SHAFT CRACK MONITORING VIA TORSIONAL VIBRATION ANALYSIS; PART 2 – FIELD APPLICATIONS

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ABSTRACT

Torsional vibration analysis has shown the potential to provide health diagnostics for rotating equipment. Previously, controlled laboratory experiments have demonstrated the ability to detect and track small changes in the health of a shaft with a progressively grown fatigue crack. The work presented in this paper discusses the practical issues and challenges of moving this technology to field applications. Four general categories are discussed; 1) torsional finite element modeling; 2) field instrumentation; 3) torsional signature analysis, and; 4) lessons learned. The use of a torsional FEM to identify suitable encoder placement in relation to the crack location and the expected changes in the modal frequencies due to crack propagation are discussed. Field implementation issues examined include encoder options, transducers and mounting considerations. Signature analysis topics are order content removal and separation of lateral shaft vibration from the torsion. The lessons learned sections describes the interpretation of test results from several applications including variation in torsional modal frequencies due to normal operation and the difficulty of judging the root cause of an observed frequency shift. The paper concludes with an assessment of the use of torsional vibration for rotating equipment health monitoring.

INTRODUCTION

Part 1 of this work [1] explored the ability of torsional vibration signature analysis to trend the growth of a fatigue crack in a shaft rotating in a laboratory test stand. The laboratory tests showed there is an identifiable natural frequency change that can be tracked in relation to the fatigue crack depth. The first torsional natural frequency decreased nonlinearly with respect to the diametric crack depth. Changes in natural frequency in the range of 0.1-0.2 Hz were identifiable by a visual inspection of the torsional spectrum. The laboratory results demonstrated the ability to measure and trend torsional natural frequencies as a potential diagnostic to monitor the formation and growth of cracks in shafting systems.

There are potential advantages of using torsional natural frequency over lateral natural frequency trending for detecting and monitoring cracks in direct-drive machine shafts. A natural frequency shift in a lateral mode may be caused by anything which changes the boundary conditions between the rotating and stationary components, such as seal rubs, changes in bearing film stiffness from small temperature changes, thermal growth and misalignment. Therefore, if a shift in a lateral modal frequency is observed, it is difficult to pinpoint the root cause. These boundary condition influences are inconsequential in the torsional domain. Hence, the root cause of an observed shift in a torsional natural frequency must involve changes to the rotating element, such as a crack or perhaps a coupling degradation.

Based upon this potential, torsional monitoring has been implemented in several rotating equipment installations. The transition of any technology from the laboratory to the field always presents many pragmatic challenges. The work presented in this paper will discuss issues that have been encountered and addressed for the use of torsional vibration monitoring for shaft crack monitoring. The items can be generalized into four categories: 1) the need and use of a torsional finite element modeling to define modal sensitivities to cracks and guide placement of the instrumentation; 2) field instrumentation challenges associated with limited accessibility, shaft encoder mounting, transducer mounting and mount dynamics; 3) specialized data processing schemes are discussed that minimize the degrading effects of shaft encoder geometry variation, running speed changes and transverse shaft vibration on the torsional signature, and; 4) lessons learned from torsional monitoring in the field have shown that the root cause of an observed torsional natural frequency is not always directly apparent and that the inherent equipment operation may not excite the desired crack sensitive modes.

These topics will be discussed from an experiential point of view. Data from various torsional vibration monitoring installations will be used to illustrate the issues and the outcome of any corrective action that may have been implemented. The paper concludes with an assessment on the use of torsional vibration as a diagnostic to monitor shaft cracking.

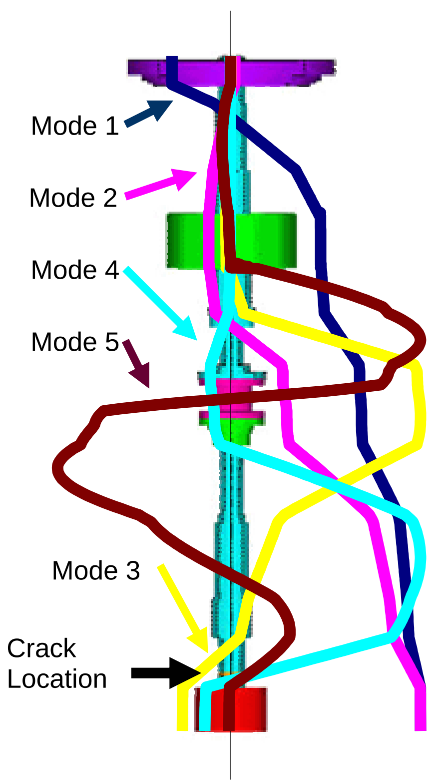

Figure 1. Torsional mode shapes from a Finite Element Model for vertical rotor.

