APPLICATION OF TORSIONAL VIBRATION MEASUREMENT TO BLADE AND SHAFT CRACK DETECTION IN OPERATING MACHINERY

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Abstract: The primary goal of the this paper is to summarize field demonstrations of the feasibility of detecting changes in blade and shaft natural frequencies (such as those associated with a blade or shaft crack) on operating machinery using non-contact, non-intrusive measurement methods. This paper primarily addresses the results of application of this non-intrusive torsional vibration sensing to: a large wind tunnel fan; a jet engine high-pressure disk; a hydro station turbine; and to large induced-draft (ID) fan motors at a coal-fired power plant. During the operation of rotating equipment, torsional natural frequencies are excited by turbulence, friction, and other random forces. Laboratory testing was conducted to affirm the potential of this method for diagnostics and prognostics of blade and shafting systems. Field installation at the NASA Ames National Full-Scale Aerodynamic Facility (NFAC) reaffirmed the ability to detect both shaft and blade modes. Field installation at a hydro power station demonstrated that the few shaft natural frequencies were visible, and correlated well with finite element results. Field installation on the ID fan motors also showed the first few shaft torsional modes. Finally, installation on a high-pressure (HP) disk in a jet engine test cell at General Electric Aircraft Engines demonstrated that the fundamental mode of the turbine blades was clearly visible during operation. The results of these field tests have resulted in high confidence that this technique is practical for diagnosing and tracking shaft and blade cracks.

Key words: Shaft cracking; blade cracking; condition-based maintenance; failure prediction; torsional vibration.

Background: The detection of blade and shaft natural frequencies in the torsional domain requires that the signal resulting from excitation of the rotating elements by turbulence and other random processes is measurable. If measurable, these natural frequencies may be tracked to determine any shifting due to cracking or other phenomena affecting torsional natural frequencies. Difficulties associated with harvesting the potentially very small signals associated with blade shaft vibration in the torsional domain could render detection infeasible. Thus, transduction and data acquisition must be optimized for dynamic range and signal to noise ratio [1, 2, 3]. An overview of the laboratory results may be found in [5,6]

The advantage of using shaft torsional natural frequency tracking over shaft lateral natural frequency tracking for detecting cracks in direct-drive machine shafts is twofold:
• A shift in natural frequency for a lateral mode may be caused by anything which changes the boundary conditions between the rotating and stationary elements: seal rubs, changes in bearing film stiffness due to small temperature changes, thermal growth, misalignment, etc. So, if a shaft experiences a shift in lateral natural frequency, it would be difficult to pinpoint the cause as a cracked shaft. However, none of these boundary conditions influence the torsional natural frequencies. So, one may say that a shift in natural frequency in a torsional mode of the shaft must involve changes in the rotating element itself, such a crack, or perhaps a coupling degradation.

• Similarly, finite element modeling of the rotor is simplified when analyzing for torsional natural frequencies: these boundary conditions, which are so difficult to characterize in rotor translational modes, are near non-existent in the torsional domain for many rotor systems. This means that characterization of the torsional rotordynamics is more straightforward, and therefore likely to better facilitate diagnostics.

Detection of the small torsional vibration signals associated with blade and shaft natural frequencies is complicated by transducer imperfections and by machine speed changes. The use of resampling methods has been shown to facilitate the detection of the shaft natural frequencies by: (1) correcting for torsional transduction difficulties [2] resulting from harmonic tape imperfections (printing error and overlap error); and (2) correcting errors as the machine undergoes gradual speed fluctuation [3, 4]. In addition, correction for more dramatic speed changes was addressed in [4]. These corrections made laboratory testing quite feasible.

**Transducer setup and methodology:** The transducer used to detect the torsional vibration of the shaft included a shaft encoded with black and white stripes, an infrared fiber optic probe, an analog incremental demodulator and an A/D converter. The implementation of the technique under laboratory conditions was previously presented in [2, 3]. Figure 1 shows a schematic of the transducer system.

![Figure 1: Schematic of transducer setup for torsional vibration measurement](image)
Field implementation:

**NASA Ames National Full-Scale Aerodynamic Complex (NFAC) Fans:** The NFAC facility is the largest wind tunnel in the world, able to test a full-scale 737 in the largest section (see Figure 3). The fans used to drive the wind tunnel, shown in Figure 4, have experienced some blade cracking in the past, and the repairs made continued inspection unpractical. NASA decided to use modal impact testing of the blades to track the natural frequencies of the blades to detect shifts that might be associated with cracking. However, it was determined that the frequency might shift as much as ten percent simply by rotating the rotor $360^\circ$ and retesting the same blade. Since this was on the order of the expected shift due to cracks, other methods were considered. A feasibility study was conducted using torsional vibration on an operating fan.

