White Paper
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High Frequency Vibration Analysis

The emphasis in this paper is the capture and analysis of stress waves introduced into rotating machinery by events such as impacting, fatiguing, and friction. The stress wave events introduced into rotating machinery are mostly flexural waves (also referred to as bending or S waves). These waves propagate away from the initiation site introducing a short term ripple on the surface of the machine. An absolute motion sensor (such as an accelerometer or ultrasonic sensor) with sufficient bandwidth and sensitivity will detect the short term wave activity providing important information for assistance in fault detection and severity assessment. Brief discussions on the generation of stress waves and their propagation as waves away from the initiation site are presented. The theoretical predictions are briefly verified by some simple experimental results. From the theoretical and experimental results, the expected frequency range over which the dominant energy appears for the various stress wave causing events are identified. Case studies employing both ultrasonic sensor and an accelerometer (using the PeakVue methodology) are presented.

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Process Management

Introduction

Vibration Analysis is the analysis of the motion introduced by the disturbance of a body (machine) about its equilibrium position. The disturbance can be a) the motion of the body on a macroscopic scale or b) the disturbance of a small section of material within the body on a microscopic scale. The macroscopic motion can be introduced by forces such as unbalance, misalignment, et al. The microscopic motion can be introduced by disturbance of material within the body due to events such as impacting, fatiguing, and friction. Microscopic vibrations are generally short term (order of fractional milliseconds) transient events occurring at random or at periodic rates. The frequencies excited by the microscopic events are high frequency (typically in the kHz to several kHz range) relative to those excited by macroscopic vibration (kHz or less). In this paper, the focus is direct to the analysis of the microscopic high frequency vibrations.

When impacting, fatiguing or friction events occur; energy is deposited in the machine in a very small region introducing motion away from equilibrium in a short interval of time. This energy is dissipated by stress waves which travel away from the initiation site at the speed of sound within the metal. As the stress waves propagate away from the initiation site, they introduce a small short term ripple on the surface of the machine (metal). An absolute motion sensor with sufficient bandwidth and sensitivity (such as an accelerometer) will capture these events.

In the following sections of this paper, the discussion will focus on the introduction of microscopic vibration (stress wave activity) in an elastic medium from various sources, the propagation of the stress waves form the initiation site to the detection site, the detection of the stress waves, and analysis methodologies. Several case studies are presented to demonstrate various points emphasized in the formal discussions.

Stress Wave Activity in Rotating Machinery

Stress Waves can be generated in any elastic medium. The primary interest focused on in this study is rotating machinery. Stress waves accompany the sudden displacement of small amounts of material in a very short time period.[1,2] In rotating machinery, this occurs when impacting, fatigue cracking, scuffing, abrasive wear, etc. occurs. The most frequent occurrences of stress wave generation in rotating machinery are observed in fault initiation and progression in both rolling element bearings and in gear teeth. The generation of stress waves from various events commonly experienced in rotating machinery is discussed in the next subsection. This will be followed by a discussion on the propagation of the stress waves from the initiation site to the detection site. The detection of the stress waves are discussed in section 2.4.

Generation of Stress Waves

A quantitative framework for the generation and detection of stress waves can be developed using the Hertz theory for metal-to-metal impacting[3] and wave theory[4] for propagation of stress waves in metal. The dominant stress waves generated by a metallic sphere impacting on a relatively large plate are bending waves[*] of half-period equal to the duration of the impact. Olma[5] carried out a parametric study using the Hertz theory of impacting to establish an expected frequency band which may be excited due to impacting from metallic balls of varying size and speed on large metallic plates. Graphical representations of his results are presented in Figure 1. The impacting object was assumed to be spherical; therefore, the diameter for a spherical steel ball is presented in Figure 1 in lieu of the mass. The key parameter establishing the frequency range where energy would be expected to be excited from an impacting object is the contact time. Small diameter balls will have less contact time than will larger balls (see Figure 1). Shorter duration contact times (smaller diameter balls) will lead to a higher frequency band being excited than will longer contact times (larger diameter balls).

[*]Bending waves are also referred to as flexural waves and S waves.

To experimentally explore the basic concept of frequency bandwidth being inversely correlated with ball size, a high frequency accelerometer was placed on a reasonably large metal plate in the proximity where impacting was carried out. The spectrum was captured with an 80 kHz bandwidth using the “Peak Hold” averaging mode. The results for balls of diameter of 0.5 in. and 1.5 in are presented in Figure 2. The captured signal, presented in Figure 2, is very similar to what would be expected from the Hertz wave impact theory (illustrated in Figure 1).

