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ON THE FEASIBILITY OF BLADE CRACK DETECTION THROUGH TORSIONAL VIBRATION MEASUREMENTS

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Abstract: The primary goal of the development project was to demonstrate the feasibility of detecting changes in blade natural frequencies (such as those associated with a blade crack) on a turbine using non-contact, non-intrusive measurement methods. The approach was to set up a small experimental apparatus, develop a torsional vibration detection system, and maximize the dynamic range and the signal to noise ratio. The results of the testing and analysis clearly demonstrated the feasibility of using torsional vibration to detect the change in natural frequency of a blade due to a change in stiffness such as those associated with a blade crack.

Keywords: Torsional vibration; failure prediction; blade cracking; condition based maintenance.

Introduction: This detection of blade modes in the torsional domain requires that (1) the blade modes to be detected couple with torsion; and (2) that the signal resulting from excitation of the blades by turbulence and other random processes is measurable. It is probably safe to say that, theoretically, condition (1) is always met. However, the difficulties associate with harvesting the potentially very small signals associated with blade vibration in the torsional domain could render detection not feasible. Thus, transduction and data acquisition must be optimized for dynamic range and signal to noise ratio.

The three shaft wheel configurations employed also serve as partitions for this review of the effort:

- a four threaded pin system to develop and optimize the transduction method and to demonstrate basic feasibility
- an eight threaded pin system to investigate coupling of the pin bending frequencies with shaft torsion
- an eight rectangular bladed system to attempt to detect changes in the natural frequency of a single blade by the introduction of a “crack.”

The transducer included an infrared fiber optic probe, an analog incremental demodulator, and an A/D converter. Signal processing was performed using a combination of tools. A finite element model of the rotors was developed to support interpretation of the test results and to facilitate characterization of the coupling phenomena.

Transducer Development: Several transduction methods were considered. However, preliminary estimates of the torsional vibration relative to the shaft motion indicated that the signal might be over 100 dB below other motion associated with the shaft rotation. This excluded several transduction methods. In addition, for use in the electric power generation industry, the transducer would have to be non-intrusive. It was decided that the use of a black and white tape and a fiber-optic probe would be the most suitable [1]. Figure 1 shows a schematic of the transducer system.

The demodulation technique used was analog. The incremental demodulator included a high-pass pre-filter, a TTL (transistor-transistor logic) circuit and a demodulator circuit. This device had a maximum signal to noise ratio of over 100 dB, and was thus the most suitable for the task.

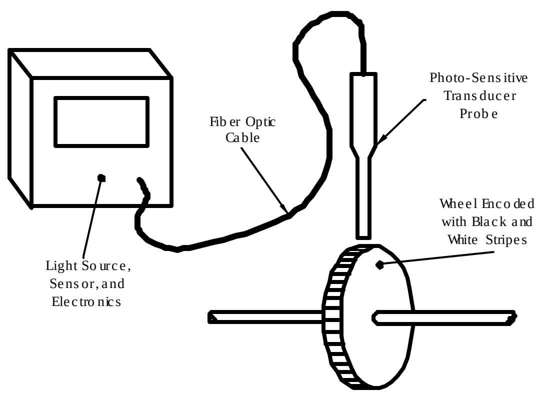

Figure 1: Sensor system for torsional vibration

Figure 2 shows a simulation of a square wave associated with a constant speed rotor (top), and a square wave modulated by torsional vibration associated with blade vibration (bottom). The modulation is, of course, greatly exaggerated to demonstrate the effect. Actual modulation would not be visible in the time domain. Figure 3 shows a schematic of the transduction method.

