Resonance phenomenon in a wind turbine system under operational conditions

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ABSTRACT:
A prototype of wind turbines in 5 megawatt class was built and tested from 2007 to October 2009 in the first German offshore wind energy test field in the North Sea. In order to investigate dynamic behaviors under a complex state of loads, a continuous dynamic monitoring system was implemented by Federal Institute for Material Research and Testing (BAM). It recorded structural responses and environmental/operational variables from November 2007 to October 2009.
This paper presents significant resonance phenomenon due to the interaction in the tower-nacelle system under operational conditions. Modal parameters are automatically estimated by the poly reference Least Square Complex Frequency domain (p-LSCF) method. Campbell plot demonstrates that a three-blade passage frequency and its multiples \( f_{3n} \) match with the natural frequencies of the wind turbine system in several modal orders. The damping estimates decrease and the vibration amplitude increase significantly. A control system is necessary to minimize the excessive vibrations.

KEY WORDS: Wind Turbine; Tower-Nacelle system; Resonance; Continuous dynamic monitoring; Automated operational modal analysis.

1 INTRODUCTION
Structural Health Monitoring (SHM) technology is receiving considerable interest in the field of wind power generation. Implementation of an SHM system not only provides an efficient health indicator for early damage detection but also assists to understand the dynamic behaviors of the wind turbine system under normal operational conditions. With the increasing size of the wind turbine for harvesting more energy, the dynamic interaction between the different structural components is still not sufficient, though such an issue has attracted considerable research attentions.
Theoretically, Gasch and Twele suggest that the significant resonance phenomenon of the wind turbine system will be observed when the structural frequencies agree with the frequency resulted from mass unbalance of blades and the harmonic frequencies due to blade passage of tower [1]. In [2], Brasil et al simplify a wind turbine system as an unbalanced rotor on a supporting tower and so-called Sommerfeld Effect is addressed. Murtagh et al perform a time-domain forced vibration analysis of the wind tower considering the dynamic interaction between the tower and the blades [3]. Recently, Liu performs the numerical vibration analysis by further considering the wind turbine system as a blade-carbin-tower coupling system [4]. Staino and Basu proposed a multi-modal mathematical model describing the dynamics of flexible rotor blades and their interaction with the turbine tower, taking the variable rotor speed into account [5]. Nevertheless, nearly no prior works focus on experimental investigations of the dynamic interaction in the tower-nacelle system under normal environmental/operational conditions using real measurements within several years, though they are critical important for design verification, optimization and structural operation maintenance.

This paper mainly describes the experimental investigation on the dynamic behaviors of a wind turbine system with purpose of revealing the significant resonance phenomenon in the tower-nacelle system. It is divided into four main parts: The first part generally introduces the wind turbine system in 5 megawatt class. The second part presents a continuous dynamic monitoring system integrated with automatic operational modal analysis algorithm on the basis of the poly reference Least Squares Complex Frequency domain (p-LSCF) method. Subsequently, correlation analysis is performed between the estimated modal parameters as well as the environmental/operational variables during two years. It is observed that the three-blade passage frequency and its harmonic frequencies \( f_{3n} \) cross the several structural natural frequencies. Meanwhile, the damping estimates decrease and vibration amplitude increases. Finally, discussions of the experimental results and conclusions are presented.

2 A PROTOTYPE OF AREVA MULTIBRID M5000
A prototype of wind turbines in 5 megawatt, Areva Multibrid M5000 (Figure 1(a) and (b)), was built and tested from 2007 in the first German offshore wind energy test field in the North Sea, preparing for the production of the commercial offshore wind power system. Table 1 lists the main characteristics of the wind turbine [6]. In design phase, it is assumed that the generation of electrical power increases...
gradually when the wind speed varying from 4m/s to 12m/s and be stable around 5 megawatt as the wind velocities are larger than 12m/s. When the wind velocity is above 25m/s, the rotor blades will stop working in order to avoid potential damages caused by excessive wind loads. Under normal operational conditions, the direction of nacelle changes automatically with the wind direction in order to produce the maximum rotation speed of rotor blades. According to statistical analysis in design phase, the main wind direction (MWD) is along the Southwest and the secondary wind direction (SWD) mainly distributes perpendicular to the MWD, as shown in Figure 1 (c).

