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USE OF ACOUSTIC INTENSITY SPATIAL DISTRIBUTION & FREQUENCY SPECTRA FOR MACHINERY CONDITION MONITORING

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Abstract:
Evaluation of operationally-induced vibration is useful for machinery health diagnosis and prognosis. However, obtaining vibration at many locations required for adequate diagnostic information is time-consuming because of the need to move probes to locations that may be difficult to access, especially on large or hot machinery. In addition, measurement of structural vibration is dependent upon consistency of probe attachment integrity on typically rough uneven surfaces. The authors use a microphone pair in which the microphones are phase-matched to allow independent determination of sound pressure and acoustic particle velocity, such that sound intensity measurements can be made on all line-of-sight locations on a rotating machine surface, including exposed rotating components. This approach has been demonstrated useful even when the probes are located at a significant distance from the machine. Novel calibration and signal processing techniques allow the sound intensity spectra gathered at various locations to be used to determine local excitation forces as well as responses (displacement, velocity, and acceleration) within the machine. The forces and responses can be used in deterministic tribological and fatigue models to indicate impending problems, and to predict remaining useful life. This paper explains the authors' procedure, and provides seeded fault case histories to illustrate successful application in the diagnosis and prognosis of a electric motor/ gear driven fan rig in a reverberant environment.

INTRODUCTION

Vibration interpretation has been used as a key tool for machinery diagnostics for many decades now [10], [3], [5], [8]. With the advent of signal analysis by Fast Fourier Transform (FFT), the evaluation of operationally-induced vibrations became particularly useful [4], [7], since complicated signals in the time domain could be interpreted in terms of their frequency content versus the expected strength of the vibration signal at various discrete frequencies and while creating a given "operating deflection shape (ODS) pattern. For example, a given level of imbalance at one end of the machine would be expected to produce an increased, quantifiable vibration level at a frequency of exactly one times running speed, and to exhibit much stronger motion at the bearing housings, versus vibrational motion of the main casing of the machine. Furthermore, when combined with a deterministic model of force levels implied by the detected response levels, they can provide sufficient quantitative information that it becomes practical to perform prognosis through the use of tribological and fatigue/ fracture mechanics analytical methods.

However, obtaining vibration at the many locations required for an adequate application of such a detailed diagnostic/ prognostic approach is time-consuming because of the need to move probes to often difficult-to-reach locations (especially on large or very hot machinery), and because of the dependency of data quality upon the consistency of probe attachment integrity (particularly on typically rough uneven surfaces). Therefore, the authors have developed an acoustically-based approach which uses a high-fidelity sound intensity probe. The approach has been demonstrated successfully when the sensing microphone is located at significant distances from the machine. The research discussed in this paper applied the proposed approach to an actual machinery train, and demonstrated the capability of the approach to perform reliable diagnostics and prognostics for a seeded fault which represented a typical compressor or fan failure mode.

RESEARCH GOAL

The goal of the presented research is to eventually develop a system to provide a non-intrusive method for machinery operators and maintenance personnel to be aware of the mechanical health of their machinery, so that they can make appropriate Operation & Maintenance decisions. Such information could include identification of the components in need of repair or replacement, conveyed in easily understood form.

TEST RIG APPLICATION

Figure 1 shows the test rig constructed by the authors, using company Internal Research & Development funding. The test rig consists on a rotary compressor/ fan shown on the left, driven by a variable speed electric induction motor shown on the right. Between is a solid drive shaft supported by a pair of pillow-block grease-lubricated journal bearings, with a flexible spiral flex-joint all-metal coupling at each end. The radial fan impeller is rigidly mounted to the cantilever-supported pinion shaft, such that the fan impeller is cantilevered within its single volute and housing. The fan suction is open to lab air, and the discharge vents to the open lab air.

