/s, 8 m/s, 12 m/s, 16 m/s and 20 m/s. Windseed 2, 12 % TI.



Figure 7. Reduction of STD in tower fore aft vibration caused by the TMD mounted to the tower top (blue) and Weibull distribution of mean wind speed for the site (red).

amount of vibrational energy is located as a sharp peak around 1p, which is approximately $0.6 f/f_1$ for 4 m/s mean wind speed and $0.8 f/f_1$ for 8 m/s mean wind speed. Similar to the tower bending moment the low frequency components below are $0.2 f/f_1$ are caused by the slow change in wind speed. The very limited influence of the TMDs is to be expected for a mean wind speed below rated as the tuning frequency f_1 is set to 1p at rated speed.

With a mean wind speed above rated the vibration energy concentrates around the tuning frequency of the TMDs as well as the higher harmonics 2p and 3p of the 1p frequency. Additionally blade vibration around $0.4 f/f_1$ is caused by the pitch control. However the mitigation of the flapwise blade vibration around tuning frequency remains very limited. Here the limiting factor lies in the high aerodynamic damping of the rotor in flapwise direction as well the complex load situation at the turbine rotor which couples flapwise and edgewise blade deflection. Additional investigation on the impact of the TMDs on the edgewise bending moment showed negligible effects due to the small TMDs masses. Figure 9 shows the reduction of STD in blade vibration due to the implementation of the TMDs for each mean wind speed. Overall the performance of the TMD concept is shown to be very limited with a maximum STD reduction of approximately 2%. The implementation of TMDs at the rotor blades to mitigate flapwise vibration reduces the fatigue DELR of the flapwise bending moment at the blade root by 0.21%.

5. Conclusion

The aim of this contribution is to investigate the performance of two TMD concepts on mitigating the structural response of the turbine tower and rotor blades. The wind turbine under consideration is an industrial 3 MW wind turbine designed by W2E Wind to Energy. The basis for the investigation is an experimentally validated, highly detailed multibody model of the turbine prototype. First a TMD is mounted at the tower top and tuned to the first eigenmode of the turbine in fore aft direction. In a second step a TMD is mounted to each rotor blade and tuned to the 1p frequency of the flapwise blade vibration. The performance evaluation of the TMD covers the whole operating range of the wind turbine comprising 84 DLC. The limiting factor for the TMD performance to mitigate turbine fore aft vibration becomes clear by investigating the frequency of the TMD for the major part of the operating range of the wind turbine. The overall performance of the TMDs mounted at the rotor blades is very limited due to high aerodynamic damping and the complex coupling of flapwise and edgewise vibration. Future research will include the implementation of active TMDs to mitigate the turbine fore aft vibration.

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Figure 8. Selected time series (left) and corresponding frequency spectra (right) of the flapwise bending moment at the blade root with and without blade TMDs (tuning frequency f_1) for the DLC (top down) 4 m/s, 8 m/s, 12 m/s, 16 m/s and 20 m/s. Windseed 2, 12 % TI.



Figure 9. STD reduction in blade flapwise vibration caused by the TMDs at the turbine blades (blue) and site-specific Weibull distribution of mean wind speed (red).