### Table 1. Technical specifications of the prototype of Areva Multibrid M5000

<table>
<thead>
<tr>
<th>General</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Rated power</td>
<td>5000kW</td>
</tr>
<tr>
<td>Design life time</td>
<td>20 years</td>
</tr>
<tr>
<td>Cut-in wind speed</td>
<td>4m/s</td>
</tr>
<tr>
<td>Rated wind speed</td>
<td>12.5m/s</td>
</tr>
<tr>
<td>Cut-out wind speed</td>
<td>25m/s</td>
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</table>

<table>
<thead>
<tr>
<th>Tower</th>
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</thead>
<tbody>
<tr>
<td>Type</td>
<td>Tubular tower</td>
</tr>
<tr>
<td>Height</td>
<td>67m</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Rotor</th>
<th></th>
</tr>
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<tbody>
<tr>
<td>Rotor diameter</td>
<td>116m</td>
</tr>
<tr>
<td>Number of blades</td>
<td>3</td>
</tr>
<tr>
<td>Lowest rotation speed</td>
<td>4.5rpm</td>
</tr>
<tr>
<td>Rated rotation speed</td>
<td>14.8rpm</td>
</tr>
<tr>
<td>Highest rotation speed</td>
<td>14.8rpm±10%</td>
</tr>
</tbody>
</table>

3 CONTINUOUS DYNAMIC MONITORING SYSTEM

3.1 Integrated system for monitoring and assessment

In the context of national research project IMO-WIND, an integrated monitoring system for supervision of all components of wind turbines is developed. It is composed of two individual data measurement, acquisition and transfer systems, installed on both the rotor blade and the support structure with purpose of experimental verification of design assumption [7-9], implementation of a risk-based structural assessment procedure [10,11] as well as development of a structural health monitoring system [12-14].

The dynamic responses of the tubular steel tower are recorded by 8 accelerometers mounted on its internal surface as shown in Figure 1 (b). According to the wind direction, 8 accelerometers are classified in two groups: One consists of y1, y3, y5 and y7 along the MWD and another one is composed by y2, y4, y6 and y8 along the SWD. The structural responses were measured synchronously by signal acquisition equipments HBM MGCplus and were recorded continuously with a sample rate of 50Hz from 1st November 2007 to 31st October 2009. Only the first 8192 sampling points acquired by each accelerometer at beginning of each hour are saved in the centre computer and transferred to BAM. The typical acceleration signals recorded by two groups of accelerometers are plotted in Figure 2.

In order to evaluate the vibration amplitude of the tubular tower, the corresponding Root Mean Square ($RMS$) values on the basis of 8192 sampling points recorded at the beginning of each hour from every accelerometer are calculated as follows:

$$RMS = \sqrt{\frac{1}{8192} \sum_{i=1}^{8192} (x_i)^2}$$  \hspace{1cm} (1)

where $x_i$ is the amplitude of each acceleration sampling point. If the maximum $RMS$ value calculated from 8 accelerometers is defined as $Rmax$, the ratios between the $RMS$ values computed by accelerations recorded by 8 sensors and the $Rmax$ are shown in Figure 3.

![Figure 1](image1.png)  
(a) General overview  
(b) Scheme of wind turbine and positions of 8 accelerometers  
(c) Plane view of the wind turbine and wind direction  

Figure 1 The prototype of wind turbine M-5000

![Figure 2](image2.png)  
(a) Acceleration signals along the MWD  
(b) Acceleration signals along the SWD  

Figure 2 Typical acceleration signals acquired with 8192 sampling points
An environmental/operational measurement station was installed on the hub of wind turbine by the owner AREVA Wind GmbH [9]. From November 2007 to October 2009, the variables such as temperature, wind speed, rotation speed of blades, pitch angle of blades and orientation of nacelle are also recorded simultaneously at the beginning of each hour for 8192 points with sampling frequency 1Hz. In order to investigate the environmental/operational influences on the dynamic properties of the wind turbine, the mean values of every 8192 samples of different environmental/operational factors are considered. The variation and statistical analysis of each environmental/operational factor are presented in [13]. In particular, it is observed from Figure 4 that 23.9% of the measured rotation speeds fall in the range from 7.5-8 rpm during two years.