![Figure 3: National Full-Scale Aerodynamic Complex (NFAC) at NASA Ames](image)
The shaft was encoded with zebra tape, as shown in Figure 5, and testing was performed during operation of the fans.
The results are shown in Figure 6. The first two peaks correspond to overall shaft torsional frequencies, based on simple dynamic models developed by the vendor in the 70s. The third peak, near 13.5 Hz, corresponds to the blade group. Of particular interest is the tight packing of the fifteen blade modes, especially knowing that the impact testing on a single blade could vary by 1 Hz or more during a single test session. We believe that the variable pitch mechanism (VPM) does not provide a repeatable boundary support for the blades when it is not operating, but provides uniform and repeatable results during operation, when the VPM is preloaded.

Subsequent to this testing, impact testing was performed on a blade with the oil pump operating to attempt to allow torsional coupling. The resulting natural frequency was found to be about 14.6 Hz, as shown in Figure 7.

This testing demonstrated the ability of this measurement system to detect blade natural frequencies during operation of the NFAC fans.
Figure 7: Results of Modal Testing of Single Blade

**Power plant implementation:** The methodology was implemented on two power plant machines: one a hydroelectric plant turbine generator that has experienced cracking on its newly redesigned turbine rotor; the other a motor on an induced draft (ID) fan at a super-critical coal-fired plant that has experienced cracking of the web-shaft welds.

**Hydro turbine:** The methodology was implemented on a hydroelectric plant turbine generator that has experienced cracking on its newly redesigned turbine rotor. The hydro plant consists of five 3 MW electric turbine generators sets. The plant was originally built in about 1910, but it has recently been redesigned to eliminate an underwater, wooden (lignum vitae) bearing and improve efficiency. The layout of a unit is shown in Figure 8 and Figure 9.
However, in the last five years, three of the newly designed turbine rotors have experienced severe cracking. Instrumentation and analysis was performed on one of the units that had not experienced cracking to demonstrate the feasibility of detecting shaft natural frequencies. Figure 10 shows the optic probe, tachometer, and encoded tape placement.

The data was analyzed using the double resampling technique [3,4] to eliminate the adverse effects of the presence of running speed and its harmonics on frequency identification. The results of four test runs are shown in Figure 11. Note the peaks at about 16 Hz and 41 Hz. These correspond well to the finite element model torsional frequencies of 16 Hz and 40 Hz.
The frequencies below 5 Hz are somewhat enigmatic. Since the operating speed of the unit is 300 RPM, or 5 Hz, it was at first assumed that these frequencies correspond to fluid whirl, which generally occurs at speeds between 0.42 and 0.48 times operating speed \[10\]. However, the shaft lateral vibration data exhibited none of the signs of whirl. In addition, the three closely spaced subsynchronous peaks were stable and repeatable from run to run, as seen in Figure 12. Such stability and repeatability for three closely spaced frequencies does not correspond to the whirl phenomenon. In addition, similar spectral components have since been observed on hydro units at other sites. So, we hypothesize that these subsynchronous frequencies corresponds to the “rigid body” torsional mode on torsional springs corresponding to the bearing film stiffness in shear. Further investigation will be necessary to confirm this and to clarify the significance of these spectral components.
Several issues arose during the on-site data acquisition and analysis. Figure 13 shows some of the data of Figure 12 along with runs that had significant distortion due to tape errors. When the tape was changed, or even the axial location of the transducer was changed on the same tape, the spurious frequencies shifted. These spurious frequencies seem to be related to the encoded tape, and often interfered with the identification of shaft natural frequencies. Investigation into this phenomenon is continuing.

**Figure 13: Torsional spectra showing encoded tape error spectral content**

**ID fan motors:** The motors on the fossil-fired induced draft fan were constructed using rectangular cross section webs from the shaft to the rotor coil supports. The square end on the web was then welded to the circular shaft without machining to match the contours. The result has been a number of failures of the motors due to failure of the web welds. Two of these motors were instrumented to detect shaft natural frequencies and establish a baseline to track the changes that may be associated with web weld failure. Figure 14 shows the fan motor.

**Figure 14: ID fan motor: (a) Motor housing; (b) scaled with minivan**
Installation of the tape was more difficult on the ID fan than on the hydro unit due to the shaft size and the tight quarters. Figure 15 shows the installation of the transducer system.

