Received: 23 November 2017 | Revised: 9 May 2018 | Accepted: 17 May 2018
DOI: 10.1002/we.2214

SHORT COMMUNICATION

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Structural parameter identification of a 2.4 MW bottom fixed wind turbine by excitation test using active mass damper

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Funding information
New Energy and Industrial Technology Development Organization (NEDO)

Abstract

While the structural damping is an important parameter in the seismic resistant design, several problems exist in the current design codes such as that the recommended values vary largely between design codes, and that the second mode damping ratio which is required in the design is usually not described. In order to evaluate the damping ratios for both first and second modes of MW-size wind turbines, a series of excitation tests using an active mass damper were performed in this study on a 2.4 MW offshore wind turbine. First, the sinusoidal test, which gives accurate and reliable results for linear systems, is performed, and results show that the damping ratio for the fore-aft first mode is 0.2% and the fore-aft second mode is 2.4% for the target wind turbine. Next the free decay test, which is applicable to systems with the effect from the aerodynamic damping, is performed, and results show that the damping ratios obtained for the fore-aft first and second mode are similar to those from the sinusoidal test. The damping ratio is 1.2% for the side-side first mode and 3.2% for the side-side second mode. Finally, an empirical formula for the damping ratios of first mode is proposed for wind turbines with steel towers using the results from the previous researches and the excitation tests in this study.

KEYWORDS

damping ratio, excitation test, parameter identification, active mass damper, seismic resistant design

1 | INTRODUCTION

The structural damping is an important parameter in the seismic resistant design and is usually determined according to the recommendations in the design guidelines. However, there are several issues regarding current design codes. First, the recommended values of the structural damping for wind turbines seismic designs vary largely among different design codes. For example, the recommended practice for wind turbines issued by American Wind Energy Association and American Society of Civil Engineers (ASCE and AWEA)[1] recommends 1% of damping ratios for both first and second modes which follows IEC61400-1.[2] The guidelines for design of wind turbine support structures and foundations published by Japan Society of Civil Engineers[3] recommends the use of 0.8% for wind turbines with gearboxes and 0.5% for those without gearboxes, while the German guideline for wind turbines[4] recommends the value as 0.25% for steel structures. One explanation for this variation would be that the experiment data each design code based on is obtained from different sizes of wind turbines. The recommended practice by ASCE/AWEA adopts the result of an excitation test for a 100-kW wind turbine,[5,6] and the values in the guideline of Japan Society of Civil Engineers are based on the results of the human excitation test for a 400-kW wind turbine[7] and the ambient vibration analysis for a 500-kW turbine. The German recommendations are simply based on the material damping of steels. Another problem with these design codes is that the evaluation of the damping ratio of second mode is assumed to be the same with the damping ratio of first mode. This assumption is made because of the lack of experimental data for the damping ratios of second mode of wind turbines, because the frequency of second mode is usually in the range where it is difficult to excite by human excitations or emergency stops.

In this study, excitation tests are performed for a 2.4-MW horizontal axis wind turbine using an active mass damper (AMD) to identify the structural parameters for first and second modes. Two types of excitation test are performed to evaluate the damping ratios in both fore-aft and side-side directions. Finally, an empirical formula is proposed for the estimation of the structural damping of first mode for wind turbines with steel towers by using data from the excitation tests in this study and previous researches.