TORSIONAL FINITE ELEMENT MODELING

Rotor torsional finite element modeling is simplified compared to the lateral domain due to the difficulty of characterizing boundary conditions. The boundary conditions are nearly non-existent in the torsional domain for many rotor systems. This means that the characterization of the torsional rotordynamics is more straightforward and more likely to facilitate better diagnostics.

A torsional model for the targeted equipment is useful in several aspects. The growth of a crack at a specific location in a shaft will not affect all modal natural frequencies similarly. Certain torsional modes can show great sensitivity to small crack growth while others can be almost completely immune. Obviously, it is important to identify the potentially sensitive modes, in order to select feasible measurement locations, well away from any nodal points. The relations of the shaft crack growth to modal frequency changes are highly dependent on the equipment and can not be generalized. Therefore, the torsional dynamic model is critical to correlating a natural frequency shift to crack growth.

Figure 1 shows a Finite Element Model for a vertical rotor and the first five torsional mode shapes. Evaluation of the torsional mode shapes is useful to establishing a suitable shaft crack implementation package. Rotor designs that have a propensity for crack formation usually develop them at specific locations. Historical inspection records can usually

Mode Number	Natural frequency (Hz)	Center (Hz)	Bottom (Hz)
1	162.2	152.5	160.7
2	494.5	492.2	488.3
3	570.3	570.3	554.6
4	845.9	787.4	842.9
5	920.9	912.2	918.8
6	1463.4	1463.3	1463.4
7	1872.0	1872.0	1872.0
8	2160.1	2157.4	1816.0
9	2233.6	2233.6	2233.6
10	2412.1	2412.1	2412.1
11	2958.4		2958.4
12	3357.9		3357.9
13	3735.5		3132.9

Table 1. Torsional natural frequencies for a 60% diametric cut placed at the center line shaft and near an end.

be used to identify the critical locations. For example, assume the equipment has had a history of cracks developing near the location indicated on Figure 1. The closer the crack is to a modal nodal point the greater the change in natural frequency as the crack grows. Mode 4 would be an excellent monitoring choice, since a node is located in close proximity to the crack location. Mode 3 would also show good sensitivity. Therefore, it would be necessary to measure the frequencies associated with these two modes. The shaft encoding device must be placed at a position along the shaft that the response of these two target modes is high. The optimal location would be approximately one third of the distance from the bottom in the region where the two mode shapes cross each other. Next, accessibility to the rotating shaft must be considered. It is conceivable that access to the rotating shaft at this ideal location is not possible and an alternative position must be selected. When evaluating candidate locations the guiding principle is to keep the encoder away from nodal points of the modes identified to be sensitive to a crack location.

The finite element model can also be used to evaluate the sensitivity of the torsional modes to a localized decrease in stiffness (e.g., a crack). Due to the difficulty of modeling a closed fatigue crack an open cut has been found to be effective in evaluating general changes to the torsional dynamics. Table 1 shows the expected torsional natural frequency shifts for a 60% diametric cut placed in a rotor at two locations (Note, results are not from the same rotor as depicted in Figure 1). For a crack in the center, mode 1 shows a drop of 10 Hz while mode 4 has a decrease of almost 60 Hz. The other modes show little to no frequency change to the crack. Modes 8 and 13 would be the best modal frequencies to monitor if the targeted crack were near the bottom of the shaft.

A rotor dynamic finite element model is an invaluable design tool while developing an effective condition based torsional monitoring package. The model development effort costs are offset by the increased probability of success. In many cases, field results differ from the Finite Element analysis and thus model verification is essential.

FIELD INSTRUMENTATION

Most of the field applications to date have focused on rotating equipment already in service. All the torsional vibration instrumentation must be retrofit to existing equipment designs and therefore, work within their respective installation constraints. The torsional vibration instrumentation consists of two primary components; 1) an incremental shaft encoder, and; 2) a sensor to detect the passage of the encoder segments. The installation of both components in a retrofit environment can affect the ability to acquire high quality torsional vibration sufficient for crack monitoring.