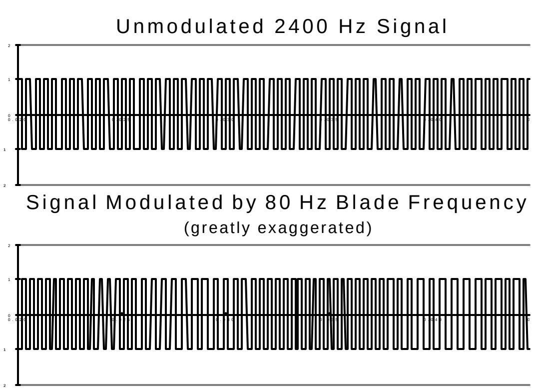

Figure 2: Simulated time wave form from a constant speed rotor (top) and modulated by blade vibration (bottom, greatly exaggerated)

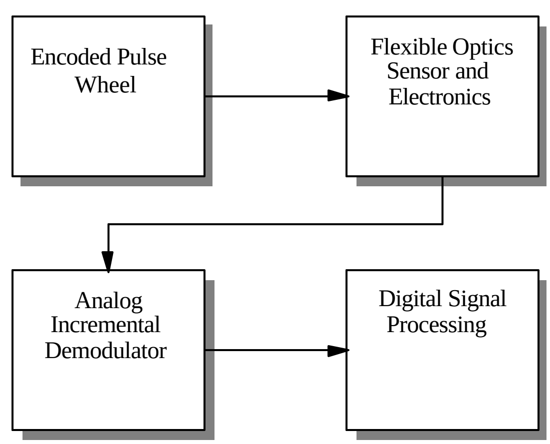

Figure 3: Schematic of transduction system

Although digital demodulation is feasible, it would require very large blocks of data at relatively high data acquisition speeds. Coupled with the dynamic range requirements, these requirements pushed digital demodulation out of the practical range. The analog incremental demodulator included a high-pass pre-filter, a TTL circuit and a demodulator circuit. This device had a maximum signal to noise ratio of over 100 dB, and was thus the most suitable for the task.

Initial Feasibility Demonstration: Figure 4 shows the apparatus used in demonstrating the feasibility of detection blade vibration. The rotor included a wheel with four threaded rods to simulate blades. A thin aluminum wheel (the pulse wheel), three inches in diameter, was used for the black and white striped tape. The entire rotor kit was enclosed in a plastic safety case, with the fiber optic probe penetrating the case through the top to the pulse wheel.

Figure 5 shows five overlaid demodulated spectra for the rotor without the threaded pins installed. Each spectrum was obtained with the rotor at a slightly different speed. Note the plethora of spectral peaks. These peaks were associated with the motor speed control, and caused some confusion at first. However, it was clear that they could not be associated with structural vibration due to the extremely low damping and the non-linear shifting with rotor speed. Note that, buried below these peaks, there is a peak at about 222 Hz. It was determined by experimentation and finite element dynamic analysis that this was the fundamental torsional mode of the rotor system.