2 | DESCRIPTION OF 2.4-MW WIND TURBINE AND THE EXCITATION TEST

2.1 | Outline of the wind turbine

The wind turbine targeted in this study is a pitch-regulated MHI 2.4 MW wind turbine located at 3.1 km offshore Choshi, Japan. Location of the wind turbine is shown in Figure 1, and the outline of the 2.4-MW wind turbine and the equipped sensors are shown in Figure 2. The hub height is 80 m above sea level, and the rotor diameter is 92 m. The wind turbine and the steel tower are supported with a gravity foundation up to 10.83 m. Five accelerometers are installed at 5 heights of the tower: 10.5, 26.5, 45.5, 48.9, and 64.7 m from the tower base height of 10.08 m. The accelerometer at 45.5 m height is used for data analysis of the forced excitation tests considering that the second mode vibration at the location is more pronounced than the top accelerometer which is more suitable for evaluation of the first mode. An AMD is installed at 55 m height of the tower for the purpose of vibration control. It is used for the generation of the excitation forces in this study. A photo of the AMD is shown in Figure 3. The damper is movable in 2 directions, and the motions are controlled with user-defined signals inputted to the controller of the AMD equipped at the tower base. The primary parameters of the AMD are shown in Table 1. The frequency range of excitation capability of the AMD is 0.1 to 4 Hz, which covers the tower vibration frequencies up to the second mode. The field excitation tests were carried out at 2 periods when the weather state was calm; February 21st to 22nd, 2014 with the mean wind speed in the range of 4 to 6 m/s, and October 28th to 29th, 2014 when the mean wind speed was in the range of 5 to 8 m/s. Environmental conditions during the excitation tests are summarized in Table 2. The pitch angles of blades were set at the feathering condition throughout both excitation tests described below.

2.2 | Sinusoidal vibration test

In sinusoidal vibration test, excitation is performed for a range of frequencies including the resonant frequency, and modal damping ratios are obtained by fitting the measured amplitudes and phases of the acceleration to the theoretical equation of modal acceleration of linear system. The theoretical equation is derived by decomposing the equation of motion of N degrees of freedom system (Equation 1) into Equation 3 by using modal decomposition shown in Equation 2.

$$[M]\{\ddot{u}\} + [C]\{\dot{u}\} + [K]\{u\} = \{f(t)\} \tag{1}$$

$$u^i = \sum_j q_j \phi_j^i \tag{2}$$

$$M_j q_j + C_j q_j + K_j q_j = \phi_j^{AMD} |f(t)| \tag{3}$$

FIGURE 1 Outline of the location of the target wind turbine [Colour figure can be viewed at wileyonlinelibrary.com]

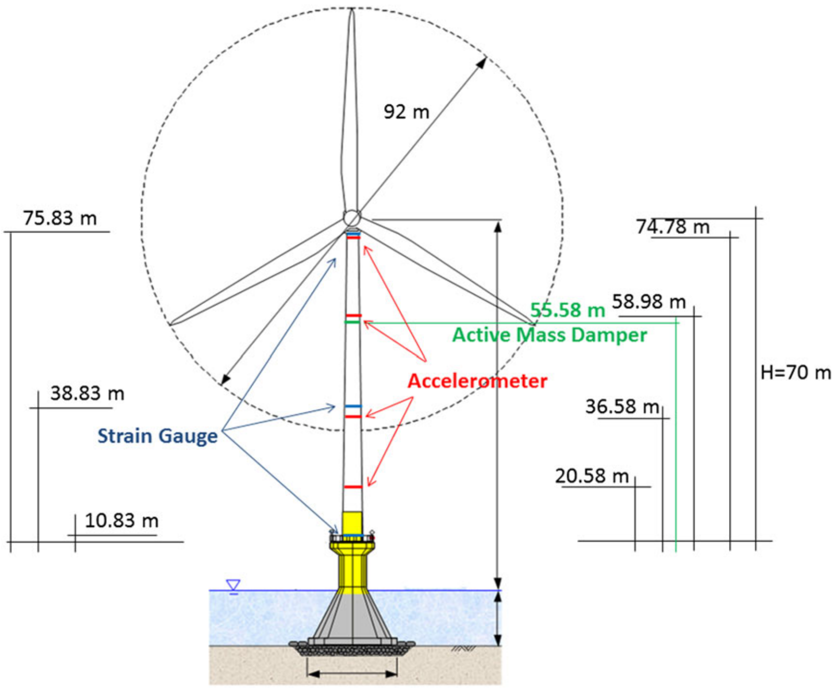

FIGURE 2 Arrangements of the target wind turbine and measurement equipment [Colour figure can be viewed at wileyonlinelibrary.com]

