Detection and Identification of Damaged Blade without Stopping the Wind Turbine

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Abstract

It was previously shown that rotor anisotropy caused by blade damage affects the periodic mode shapes of the operating wind turbine. This phenomenon can be utilized for detecting and identifying the damaged blade. The previously suggested approach assumed that the vibration sensors are placed on each blade, which is not practical. In addition, it required modal identification, which significantly complicated the process of damage detection and localization.

The current study presents another approach to blade damage detection and localization; the approach utilizes the same physical phenomenon but requires only one vibration sensor located at the tower top. The new approach uses basic signal processing and does not require any modal analysis.

1 INTRODUCTION

Nowadays, it is apparent that automatic monitoring of wind turbine blades is a significant element of cost efficient green energy production [1], [2]. Indeed, the blades of a modern wind turbine are extremely complex structures, constituting a significant part of the entire wind turbine cost. The blades are subjected to harsh weather conditions and must collect and transmit the megawatts of energy to the shaft. Even a small blade fault, if not noticed and repaired, may develop into a big one, which may lead to an expensive blade repair and long downtime. The present approach to blade-health monitoring is regular visual inspections that take place every one-to-two years and do not guarantee timely damage detection.

This explains the interest of the SHM society in the automatic monitoring of wind turbine blades. Many methods have been suggested, see [3] for the review. Utilizing the change in modal parameters to indicate damage is one of the popular approaches, though the low sensitivity of the natural frequencies to damage is well known. In his previous paper [4], the author suggests using the shapes of the backward and forward whirling in-plane modes as a damage indicator, which demonstrate much higher sensitivity compared to the natural frequencies.

The main drawbacks of the above-mentioned approach is that obtaining rotor mode shapes requires fitting the blades with vibration sensors (accelerometers). This significantly complicates the monitoring system. The presented study proposes a novel approach to overcome this drawback: it uses a signal from a single sensor mounted on the top of the tower (more precisely, in the nacelle) in side-to-side direction. In addition, this approach does not
require modal identification and uses simple signal processing.

The paper is organized as follows: section 2 provides a brief theoretical insight into modal decomposition of systems with a rotating rotor; section 3 explains the physical phenomenon used for detection and identification of rotor anisotropy; section 4 introduces the new approach; and finally, section 5 demonstrates the approach on synthesized data.

2 THEORETICAL BACKGROUND

As it was mentioned in the introduction, the presented approach does not require experimental modal identification. However, to understand the idea behind the method, it is beneficial to apply modal decomposition to the vibrations of a system with a rotating rotor.

The equation of motion of a (linear) structure with rotating rotor is given by

\[
M(\psi(t)) \ddot{y}(t) + G(\psi(t)) \dot{y}(t) + K(\psi(t)) y(t) = 0, \tag{1}
\]

where \(M\), \(G\) and \(K\) are the mass, damping/gyroscopic and stiffness matrices respectively, which depend on rotor azimuth angle \(\psi(t)\), varying in time. Vector \(y(t)\) describes the displacement of the system’s \(N\) degrees of freedom. If the rotor has a constant angular speed, \(\Omega = \dot{\psi}(t) = \text{const}\), (1) is the equation of motion of a linear periodic time variant (LPTV) system.

In general, the conventional modal decomposition is not applicable to LPTV systems since the system is not time invariant. For LPTV systems, the Floquet theory [5] suggests a decomposition similar to the modal decomposition but involving periodic mode shapes:

\[
y(t) = \sum_{r=1}^{N} b_r u_r(t) e^{\lambda_r t}, \tag{2}
\]

where \(b_r\) is the mode participation factor, and \(\lambda_r\) is the eigenvalue defining the natural frequency and damping. The periodic mode shape \(u_r(t)\) can be expanded into an infinite number of Fourier components:

\[
u_r(t) = \sum_{n=-\infty}^{+\infty} C_{r,n} e^{jn\Omega t}, \tag{3}
\]

where the complex-valued vector of Fourier coefficients \(C_{r,n}\) defines the magnitude and phase of each component. Expression (2) can be written as a double summation:

\[
y(t) = \sum_{r=1}^{N} b_r \sum_{n=-\infty}^{+\infty} C_{r,n} e^{(\lambda_r + jn\Omega)t}, \tag{4}
\]

where the first sum represents the modal decomposition and the second sum is the Fourier decomposition of the periodic mode shapes.

To illustrate the above-mentioned, let us consider a simple system introduced in [4]: a three-blade rotor mounted on a mass supported by vertical and horizontal springs. Each blade is modelled as an assembly of two bars, connected by an angular spring, and a lumped mass at the end. The system has six degrees of freedom (DOFs) and is described by vector \(\mathbf{y} = [x_C, y_C, \phi_1, \phi_2, \phi_3, \theta]^T\), where \(x_C\) and \(y_C\) are the horizontal and vertical displacement of the mass, \(\phi_1, 3\) are the in-plane angular vibration of each blade and \(\theta\) is the angular shaft displacement. If the rotor speed, geometry, mass and stiffness properties are known, the Floquet analysis can be performed to analytically obtain the system’s eigenvalues and Fourier coefficients [5]; the detailed computation procedure and values of the parameters are given in
For the experimental identification of LPTV systems, the technique proposed in [6] can be employed.