The contact time for the 1.5″D ball is estimated to be 100 microseconds (☐s) as per Figure1B. For 100 ☐s contact time, the bandwidth to 10 dB attenuation would be around 10 kHz (Figure 1A). This bandwidth is consistent with that observed for the 1.5″ D ball shown in Figure 2.[†] The contact time for the 0.5″ D ball would be in the 30–40 μs range (see Figure 1A). For this contact time, the bandwidth to the 10 dB attenuating level is estimated to be in the 30 kHz range(Figure 1B). This is very consistent with the impacting spectral data for the 0.5″ D ball presented in Figure 2.

In addition to impacting as a source for stress wave activity in rotating machinery, friction (metal rubbing metal) and fatigue cracking must also be considered. The frequency band excited in the stress wave packets will still generally follow the Hertz theory if the equivalent contact time can be approximated (the time where material movement is present on a microscopic scale).

For the case of friction, two examples are presented in the form of spectral data. The first is for a reasonably small diameter bearing having rollers of about 3/8″D running at about 3600 RPM where lubricant was sparse. The second is from a split ring pedestal bearing with rollers in the 1.0 to 1.5 inch diameter range turning at 700 RPM, which definitely had friction occurring as confirmed by large black spherical particles found in the oil lubricating the bearing.

The spectral data from the smaller bearing are presented in Figure 3. Here, most energy is in the

[†]The increased activity around 15 kHz in the spectral response for the 1.5″ Diam. ball in Figure 2 is from structural resonance. This response was there and stationary for several different impacting balls of various diameters

20 kHz to 30 kHz range primarily due to friction (lower limit has increased from 10 kHz to 20 kHz relative to ½″D ball impacting data presented in Figure 2). The reduction in the spectral data in the 30 kHz range is most probably due to the sensor mounting, and not due to absence of energy. Since the lower frequency has shifted upwards relative to the ½″ D ball by a factor of 2, the higher frequency most probably also increased by a factor of 2 (60 kHz versus observed 30 kHz). This would place the equivalent contact time during stress wave generation in the 20–25 μsec range.

The spectral and time waveform data captured for the larger bearing turning around 700 RPM are presented in Figure 4. The time waveform is for a period of 40 msec where one large event (Pk-Pk g level of 216 g's) was captured. The energy from this "friction" event is concentrated in the 5 to 18 kHz range, which is about a factor of 3 times greater than that observed from the impacting of the1.5″D ball (see Figure 2).

The spectral data presented in Figures 2, 3 and 4 from impacting and friction events were displayed on a linear scale and hence showed a reasonably small frequency band where energy is present. Referring to Figure 1, theoretically the energy is broadband, but with increasing attenuation with increasing frequency with a picket fence effect (nodal points). The spectral data presented in Figure 4 with a linear amplitude scale is displayed in Figure 5 on a log scale. Here the broadband nature with increasing attenuation with increasing frequency and the picket fence effect are apparent. From the comparisons of the energy distribution versus frequency within stress wave generating events, it is concluded they are similar with the exception that the equivalent contact time for friction is less than that caused by impacting by a factor of 2-4 times. This translates to the frequency band where the dominant energies reside will be higher (factor of 2–4) for friction generated events than those from impacting events.

The third major event for stress wave generation in rotating machinery is fatigue cracking. No specific data is readily available to quantify the equivalent contact time; however, past experiences have suggested the duration (and hence, contact duration) of these events is less than that from friction-generated events. They would likely still be in the several μsec ranges (not the nanosecond range); hence the dominant energy would be in same range as friction activity.