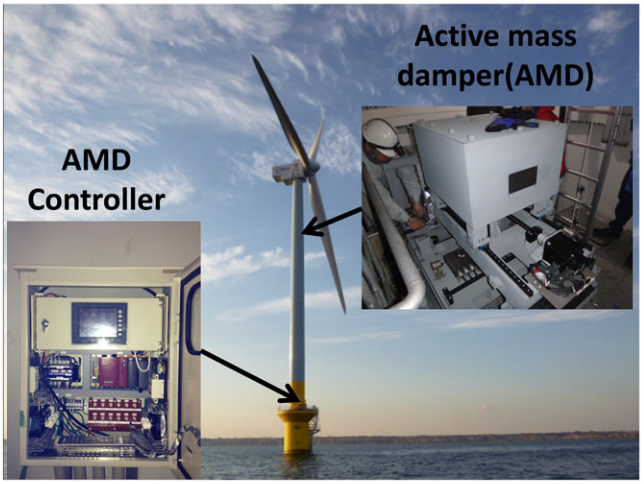

FIGURE 3 Outline of the target wind turbine and the active mass damper [Colour figure can be viewed at wileyonlinelibrary.com]

TABLE 1 Primary parameters of the active mass damper

	Fore-Aft Direction	Side-Side Direction
Movable mass	1700 kg	1300 kg
Effective stroke	190 mm	290 mm
Maximum speed	0.83 m/s	0.83 m/s
Frequency range	0.1-4 Hz	0.1-4 Hz
Actuator power	Maximum: 4400 N Continuous: 1900 N	Maximum: 4400 N Continuous: 1900 N
Acceleration	Maximum: 2.58 m/s^2 Continuous: 1.11 m/s^2	Maximum: 3.38 m/s^2 Continuous: 1.46 m/s^2

TABLE 2 Environmental conditions during the excitation tests

Date	2014 Feb., 21-22	2014 Oct., 28-29
Mean wind speed	4-6 m/s	5-8 m/s
Mean wind direction	345-350 deg.	20-25 deg.

where u^i is the displacement of ith node, q_j is the modal displacement of jth mode, ϕ_j^i is the value of the modal shape of jth mode at ith node, M_j is the modal mass, C_j is the modal damping, and K_j is the modal stiffness of jth mode. Considering that the excitation force is applied at 1 node in this study, the external force vector $\{f(t)\}$ can be expressed as the right-hand term of Equation 3 using the magnitude of mode shape at the height of AMD ϕ_j^{AMD}.

When the excitation force $|f(t)|$ is harmonic and is expressed as Equation 4, Equation 3 can be solved analytically, and the acceleration at i-th node can be derived as Equation 5,[8]

$$|f(t)| = f_a \sin\overline{\omega}t \tag{4}$$

$$\frac{\ddot{u}^i}{f_a} = \phi_j^i \phi_j^{AMD} \cdot \frac{-\overline{\omega}^2}{K_j\sqrt{\left(1-\beta_j^2\right)^2 + \left(2\xi_j\beta_j\right)^2}} \cdot \sin\left(\overline{\omega}t - \tan^{-1}\frac{2\xi_j\beta_j}{1-\beta_j{}^2}\right), \quad \beta_j = \overline{\omega}/\omega_j \tag{5}$$

where ω_j is the natural frequency and ξ_j is the damping ratio of the jth mode. In Equation 5, the parameters ϕ_j^i and K_j have to be determined. In this study, an in-house FEM code[7] is used for the eigenvalue analysis of wind turbines. In the FEM model, the gravity foundation of the wind turbine is assumed as rigid and fixed, and the tower and the rotor-nacelle assembly are modeled with the concentrated mass and the beam elements. The modal shapes for the first and second modes are compared with the measured acceleration amplitudes from the 5 sensors as shown in Figure 4. The calculated modal shapes show favorable agreement with the measurement for both first and second modes. Finally, by fitting Equation 5 to the amplitudes and phases of the measured acceleration, the natural frequency ω_j and the modal damping ratio ξ_j are evaluated.