### Automated operational modal analysis

In order to interpret the massive vibration signals automatically and manage the analysis results robustly, a signal processing and management software system is developed. It consists of four main functions: automated OMA on the basis of the poly-reference Least Squares Complex Frequency domain (p-LSCF) method and data driven Stochastic Subspace Identification (DATA-SSI) approach, investigation of environmental/operational influences on dynamic properties, feature extractions and management as well as visualization of processing results.

Comparison of structural properties estimated by both p-LSCF and DATA-SSI methods demonstrate that the former can provide better modal estimates [12]. Current section introduces the automated p-LSCF algorithm implemented for continuous dynamic monitoring.

Assuming the wind turbine is a linear and time-invariant physical system within the data window considered (8192 points in the present application), positive output spectra \( S^+(j\omega) \) estimated by measured output responses can be modelled in Right Matrix Fraction Description (RMFD) form as:

\[
S^+(j\omega_j) = B_r(j\omega_j)(A_r(j\omega_j))^{-1}
\]  
(2)

where frequency points \( f = 0, 1, 2, \ldots, N_f \), \( B_r(j\omega_j) \) is the numerator matrix polynomial and \( A_r(j\omega_j) \) is the denominator matrix polynomial. Both of them are defined as:

\[
B_r(j\omega_j) = \sum_{\omega_j}^{p} \Omega_r(\omega_j) \beta_r
\]
(3)

\[
A_r(j\omega_j) = \sum_{\omega_j}^{p} \Omega_r(\omega_j) \alpha_r
\]

in which \( p \) is the user defined polynomial order and \( \Omega_r(\omega_j) \) are the polynomial basis function defined as:

\[
\Omega_r(\omega_j) = e^{-i\Delta\omega_j}
\]
(4)

where \( \Delta \) is the sampling period. The polynomial coefficients \( \beta_r \) and \( \alpha_r \) are the parameters to be estimated and are assembled in following matrices:

\[
\beta_r = \begin{pmatrix} \beta_{\theta_0} \\ \beta_{\theta_3} \\ \vdots \\ \beta_{\theta_p} \end{pmatrix}, \quad \alpha_r = \begin{pmatrix} \alpha_0 \\ \alpha_1 \\ \vdots \\ \alpha_p \end{pmatrix}, \quad \theta = \begin{pmatrix} \beta_1 \\ \beta_2 \\ \vdots \end{pmatrix}
\]
(5)

where \( l \) is number of measured outputs.

By fitting the measured output spectra \( S^+(j\omega_j) \) with this model by coefficient \( \theta \) at each frequency point \( \omega_j \), structural modal parameters will be estimated by p-LSCF algorithm [15, 16]. In current research, the half positive spectra matrix \( S^+(j\omega_j) \) is estimated by weighted correlogram method, with maximum frequency points \( N_f = 1024 \). The polynomial orders are considered as \( p = 0, 1, 2, \ldots, 100 \). The number of output responses \( l \) is 4 because the signals recorded from 8 accelerometers are classified as two groups according to the wind direction.

In practice, modal parameters are identified by picking stable poles from a stabilization diagram. The automatic operational modal analysis procedure consisting of the construction of a stabilization diagram and of the selection of stable poles is well described in [17].

Table 2 and Table 3 list the mean value and the standard deviation value of the estimated modal parameters as well as the corresponding frequency values calculated by finite element model. From November 2007 to October 2009, the modal parameters of modes around 0.41Hz, 3.26Hz, 4.02Hz, 6.47Hz, 7.50Hz, 8.15Hz, 12.16Hz and 21.81Hz are successfully identified using automatic p-LSCF method on the basis of the structural responses recorded by accelerometers.
y1, y3, y5 and y7. On the contrary, only dynamic properties of modes around 0.42Hz, 3.26Hz, 4.02Hz and 6.48Hz are estimated by accelerometers y2, y4, y6 and y8. The possible reason is that the direction of y1, y3, y5 and y7 is identical with the MWD while the orientation of y2, y4, y6 and y8 is perpendicular to the MWD. Low excitations along the SWD may cause the difficulties for identification of the modal parameters in higher modes.