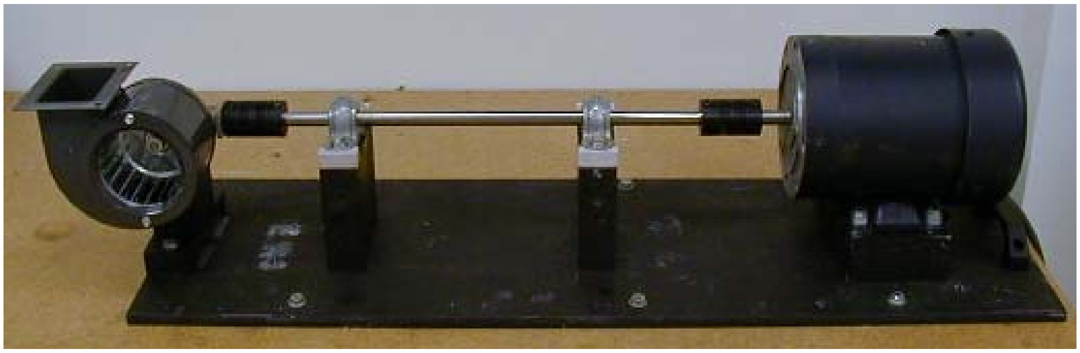

Figure 1: Radial fan/ motor test-rig developed for subsequent research

Figure 2 shows vibration test data superimposed on a CAD (computer-aided design) model created with mechanical modeling and design software. The particular situation exhibits the excitation of a motor/ frame structural mode at 79.5 Hz. This is an example of a vibration pattern that was matched between vibration probes (in a “birth certificate” calibration test) versus acoustic intensity data in a later step of the process. A baseline ODS (Operating Deflection Shape) was obtained using high-frequency-response accelerometer vibration probes at many locations. The combined vibration and acoustic tests are performed only once, at installation, and after this only acoustic measurements are used to determine the combined acoustic and equivalent vibration ODS pattern, and its implication concerning potential machinery anomalies..

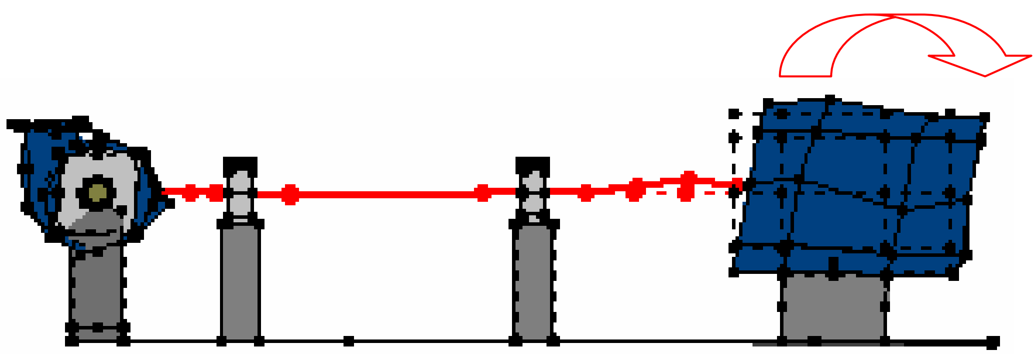

Figure 2: ODS measurements show motor support excited at 79.5 Hz

A baseline acoustic intensity map of the compressor train was recorded at the same time as the vibration data to establish a one-to-one relationship between the vibration and acoustic amplitudes across the full spectrum of frequencies of interest. Acoustic intensity is a vector quantity [1], [2], [6], [9], [11], with the vector direction being identical with the line of centers of two front-to-back microphone heads in the author's custom-designed sound intensity probe. The microphones were selected for their low expense [12] and minimal cross-section, and they were attached into an assembly using three stiff but thin rods, to minimize the interference of the probe structure with acoustic intensity signals. The intensity probes were calibrated using sinusoidal excitation across the frequency range of interest, through use of a laboratory-quality amplifier and loud speaker.

One important difference between such an acoustic measurement and a vibration measurement with an accelerometer is the transmission loss as the energy travels through air [2], and this was compensated for with appropriate analytical functions dependent upon the temperature of the air, and the distance of the probe assembly to the measured surface. The acoustic intensity map provided data for each discrete point of focus, over the entire frequency span of interest, creating an acoustic spectral ODS that was able to be translated into a vibration spectral ODS, aided by the "birth certificate" vibration acceleration vs. acoustic intensity calibration, which in turn was a function of location and frequency.

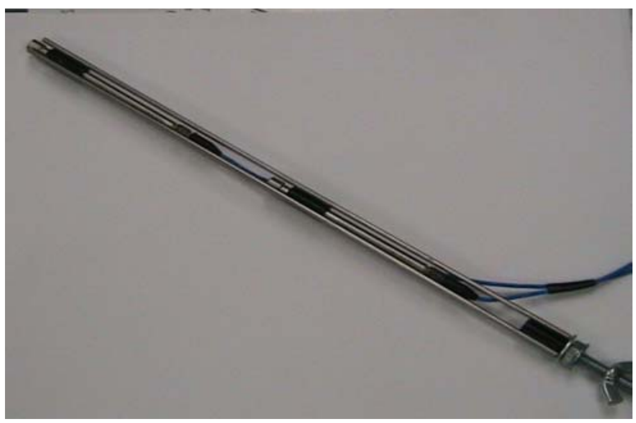

Figure 3: A first generation acoustic intensity probe assembled by MSI

Figure 3 shows the acoustic intensity probe assembled by the authors. The intensity probe was a product of two phase-matched and magnitude calibrated microphones. The intensity was then calculated by the following equation [1], [2]: $Intensity = -\frac{\mathrm{Im}(G_{12})}{\omega\rho_0\Delta r}$. This equation only requires the imaginary part of the cross-spectrum G_{12} from an FFT measurement set, the frequency ω at a given point in the spectrum, the density of air ρ_0, and the distance between the two microphone heads Δr. The intensity probe was moved from location to location and measurements were started and stopped manually. The result of the acoustic intensity signature is shown in Figure 4, where this example plot is at the fan blade pass frequency.