2.3 | Free decay test

Although the sinusoidal vibration test gives high accuracy in the estimation of the structural parameters, it is only applicable to linear systems where the structural damping is dominant, and the modal decomposition including the damping matrices in Equation 3 is acceptable. In order to estimate the damping ratios in the side-side direction where the direction of the excitation aligns with the blade flap-wise direction, free-vibration tests are performed considering that the aerodynamic damping is larger in these cases. To generate a free decay condition, the AMD is ordered a sudden stop after the steady state of tower vibration at the natural frequency is achieved. The damping ratio for the excited vibration mode is then estimated using the envelope function of the free decay of the vibration shown in Equation 6. In order to eliminate the effect of measurement noises, a band-pass filter is applied to the measured acceleration signals to extract the free decay vibration of the target frequency.

$$g(t) = A\exp\left(-\omega_j\xi_j t\right) \tag{6}$$

3 | RESULTS OF EXCITATION TESTS

3.1 | Sinusoidal vibration analysis

Excitations using the AMD were performed for several frequencies around the first and second mode, and amplitudes and phase angles of the measured accelerations were used to fit the theoretical equation to obtain the natural frequency and the modal damping. The results of the

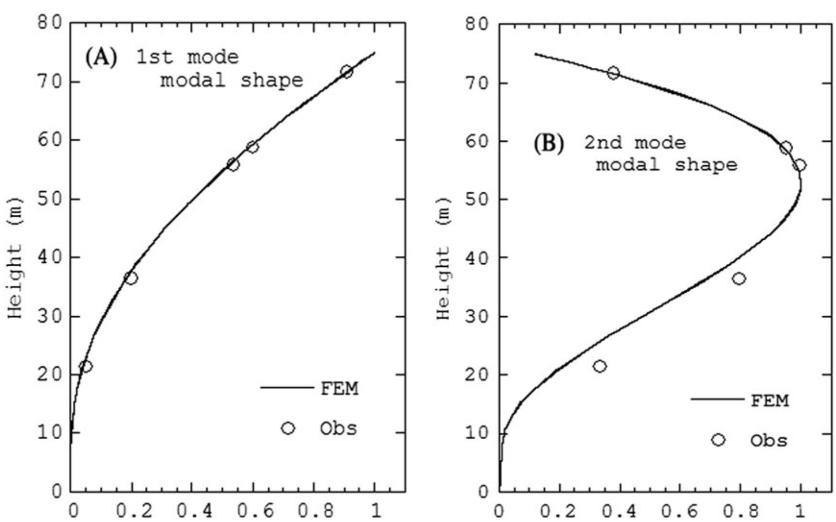

FIGURE 4 Comparison of measured and calculated mode shapes

measurement and the fitted analytical solution are shown in Figure 5 for the amplitude ratios and phase angles of the acceleration to the input force. It is found that the analytical solution fits well around the natural frequencies for the amplitude and the phase angle for first and second modes. The parameters used for the fitting are 0.35 Hz for the natural frequency and 0.2% for the damping ratio for the first mode and 2.98 Hz for the natural frequency and 2.4% for the damping ration for the second mode.

3.2 | Free decay test

Measured acceleration signals and the fitted curves for the free decay tests are shown in Figure 6 for the fore-aft and side-side directions. Figures show that the free decay of the measured acceleration is fitted well with Equation 6 for both first and second modes. For the fore-aft direction where the vibration is aligned with the blade edge-wise direction, the aerodynamic damping is small. The damping ratios for the for-aft excitations are estimated as 0.2% for the first mode and 2.4% for the second mode. These results agree with those obtained from the sinusoidal vibration test. For the side-side excitation where the vibration is aligned with the blade flap-wise direction, aerodynamic damping is indispensable. The estimated damping ratios are 1.2% for the first mode and 3.2% for the second mode in this excitation direction, which are larger than the results in the fore-aft direction for both modes. Summary of the results from the sinusoidal vibration test and the free decay test are shown in Table 3.