An incremental encoder (i.e., timing gear) must be attached to the shaft. For example, a toothed ring type encoder requires a bolted flange for its physical attachment to the shaft. Therefore, the choice of mounting locations is predetermined. Even for more flexible encoding schemes such as adhesive “zebra tape”, which does not require a flange, simple accessibility to open sections of the rotating shaft can be limited. Because of these location constraints the available torsional sensing location may not be ideal and can potentially affect the ability to sense the response of the desired modes.

Four examples of retrofit installations to measure torsional vibration are shown in Figure 2. Figure 2A shows a timing gear that used a Hall Effect transducer (not visible) to sense the passage of tooth segments. A timing gear was available on the high speed turbo-machinery, already being used for speed measurement. Hence, the

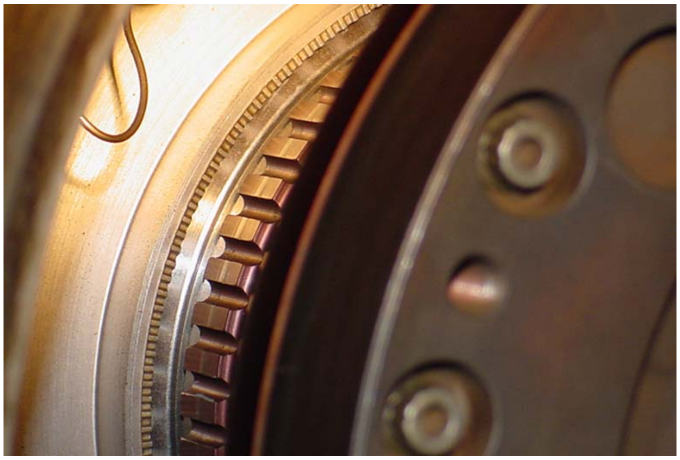

Figure 2A. 60 tooth gear encoder installed on high speed turbo machinery.

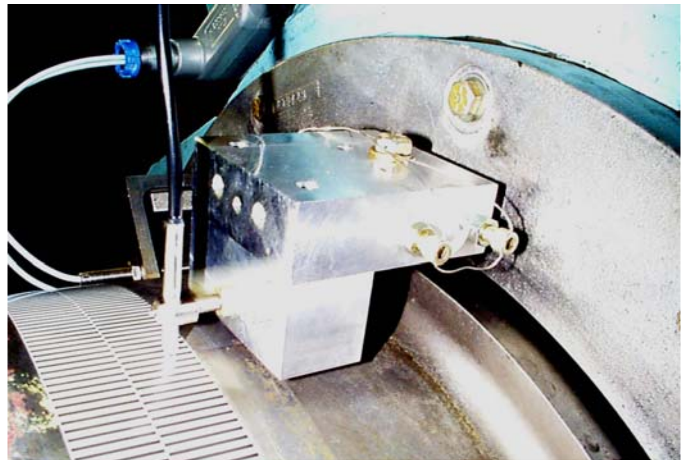

Figure 2B. Zebra tape encoder and fiber optic transducer on a large diameter shaft (22 inch).

Figure 2C. Zebra tape encoder and fiber optic transducer.

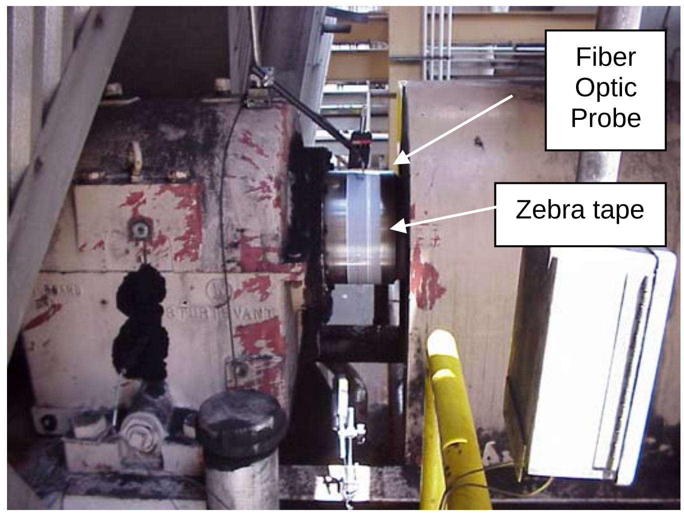

Figure 2D. Zebra tape encoder and fiber optic transducer in limited rotating shaft access installation.

adaptation to measure torsional vibration was straightforward. Both installations in Figures 2B and 2C used adhesive backed tape with optical passage sensors. The relatively open accessibility to the rotating shaft in these examples affords flexibility in placing the encoding tape at optimal measurement locations to enable the measurement of a sensitive mode. By contrast, the installation shown in Figure 2D has very limited access to the rotating shaft. Hence, in this retrofit application the encoding device may be located at a less than ideal location and hinder the measurement of important crack sensitive modes. However, the hardware design presents no other options and the choice is to either use whatever is available or to not try at all.