Table 2 Statistical analysis of the modal parameters estimated by accelerometers y1, y3, y5 and y7

<table>
<thead>
<tr>
<th>$f_{FE}$ (Hz)</th>
<th>Eigen frequency</th>
<th>Damping ratio</th>
<th>MAC value</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Mean value (Hz)</td>
<td>Std</td>
<td>Mean value (%)</td>
</tr>
<tr>
<td>0.41</td>
<td>0.41</td>
<td>0.007</td>
<td>0.96</td>
</tr>
<tr>
<td>0.42</td>
<td>3.51</td>
<td>0.056</td>
<td>1.73</td>
</tr>
<tr>
<td>3.55</td>
<td>4.09</td>
<td>0.078</td>
<td>1.51</td>
</tr>
<tr>
<td>4.09</td>
<td>6.62</td>
<td>0.086</td>
<td>1.12</td>
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<td>6.62</td>
<td>7.49</td>
<td>0.043</td>
<td>0.36</td>
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<tr>
<td>7.49</td>
<td>8.28</td>
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<td>0.39</td>
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<tr>
<td>8.28</td>
<td>12.72</td>
<td>0.042</td>
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<tr>
<td>12.72</td>
<td>20.75</td>
<td>0.032</td>
<td>0.11</td>
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Table 3 Statistical analysis of the modal parameters estimated by accelerometers y2, y4, y6 and y8

<table>
<thead>
<tr>
<th>$f_{FE}$ (Hz)</th>
<th>Eigen frequency</th>
<th>Damping ratio</th>
<th>MAC value</th>
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<td>Mean value (Hz)</td>
<td>Std</td>
<td>Mean value (%)</td>
</tr>
<tr>
<td>0.41</td>
<td>0.41</td>
<td>0.008</td>
<td>1.19</td>
</tr>
<tr>
<td>0.42</td>
<td>3.31</td>
<td>0.057</td>
<td>1.91</td>
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<tr>
<td>3.31</td>
<td>4.09</td>
<td>0.076</td>
<td>1.50</td>
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<tr>
<td>4.09</td>
<td>6.62</td>
<td>0.082</td>
<td>1.12</td>
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4 RESONANCE PHENOMENON

A particular environmental/operational influence for a wind turbine system is the resonance phenomenon caused by the passage of each blade over the tower. In order to experimentally reveal such phenomenon, the Campbell diagram is presented by plotting all identified frequency estimates against the measured rotation speed of blades [1]. Afterwards, the variation of the damping estimates around the fundamental mode is also investigated. The vibration amplitudes evaluated by the RMS/Rmax values from 8 accelerators are presented. Besides, the resonance phenomenon in higher modes is also discussed.

4.1 Campbell diagram

In a tower-nacelle system, a loading impulse is resulted from each blade passing the tower. For a three-bladed wind turbine, the tower vibrations are excited by 3f and its multiples:

$$f_{3n} = 3nf \quad (n=1,2,3,...)$$  \hspace{1cm} (6)

where $f$ is frequency of the rotation speed of blades. If these frequencies $f_{3n}$ are close to structural frequencies, especially the fundamental frequency of the tower, the resonance will be activated.

Campbell diagram is used to illustrate the influence of the rotational speed on the eigen frequencies of the wind turbine system. Both analytical and experimental Campbell diagrams are shown in Figure 5. The analytical Campbell diagram is calculated by Equation 6 and the experimental one is produced by plotting the measured rotation speeds of blades against the frequencies estimated by the dynamic responses acquired by two groups of accelerators along MWD and SWD from November 2007 to October 2009. Except for the structural frequencies, blades passage frequency 3f and its harmonics 6f…18f are also observed in both Figure 5 (a) and (b). It is clearly notified that the blade passage frequency 3f increases with the rotation speed rising from 5rpm to 14.9rpm and crosses the identified fundamental frequency of the wind turbine system (0.41Hz) as the rotation speed approaching 8.0 rpm (3*8.0rpm=3*8.0/60=0.40Hz). The blades passage frequency 3f matching with fundamental frequency 0.41Hz unavoidably leads to the resonance of the tower, which can be further illustrated by the variation of the damping values and the RMS/Rmax values of different accelerometers.