The system depicted in Figure 1 has six DOFs and hence, six periodic modes. Each mode has an infinite number of Fourier components. However, only few components are significant, the components with small Fourier coefficients $C_{r,n}$ can be discarded. In practice, the modes are named after the dominating Fourier component, which also resembles the mode shape of the system at a standstill.

The mode visualization is exemplified in Figure 2. Two modes are shown, backward (BW) whirling mode and forward (FW) whirling mode. The normalized values of $C_{r,n}$ (for selected DOFs) are shown as magnitude (bottom) and phase (top). The complexity plots of the dominant whirling component are shown in the insets.

3 SENSITIVITY OF WHIRLING MODES TO ROTOR ANISOTROPY

In [4], a change in the shape of the dominant whirling component of the BW and FW whirling modes is suggested as a blade damage indicator. The idea is demonstrated in Figure 2: the left column displays the isotropic (undamaged) rotor, whilst the right column displays the rotor with 3% reduced stiffness of the third blade. Both the magnitude (marked by “B”) and the phase (marked by “A”) are affected, which allows damage detection and localization. Study [4] shows that this damage indicator is significantly more sensitive than the natural frequency and demonstrates how it can be obtained from measured vibrations. However, to calculate this damage indicator, it is necessary to have vibration sensors mounted on each blade (preferably accelerometers at the blade tips), which is a demanding technical task in real life. In addition, the method suggests using OMA in order to extract the whirling component of the modes. Though a simplified way of conducting OMA will suffice, this task will significantly complicate the damage index calculation and make its automation more difficult.

The idea of using tower top vibrations instead of blade tip vibrations is originally suggested in [4]. It was noticed that the secondary horizontal (side-to-side) tower component of the BW and FW whirling modes is significantly affected by the rotor anisotropy (marked by “C” in Figure 2b and Figure 2d). Study [4] lists two practical problems of utilizing this phenomenon as a damage indicator: it does not identify which blade is damaged, and the frequency of the secondary component of one whirling mode almost coincide with the frequency of the primary side-to-side components of the other whirling mode. To clarify the latter, the secondary component of the BW mode has the frequency 1.03Hz, which almost coincides with the frequency of the primary side-to-side component of the FW mode (1.06Hz). Similarly, the
secondary side-to-side FW mode component (0.74 Hz) almost coincides in frequency with the primary side-to-side component of the BW mode (0.71 Hz). If the system was entirely symmetric (namely, $k_H = k_V$ in Figure 1), these frequencies would be equal.

4 IDENTIFICATION OF THE DAMAGED BLADE

A thoughtful examination of the analytically computed Fourier components reveals the correlation between the index of the damaged blade and the phase of the Fourier coefficients. Figure 3 demonstrates this observation, comparing the phase of the secondary side-to-side Fourier coefficient with the phase of the primary one (which is set to zero for the convenience of comparison). The secondary side-to-side component is selected due to its high sensitivity to the rotor anisotropy.

Figure 2. Rotor related modes obtained via Floquet analysis (from [6]), top: phase, degrees; bottom: normalized magnitude: a,b) BW whirling mode; c,d) FW whirling mode. Left column: isotropic rotor; right column: anisotropic rotor with 3% reduced stiffness of the third blade. The insets show the complexity plots of the dominant whirling component.
Figure 3. Magnitude and phase of the Fourier components of the BW (left) and FW (right) whirling modes. Only the side-to-side DOF of the tower \((x_C)\) is shown. a,b) Magnitude; Phase, degrees: c,d) Blade #1 damaged; e,f) Blade #2 damaged; g,h) Blade #3 damaged. The red squares indicate the affected Fourier component.

From Figure 3 it becomes apparent that there is a correlation between the phase of the secondary side-to-side Fourier coefficients and the index of the damaged blade. The observations are summarized in Table 1.

<table>
<thead>
<tr>
<th>Index of the damaged blade</th>
<th>BW whirling mode</th>
<th>FW whirling mode</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>(180^\circ)</td>
<td>0</td>
</tr>
<tr>
<td>2</td>
<td>(60^\circ)</td>
<td>(120^\circ)</td>
</tr>
<tr>
<td>3</td>
<td>(-60^\circ)</td>
<td>(-120^\circ)</td>
</tr>
</tbody>
</table>

Table 1. Phase of the secondary side-to-side Fourier component

If it was possible to reveal the phase information from the vibration signal measured at the top of the tower, one could identify the damaged blade.

5 EXPERIMENTAL APPROACH

To address the problem formulated in the previous section, we suggest using same approach as utilized in the harmonic OMA time domain (H-OMA-TD) method, [6], [7]. The measured
signal is correlated with its modulated version, which is based on the rotor azimuth. It is crucial to use the same azimuth origin as in the Floquet analysis as the offset from the origin is used to identify the damaged blade.

The procedure consists of the following steps:

1. While the turbine is operating, the tower top vibration (in side-to-side direction) and rotor azimuth are recorded. For the simplified structure in Figure 1, it is $x_C(t)$ and $\psi(t)$.