Sensors must be used to capture the passage of the encoder segments to determine the torsional vibration. Furthermore, the transducers usually must be placed in close proximity (0.010 – 0.050 inches) to the encoding device to accurately sense the segment passages. The availability of sensor mounting locations is highly equipment dependent and may require long reaches to be used as seen in Figures 2C and 2D. These mounting demands can create some adverse conditions that may potentially affect the measured torsional vibration signature.

In the computation of the torsional vibration, the encoder passage times are assumed to be only due to rotation of the shaft. Any lateral movement between the sensor relative to the encoder can affect the passage times. In particular sensor vibration tangential to the shaft can significantly affect the passage times. If a sensor mount is too compliant, its resonances can be excited by the inherent machinery vibration and produce superfluous artifacts in the torsional spectra. An example of this behavior is illustrated by the spectra in Figure 3. The torsional

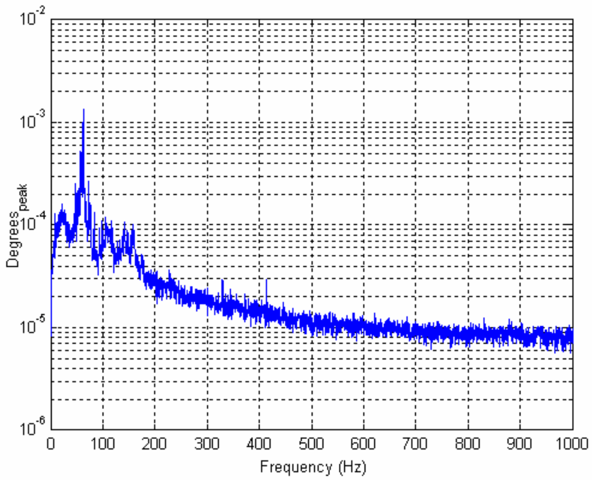

Figure 3A. Torsional vibration spectrum measured with the passage sensor with a rods and magnetically mounted similar to Figure 2C and 2D.

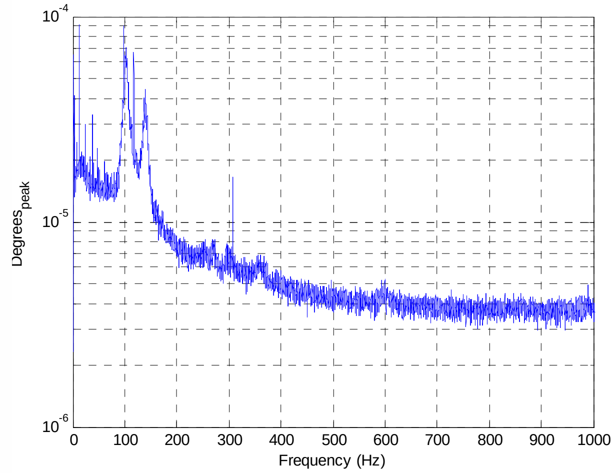

Figure 3B. Torsional vibration spectrum measured with the passage sensor bolted to the the machine base with a stiff mount similar to that shown in Figure 2B.

spectrum in Figure 3A was captured when the sensing transducer was mounted in a similar fashion to that shown in Figure 2C using a cantilevered rod anchored with a magnetic mount to the equipment. The high amplitude response between 50 – 75 Hz is related to the mounting system resonance responses. Figure 3B shows the corresponding torsional spectra when a stiff bolted mount similar to that depicted in Figure 2B was used. Note that the amplitude scales in Figures 3A and 3B are significantly different. The artifacts of the mount resonances are removed making the actual torsional responses readily apparent. The effort required to specially fabricate stiff mounts and install as rigidly as possible has proven beneficial.

Since torsional vibration is more difficult to measure than lateral vibration, extra effort is required to develop and install an effective system.