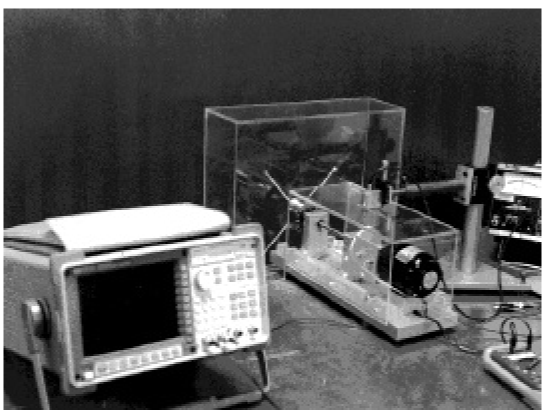

Figure 4: Feasibility demonstration apparatus

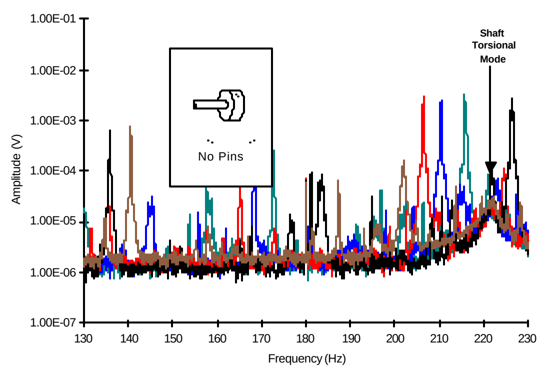

Figure 5: Initial demodulated spectrum for rotor with no threaded pins

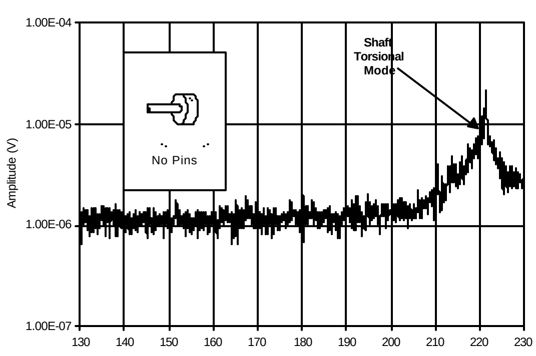

Figure 6: Data of Figure 5 after masking

Figure 6 shows the same data after post-processing by "masking", in which the minimum of the several spectra obtained at slightly different speeds was retained at each frequency point. The shaft first torsional natural frequency is now clearly distinguishable. This masking process

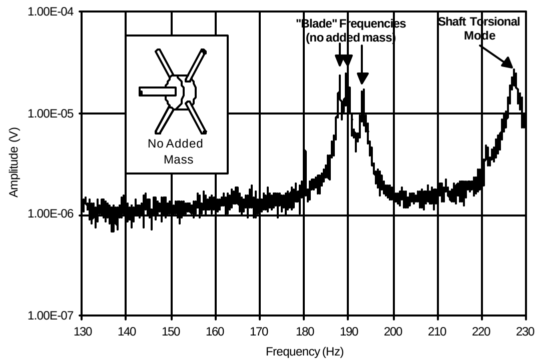

Figure 7: Demodulated spectrum of rotor with four threaded pins

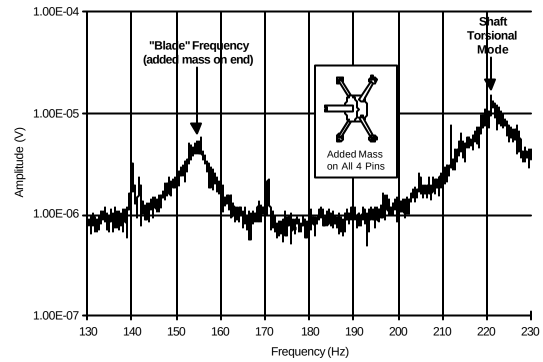

Figure 8: Demodulated spectrum of rotor with mass added to the pins

proved to be an effective way of distinguishing between structural frequencies and speed control frequencies. The remainder of the demodulated spectra are in this masked form.

The four threaded pins were next added to the rotor. Figure 7 shows the resulting demodulated spectrum. Note that there is about a 6 Hz spread between the three peaks associated with the four threaded rods. The wheel was removed and clamped in a large vise, the pins were plucked, and the natural frequency of each pin recorded. Although the pins were the same length, there was a 5 Hz spread noted in the pins, indicating differences in the boundary conditions of the pins. From this point forward, the pins were tuned in the vise before installation in the rotor kit.

Mass (in the form of nuts) was then added to the pins. Figure 8 shows the resulting demodulated spectrum, manifesting the expected reduction in the pin frequency. It was noted that the relationship between the natural frequencies in the vise and in the demodulated spectrum for the case with added mass was different than for the case without nuts.

Finally, nuts were included on two of the pins and the nuts were removed from the other two. Figure 9 shows the resulting demodulated spectrum, where the split between the two pairs of pins is evident.

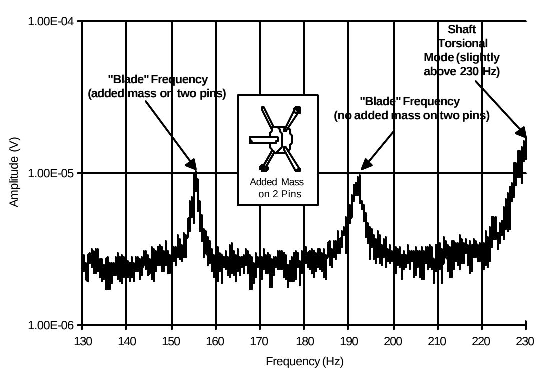

Figure 9: Demodulated spectrum of rotor with mass on two pins and no mass on two pins

On the basis of these experiments, it is clear that the feasibility of detecting blade natural frequencies by measuring torsional vibration is established.