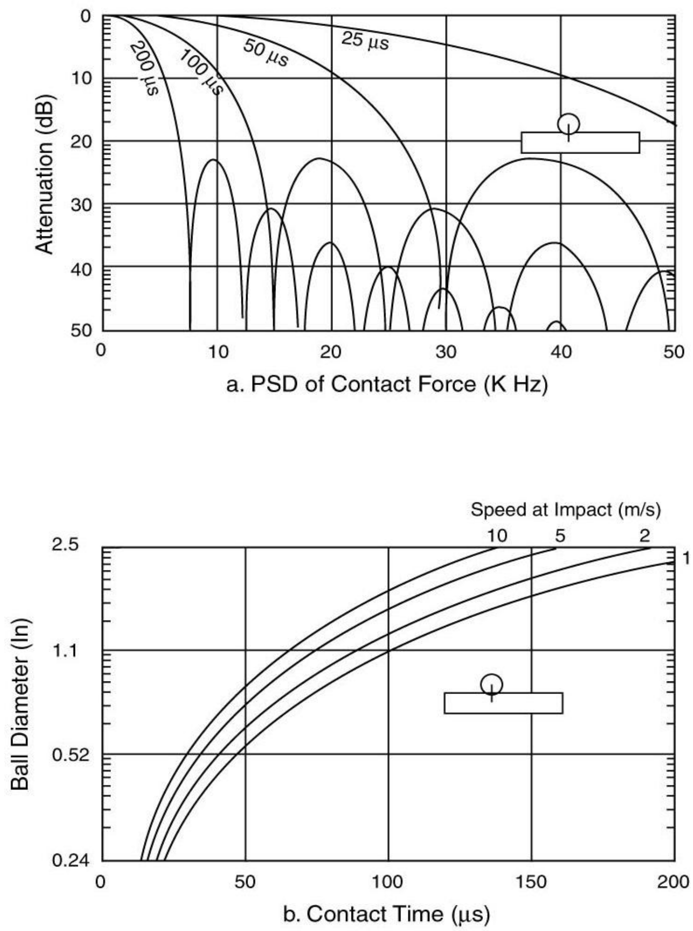


Figure 1. Hertz Theory Prediction for Metal Balls for Varying Size and Speed Impacting on Large Metal Plates.[5]

Figure 2. Spectral Data Captured for a 0.5″ Dia. and a 1.5″ Dia. Metal Ball Impacting on a Large Metal Plate.

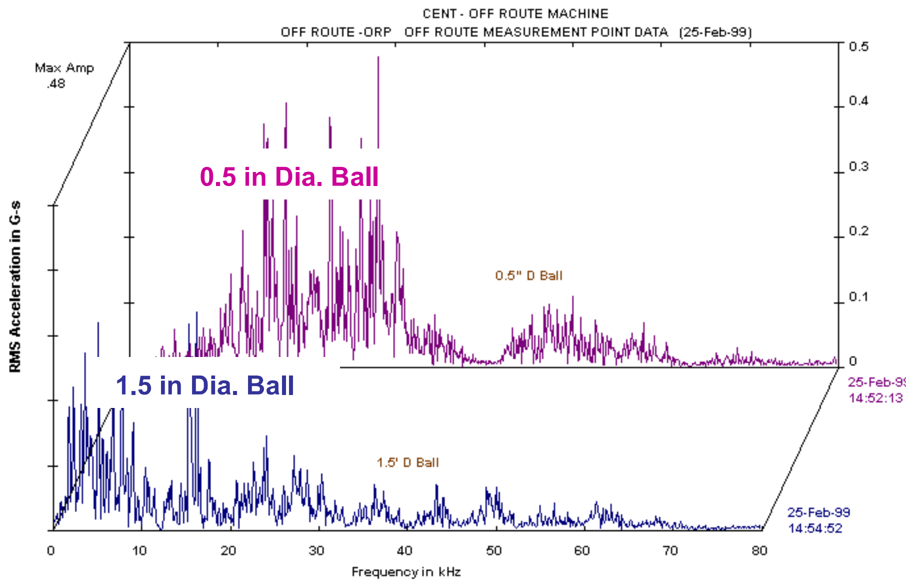


Figure 3. Spectral Data Captured on Small Machine with 3/8″ Diameter Ball Running Dry at about 3600 RPM (Note Difference in Impact & Friction Response Regions).

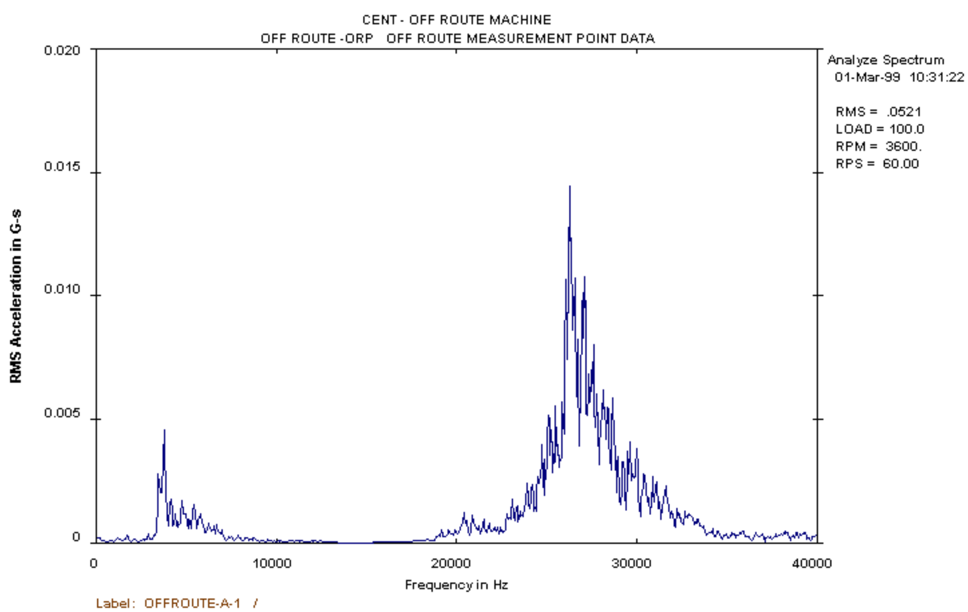


Figure 4. Spectral and Time Waveform data from a Split Ring Pedestal Bearing with Rollers in the 1.0″ to 1.5″ Diameter Range at 700 RPM with Friction.