TORSIONAL SIGNATURE ANALYSIS

The torsional data from the fatigue crack experiments in Part 1 [1] of this work were obtained in a controlled laboratory setting. The added difficulties of acquiring data in the field environment can potentially induce corrupting artifacts in the spectra. Use other than precision optical encoders can produce high amplitude order content synchronized with the shaft rotation. Changes in running speed can cause a smearing in the frequency domain. Lateral vibration of the shaft can induce superfluous torsional spectral content. This undesired content can further hamper the identification and tracking of the natural frequencies. Care needs to be exercised and corrective processing applied, when possible, to minimize these artifacts to produce highest quality spectra possible.

Both analog and digital demodulation methods can be used to produce the time variant torsional vibration from the shaft encoder signal. Both methods are susceptible to containing high level shaft synchronized (order) content induced from geometric irregularities in the encoder. For example, the digital processing method to determine the torsional vibration via the Time Interval Method [3] requires an accurate reference signal. The reference signal is assumed to; 1) have equal encoder segments and 2) operate with a constant shaft rotational speed, sans any torsional vibration. Unless a high precision device such as an incremental optical encoder is used, assumption 1 is violated. Other than synchronous motors, the running speed does not remain constant, so assumption 2 is also questionable. These conditions create difficulties in establishing a suitable set of reference times to calculate the torsional vibration for a non uniform encoder (i.e., zebra tape). These deviations from the ideal conditions introduce computational artifacts in the form of extraneous order content as seen in Figure 4A. Similar order content can be produced with analog demodulation. [4]. Corrective algorithms have been developed that are able to compensate for encoder imprecision and running speed variation for both the analog and digital demodulation methods [5]. Figure 4B shows the torsional spectrum from a high speed turbine (Figure 2A) when encoder characterization and constant time resampling algorithms are applied to compensate for the timing gear geometric

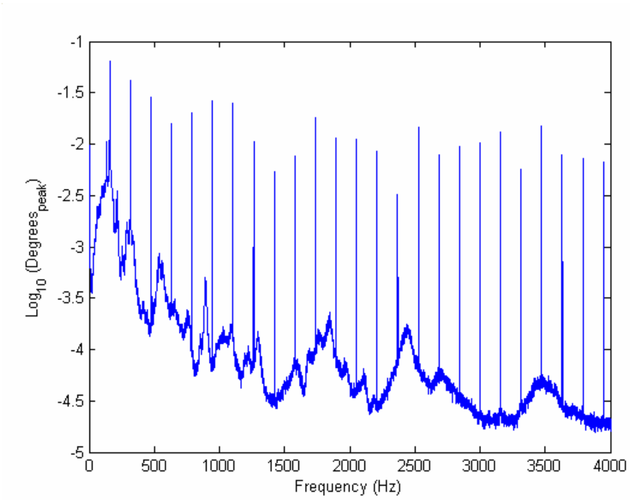

Figure 4A. Torsional vibration of high speed turbo-machine with a gear encoder (Figure 2B). .

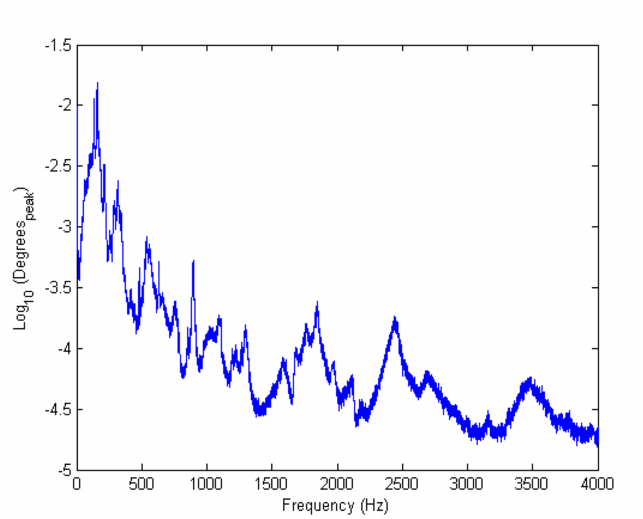

Figure 4B. Torsional vibration spectrum of high speed turbo-machine with a gear encoder (Figure 2B) with encoder correction and constant time resampling algorithms applied.

irregularities and variation in running speed. The processing considerably enhances the spectral quality aiding the extraction of the desired features.