4 | AN EMPIRICAL FORMULA FOR THE STRUCTURAL DAMPING OF FIRST MODE FOR WIND TURBINES

It is known in the field of architectural engineering that for the structures such as high-rise buildings and chimneys there is a correlation between the natural periods and the damping ratios. This relationship is useful in estimating the structural damping in the seismic resistant design when the direct measurement is not available. As shown in previous studies, the measured damping ratios depend on the sizes of wind turbines, the first mode modal damping ratios for wind turbines with steel tower is assumed to correlate with the natural period in this study. Figure 7 shows the relationship of the damping ratios of the first mode and the natural periods obtained in both present and previous studies using forced excitation tests. It is found that the modal damping ratios of the first mode decrease with the increase of the natural periods. By fitting these experimental data with an exponential function, the empirical formula of the modal damping ratios of the first mode of wind turbines with steel towers is obtained as:

$$\zeta = 2.0e^{-1.3T} + 0.15 \tag{7}$$

where ζ is the damping ratio (%) and T is the natural period (s) of the first mode vibration. The contribution of the soil damping to the modal damping ratios of the first mode in the present and previous studies is negligible, Equation 7 can also be used for the structural damping ratios of the first mode of wind turbines with steel towers.

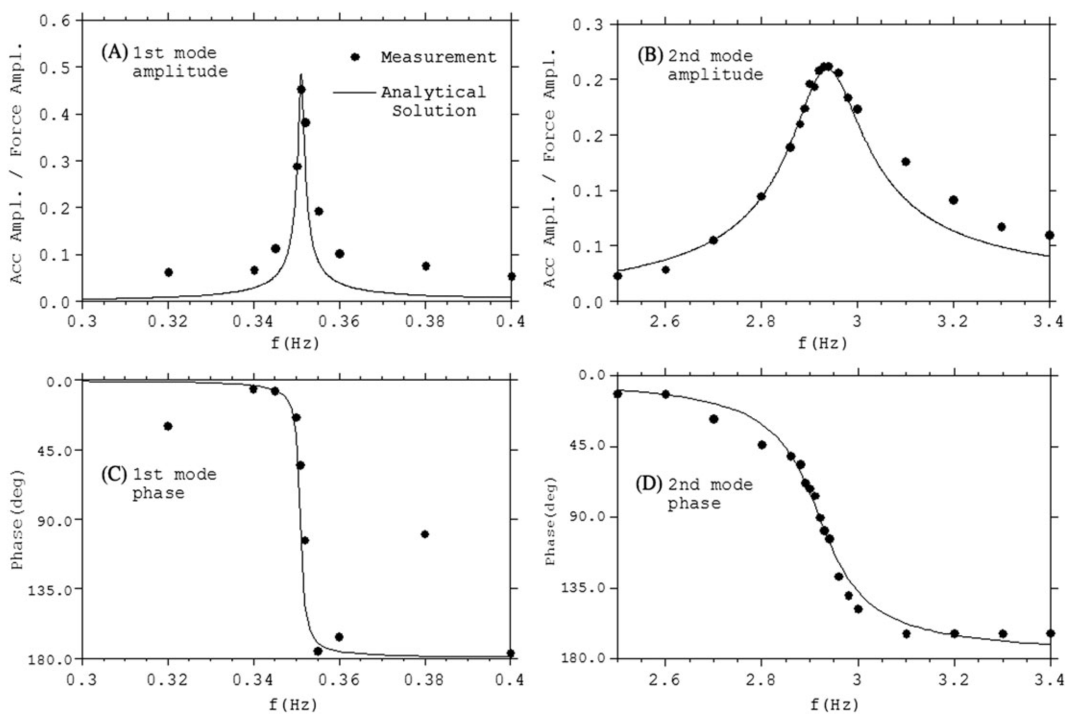

FIGURE 5 Comparison of measured acceleration with the analytical solution for the sinusoidal test