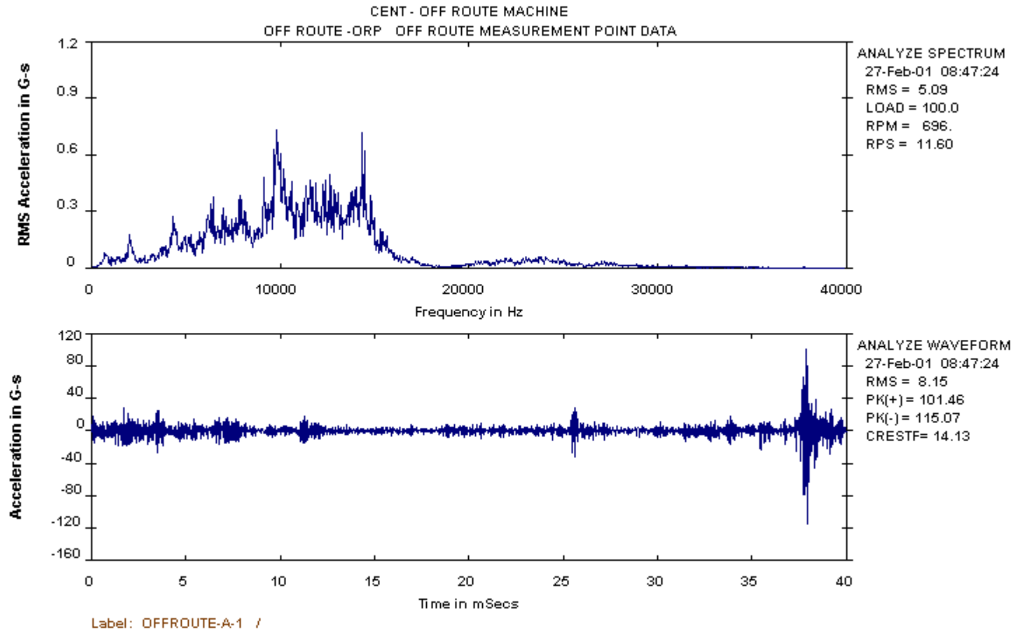
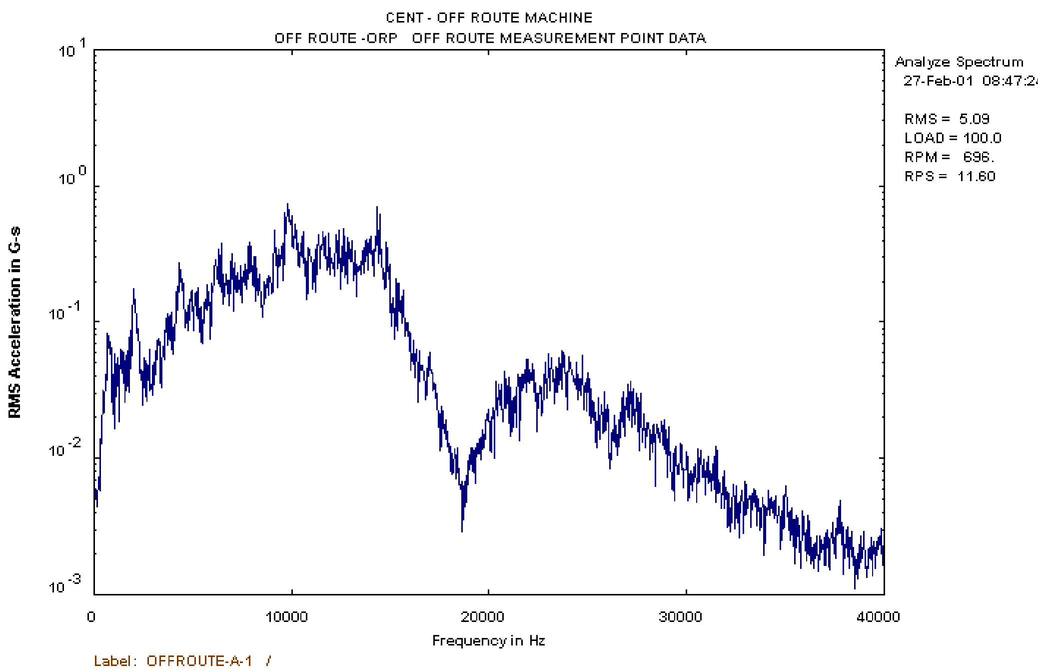


Figure 5. Spectral Data for Figure 4 with Log Amplitude Scale.