Coupling Investigation: The measurement of blade vibration in the torsional domain is totally dependent on the coupling of the blade bending vibration with torsion. In addition, the relationship between the blade frequencies in the vise and in the demodulated spectrum was not easily modeled. To clarify the coupling issue, a second rotor was used. This rotor, shown in Figure 10, consisted of a rotational mass with eight threaded pins to facilitate adjustment of the masses and, therefor, the pin frequencies.

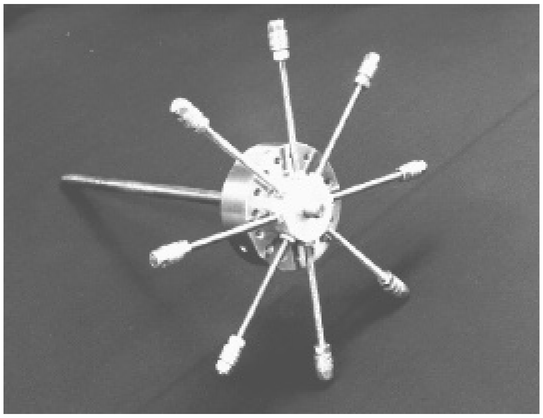

Figure 10: Photo of eight threaded pin rotor assembly

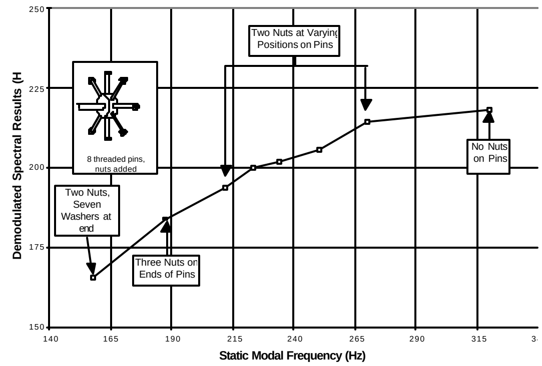

Figure 11: Test results: first operating coupled mode vs. static pin mode of the eight threaded pin configuration with various nut locations

The wheel was mounted in the vise , and the pins were tuned to the same frequency (± 0.5 Hz). The wheel was then mounted in the rotor, and the demodulated spectrum was generated. Mass was added to the pins, and the process was repeated. Then, the mass was shifted along the length of the pins to give as large an overall shift in pin frequency as possible. The peak in the demodulated spectrum for each case is plotted against the frequency obtained in the vise (termed the static modal frequency) in Figure 11.

To further facilitate the understanding of the coupling phenomenon, a beam finite element model (FEM) of the rotor was developed (see Figure 12). The boundary conditions of the pins at the wheel were adjusted until the frequency extracted from the FEM (with wheel fixed) matched the static modal frequency. Modal extraction was then performed on the entire model, which included the stiffening effects of rotation on the pins. The results are shown in Figure 13 and tabulated in Table 1. Table 2 shows the comparison of the FEM and the test frequencies.

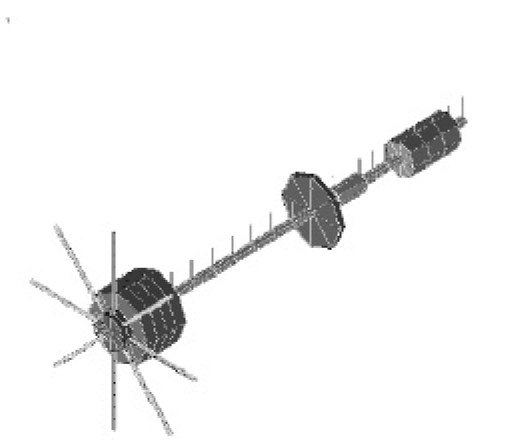

Figure 12: FEM model Geometry of the rotor

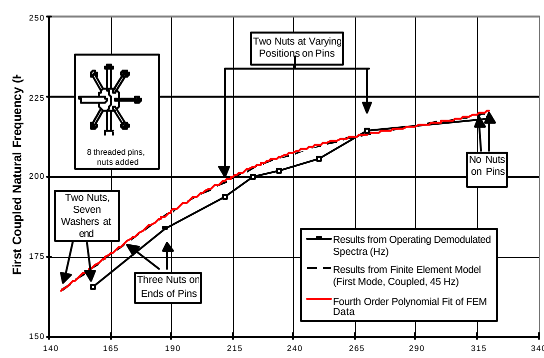

Figure 13: Comparison of FEM and test result for the first coupled mode

Table 1: Comparison of static modal frequency and first coupled mode - test and FEM