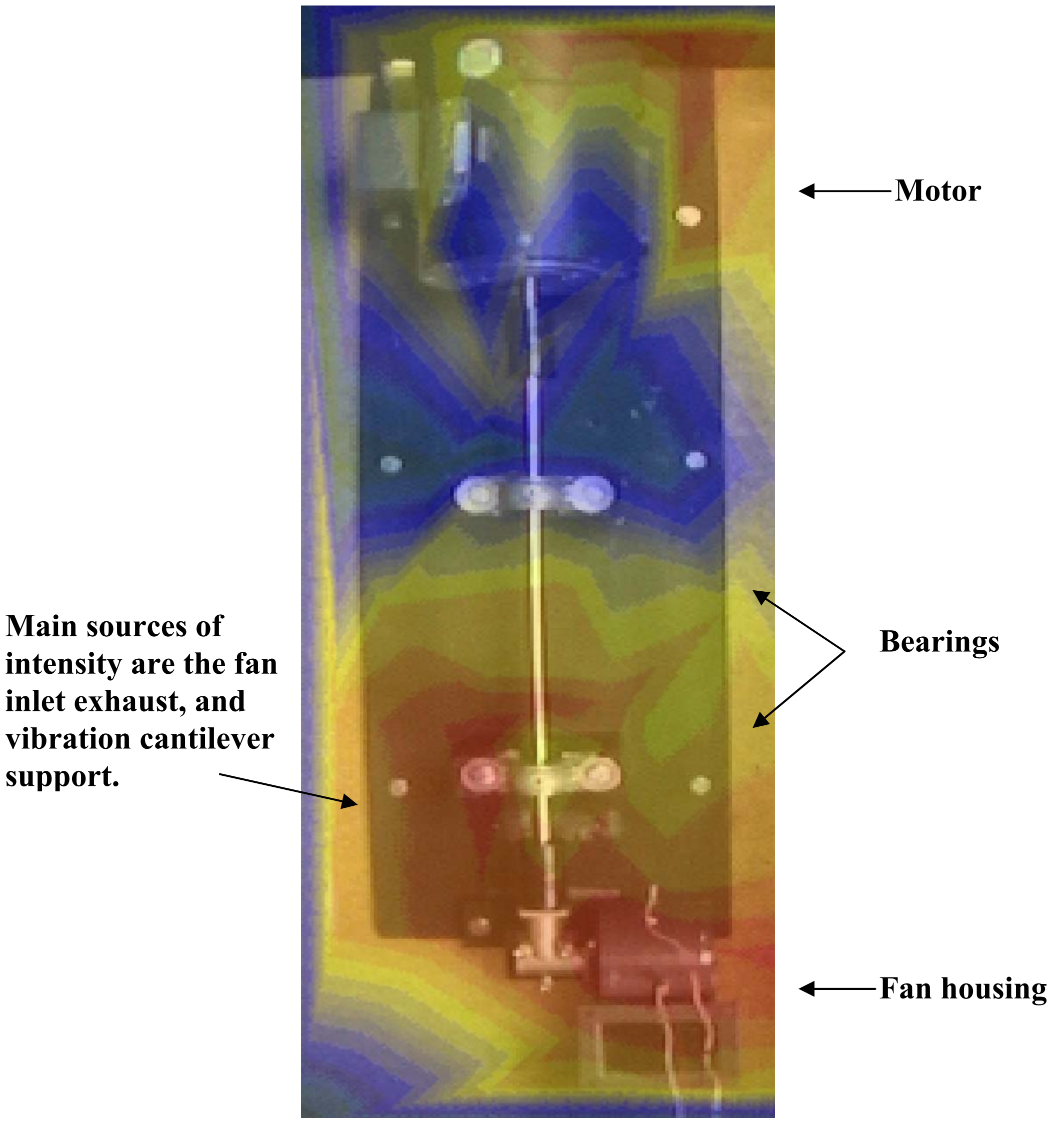

Figure 4: Acoustic intensity baseline map overlaid on a photograph of the rig

The vibration/ acoustic correlation technique was demonstrated successfully on the laboratory test-rig. Once the key operational modes were calibrated by test and the author's analytical evaluation procedures, the proposed method was shown to be capable of predicting the quantitative excitation source of any vibration anomalies. In a fully developed automated system, such estimates would be made automatically in real-time, easing the burden on the optimization algorithm, and would always provide an accurate representation of the current condition of the fan/ motor system, in near real-time.