Propagation of Stress Waves to Detection Site

Bending waves are the most dominant stress waves generated in events experienced on rotating machinery.[3,5,6] The velocity at which bending waves propagate away from the initiation site is proportional to the square root of frequency. Thus there will be dispersion within the stress wave packet when viewed (detected) at locations removed from the initiation site. The amplitude of the event will decrease and the duration of the event will increase as the observation point is further removed from the initiation site.

In addition to the diffusion of the stress wave packet, the attenuation of the stress waves are also frequency dependent[4,7] with the higher frequency components attenuating more rapidly than the lower frequency components as the observation (measurement) point is removed from the point of origin. This is especially true when stress wave packets must cross physical interfaces such as from shaft (or inner race of a bearing) to the observation point on the outer structure of the machine.

The significant attenuation of concern is that associated with crossing physical boundaries. To demonstrate this effect and to gain an approximate confirmation of the attenuation and its frequency dependence, a multi-frequency ultrasonic sensor[**] was a) mounted on the inboard shaft of a stationary 15 HP motor; and b) then on the motor housing. The response was measured while separately impacting the shaft with a 1.5″ D ball and with a 0.5″ D ball. The response using the three narrow bands of 4 kHz, 30 kHz, and 150 kHz in dB were recorded and are presented in Table 1.

[**]The CSI Model A700080 sensor modified to include 150 kHz band was used.

TABLE I
RESPONSE OF NARROW FREQUENCY BAND SENSOR TO IMPACTING BALLS ON SHAFT (dB)

Ball Size	4 kHz		30 kHz		150 kHz	
	Shaft	Housing	Shaft	Housing	Shaft	Housing
0.5″	77	42	46	32	32	0
1.5″	80	57	43	28	5	0

Looking at the response at 30 kHz, an attenuation factor of about 14 dB (factor of 5) in the signal in passing from the shaft to the motor housing (see Table I). This is consistent with observations made in the past and with approximate theoretical predictions. The signal at 150 kHz did not transmit from the motor shaft to the motor housing for the 0.5″D ball impacting. Thus the attenuation exceeds 30 dB (factor of 30) at this frequency. For the 4 kHz range, the apparent attenuation from the shaft of approximately 25 to 35 dB (77-42 dB for the 0.5″D; and 80-57 dB for the 1.5″D) to the outer housing exceeds expectations based on both theory and on experience (one would only expect 6-10 dB attenuation). The most probable explanation for this result is that the stress waves maybe exciting the natural frequency (mode) of this shaft, thereby introducing excessive motion in the proximity of the sensor mount on the shaft.

Based on the above discussion on the stress wave propagation from the initiation site to the probable observation point on the outer surface, the higher frequency components will be attenuated more rapidly than will the lower. Even though frequencies exceeding ~100 kHz are generated for impacting of small balls (<0.5″ D), friction events, and fatigue cracking, the probability of that energy transmitting to a location where a sensor could be mounted is small at such a high frequency.

Detection of Stress Waves

Due to the dispersion and attenuation of the stress wave packet, it is desirable to locate the sensor as near to the initiation site as possible. This generally will be near or on the bearing housing (preferably in the load zone). Stress waves will propagate in all directions. Hence the selection of axial, vertical, or radial is less of an issue than is mounting the sensor in or near the load zone (relative to normal vibration monitoring). The sensor can be an accelerometer or an ultrasonic sensor. The advantage of an accelerometer (a broadband sensor) is that the primary energy from stress waves initiated from impacting, fatiguing, and friction can be captured. The ultrasonic sensor typically is a narrow bandwidth sensor. The frequency at which the ultrasonic sensor is tuned may be greater than the dominant frequency range for a particular event type, but the sensor could still capture the event due to the "picket fence" effect (see Figure 1).

Analysis of Stress Waves

The sensors employed for the detection of stress waves generally are an accelerometer or an ultrasonic sensor. The accelerometer is responsive to a broad frequency range (covering macroscopic and microscopic vibration). The ultrasonic sensor is responsive in a narrow frequency band around a fixed center frequency. The objectives in Vibration Analysis relative to stress wave activity are:

1. Detect the presence of stress waves;
2. Identify the periodicity (or lack thereof) of the stress wave events;
3. Identify the source;
4. Assist in severity assessment of the fault.

In the next subsection, the use of an ultrasonic sensor for vibration analysis of stress waves is discussed. This will be followed using an accelerometer as the sensor for vibration analysis of stress waves.

Stress Wave Analysis with an Ultrasonic Sensor

An ultrasonic sensor has a narrow frequency band centered on a fixed frequency. The fixed frequency is set by the supplier of the sensor. Referring to the data in Figures 3–5, the center frequency would optimally be set in the 1–40 kHz range to optimize the sensor response to impacting, friction and fatiguing type events.