(a) Eigen frequencies estimated by dynamic responses along the MWD against rotation speed

(b) Eigen frequencies estimated by dynamic responses along the SWD against rotation speed

Figure 5 Analytical and experimental Campbell diagram

4.2 Damping estimates

For the wind turbine system under operational conditions, the total damping $\xi_{total}$ estimated by OMA can be divided into the structural damping $\xi_{struc}$ and the aerodynamic damping $\xi_{aero}$ as:

$$\xi_{total} = \xi_{struc} + \xi_{aero}$$
Structural damping $\xi_{\text{struct}}$ is a measurement of energy dissipation in the wind turbine system. Aerodynamic damping $\xi_{\text{aero}}$ develops from the interaction between the wind and oscillating rotor blades along the direction of the wind. The relative speed between wind and rotor blades determines the aerodynamic load that further affects the structure: the rotor blades that are moving along the wind direction experience an increased wind load that will counteract the tower motion. While the rotor blades move backward, the aerodynamic force reduces with the tower motion. Such effect is associated with the velocity term in equation of motion and thus is termed as aerodynamic damping. It is dependent on wind velocity, rotation speed of blades and other factors such as eigen frequency, geometrical conditions and the kind of the flow around the blades etc. Under operational conditions, aerodynamic damping $\xi_{\text{aero}}$ may become negative and structural vibrations are amplified when resonance occurs [18, 19].

Variation of the experimental estimated damping values $\xi_{\text{total}}$ of the fundamental mode, taking in account of the influences of harmonic mode, can also illustrate the resonance phenomenon of the wind tower. Figure 6 (a) shows the frequency estimates corresponding to the fundamental mode and the harmonic mode excited by $3f$. In order to characterize the variation of the estimated damping values $\xi_{\text{total}}$ at different operational conditions, the frequency samples shown in Figure 6 (a) are artificially divided into four clusters in different colors, according to different rotation speeds and frequency ranges: The first two of them are decomposed of the samples (in blue and in red) that reflect the structural frequencies under both the low (0-1.0 rpm) and the high (14.0-14.9rpm) rotation speeds, without any influences from the harmonic frequencies due to passage of rotor blades. The third cluster consists of the frequency estimates (in black) within 0.4Hz to 0.45Hz as well as the rotation speed varying from 1.0 rpm to14.0 rpm. They are excited by both normal operational loads and blades passing by the tower. The fourth cluster is defined with the frequency samples (in gray) that are smaller than 0.4Hz or larger than 0.45Hz as the rotation speeds fall in the range from 1 rpm to 8 rpm. Approximately, most of them are associated with the excited harmonic mode $3f$.

The corresponding identified damping values $\xi_{\text{total}}$ of these frequency estimates in the range from 0.4Hz to 0.45Hz are plotted against the rotation speeds in Figure 6 (b). To better characterize the variation of damping ratios with the rotation speed, the mean values of the damping estimates $\xi_{\text{total}}$ in different clusters within the interval of 0.5 rpm are calculated and are plotted in Figure 6 (c). As the rotation speed changes from 1 rpm to 5 rpm, the mean values are not included because only a few samples scatter in this range. In Figure 6 (b), a clear gap of the variation of the damping ratios $\xi_{\text{total}}$ is observed in the vicinity of 8 rpm where the blades passage frequency $3f$ crosses the fundamental frequency of the wind turbine system. It can be explained by the variation of the mean damping values with an interval of 0.5 rpm shown in Figure 6 (c). When the rotation speed varies from 0 to 0.5 rpm, then mean value of the damping estimates is 0.76% that may approximately represent the structural damping $\xi_{\text{struct}}$ and the aerodynamic damping $\xi_{\text{aero}}$ may be ignored because the rotation speed is quite low. With the blades begin to rotate from 5 rpm, the mean value of damping estimates $\xi_{\text{total}}$ (5-5.5rpm) jumps to 2.28 due to the activation of the aerodynamic damping $\xi_{\text{aero}}$. As the rotation speed approach to 8 rpm, the estimated damping values decrease rapidly because of the variation of the aerodynamic damping. When the rotation speed changing from 7.5 rpm to 8 rpm, the mean value of the damping estimates is only 0.53% and even smaller than the corresponding mean damping value 0.76% as the blade operates under low speed from 0 rpm to 0.5 rpm. It may result from the negative aerodynamic damping when the resonance occurs, which further reduces the estimated damping $\xi_{\text{total}}$. Afterwards, the damping estimates $\xi_{\text{total}}$ increase gradually with the increasing rotation speed over 8 rpm due to the increasing aerodynamic damping $\xi_{\text{aero}}$. 