	Test Results			FEM Results		
Location of Double-Nut Configuration	Static Modal Results (Hz)	Results from Operating Demodulated Spectra (Hz)	% Difference	Static Modal Results (Hz)	First mode, coupled, 45 Hz speed	% Difference
2 nuts, 7 washers on end	157.5	165.47	5.1%	145.2	164.5	13.3%
3 nuts on end	187.5	183.9	-1.9%	169.9	178.4	5.0%
2 nuts at varying positions	211.5	193.6	-8.5%	195.3	191.8	-1.8%
"	223.5	199.8	-10.6%	227.4	204.1	-10.3%
"	234.0	202.0	-13.7%	261.9	212.0	-19.1%
"	250.0	205.8	-17.7%	292.6	216.3	-26.1%
"	270.0	214.5	-20.6%	311.3	218.5	-29.8%
No nuts	320.0	218.3	-31.8%	317.5	220.2	-30.7%

Table 2: Comparison of static modal frequency and first coupled mode - test and FEM

Location of Double-Nut Configuration	Static Modal Results (Hz)	Test Results from Operating Demodulated Spectra (Hz)	FEM ResultsFirst mode, coupled, 45 Hz speed *	% Difference
2 nuts, 7 washers on end	157.5	165.47	171.5	3.6%
3 nuts on end	187.5	183.9	187.9	2.2%
2 nuts at varying positions	211.5	193.6	198.6	2.6%
"	223.5	199.8	202.9	1.6%
"	234.0	202.0	206.1	2.0%
"	250.0	205.8	209.9	2.0%
"	270.0	214.5	213.3	-0.6%
No nuts	320.0	218.3	220.6	1.1%

* Based on fourth-order polynomial fit of FEM data

Table 3 shows a comparison of the static frequencies (in the vise) and the demodulated frequencies (in the rotor). Note that the demodulated frequencies are below the static frequencies for most cases, but for the case of the highest mass, the demodulated frequency is above the static frequency.

Table 3: Results of change in mass (size and location) - FEM and test

	Test Results			FEM Results		
Location of Double-Nut Configuration	Static Modal Results (Hz)	Results from Operating Demodulated Spectra (Hz)	% Difference	Static Modal Results (Hz)	First mode, coupled, 45 Hz speed	% Difference
2 nuts, 7 washers on end	157.5	165.47	5.1%	145.2	164.5	13.3%
3 nuts on end	187.5	183.9	-1.9%	169.9	178.4	5.0%
2 nuts at varying positions	211.5	193.6	-8.5%	195.3	191.8	-1.8%
"	223.5	199.8	-10.6%	227.4	204.1	-10.3%
"	234.0	202.0	-13.7%	261.9	212.0	-19.1%
"	250.0	205.8	-17.7%	292.6	216.3	-26.1%
"	270.0	214.5	-20.6%	311.3	218.5	-29.8%
No nuts	320.0	218.3	-31.8%	317.5	220.2	-30.7%

Detecting a Single Blade "Crack": The final phase of the feasibility study was to determine the feasibility of detecting a shift in the frequency of a single blade. The rotor was modified to have 8 rectangular pins to represent blades. The blades were tuned in the vise so that each had a frequency of 168.5 Hz. Figure 14 shows a photo of the wheel. The wheel was then installed in the rotor kit and operated as before. The resulting demodulated spectrum is shown in Figure 15.