An example of the practical application of this method would be if curve-fitting identifies an increased vibration magnitude at 1X running speed. The parameter with the highest probability for contributing to this is an imbalance, and the approximate axial location for the imbalance would be identified by increases in certain locations of the acoustic map. The amount of imbalance and its precise location would be determined by the force amount and location required to produce an acoustic ODS that matched a calibrated vibration ODS.

In order to demonstrate the feasibility of this step of the process, the authors introduced seeded faults into the test rig and proved the ability of the proposed method to detect a spectral change and response location, coupled with time-frequency statistics for the detection of impulsive anomalies. In separately funded research, to be reported in future papers upon approval of the funding agency, seeded faults were applied and successfully detected:

Imbalance – Introduced various levels of imbalance to the fan wheel, and then correlated values to predict bearing loads and life.
Bearings – Identified healthy, deteriorated, and severely damaged bearings
Gearbox – Slightly damaged a gear tooth and uniquely diagnosed this situation
Vane Pass – Introduced foreign debris into the fan volute, and identified the source of the resulting acoustic intensity.

The imbalance example was demonstrated by adding an imbalance to the fan wheel on the test rig. The fan housing acoustics was calibrated, and then used to determine the 1x frequency of the response (characteristic of imbalance), the location of peak response as the fan area, and the ODS motion indicative of a quantitative level of fan wheel imbalance. Various levels of imbalance were installed, and the match was excellent between the acoustic determination of vibration displacement at a selected housing location (indicative of imbalance level through analytical relationships that had been validated by test) and the actual imbalance level.

When the cause and its associated force and (depending on the component) running clearance deflections are determined, fatigue and tribological models can then be exercised to predict remaining useful life. For example, in the imbalance case, the forces on the bearings supporting the compressor wheel were calculated, using as input the acoustically determined vibration response at key locations, such that these forces in turn served as an input to a tribological model which predicted the remaining life of the impeller bearings.

The blade pass seeded fault was completed by introducing a slight interference between the blades and volute tongue to simulate foreign object debris. A map across the machine train surface area was created in the same fashion as the baseline acoustic intensity measurement, so that the changes could be identified. The result of the contour map recorded with the seeded fault in place is shown in Figure 5, overlaid on a picture of the test-rig. This map is shown at the vane pass frequency and identifies the discharge of the compressor as the main source for acoustic intensity at this frequency. The motor also shows a relatively high vane pass frequency magnitude, which suggests an increase in the motor torque pulsations corresponding to the blade interference frequency and associated torque pulsations.

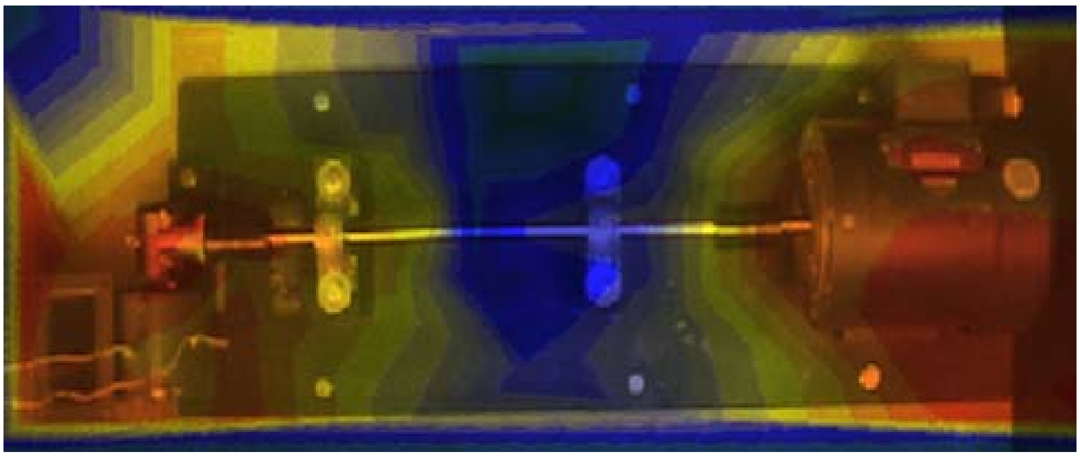

Figure 5: Acoustic intensity contour map created with vane pass seeded fault. Note that the intensity contour colors on this plot represent considerably higher (by about 25 dB) levels than the corresponding colors in the baseline plot of Figure 2.

CONCLUSIONS

The research produced a proof-of-principle system that demonstrated the feasibility of the proposed acoustic-based diagnostic/ prognostic approach. Acoustic intensity and sound pressure level measurements were able to provide the sole information necessary for diagnosis and prognosis for the various major components of a fan/ motor rig assembly.

ACKNOWLEDGEMENTS

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