Peak to peak lateral shaft movement up to 10 mils is fairly common and on occasion, depending on shaft size, as high as 25 mils occurs during normal operation of rotating plant equipment. As previously mentioned, any lateral movement of the sensor relative to the shaft can affect the recorded passage times and consequently the torsional vibration signature. Through the use of multiple passage sensors (minimum of 3) located around the encoder circumference the lateral motion effects can be separated from the torsion [6,7]. The implementation requires synchronized resampling of the three channels of data and a solution of three simultaneous algebraic equations at each discretized time interval which increases the computational demands considerably. A three probe configuration has been implemented and proven effective in removing the artifacts of the lateral shaft movement from the torsional signal. The enhanced spectra allowed the desired torsional spectral to be more easily identified and tracked.

The measurement of torsional vibration is inherently more difficult than lateral vibration. Compensation methods can be implemented that minimize the unwanted artifacts in the torsional vibration signal. The compensation causes the data processing to be more involved with a corresponding increase in the computational load. Overall, the compensation algorithms have proven to be effective in improving quality of the torsional signal.

PRACTICAL LESSONS LEARNED

Natural frequency shifts may be observed in the torsional spectra but, the cause of the shift may not necessarily be a cracked shaft. Inherent operation of the rotating equipment can cause change in torsional frequencies. For example, as the temperature changes the components expand and contract accordingly. The thermal induced geometry changes can cause variations in the torsional dynamics. Therefore, it becomes difficult to delineate whether an observed change in a modal torsional frequency is due to normal operation or the growth of a shaft crack.

An example of the difficulty of making machinery health judgments based on natural frequency shifts is found in [2]. A torsional monitoring system was installed on a hydro turbine generator. The torsional natural frequencies were trended over a two year period and a drop was ultimately observed. A decision was made to take the unit off line and inspect for shaft cracks. The inspection did not find a structural flaw in the shaft and the frequency decrease was attributed to debris collection against the impeller and in the seals.

To sense the torsional natural frequencies requires that they be excited in some manner. Because the data is typically acquired under nominal operating conditions, the torsional excitation is a result of the inherent machinery operation. The spectral content of the torsional excitation in turbo-machinery tends to be broadband and not as strong as experienced in internal combustion engines. Therefore, there is a risk that the inherent torsional excitation may be of insufficient amplitude so that the target modes of interest, as identified by the finite element analysis, may not be excited. In this case, regardless of the sensitivity of the modal frequencies to crack formation and growth, the method will be ineffective.

SUMMARY

Unexpected driveline shaft failures continue too occur. To date, the development of an effective on-line shaft crack detection monitoring system has been elusive. Current shaft crack detection monitoring system technology is primarily based on lateral vibration analysis. However, these methods often provide little warning in the plant environment. Laboratory experiments have shown that torsional vibration analysis has the potential to provide the desired structural health diagnostics for rotating shafts. The torsional domain tends to be more immune to changes in boundary conditions (i.e., seal rubs, bearing stiffness) and hence can provide more robust diagnostic features. These potential advantages have motivated further development in an effort to bring the concept to practical fruition.

This work has described some of the pragmatic issues related to transferring the torsional based monitoring method from the laboratory to the field. Torsional vibration is inherently more difficult to measure than lateral vibration. Hence the field instrumentation set up and data acquisition is more demanding. Furthermore, specialized data processing is necessary to compensate for a number of corrupting artifacts that may be induced. Ultimately, when properly implemented a high quality torsional signal can be acquired.

Shaft cracking diagnostic efforts to date have focused on trending shifts in torsional natural frequencies. As with all natural frequency based diagnostics systems, the root cause of an observed shift is not always easily identified. Furthermore, when a crack is identified as the cause of a frequency shift the relationship to crack depth is not easy to correlate. This correlation is required to calculate the Remaining Useful Life (RUL) of the shaft. Accurate knowledge of the RUL allows the repairs to be scheduled at least disruptive times. Research to this end is needed.

Despite the various identified difficulties in using torsional vibration analysis for shaft cracking diagnostics the upside potential remains good. Natural frequency trending is a rudimentary machinery diagnostic feature. The torsional vibration signal appears ripe with other features that may provide an even more sensitive indicator for early shaft crack detection and monitoring purposes. Fusion of lateral vibration diagnostics with torsional vibration features could provide enhanced more robust crack indicators. Work to this end is continuing.

ACKNOWLEDGEMENTS

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