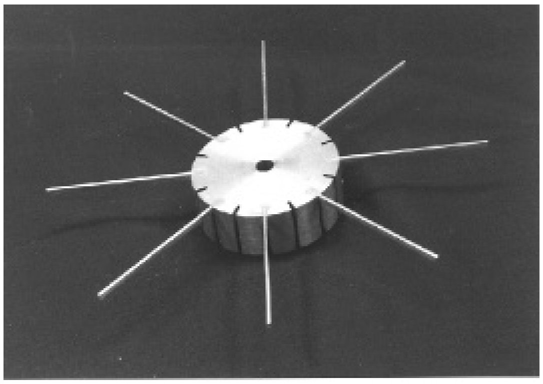

Figure 14: Rotor wheel for detecting single blade "crack"

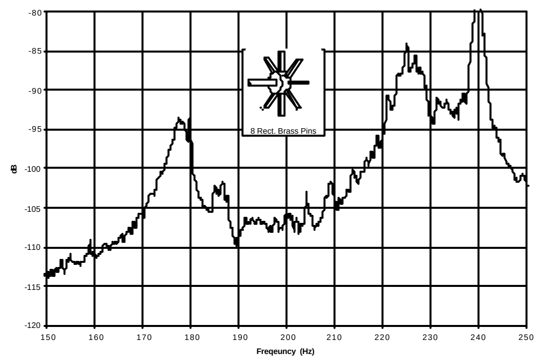

Figure 15: Results of Brass Pin Test with All Pins Tuned to 168.5 Hz (Masked)

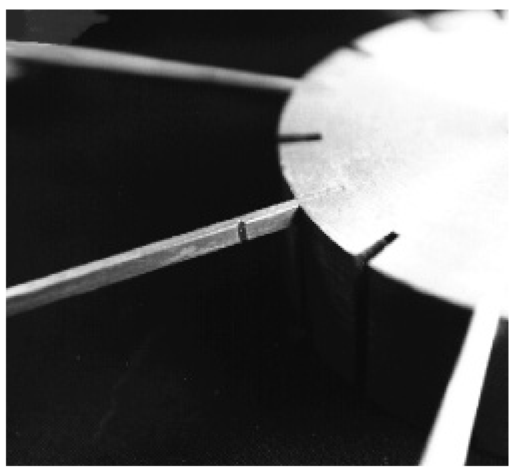

Figure 16: Close-up Showing V-Notch in Rectangular "Blade" corresponding to single blade frequency of 162 Hz (deepest cut)

A cut was then made in a single pin to represent the "crack" resulting in an initial frequency shift from 168.5 Hz to 167 Hz. The rotor was then operated, and demodulated spectra generated. The process was repeated for deeper cuts and, accordingly, large frequency shifts for the single blade. Figure 16 shows the deepest cut, corresponding to a single blade frequency of 162 Hz, a 6.5 Hz shift out of the group frequency of 168.5 Hz.

Figures 17 through 20 show the resulting demodulated spectra. With some imagination, the group peak may be seen to broaden with only a 1% difference in frequency (167 Hz vs. 168.5 Hz) between the single blade and the group (Figure 17). With about a 2% difference, the peak associated with the "cracked" blade becomes evident (Figure 18). As the cut deepens, the peak becomes more distinct (Figure 19), Until, at and 20). Upon disassembly, it was noted that several of the pins, which were pressed into the wheel and epoxied, had become loose. This may shed some light on the increased “noise” in the spectrum of Figure 20.