A common ultrasonic sensor used for stress wave vibration analysis has a center frequency of approximately 25 kHz. This sensor would be responsive to the primary lobes of activity for both impacting and friction from the small balls (see Figures 2–4) and the secondary lobes for the larger ball. Thus the sensor would theoretically meet the first objective of detecting the presence of stress waves. If a data block of sufficient duration is captured (typically containing 15 or more revolutions of the machine being monitored), an autocorrelation analysis would suffice to establish no periodicity (2nd objective). A spectral analysis could identify the rate of any periodic activity (identifying the source). The level of the signal (g-level) could not be relied upon to establish severity of the fault in a consistent manner.

Stress Wave Analysis with an Accelerometer

Accelerometers typically used for vibration analysis when stud mounted on a clean flat surface have a bandwidth from the low Hertz to sensor resonance (typically in 25-35 kHz range). Thus the sensor covers the macroscopic and a significant percentage of the microscopic vibration frequency range. For macroscopic vibration analysis, the signal is band limited by a low pass (anti-aliasing) filter which generally is in the fractional to a few kHz range. For microscopic vibration analysis, the signals from the accelerometer due to microscopic events are separated from macroscopic vibration by routing the signal from the accelerometer through a high pass filter. The upper frequency for the microscopic analysis is dependent on the mounted sensor natural frequency.

For a commonly-used accelerometer used for route based data acquisition, the mounted resonances are typified by the measured frequency response curves presented in Figure 6[††] for cases of 1) sensor stud mounted (around 30 kHz), 2) sensor attached to flat rare earth magnet and placed on clean smooth flat surface (around 15 kHz) and 3) sensor attached to dual rail magnet and placed on clean curved surface (around 9 kHz). Experience has demonstrated that use of a flat magnet mounting provides adequate capability for detecting friction and fatiguing on industrial machinery if this magnet is mounted on a clean, flat surface (allowing it to measure up to approximately 15 kHz).

Data Reduction Methodology

The dominant vibration analysis occurs in the spectral (frequency) domain. This requires the capture of a time waveform (digital) which generally is transformed into the frequency domain via an FFT analysis. In addition to the spectral analysis, the transforming of the time waveform into the autocorrelation function provides great assistance in diagnostics. Autocorrelation allows an analyst to determine what are the dominant periodic events within either a normal or stress wave analysis waveform (in doing so, it can often clean up a waveform allowing an analyst to see what sources are the main contributors to such waveforms).

The first task is the construction of the time waveform of sufficient duration (should contain 15+ revolutions of the machine being monitored). To acquire a direct capture of the waveform associated with microscopic vibration, a high sampling rate would be required (such as 50,000 samples/sec or higher) for a significant time (15+ revolutions), which is not practical. Hence an alternative method of obtaining an appropriate time waveform is required.

Two methods of obtaining the time waveform for stress wave vibration analysis are briefly discussed here. The first is a commonly used Amplitude Demodulation technique. The second is a peak value (PeakVue™) methodology employed by Emerson.

[††]The frequency response curves were provided by IMI Sensors a PCB Piezotronics Division located in Depew, NY.