References

- A.Staino and B. Basu. Emerging trends in vibration control of wind turbines: a focus on a dual control strategy. Phil.Trans.R.Soc.A, 373: 20140069, 2014.
- [2] E. A. Bossanyi. Individual Blade Pitch Control for Load Reduction. Wind Energy, 6:119 128, 2003.
- [3] A.Staino, B. Basu. Robust Constrained Model Predictive Control for Flapwise Vibration Mitigation in Wind Turbines. Proceedings of the 8th International Conference on Structural Dynamics, Proceedings of the 8th International Conference on Structural Dynamics. Leuven, Belgium, 2011. Leuven, Belgium
- [4] Z. Zhang, S. R. K. Nielsen, F. Blaabjerg and D. Zhou. Dynamics and Control of Lateral Tower Vibrations in Offshore Wind Turbines by Means of Active Generator Torque. Energies, 7:7746 - 7772, 2014.
- [5] J. Licari, C. E. Ugalde-Loo, J. B. Ekanayake and N. Jenkins. Comparison of the performance and stability of two torsional vibration dampers for variable-speed wind turbines. Wind Energy, 18:1545 - 1559, 2015.
- [6] M. A. Lackner and M. A. Rotea. Passive Structural Control of Offshore Wind Turbines. Wind Energy, 14:373
 – 388, 2011.
- [7] O. Altay, F. Taddei, C. Butenweg and S. Klinkel. Vibration Mitigation of Wind Turbine Towers with Tuned Mass Dampers. Wind Turbine Control and Monitoring. Springer International Publishing Switzerland, 2014.
- [8] M. A. Lackner, M. A. Rotea and R. Saheba. Active Structural Control of Offshore Wind Turbines. 48th AIAA Aerospace Sciences Meeting Including the New Horizons Forum and Aerospace Exposition. Orlando, Florida, 2010.
- [9] Y. Hu, M. Z. Q. Chen and C. Li. Active structural control for load mitigation of wind turbines via adaptive sliding-mode approach. Journal of the Franklin Institute, 354:4311 - 4330, 2017.
- [10] Z. Zhang. Passive and Active Vibration Control of Renewable Energy Structures. Ph.d.-serien for Det Teknisk-Naturvidenskabelige Fakultet, Aalborg Universitet. Aalborg Universitetsforlag, 2015.
- [11] Z. Zhang, J. Li, S. R. K. Nielsen and B. Basu. Mitigation of edgewise vibrations in wind turbine blades by means of roller dampers. Journal of Sound and Vibration, 333:5283–5298, 2014.
- [12] B. Fritzgerald, B. Basu and S. R. K. Nielson. Active tuned mass dampers for control of in-plane vibrations of wind turbine blades. Structural Control and health monitoring. Wiley Online Library, 2013.
- [13] J. Arrigan, V. Pakrashi, B.Basu and S. Nagarajaiah. Control of flapwise vibrations in wind turbine blades using semi-active tuned mass dampers. Structural Control and Health Monitoring. 18:840851, 2011.
- [14] T. Ikeda, Y. Harata and Y. Sasagawa. Vibration suppression of wind turbine blades using tuned mass dampers. ASME 2014 international design engineering technical conferences and computers and information in engineering conference. Buffalo, NY, 2014.
- [15] J. Jonkmann, S. Butterfield, W. Musial and G. Scott. Definition of 5 MW reference wind turbine for offshore system development. TP 500-38060, National Renewable Energy Laboratory, 2008.
- [16] A. Schulze, J. Zierath, R. Rachholz1, S.-E. Rosenow, R. Bockhahn, C. Woernle. Modelling and Validation of a 3MW Wind Turbine as a Basis for Structural Optimisation. ECCOMAS Thematic Conference on Multibody Dynamics. Prague, Czech Republic, 2017.
- [17] SIMPACK. SIMPACK 9.7 Documentation. Simpack AG, Gilching, Germany, 2014.
- [18] J.M. Jonkman, G.J. Hayman, B.J. Jonkman, R.R. Damiani. AeroDyn v15 Users Guide and Theory Manual. National Renewable Energy Laboratory, 2016.
- [19] IEC. IEC 61400-13 Ed. 1, Wind Turbine Generator Systems-Part 13: Measurement of Mechanical Loads. International Electrotechnical Commission, Genf, Switzerland, 2001.
- [20] J.P. Den Hartog. Mechanical Vibrations. McGraw-Hill, New York, 1956.
- [21] Germanischer Lloyd. IV Rules and Guidelines Industrial Service, Guidelines for the Certification of Wind Turbines, 2010.