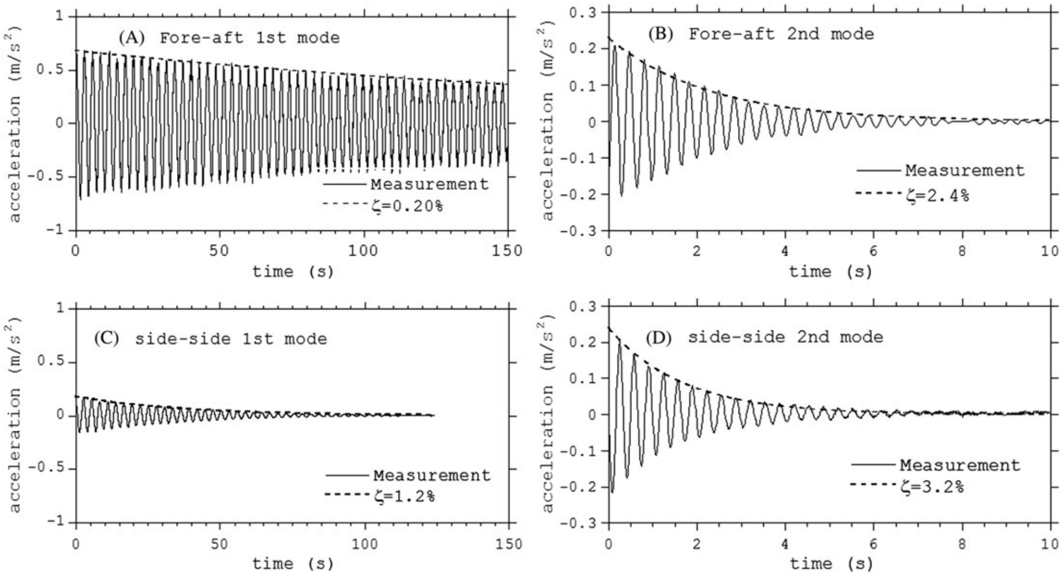

FIGURE 6 Measured acceleration and fitted result for the free decay test

TABLE 3 Summary of measured modal damping ratios for the first and second modes

	X-1st	X-2nd	Y-1st	Y-2nd
Sinusoidal test	0.2%	2.4%	-	-
Free decay test	0.2%	2.4%	1.2%	3.2%

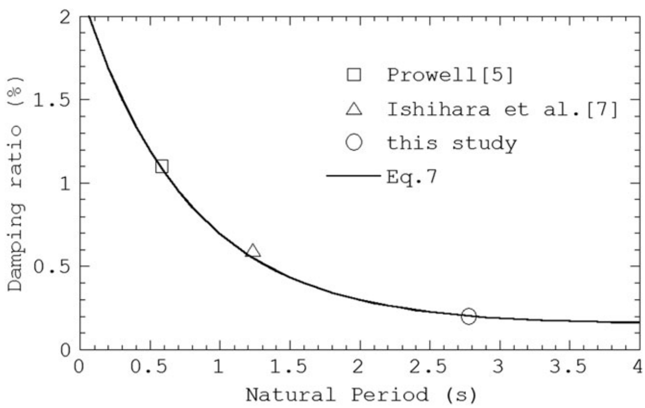

FIGURE 7 The structural damping ratio against natural period for wind turbine first mode

5 | CONCLUSIONS

In this study, a series of excitation tests are performed using an AMD on a 2.4-MW offshore wind turbine. Following conclusions are drawn.

1. The sinusoidal test for the 2.4-MW wind turbine shows that the damping ratio for the fore-aft first mode is 0.2% and that for the fore-aft second mode is 2.4% .
2. The free decay test illustrates that the damping ratios for the fore-aft first and second modes are similar to those obtained with the sinusoidal test, while the damping ratio for the side-side first mode is 1.2% and that for the side-side second mode is 3.2% due to the aerodynamic damping from the blades.
3. The damping ratios for the first mode of wind turbines with steel towers decrease as the natural periods increase and an empirical formula for estimation of the structural damping ratio of first mode is proposed based on the results from the previous researches and the excitation tests in this study.

ACKNOWLEDGEMENTS

This research is carried out as a part of a project funded by the New Energy and Industrial Technology Development Organization (NEDO), Japan, and the field tests were carried out with cooperation from Tokyo Electric Power Company. The authors wish to express their deepest gratitude to the concerned parties for their assistance.

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How to cite this article: Oh S, Ishihara T. Structural parameter identification of a 2.4 MW bottom fixed wind turbine by excitation test using active mass damper. *Wind Energy*. 2018;1–7. https://doi.org/10.1002/we.2214