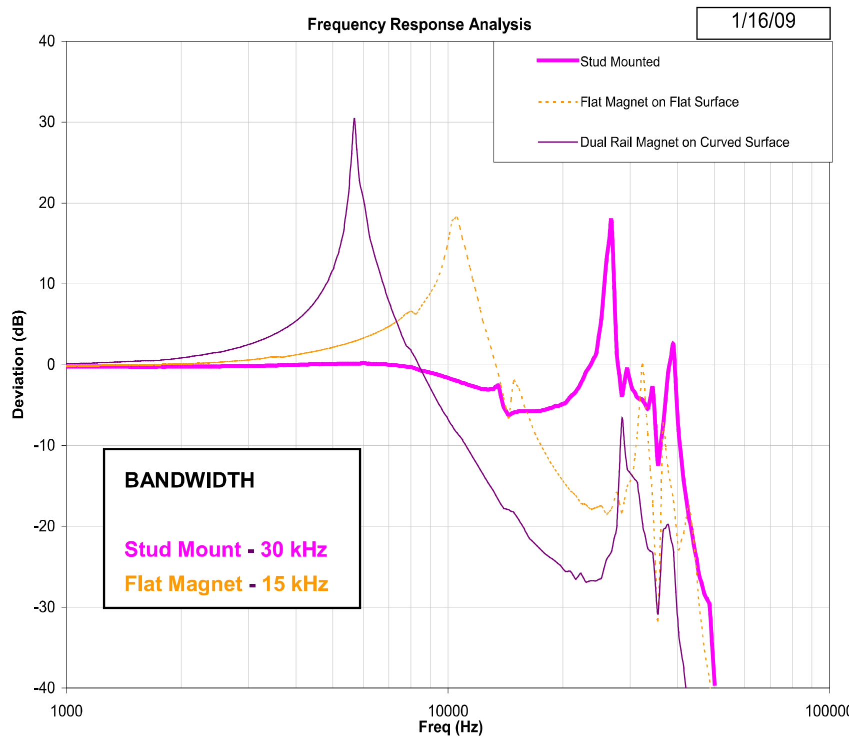


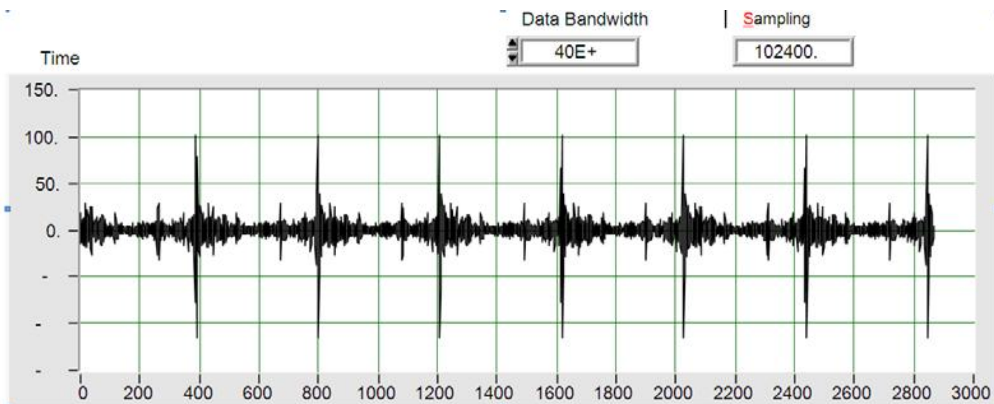
Figure 6. 50kHz Frequency Response for Accelerometer a) Stud Mounted, b) Flat Magnet Mounted, and c) 2-Rail Magnet on Curved Surface Mounted.

The procedure for the demodulation methodology is to route the signal from the accelerometer through 1) a high pass filter, 2) full wave rectification, 3) low pass filter to separate the modulating signal from the “carrier,” and 4) convert the resultant signal into a digital signal with bandwidth exceeding the highest probable fault frequency with a duration sufficient to capture 15+ revolutions.

The PeakVue methodology is similar to the demodulation methodology with the exception that the signal never goes through a low pass filter. That step is replaced by an absolute peak value extraction. The essence of the procedure is to 1) pass the signal through a high pass filter, always sample the signal at a sampling rate consistent with the 40 kHz bandwidth ($2.56 \times 40{,}000 = 102{,}400$ samples/sec), construct the peak value time waveform by absolute decimation by the factor of 40,000/(analysis F_{max}) sequentially until the waveform block is complete.

To demonstrate the effect of the low pass filter (could be the anti-aliasing filter set by the analysis bandwidth), consider the time waveform presented in Figure 7. The waveform was constructed by capturing a waveform from a machine with significant friction activity with an F_{max} set at 40,000 Hz and 1600 lines (4096 data points). The captured waveform was then repeated seven times to make up the test waveform presented in Figure 7. Using this test waveform, the waveforms for both demodulation and PeakVue were constructed for an analysis bandwidth of 1000 Hz, 200 Hz, and 50 Hz. The results are presented in Figure 8. The attenuation which the low pass filter (here it is the anti-aliasing filter) has is obvious. The demodulation process lost at least 75% of the signal relative to the equivalent PeakVue data. On the other hand, the attenuation introduced in the PeakVue methodology is nonexistent.

Figure 7. Waveform Used to Demonstrate the PeakVue Methodology Versus the Classic Demodulation Methodology Presented.



Case Studies

Three case studies are selected to emphasize key points brought out in the body of this paper. The first case study employs a multi-frequency ultrasonic sensor for the detection and location of a bearing fault on a conveyor belt used to transport material at a mining facility. The second case study is the comparison of demodulation and PeakVue data from a gearbox driving a rock crusher. Demodulation and PeakVue data were acquired in a route based condition monitoring program. The third case study was selected to emphasize the importance of maintaining discipline in sensor mounting in Route Based vibration condition monitoring.

Ultrasonic Sensor for Bearing Fault Detection

A multiple frequency ultrasonic sensor with three center frequencies of 4 kHz, 30 kHz and 40 kHz was used to monitor two conveyor belt systems where each has a length of 0.5 mile and width of 4 feet. An outline drawing representative of the conveyor belts is presented in Figure 9. The inboard and outboard bearings were monitored on each of the 6 rolls on each conveyor belt (total of 24 bearings). Of the 24 monitored bearings, 8 were identified as having probable bearing faults. The 8 bearings were replaced and 7 of the 8 were visibly verified to be defective (no means for estimating severity of faults from the data since no trend data was available).