\[
\xi_{\text{total}} = \xi_{\text{struct}} + \xi_{\text{aero}} \quad (7)
\]
4.3 Vibration amplitude of the wind tower

Figure 7 shows the ratios $RMS/R_{\text{max}}$ computed on the basis of structural responses recorded by 8 accelerometers installed at the different positions.

$RMS/R_{\text{max}}$ evaluated by $y5$-$y8$ (Figure 7 (e)-(h)) are subject to less pronounced resonance effect. It is easily understood because the fundamental mode reflects the bending mode of the wind tower. Figure 8 (a) shows the dominated fundamental mode shapes [14] estimated by the measurements during 2 years along the MWD and the SWD, agreeing with the calculated bending mode in Figure 8 (b). The excited bending mode leads to the relative significant vibration on the top of the tower.

4.4 Resonance phenomenon in wind blades

Figure 9 shows the first flapwise mode of the wind blades calculated by a numerical model [9]. It is found in Figure 10 that the harmonic frequency $6f$ meets with the frequency estimates of such mode around 13.5rpm. It may be concluded that the potential resonance phenomenon also occurs in the wind blades under operational conditions.

4.5 Resonance phenomenon in higher modes

Figure 11 describes the long term trend of the frequency estimates around 7.50Hz during two years. In Figure 11 (a), the identified frequencies mainly scatter in two parts. One of them reflects the annual fluctuation under low and middle rotation speed (0-1 rpm in blue and 1-14 rpm in black), and another one focus in the range from around 7.45Hz to 7.50Hz as the wind blade spins with higher speed (14-14.9 rpm in red). Such variation may be explained by Figures 11 (b)-(d).
On one side, as the measured wind speed changes from 0 m/s to about 10 m/s, the rotation speed of blades varies from 0 rpm to about 14 rpm. Under these operational conditions, the frequency estimates are only subjected to the temperature influence as illustrated in Figures 11 (b). On the other side, with the wind speed rising above about 10 m/s, the rotation speed of blade increases gradually from 14 rpm to the maximum rotation speed 14.9 rpm (Figures 11 (c) and (d)). Under such conditions, the identified frequencies drop dramatically due to the resonance frequency $30f$ ($30 \times 14.9 \text{ rpm}/60 = 7.45 \text{ Hz}$) induced by the blades passing by the tower. From Figure 11 (b), it is observed that the resonance frequencies are not subjected with the influences of temperature.

![Variations of frequency estimates around 7.50Hz](image)

(a) Variations of frequency estimates around 7.50Hz
(b) Frequency estimates vs temperature
(c) Frequency estimates vs rotation speed
(d) Frequency estimates vs wind velocity

Figure 11 Environmental/operational influences on the variation of frequency estimates around 7.50Hz

Similar resonance impacts are also observed for the identified frequencies around 8.15Hz plotted in Figure 12. When the rotation speeds are higher than 14 rpm, the frequency estimates are excited by the blades passage frequency $33f$ ($33 \times 14.9 \text{ rpm}/60 = 8.20 \text{ Hz}$). While the rotation speed is smaller than 14 rpm, the identified frequencies are mainly affected by the temperature variations as illustrated in Figure 12 (b).