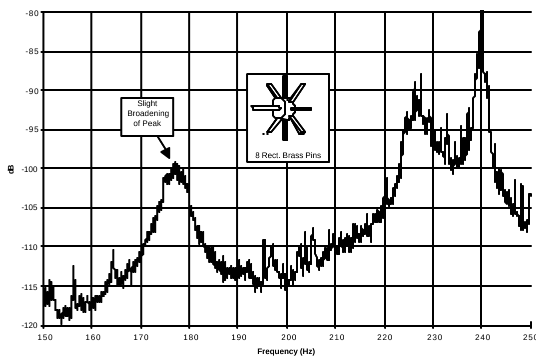

Figure 17: Results with One Notched Pin, Static Frequency of 167 Hz

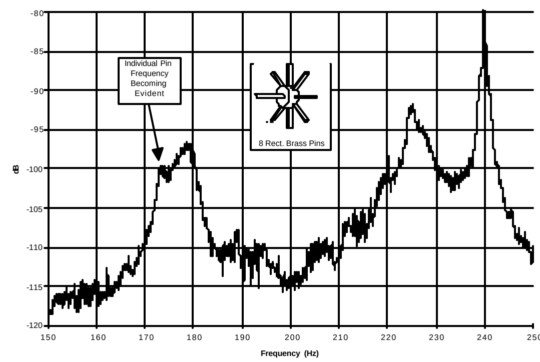

Figure 18: Results with One Notched Pin, Static Frequency of 165.5 Hz

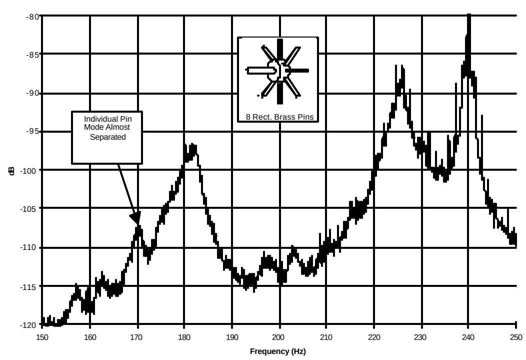

Figure 19: Results with One Notched Pin, Static Frequency of 164.5 Hz

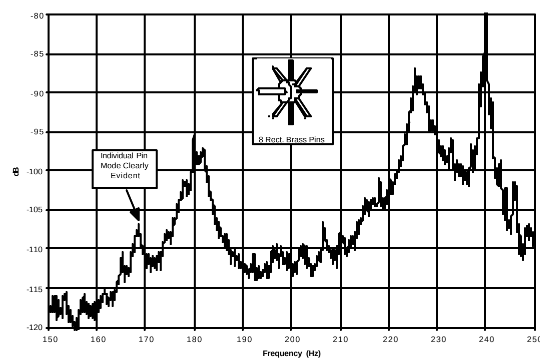

Figure 20: Results with One Notched Pin, Static Frequency of 162 Hz

Table 4 summarizes the results of the “crack” detection feasibility study. Note that the frequency shift in the demodulated spectrum was higher than the shift in the static frequency determined by plucking the blade in the vise. In general, this difference was larger by a factor of about two.

Table 4: Summary of Brass Pin Tests with One Notched Blade

	Static Modal Test Frequencies (Hz)			Operating Demodulated Spectra (Hz)		
Condition	Group	V-Notched	Difference	Group	V-Notched	Difference
Tuned	168.5	NA	0	181.7	181.7	0.0
V-Notch 1	168.5	167	-1.5	177.7	175.3	-2.4
V-Notch 2	168.5	165.5	-3	179.3	173.6	-5.7
V-Notch 3	168.5	164.5	-4	181.7	170.2	-11.5
V-Notch 4	168.5	162	-6.5	181.9	168.6	-13.4

Summary: Testing and analysis was performed on a laboratory rotor to establish the feasibility of detecting shifts in blade natural frequencies by measuring torsional vibration in a non-intrusive way. The results were encouraging:

- Blade bending natural frequencies can be detected by measuring torsional vibration of the rotating assembly.
- An uderstanding of the coupling of the blade modes with rotor torsion is necessary to identify blade frequencies in a complex machine. Finite element was demonstrated to be an adequate tool for clarifying coupling issues.
- The modeling of the rotor highlighted the major advantage of using torsional natural frequency detection over using translational vibration: boundary conditions, which are so difficult to characterize in rotor translational modes, are near non-existent in many rotor systems. This means that characterization of the torsional rotordyanamics of a system is much more straightforward, and therefore likely to better facilitate diagnostics.
- It was demonstrated that small shifts in natural frequency (<2%) of one blade in eight is detectable using the methods described.

The results of the testing and analysis clearly demonstrated the feasibility of using torsional vibration to detect the change in natural frequency of a blade due to a change in stiffness such as those associated with a blade crack. The methodology described is now being tested for use in industrial turbines and fans.

Acknowledgement:

The research and development work summarized here was sponsored by the Southern Company. Particular thanks goes to Mr. Robert Moore, Vice President of Gulf Power, whose vision for condition based maintenance provided the stimulus for this work.

Reference:

1. Vance, John M., *Rotordynamics of Turbomachinery*, John Wiley & Sons, New York, 1988, pp. 377ff.