2. The vibration signal is modulated:

$$x_{C,K}(t) = x_C(t)e^{-jK\psi(t)}, \quad (5)$$

Since the Fourier components of interest are separated by $2\Omega$, one has to use $K = \pm 2$. Note, the signal $x_{C,K}(t)$ is complex. Modulation will shift the frequency spectrum by $\pm 2\Omega$.

3. Coherence functions between the original signal $x_C(t)$ and its modulated versions $x_{C,2}(t)$ and $x_{C,-2}(t)$ are computed.

4. The peaks of the coherence function (outside the rotor harmonics) indicate the rise of the secondary side-to-side component, and the appearance of rotor anisotropy.

5. To identify the damaged blade, the cross-spectra between the signal and its modulated versions are computed. The phase of the cross-spectrum functions computed at the frequencies where the coherence functions have the peaks will pinpoint the damaged blade.

5.1 Demonstration

To demonstrate the proposed approach let us use the simulated experiment: the simplified rotor system shown in Figure 1 is loaded with uncorrelated random forces applied to all DOFs. The forces model the wind excitation; the author acknowledging that real wind excitation has a much more complicated nature, however at the frequency ranges between rotor harmonics, such a modeling can be considered sufficiently realistic, [8].

After replacing the right-hand-side of the equation of motion (1) by the time histories representing the forces, the equation is numerically integrated. The resulting response time histories replicate the data from the vibration sensors, as from a real experiment. The simulation is conducted for the stationary regime: $\dot{\psi}(t) = \Omega t$.

The power spectral density (PSD) of the side-to-side motion (coordinate $x_C$) is shown in Figure 4a for three cases: isotropic (undamaged) rotor, rotor with blade #2 stiffness reduced by 3% and rotor with blade #2 stiffness reduced by 5%. Except the clear peaks at the first and second rotor harmonics due to the rotor unbalance, there are no visual differences between the spectra of the undamaged and damaged systems.

The magnitudes of the normalized Fourier coefficients for BW and FW whirling modes shown in Figure 4b. The phenomenon in focus is the rise of the secondary side-to-side Fourier components due to rotor anisotropy, the corresponding coefficients are denoted by the red squares. Apparently, the effect of the phenomenon is masked by the primary side-to-side motion of the counterpart mode and cannot be found by straightforward spectrum examination.

To reveal the phenomenon from the measured signal, one can modulate the signal to shift its spectrum by $\pm 2\Omega$ and correlate it with the original spectrum, as explained in Figure 5. The coherence functions between the original signal and its frequency-shifted version are shown in Figure 6a and Figure 6b, for frequency shifts $+2\Omega$ and $-2\Omega$, respectively. For the undamaged rotor, the coherence is almost zero over the entire frequency range (red line). For the rotor with
one damaged blade, there are high values of coherence at the rotor harmonics frequencies and at the frequencies denoted in Figure 5. The latter are in the focus of the study.

To identify which blade is damaged, the cross-spectrum between the original signal and its frequency-shifted version is calculated. Figure 6c and Figure 6d show the phase of the cross-spectrum at the frequencies denoted in Figure 5, for the damaged blade #2.

Figure 7 shows the phase for damaged blade #1 (top) and blade #3 (bottom). The values given in the datatips are in good correlation with Table 1. The swopping positive and negative values for the phase is explained by the arguments’ arrangement when calculating the cross-spectrum: the first argument is the original signal; the second is its frequency-shifted copy (cf. Figure 5).

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Figure 4. a) PSD of the simulated side-to-side signal, vertical dashed lines are rotor harmonics; b) normalized magnitude of the Fourier coefficients (BW and FW whirling modes) computed for blade #2 stiffness reduced by 5%.

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Figure 5. Frequency shift and the correlation between the original signal and its modulated versions. Filled triangles denote the Fourier components of the BW mode, empty triangles denote FW mode. Big triangles stand for primary components, small ones stand for secondary...
5.2 Discussion

One can argue that the peaks at the PSD (Figure 4a) at the first and second rotor harmonics already good indicators of any rotor unbalancing, and this is correct. However, finding out which blade is damaged is very difficult or impossible since the phase at the rotor harmonics (Figure 6c, 6d and Figure 7) depends on the rotor speed. In contrast, Fourier coefficients carry information about the structure, not the excitation, and do not depend on the rotor speed.

6 CONCLUSIONS

The study presents a technique for anisotropy detection of bladed rotors and identification of the blade with reduced stiffness. The technique utilizes the high sensitivity of specific Fourier components of the whirling rotor modes to blade damage. In contrast to the previous
work, the technique does not require installation of vibration sensors on the blades and only uses one vibration sensor, measuring in the side-to-side direction, installed on the top of the tower. From the implementation point of view, the latter is much more convenient. Another advantage of the proposed technique is that it does not need modal identification. Instead, it uses simple signal processing: modulation and calculation of cross-spectra and coherence functions.

The presented study only introduces the technique. Further work to investigate the method sensitivity and robustness is required.

REFERENCES