Figure 13 shows the relations between damping estimates and rotation speed for these modes. Table 4 lists the averaged values of damping estimated under low (0-1.0 rpm) and high (14.0-14.9 rpm) rotation speed on the basis of accelerometers along the MWD. For the modes are subjected to the resonance effects due to the blade passing frequencies $30f$ and $33f$ (Figure 11-12), the decreasing damping values with rotation speed varying from 14 rpm to 14.9 rpm indicates that the negative aerodynamic damping caused by the resonance.

![Damping estimates with frequencies around 7.50Hz](image)

(a) Damping estimates with frequencies around 7.50Hz
(b) Damping estimates with frequencies around 8.15Hz

Figure 13 Damping estimates versus rotation speed

<table>
<thead>
<tr>
<th>Mean value of frequencies (Hz)</th>
<th>Mean value of damping ratios (%)</th>
<th>Difference (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Low speed (0-1.0 rpm)</td>
<td>High speed (14.0-14.9 rpm)</td>
<td></td>
</tr>
<tr>
<td>3.26</td>
<td>1.51</td>
<td>+0.68</td>
</tr>
<tr>
<td>4.02</td>
<td>0.79</td>
<td>+1.34</td>
</tr>
<tr>
<td>6.47</td>
<td>1.01</td>
<td>+0.58</td>
</tr>
<tr>
<td>7.50</td>
<td>0.43</td>
<td>-0.37</td>
</tr>
<tr>
<td>8.15</td>
<td>0.39</td>
<td>-0.20</td>
</tr>
</tbody>
</table>

5 DISCUSSIONS

Figure 4 in section 2 remind that the rotation speed of rotor varying from 7.5 rpm to 8 rpm account for the 23.9% of all measurements during two years. Under such range, the resonance occurs and results in the excessive vibration of the wind tower as shown in Figure 7. It indicates that the wind tower is easily to be stuck in the resonance. Such phenomenon can be explained by the Sommerfeld Effect [2,13].

![Frequency spectrum based on strain measurement](image)

Figure 14 Frequency spectrum based on strain measurement

Moreover, it is suggested in [1] that the resonance, caused by the period excitation (1f) due to both the mass unbalance of blades and the harmonic frequencies (3f, 6f, 9f, ...) resulted from blades passage of tower, play a big role during operation of wind turbines, and the former may lead to more significant
lateral vibration. Using the acceleration measurements in current paper, the period excitation (1f) due to the mass unbalance is not observed. However, it is shown in Figure 14 that both 1f and 3f are clearly observed in the frequency spectrum based on the strain measurement [7-9]. In the future, the automated p-LSCF method will be applied to the strain measurement in order to detect the 1f effect.

Finally, according to the numerical results shown in section 4.4, the resonance phenomenon of the blades due to the 1st flapwise frequency meets with 6f excitation is observed. In the future, accelerometers will be installed on the wind blades in order to experimental determine the resonance phenomenon of the blades under operational conditions.

6 CONCLUSIONS

This paper mainly presents the resonance phenomenon of a wind turbine system under operational conditions, due to the structural frequencies in different orders meet with the blade passage frequency and its multiples f_{30}. Significant increase of the vibration amplitude of the wind tower is observed. It is caused by the fundamental frequency matching with 3f excitation as the rotation speed approaching 8rpm. Even worse, 23.9% of the measured blade rotation speeds fall in the range from 7.5-8rpm, which further aggravates the resonance of the wind tower. An active or passive control system is necessary for such a wind turbine system. Meanwhile, the 1st flapwise mode of the blades agreeing with the 6f excitation may also indicate the potential resonance of the blades. Finally, the observed resonance phenomenon in the higher modes due to the 30f and 33f excitation is useful for the explanation of the variation of the modal parameters under operational conditions.

ACKNOWLEDGMENTS

The authors are grateful to the German Ministry of Economics and Technology for the financial support of the IMO-WIND project (Grant No. 161NO326), also to the Federal Institute for Materials Research and Testing (BAM). The fellowship granted by Federal Institute for Materials Research and Testing (BAM) - 031905. Finally, according to the numerical results shown in section 4.4, the resonance phenomenon of the blades due to the 1st flapwise frequency meets with 6f excitation is observed. In the future, accelerometers will be installed on the wind blades in order to experimental determine the resonance phenomenon of the blades under operational conditions.

REFERENCES