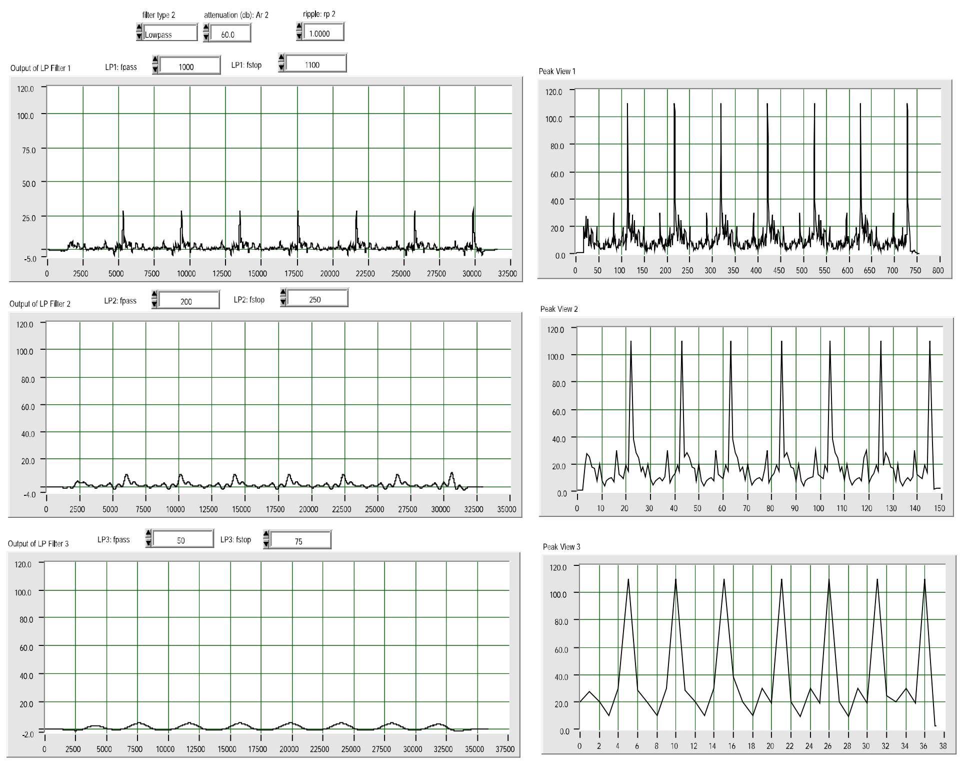


Figure 8. Demodulation and PeakVue waveforms from waveform presented in Figure 7 for F_{max} = 1000, 200 and 50 Hz.

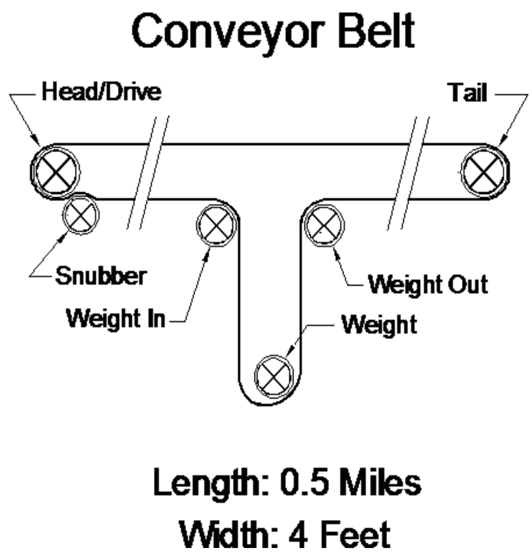


Figure 9. Conveyor Belt with Key Rolls Identified.

Representative spectral data from the ultrasonic sensor from the IB (inboard) Head Snubber of the B Conveyor System for the 4 kHz and 30 kHz channels are presented in Figure 10. The periodic activity is at the BPFO fault frequency. The spectral data from the OB (outboard) bearing of the same Head Snubber roll are presented in Figure 11. The outer race fault signature is still present in the 4 kHz OB data but not in the OB 30 kHz data. The absence of the fault signature in the 30 kHz data on the OB leads to the conclusion that the fault is in the IB bearing.

Demodulation and PeakVue Data

Data are presented in this subsection extracted from a route base condition monitoring program of a Kiln gearbox. In the transition from the demodulation to the PeakVue methodology, both sets of data were acquired and stored. The spectral data from demodulation and PeakVue methodologies are presented in Figure 12. Both sets of data show periodic activity at 4.02 Hz (running speed). The time waveforms corresponding to these two spectral data are presented in Figure 13. In the PeakVue data, the peak g-level is 1.24 g's. In the demodulation waveform data, the peak g-level is 0.27 g's. For a shaft turning at 241 RPM, the recommended Alert level in PeakVue is 1.9 g's. Thus the possible fault would have minimal impact on the monitoring program since it is below the Alert level. Since the g-level in