In addition, the butt joint misalignment of the ends of the tape appeared to be exacerbated by thermal growth of the shaft. It was observed that a space between the ends appeared after heat up of the unit. This underlap, in some cases, caused saturation and malfunction of the analog demodulator. Figure 16 shows the results for one of the fans. The first mode appears to be about 10 Hz. Once again, it was observed that changing tapes or changing the shaft axial position of the optical probe on the encode tape changed some of the spectral content above 20 Hz. It is difficult to assess the remainder of the spectrum with high confidence due to the spectral content of the tape. However, most likely the second and third modes are at about 16 Hz and 19 Hz.
**Aircraft Jet Turbine:** The HP turbine of a commercial jet engine was instrumented to determine the feasibility of detecting the blade natural frequencies during operation using torsional vibration. The testing was performed in a warm air test facility, under load at about 9400 RPM. Figure 17 shows the fully assembled commercial engine and the HP disk as installed in the warm air test facility.

![Figure 17: a) Fully Assembled Jet Engine; b) HP Disk in Warm Air Test Facility](image)

Several individual blades were tested by the engine manufacturer to determine the natural frequencies. These blades were fixtured to simulate the boundary conditions during operation. Figure 18 shows a typical spectrum of the results of the modal testing, which is summarized for the three specimen blades in Table 1.

![Figure 18: Typical Spectrum of Individual Fixtured Blade](image)
The instrumentation included a 200-stripe zebra tape and fiber optic probes. In addition, the output from a 60-tooth speed encoder was used as backup. The zebra tape did not survive the test, and the speed signal from the 60-tooth gear was used.

![Image of zebra tape and encoder](image)

### Table 1: Results of Modal Testing of Sample Blades

<table>
<thead>
<tr>
<th>Blade</th>
<th>Mode 1</th>
<th>Mode 2</th>
<th>Mode 3</th>
<th>Mode 4</th>
</tr>
</thead>
<tbody>
<tr>
<td>F12</td>
<td>2450</td>
<td>3943</td>
<td>6038</td>
<td>8409</td>
</tr>
<tr>
<td>C14</td>
<td>2413</td>
<td>3853</td>
<td>6091</td>
<td>8469</td>
</tr>
<tr>
<td>B7</td>
<td>2434</td>
<td>3875</td>
<td>5994</td>
<td>8364</td>
</tr>
<tr>
<td>Average</td>
<td>2432</td>
<td>3890</td>
<td>6041</td>
<td>8414</td>
</tr>
<tr>
<td>Std. Dev.</td>
<td>19</td>
<td>47</td>
<td>49</td>
<td>53</td>
</tr>
</tbody>
</table>

The use of the 60-tooth wheel limited the frequency range of the data to one-half the number of teeth times the speed of the shaft, or about 4500 Hz. Figure 20 shows the torsional spectrum of the HP turbine rotor. Note that torsional natural frequencies of the shafting system should change very little with speed, and coupled blade and shaft torsional modes increase very slightly with speed due to stiffening by axial force. So, we look for frequencies that are not shifting with speed. Note that the first peak at about 2400 Hz is close to the first mode from the blade modal testing.

Earlier experimental and analytical work, however, indicates that blade natural frequencies can be significantly altered by coupling with shaft torsional modes [3,4]. It was shown that for a small desktop rotor with much larger blade to rotor mass ratio the difference might be more than 30%, and may be higher or lower than rig impact testing. Although intuition might imply that for relatively small blades coupling would be less of a factor, the effects of torsional coupling should be clarified using a dynamic torsional model.
Summary and conclusions: The techniques developed for detecting torsional natural frequencies in the laboratory were implemented on a large wind tunnel fan which has experienced blade cracking, power plant machines that have experienced shaft cracking (hydro turbine generator, ID fan motor), and a jet engine which has experience cracking in the disc near at blade root. The goals of the implementation project were to demonstrate the feasibility of field application, and to establish a baseline for each class of machine. The data acquired clearly demonstrated the feasibility of field implementation, and established baseline natural frequencies for blades and shafts.

However, interference from tape related spectral content was experienced. This interference was not experienced in the laboratory due to shaft size, access, and environmental differences. It is believed that this spectral content is associated not with tape printing error or overlap, but was introduced by the installation. This may be overcome by using a more precisely fabricated encoding device (such as the 60-tooth speed encoder at the jet engine test facility), or by developing correction algorithms for the installation error.

Future work: Research to establish the size of detectable cracks in each class of machine is needed. For instance, reactor coolant pumps (RCP) in nuclear power plants have experienced severe cracking with no notice given by the crack detection instrumentation. It is important that we estimate the size of the detectable crack and the remaining useful life of the shaft at the detection threshold to establish the torsional vibration method as practical in field machines. Correction of the installation errors must be accomplished to remove ambiguity and make the technology widely accessible. This work is currently underway.
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References:


