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**DESCRIPTION OF PEAKVUE AND ILLUSTRATION OF ITS WIDE
ARRAY OF APPLICATIONS IN FAULT DETECTION AND
PROBLEM SEVERITY ASSESSMENT**

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DESCRIPTION OF PEAKVUE AND ILLUSTRATION OF ITS WIDE ARRAY OF APPLICATIONS IN FAULT DETECTION AND PROBLEM SEVERITY ASSESSMENT

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Abstract:

The primary objective of this article is to serve as a “white paper” on the PeakVue™ analysis method. PeakVue analysis is actually a measure of “stress wave” activity in a metallic component. Such stress waves are associated with impact, friction, fatigue cracking, lubrication, etc., that generate faults in various components such as rolling element bearings and gears. For example, when a rolling element impacts a defect on a bearing raceway, it will generate a series of stress waves that propagate away from the location of the defect in numerous directions. The wave propagation introduces a ripple on the machine surface that will introduce a response output in a sensor detecting absolute motion such as an accelerometer or a strain gage.

This paper is not intended to imply that Stress Wave Analysis (PeakVue Analysis) is the “panacea” of Condition Monitoring tools that should replace all the PdM tools available today in detecting and correcting machine faults. Instead, its primary purpose is to prove that Stress Wave analysis is a powerful complementary tool that can detect a range of faults and problem conditions that techniques such as Vibration Analysis alone might miss under certain conditions.

Some common defects which generate stress waves are pitting in antifriction bearing races causing the rollers to impact, fatigue cracking in bearing raceways or gear teeth (generally at the root), scuffing or scoring on gear teeth, cracked or broken gear teeth and others. The challenge becomes one of detecting and quantifying the stress wave activity relative to energy and repetition rate. This leads to the identification of certain faults and, with experience, allows evaluating severity of those faults detected.

This paper will begin describing what stress waves are and how they are measured. It will describe some of the signal processing methods used to measure stress waves and show how this differs from processing vibration signals.

An important topic of the paper will deal with the recommended measurement setups needed to ensure optimum PeakVue data is captured. This will include the proper choice of high-pass filter, analysis bandwidth (F_{MAX}), number of FFT lines, number of time domain samples, etc. The choice of such parameters may be dependent on the type of fault for which one is looking (cracked gear tooth versus generalized tooth

wear, for example) so this will likewise be discussed. Examples will be included demonstrating how certain faults might actually go undetected if the proper setup parameters are not chosen.

Another interesting section will compare PeakVue with a somewhat similar high frequency enveloping technique called "Amplitude Demodulation". Important differences in these two tools will be pointed out and several direct comparative measurements will be made to show the very real differences in results that can occur.

An important section will then show how to apply PeakVue alarm levels to various machine components and operating speeds. Studies have been conducted to determine the effect on PeakVue amplitudes of certain faults occurring on specific components and how they can generate very different amplitudes.

The paper will conclude with presentation of a series of real-world case histories that will demonstrate how each of the PeakVue techniques taught can be put to work not only to detect a wide array of machine faults, but also to assess the severity of such problems found, eventually leading to determination of what corrective actions are required to resolve these problems along with when these actions should be taken.

It is sincerely hoped by the authors that the information provided in this paper will be of great assistance in enhancing the knowledge, understanding and practical application of stress wave analysis to a variety of faults that can be detected using this powerful diagnostic tool. In so doing, it is hoped that the reader can directly apply this information contained herein so that the overall effectiveness of his Condition Monitoring Program can be significantly enhanced.

1.0 Introduction

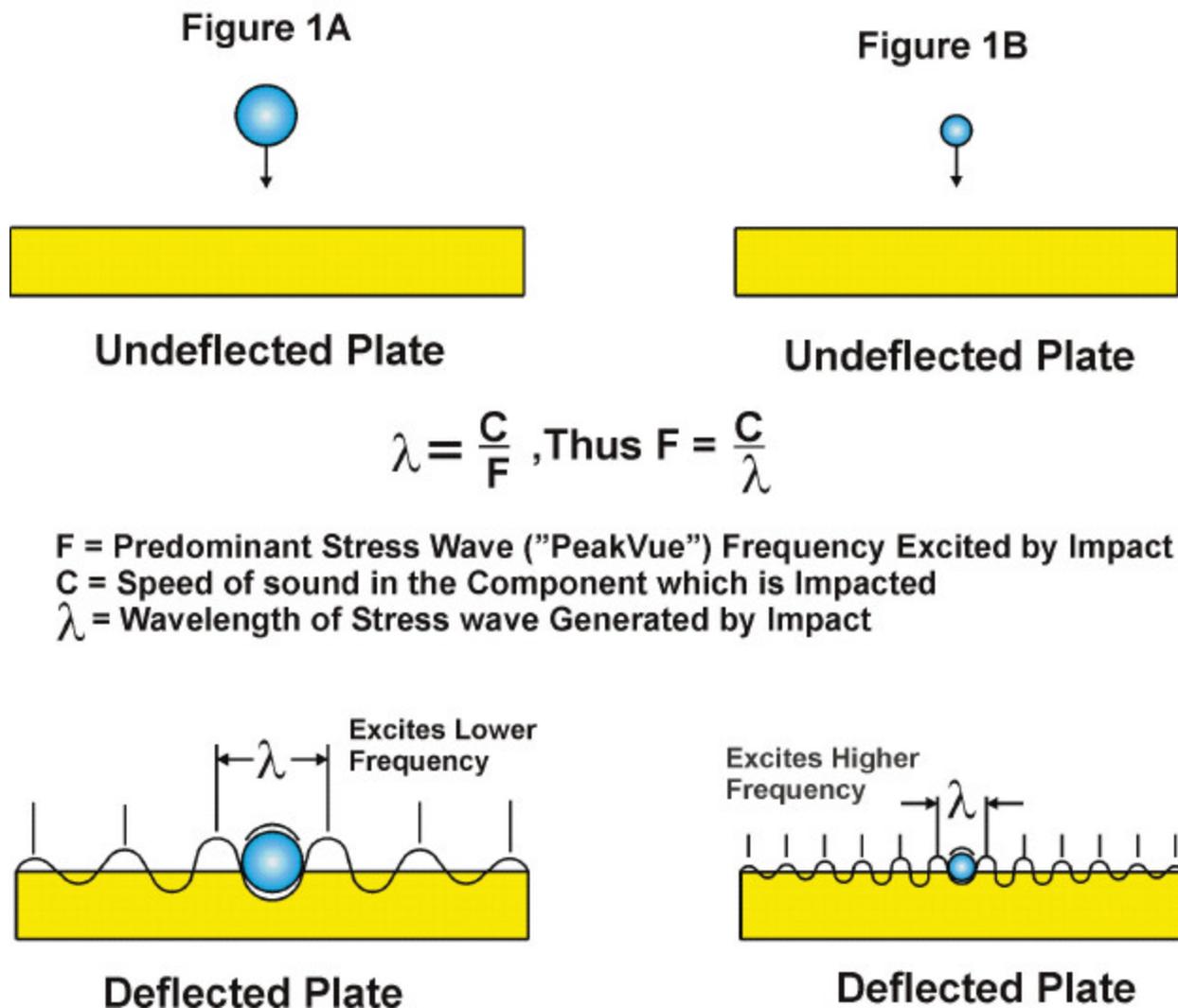
Many technical faults within industrial rotating machinery manifest themselves through modal excitation (vibration) and stress wave initiation. Modal excitation can be detected using sensors which detect absolute or relative motion. A common sensor employed for detection of absolute motion is the accelerometer. The analysis of the modal motion relative to the machine health is referred to as vibration analysis. The methodology employed in vibration analysis consists of:

1. Capture (digitally) a time waveform from a sensor for a specified time period. The signal is first passed through a high order, low-pass filter prior to digitalization. The purpose of the low-pass filter is to remove all frequency content which exceeds the Nyquist frequency (one-half the sampling rate).
2. Transform the modified discrete time waveform into the frequency domain employing FFT methodologies.
3. Look for excessive activity compared to other similar machinery or previous history on the same machine at discrete known fault frequencies.

The implicit assumption in vibration analysis is that the signal being analyzed is stationary. The spectral values are average values which are appropriate for stationary (continuous) conditions.

Stress waves in metal accompany actions such as impacting, fatigue cracking, scuffing (scoring) abrasive wear, etc. Stress wave emissions are short-term, lasting several microseconds to a few milliseconds, transient events which propagate away from the initiation site as bending (s) and longitudinal (p) waves at the speed of sound in metal. The s waves introduce a ripple on the surface which will excite an absolute motion sensor such as an accelerometer. Figure 1 illustrates how stress waves are generated by impact and propagate along a metal plate. Note that the larger impacting object excites a greater wavelength, and therefore, generates a lower stress wave frequency according to the equation given in Figure 1. The detection and classification of these stress wave packets provides an important diagnostic tool for (a) detecting certain classes of problems and (b) severity assessment.

Figure 1. Excitation of a Stress Wave within a Flat Plate due to Impact by a Larger (Fig. A) and a Smaller (Fig.1B) Spherical Object



NOTE: The speed of sound (c) in most metals ranges from approximately 5000 to 10,000 ft/sec (1520 to 3040 m/sec).

Figure 1 shows that the predominant frequencies generated by the stress waves are predominantly controlled by the ratio of the speed of sound within the media over the wavelength. Later sections will show that these frequencies are largely concentrated in the 1000 to 15,000 Hz range (largely dependent on the mass and geometry of the impacting object, the type of surface it impacts, etc. Stress waves also excite and include frequencies excited by system resonances. However, it is surprising that the contribution of such resonant responses is typically only 5% to 10% of total stress wave content.

For an accelerometer at a fixed location, the wave propagation will be a reasonable short-term transient event lasting on the order of fractional to several milliseconds. The duration of the event will be dependent upon (1) type of event e.g., stress waves from impacting will last longer than stress waves accompanying the release of residual stress buildup through fatigue cracking, (2) relative location of the sensor (accelerometer) to the initiation site, and (3) severity of the fault responsible for the stress wave emission.

In this paper, the emphasis will be placed on the use of stress wave capture and analysis as a diagnostic tool in assessing the current condition of rotating equipment relative to the presence/absence of faults which may lead to deteriorating state of operating efficiency or catastrophic failure.

Section 2 of the paper will focus on the familiar use of stress waves in nondestructive testing, the fundamental nature of stress waves, possible sources of stress waves in rotating equipment, and a description of methods which may be employed for the detection and assessment of stress waves.

Section 3 will cover methods which have proven useful for the detection and analysis of stress wave data, including the choice of optimum transducer and measurement location. This section will also include a comparison of the peak value or PeakVue™ methodology developed by CSI with alternative methods such as amplitude demodulation, along with comparisons of the methods that can be employed.

Section 4 will provide recommended measurement setups for making effective PeakVue measurements in a generic sense. Here, a table will be provided tabulating what setups should be used when making measurements on a variety of machines and operating speeds (including recommended high-pass filter, analysis bandwidth (F_{max}), mounting method, number of averages, number of FFT lines, etc.

Section 5 will include recommended PeakVue alarm levels for generic faults on industrial machinery. These alarms are based on numerous studies conducted in the field and within laboratories on a wide variety of machinery types operating at a wide range of speeds. This section will show how to apply PeakVue alarm levels to various machine components and operating speeds. A table will be provided to allow specification of alarms to the PeakVue waveforms based on the "Pk-Pk Waveform" value within the software.

Section 6 will provide a table that will show how certain bearing, gear and lubrication faults will affect both PeakVue spectra and time waveforms and will include likely amplitudes that might be present in waveforms for each of the faults.

Section 7 will include several real-world case studies illustrating both a) fault detection and b) severity assessment using PeakVue analysis.

Section 8 will consist of a summary, along with final conclusions and recommendations for making meaningful PeakVue measurements.

An Appendix will follow the main paper including special PeakVue applications. It will likewise include a detailed description of a very powerful diagnostic tool known as “autocorrelation waveforms”. These are often used to improve the signal-to-noise ratio of the PeakVue waveform in order to better reveal what fault frequencies are actually causing impact within the machinery being evaluated.

2.0 Discussion of Stress Waves

2.1 Nondestructive Testing Employing Stress Waves

Stress waves are introduced into elastic materials by many different sources. Some are purposely introduced for nondestructive testing. Generally, the intent is to detect faults within the material by introducing short bursts of stress wave activity and examining the reflected wave patterns. In structures made of steel where small flaws must be detected, high frequency (ultrasonic) stress waves (in the few megahertz range) are used since the wavelength of the introduced stress waves must be similar to the flaw size to be detected. These waves are generally introduced by piezoelectric material rigidly attached to the structure through a coupling medium. The piezoelectric material is forced to vibrate by a high frequency voltage source for a short duration which introduces stress waves into the elastic body.

Stress waves are also used for nondestructive testing of concrete structures. Here, defects of concern are much larger than in steel structures. A common method for introducing stress waves for this purpose is to drop (impact) spherical balls of varying sizes (in the 1/8” to 3/4” diameter range) on the surface. Larger balls introduce lower frequency wave packets than do smaller balls. Again, the objective is to introduce the short-term wave activity and analyze the reflected signal (referred to as the “Impact-Echo Principle”). The receiving sensor is located on the surface of the structure in the proximity of where the impacts are introduced. The primary parameters to measure are a) the time between impact and received reflection and b) the energy of the received reflection relative to that transmitted.

Yet another nondestructive methodology employing stress waves is Acoustic Emission on pressure vessels, etc.² When a material is stressed to the point where the elastic limit is exceeded, either plastic deformation or cracking occurs within a very short time period. This sudden movement of material introduces stress wave packets which are detectable using either accelerometers or ultrasonic sensors.

2.2 Quantitative Framework for Understanding Stress Waves Generated in Rotating Machinery

Stress Waves can be generated in any elastic medium. The primary interest focused upon within this study is in rotating machinery. Stress waves accompany the sudden displacement of small amounts of material in a very short time period.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. Once the stress waves are generated, they propagate away from the initiation site at the speed of sound in the particular medium (metal) being evaluated.

A quantitative framework for the generation and detection of stress waves can be developed using the Hertz theory for metal-to-metal impacting³ and wave theory⁴ for propagation of stress waves in metal. We consider the brief discussion presented below on the theory of generation and propagation of stress waves to provide insight to:

1. Selection of sensor for the detection of stress waves,
2. Identification and localization of the fault introducing the stress waves, and
3. Severity assessment of the fault.

The dominant stress waves generated by a metallic sphere impacting on a relatively large plate are bending waves^{5,6} of half-period equal to the duration of the impact. Olma⁶ 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. A graphical representation of the results is presented in Figure 2. The impacting object was assumed to be spherical; therefore, the diameter for a spherical steel ball is presented in Figure 2 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 2b). Lower contact times (smaller diameter balls) will lead to a broader frequency band being excited than will longer contact times (larger diameter balls).

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 3. The captured signal, presented in Figure 3, is very similar to what would be expected from the Hertz wave impact (see Figure 2).

The contact time for the 1.5" D ball is estimated to be 100 μ s (Figure 2b). For 100 μ s contact time, the bandwidth to 10 dB attenuation would be around 10 kHz. This bandwidth is consistent with that observed for the 1.5" D ball shown in Figure 3.^a

The contact time for the 0.5" D ball would be in the 30–40 μ s range (see Figure 2b). For this contact time, the bandwidth to the 10 dB attenuating level is estimated to be in the 30 kHz range. This is very consistent with the impacting spectral data for the 0.5" D ball presented in Figure 3.

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 (that time where material movement is present on a microscopic scale).

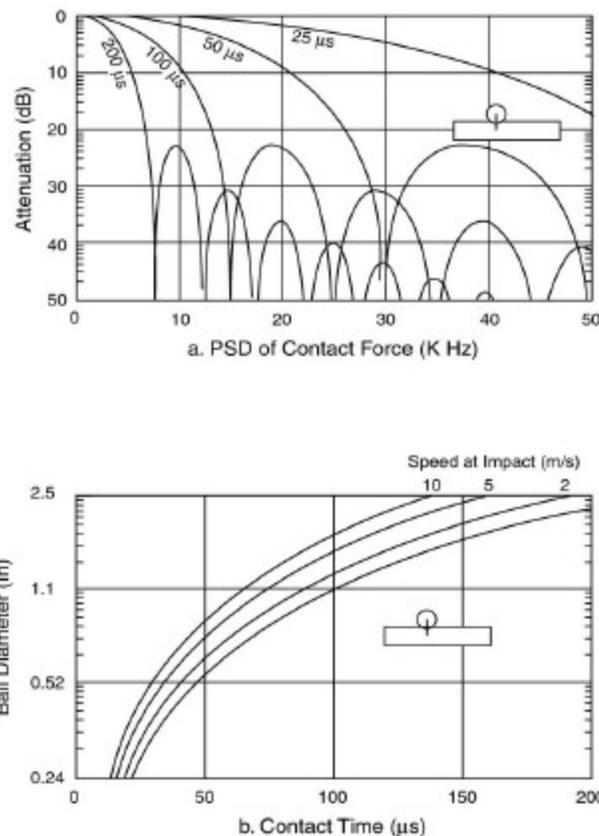


Figure 2. Hertz theory prediction for metal balls for varying size and speed impacting on large metal plates.⁶

^aThe increased activity around 15 kHz in the spectral response for the 1.5" Diam. ball in Figure 3 is from structural resonance. This response was there and stationary for several different impacting balls of various diameters.

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 evidenced by large black spherical particles found in the oil lubricating the bearing.

The spectral data from the smaller bearing are presented in Figure 4. Here, most energy is in the 20 kHz to 30 kHz range (lower limit has increased from 10 kHz to 20 kHz relative to 1/2”D ball impacting data presented in Figure 3). 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 lower frequency has shifted upwards relative to the 1/2”D ball by a factor of two, 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.

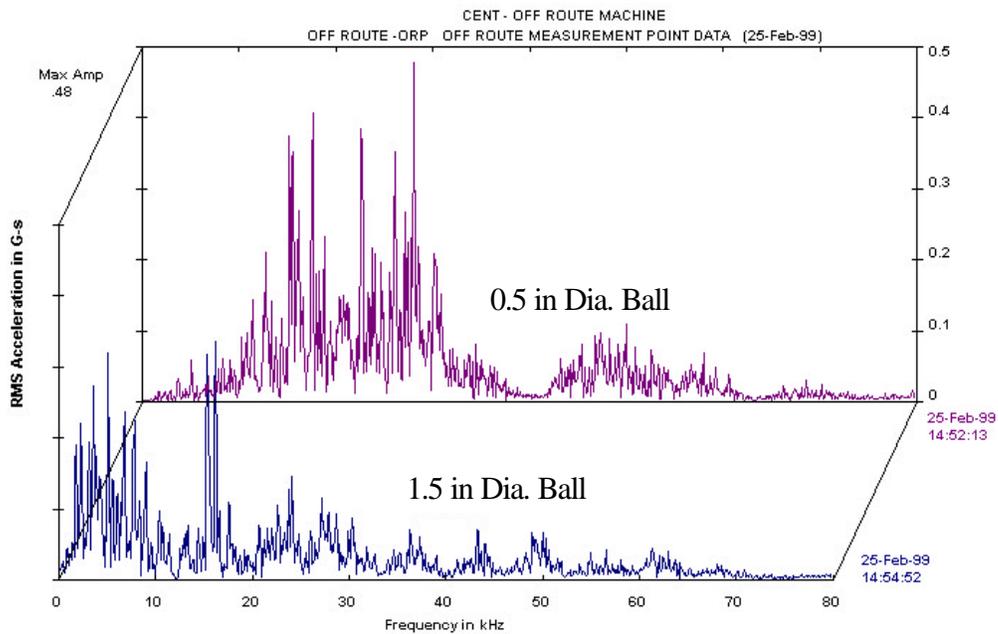


Figure 3. Spectral data captured for a 0.5” Dia. and a 1.5” Dia. metal ball impacting on a large metal plate.

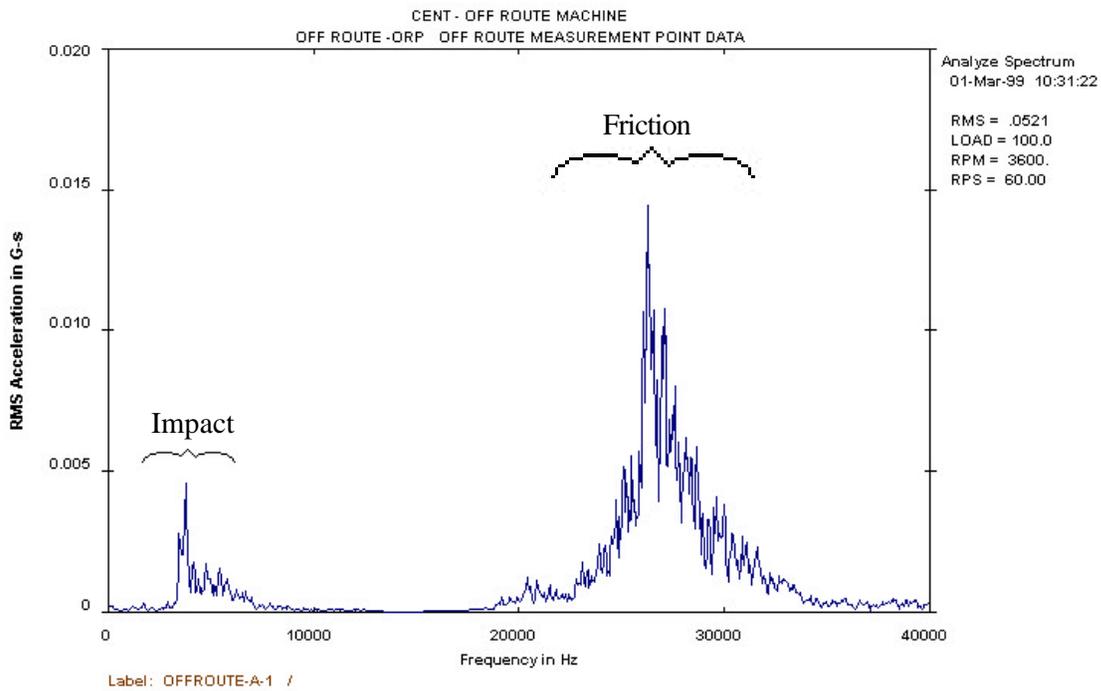


Figure 4. Spectral data captured on small machine with 3/8" Diameter ball running dry at about 3600 RPM.

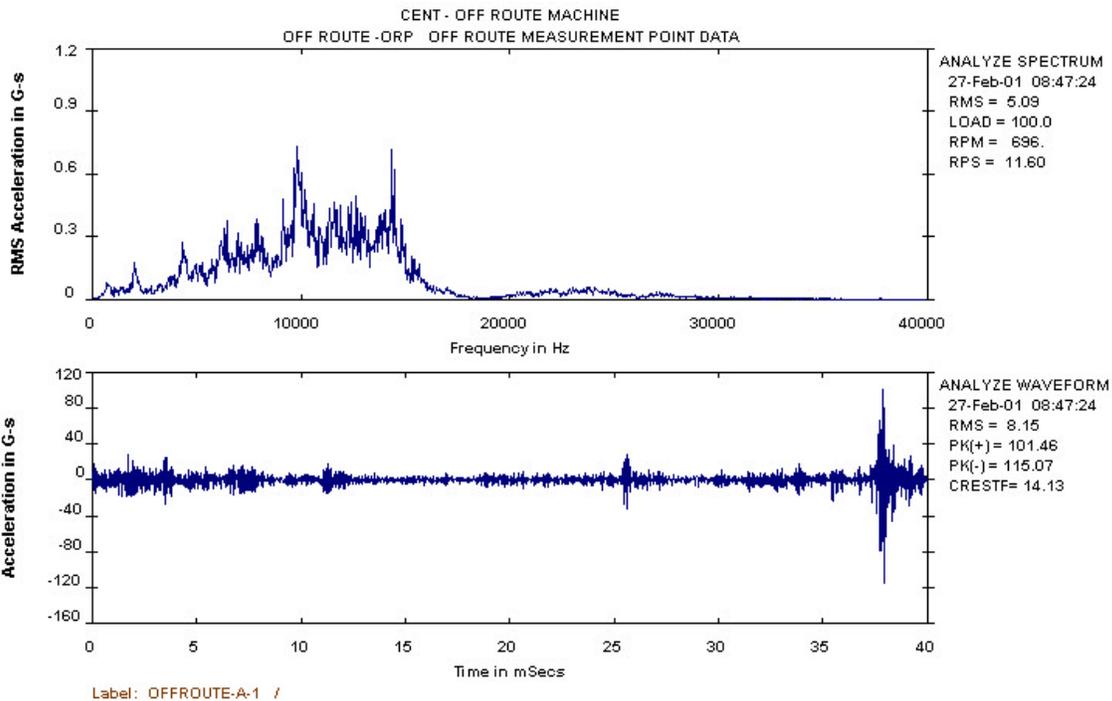


Figure 5. Spectral and time waveform data from a split ring pedestal bearing with rollers in the 1.0" to 1.5" Diameter range turning at 700 RPM with friction

The spectral and time waveform data captured for the larger bearing turning around 700 RPM are presented in Figure 5. 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 in the 5 to 18 kHz range which is about a factor of 3 greater than that observed from the impacting 1.5" D ball (see Figure 3).

The spectral data presented in Figures 3, 4 and 5 from impacting and friction events were on a linear scale and hence showed a reasonably small frequency band where energy is present. Referring to Figure 2a, 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 5 with a linear amplitude scale is presented in Figure 6 on a log scale. Here the broadband nature with increasing attenuation with increasing frequency and the picket fence effect is 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 that from impacting. This translates to the frequency band where the dominant energies resides will be higher (factor of 2-4) for friction generated events than those from impacting events.

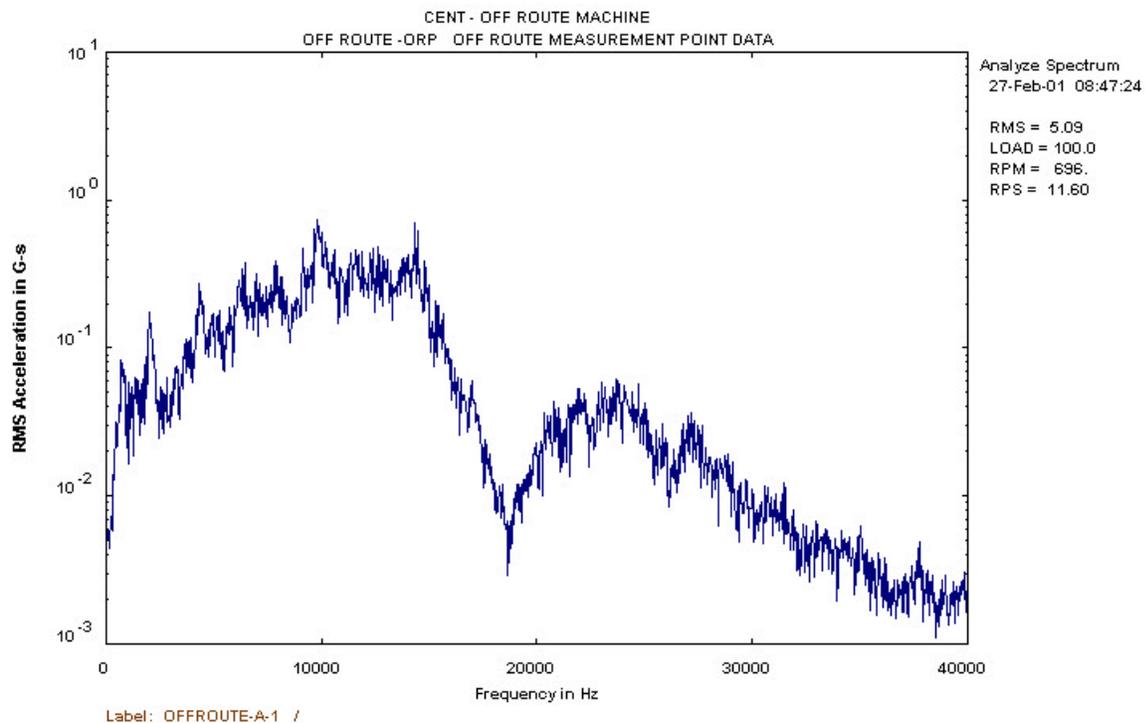


Figure 6. Spectral data for Figure 5 with log amplitude scale

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 at this time; however, past experiences have suggested the duration (and hence, contact time) of these events is less than that from friction-generated events. They would likely

still be in the few μ sec range (not the nanosecond range), hence the dominant energy would be in the few kHz range (40–80 kHz range). Again, the events are broadband and hence can be captured below or above the dominant generated frequency band.

2.3 Detection of Stress Waves Generated in Rotating Machinery

2.3.1 Introduction

In the previous section, the generation of stress waves with emphasis on the dominant frequency band excited by impacting (included effect of mass), friction and fatigue cracking was discussed. For rotating machinery, these often occur in the rotating members (rollers, inner race, outer race, gears, etc.). It is not practical to locate sensors directly on these members, hence the stress must propagate to a location where a sensor is in the proximity of the bearing housing). Hence the quantitative effect wave packet needs to be understood to assist in establishing “Alert/Fault” levels as well as provide input regarding the class of sensors and location of sensors for stress wave detection.

2.3.2 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^{4,7,8} with the higher frequency components attenuating faster than the lower frequency components as the observation (measurement) point is removed from the point of origin due to the fact that the velocity of propagation is proportional to the square root of frequency,. This is especially true when stress wave packets must cross physical interfaces such as from shaft (or inner race of a bearing) to get 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 impacting the shaft separately 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 I.

*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 is observed. This is consistent with observations made in the past and with approximate theoretical predictions. The signal at 150 kHz did not penetrate 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. As an aside, there was very little energy generated for the 1.5"D ball impacting in the 150 kHz range as evidenced by the lack of response at 150 kHz.

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 expect 6–10 dB). The most probable explanation for the event occurring here is that the stress waves are exciting the ringing vibration mode of this shaft and 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 chances of that energy transmitting to a location where a sensor could be mounted is small at such a high frequency.

2.3.3 Selecting the Sensor and Location

Due to the dispersion 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 (relative to normal vibration monitoring) of an issue than is mounting the sensor in or near the load zone with the caution that we are monitoring "waves" and hence, must always be cautious of encountering nodal points which can occur due to multi-path transmission and in the vicinity of sharp corners, etc.

The bending stress waves introduce a ripple. Hence any sensor which is sensitive to absolute motion occurring at a high rate would suffice, providing it has sufficient frequency range and amplitude resolution capabilities. Therefore, this sensor could be an accelerometer with sufficient bandwidth, an ultrasonic sensor, a strain

gauge, piezoelectric film, et al. The primary motivation behind stress wave monitoring is to acquire information for machine health monitoring which is the motivation behind vibration monitoring in a predictive maintenance program. By far, the most common sensor employed in vibration monitoring is the accelerometer; therefore the sensor of choice for stress wave monitoring is the accelerometer. The requirements for this sensor will be sufficient analysis bandwidth (frequency range), amplitude resolution and appropriate sensitivity.

The bandwidth of an accelerometer is dependent on (1) its design and (2) the manner in which the accelerometer is attached to the surface. The general effect, which different mounting schemes have on the sensor bandwidth, are presented in Figure 7 (sensor becomes entire system attached to the surface).

Typically, standard 100 mv/g accelerometers can be used for most all PeakVue measurements (even on low speed machines since the PeakVue information will still typically be above 500 Hz (30,000 CPM). There are special cases where either a higher sensitivity or lower sensitivity accelerometer might be needed to improve PeakVue measurements. For example, if the machine is at very low speeds lower than 5 to 10 RPM (certain 500 and 1000 mv/g accelerometers now have the ability of making low frequency vibration and higher frequency measurements required to detect PeakVue information). On the other hand, if a machine generates very high frequencies (above 10,000 to 20,000 Hz or greater, a special 10 mv/g, high frequency accelerometer may improve the information detected by PeakVue and compared to the high frequency vibration that is likewise taken).

Just as is the case with vibration, the transducer mounting method and the mounting surface can likewise adversely affect the PeakVue measurement (painted versus unpainted surface; flat versus curved surface, etc.). And, it will be shown that either the mounting surface or the choice of the transducer mounting method can limit how high an F_{MAX} or a high-pass filter can be chosen, thereby limiting PeakVue's ability to detect a number of faults it otherwise could reveal if required preparations of the surface and transducer mount are made that would allow higher frequency measurements to be reliably made and a greater high-pass frequency employed.

The variability in the response for the hand-held probe is significant. Hence, due to poor bandwidth and variability, this method of sensor mounting for stress wave detection should be avoided. Hand-held accelerometers provide relatively poor stress wave information due to the very limited frequency capabilities provided by a hand probe as shown in Figure 7 (note importantly that a hand held probe will miss most all data above approximately 2500 Hz where stress wave energies for most faults is typically concentrated).

Method 5, dual rail magnet, has been found to be useful for stress wave detection in many applications, with the precaution that the magnet must be placed on a clean¹ smooth surface (dual rail magnets are sometimes referred to as "2-pole

¹ Painted surfaces should be avoided. Thick paint will totally block out stress waves exciting the sensor and clearly should be removed

magnets”). There should be a minimum of dual line contact made between the magnet rails and the assumed curved surface of the machine. Limitations on using this mounting scheme will be addressed later in this paper.

A flat, rare earth magnet will capture more meaningful PeakVue data than will a dual rail magnet if mounted on a flat, reasonably clean surface. This is particularly the case if either a frequency bandwidth (Fmax) greater than approximately 3000 Hz (180,000 CPM) is taken, or if a high-pass filter greater than 2000 Hz is used. If either of these two conditions exists, actual field tests have proven that if fault frequencies are present above approximately 3000 Hz which are detected by a flat rare earth magnet, such frequencies can be missed altogether by a dual rail magnet when making PeakVue measurements.

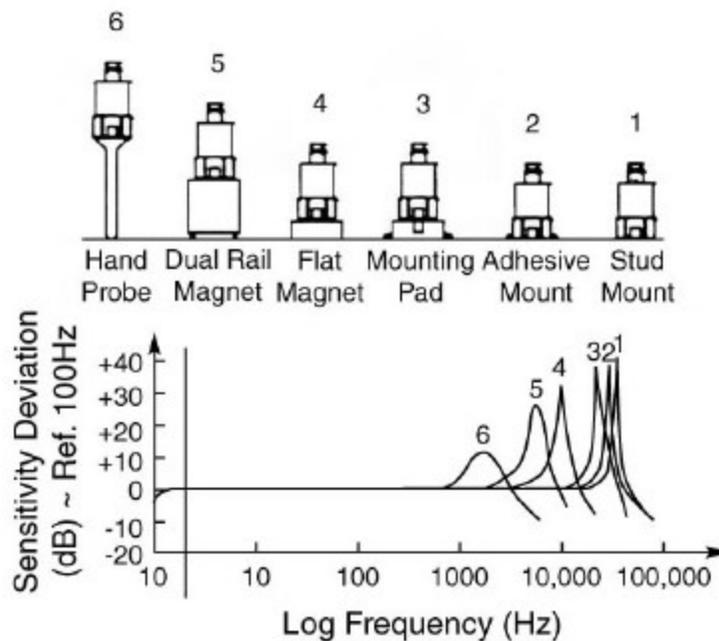


Figure 7.² Effect on frequency bandwidth of accelerometer mounting scheme to a machine surface.

² Fig. 7 is taken from the IMI Industrial Vibration Catalog, CAT.600, page 129.

3.0 Analysis of Stress Waves

3.1 Introduction

In the previous section, the initiation of stress waves with emphasis on duration (short-term transient events) and frequency content within each event for metal-to-metal impacting, friction, and fatigue cracking were globally discussed. This was followed by a discussion on the propagation of these events from the point of initiation to the point of detection. Here it was highlighted that the duration of the event at point of detection increases relative to duration of the event at the initiation site due to dispersion (speed of stress wave transmission is proportional to square root of frequency). In addition to the dispersion, the higher frequency components get attenuated much more so than do the lower frequency components, (especially when crossing interfaces)..

The previous section (section 2.3) concluded with a discussion of the effect that the attachment of the accelerometer to the surface has on the frequency response (bandwidth) of the detection system (accelerometer plus mounting hardware). Caution must be exercised on both sensor selection and mounting to ensure adequate bandwidth for reliable detection of stress waves, independent of their source.

In the remainder of this section, methods proven to be useful for the analysis of stress waves are presented. This will include comparison of the method developed by CSI referred to as PeakVue™ with a commonly employed demodulation scheme.

3.2 Capture and Analysis of Stress Waves

The analog output of an accelerometer located on a machine will include classic vibration and stress wave induced activity over the entire response bandwidth of the sensor system. Generally, the component proportional to the normal vibration will cover a lower frequency band than will the component proportional to stress wave activity. Hence the normal component is separated from the stress wave activity by routing the analog signal through a high order low-pass filter followed by the conversion to the digital domain. A sampling rate is chosen whereas the Nyquist frequency (equal to one half the sampling rate) exceeds the low-pass filter corner frequency.

On the other hand, the stress wave-induced component of the accelerometer signal is generally separated from the normal vibration by routing the signal through a high order high-pass analog filter. Prior to routine digitization of the resultant signal for further analysis, the high frequency signal must be further processed.

The most common method for further processing the high frequency data is the standard amplitude demodulation methodology. The high frequency signal (having a mean value of zero) is first rectified (half or full-wave) and then passed through a low-pass filter prior to digitization. The output of the low-pass filter will be the mean (average) value of the high frequency component input to the filter. The averaging time is dependent on the break frequency of the filter. Thus, a signal with a low peak value occurring very often (large duty cycle) could have larger mean values than a signal with

a large peak value occurring at a low rate (low duty cycle) relative to the averaging time set by the low-pass filter. This dependence on amplitude of the output signal from the low-pass filter on the low-pass filter break frequency and on the duty cycle of the event (as well as duration) makes use of the amplitude for trending for severity assessment difficult using the demodulation scheme.

The important parameters to capture from stress wave activity are:

1. Amplitude of each event,
2. Approximate time required for the detected event to occur, and
3. Rate (periodic or non-periodic) at which events are occurring with emphasis on event rate versus specific fault frequencies which are dependent on both the specific component and on machine rotational speed.

The method developed by CSI which captures peak values of the analog signal from the sensor post-passing through the high-pass filter, called PeakVue, provides the three key parameters specified above. The appropriate time resolution is accomplished by the selection of the maximum frequency, F_{max} , to obtain adequate resolutions of possible fault frequencies, e.g., an F_{max} of 3 or 4 times the inner race fault frequency when monitoring bearings. Once the F_{max} is specified, peak values will be collected at a rate of $2.56 * F_{max}$. The inverse of the sampling rate defines the time increment over which the peak value is captured. These peak values are captured sequentially until the total desired block length is accumulated. The total time in the PeakVue waveform depends on the number of shaft revolutions desired by the analyst and the block of data consists of sequential constant time intervals of peak values (the PeakVue spectrum is computed from the time block data by an FFT algorithm as are vibration spectra). For bearing fault analysis, the block time should be sufficient to provide adequate resolution on the lowest fault frequency (cage fault). This suggests a minimum of 15 revs (preferably 20) be included in the captured peak value data block .

Once a peak value time block of data is acquired, further analysis proceeds by:

1. Examination of the peak value time block of data looking at peak values incurred in a consistent pattern [the peak values are (a) trendable and (b) useful for severity assessment];
2. Analysis for repeatable impacting presence related to turning speed of the machinery through converting from time as the independent variable to frequency (generally FFT applied to compute spectral data); and
3. Analysis of peak value time block of data employing the auto correlation⁹ methodology. The primary capability of this analysis tool provides the extraction of a periodic signal from a signal consisting of significant non-periodic noise.

To illustrate these three analysis steps, a peak value (PeakVue) time block of data acquired from a roughing machine gearbox in the steel industry will be used. The impact data versus time are presented in Figure 8. This data block contains 1024 data points.

The time between each of the vertical lines in the waveform represents the time for one revolution (467 RPM = 7.78 RPS; 1 rev = $1/7.78 = .1285$ sec = 128.5 msec). The F_{max} for this acquisition was set at 200 Hz; therefore the duration of each time

increment for which peak values were captured is the inverse of 2.56 times 200 or 1.953 msec ($1/2.56 * 200 = 1/512 = .001953$ sec). Note that the time block of 2.0 sec corresponded to 15.56 revs ($2 \text{ sec} * 7.78 \text{ RPS} = 15.56 \text{ revs}$).

Examining Figure 8, note the Pk-Pk impacting value observed over this time period was approximately 11 g's ($8.70 + 2.03 \text{ g's}$). Using the generic Alarm levels (to be presented in Table IV in section 5) for this machine speed, an "Alert" level of 1.85 g's and a "Fault" level of 3.7 g's at this speed of 467 RPM (7.78 RPS) would have been preset. Thus, this machine exceeded the "Fault" level by a factor approaching 3. In addition to the level of impacting, there seems to be a repeatable pattern of increased impacting at intervals of approximately every 2+ revs. The time spacing between impacts is short relative to time per rev. This pattern in the impact time waveform is typical for a defective roller (or multiple) passing in and out of the load zone at the rate of the cage frequency (FTF).

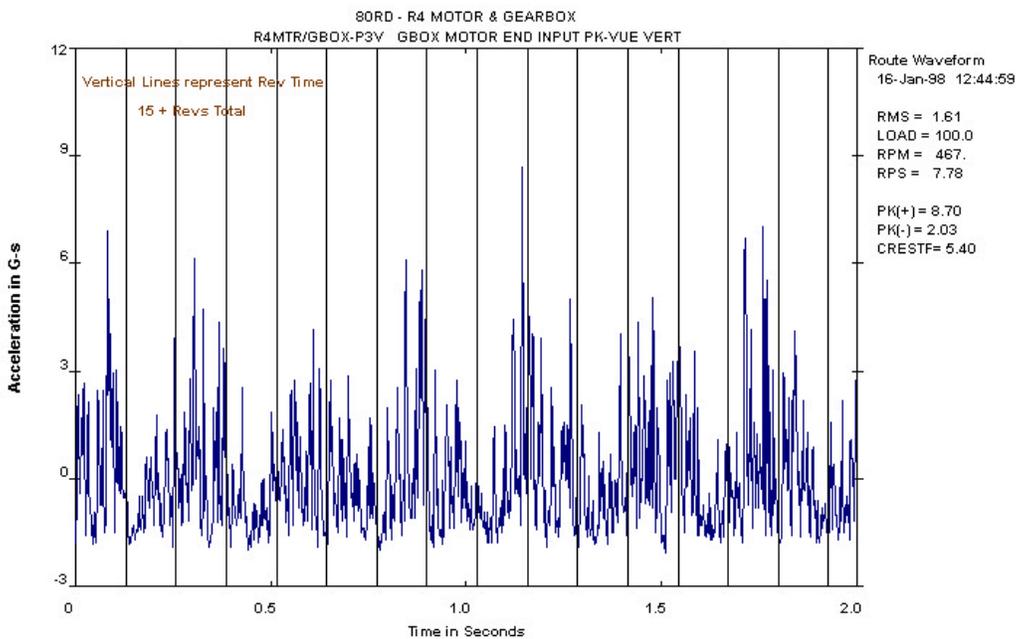


Figure 8. Impact time waveform from Roughing Machine gearbox consisting of 1024 data points at equal time increments of 1.953 msec ($F_{\max} = 200 \text{ Hz}$)

To obtain further verification of a roller defect, examine the PeakVue spectral data presented in Figure 9, computed from the impact time data block of Figure 8. In Figure 9, the roller defect at 40.6 Hz with harmonics are present ($BSF = 5.216 \times \text{RPM}$). The defect frequencies are sidebanded with cage (were clearly amplitude modulated in the impact time data block of Fig. 8). Additionally, the cage frequency at 3.429 Hz ($0.441 \times \text{RPM}$) and harmonics are easily identifiable.*

*This does not imply a cage defect is present but the impacting amplitude varies at a periodic (cage) rate.

It is not uncommon for a roller defect to manifest itself more strongly at two times roller defect since a rough area may impact twice per rev of the roller at sufficiently equal intervals of time for the impacting to be twice per rev of the roller (once on outer race and once on inner race). The spectral data in Figure 9 does not suggest two impacts equally spaced per rev of the roller since magnitude of spectra at two times fundamental roller spin (.13 g at 2 x BSF at 81.20 Hz) is significantly less than that at one times roller per rev (.28 g at 1 x BSF at 41.6 Hz).

To examine this and other aspects of periodicity, the auto-correlation coefficient^{**} was computed from the impact time waveform of Figure 8. This function is presented in Figure 10*. The independent variable here is time with a maximum value one-half of the original time (2 sec) in the impact time waveform block (1 sec in this case).

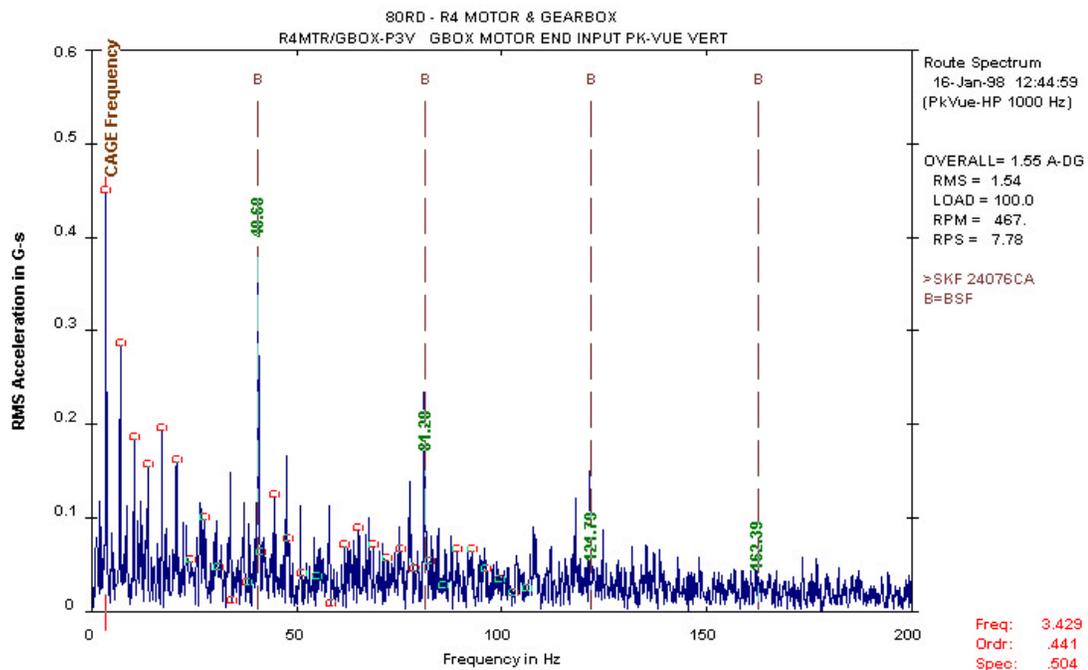


Figure 9. Spectral data computed from the impact time data block of Figure 8.

The maximum value for an auto-correlation coefficient function ranges between +1 and -1 (, see Appendix B). Perfectly correlated events will be + or -1 and totally uncorrelated events will be zero. The first event with significant correlation in Figure 10 appears at a time equal to the inverse frequency of the BSF (40.63 Hz = .0246 sec/roller rev). The highest amplitude event is at a time equal to the inverse of the cage frequency (1/3.429 = .292 sec = 1/FTF). The tricky part of reading the correlation function is to remember the independent axis is time, not frequency. For example, the time at which the second correlated event occurs in Figure 10 is the time duration between every other roller impact (the second harmonic in the spectral domain would correspond to 1/2 the time interval required for 1 rev of the roller).

*Appendix B is attached to this paper illustrating basic mathematics and properties of this function.

The important information presented in Figure 10 is confirmation that the only significant activity with repetitive occurrences are the impacts occurring once per rev of the roller which have variation in amplitude occurring at the repetition rate of the cage.

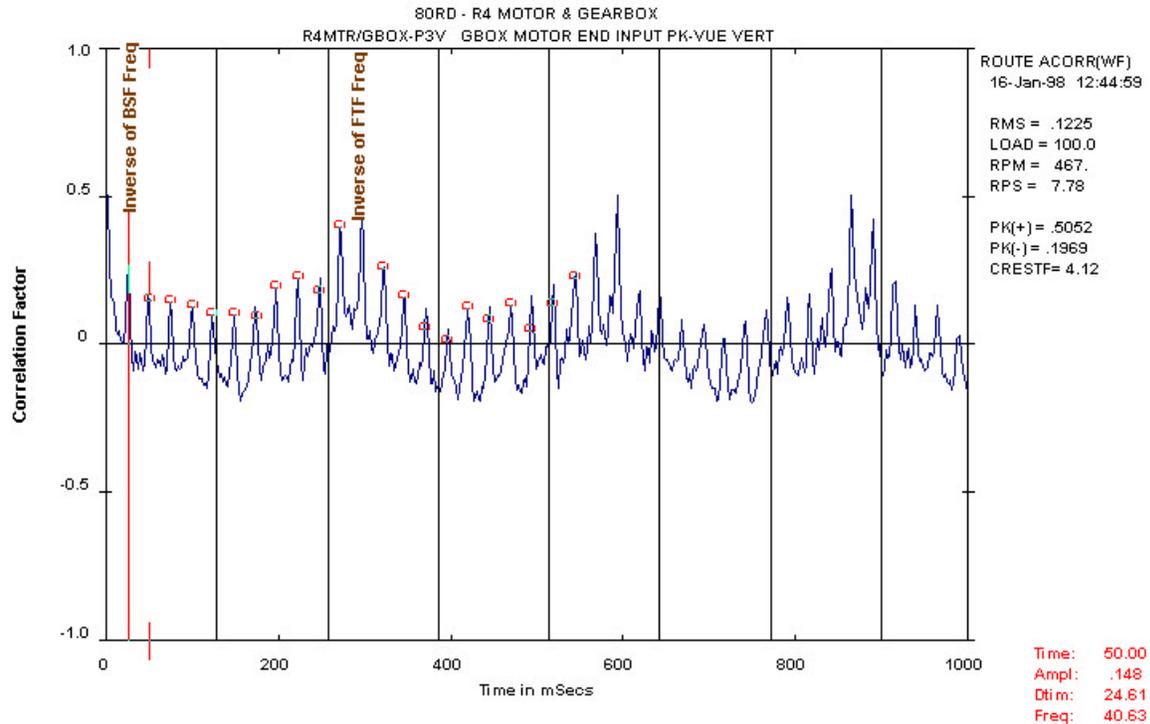


Figure 10. Auto-correlation coefficient data computed from the impact time data block of Figure 8.

3.3 Comparison between PeakVue and Demodulation

One of the main purposes for employing Amplitude Demodulation or peak value (PeakVue) analyses is to detect predominantly only the energy associated with impact or impulse events (vibration data includes energy generated by both impact and rotational sources). By removing the rotational component from both the time and spectral domains, the analyst is allowed to determine the amount of impact and/or friction-induced response, if any, is being generated in the machine being evaluated; and by using demodulation or PeakVue analysis, this allows identification of the source(s) of such impact. Unfortunately, noticeable problems have been encountered with demodulation sometimes missing the impact response almost entirely due to the following reasons:

- (a) choice of too low a frequency bandwidth that can result in significant loss of the amplitude signal (see Figure 12A);
- (b) choice of too low a bandwidth that can result in significant rotational energy still remaining in the demodulated signal (again, rotational energy must be removed to allow the analyst to examine only the amount of impact in the machine being evaluated);
- (c) choice of too high a high-pass filter that can result in loss of both the impact and rotational signals in the demodulated waveforms and spectra;

- (d) choice of too low a high-pass filter that can result in both the impact and rotational signals remaining in the demodulated waveforms and spectra (which, for all intents and purposes, defeats the very reason for taking high-pass filtered data; that is, leaving only the impact/impulse response data).

If any of these four problems are encountered when using demodulation, this will not only make it extremely difficult to apply meaningful alarms but can also prevent the analyst from even seeing a fault that is present. Ironically, if problem (d) is encountered, this can result in erroneously signaling a potential impact problem that is truly not necessarily present (with the undesirable end result in the analyst making an incorrect problem call on the machine or, possibly worse, taking unnecessary corrective actions).

Experience has indicated that when an analyst has both demodulation and PeakVue tools at his disposal (using the same analyzer/software system), it is usually best to opt for PeakVue. Of course, in no way does any of the foregoing information meant to convey that PeakVue measurements should replace vibration measurements; they should not. Instead, PeakVue data should be captured along with vibration data at each measurement location, thereby providing supplementary information that either of these tools by themselves may miss altogether.

Following below is information comparing PeakVue and demodulation tools. Following later in section 4.2 is information on recommended PeakVue measurement setup parameters; and section 5 provides information on PeakVue alarm levels that are suggested for various machine speeds and component faults.

When conducting stress wave analysis using an accelerometer as the sensor, it is generally an accepted practice to separate the stress wave activity from the normal vibration-induced activity by routing the analog signal from the sensor through a high-pass (or alternately a band-pass) analog filter. The information of interest is primarily the amplitude and dynamic behavior of the resultant signal. The signal generally consist of high frequencies (carrier) with lower frequency amplitude variation (modulation). An obvious method for extracting the amplitude variation from the carrier is a) rectification of the signal; and b) removal of carrier by passing the rectified signal through a low-pass filter (referred to as “enveloping” or “amplitude demodulation”). The output of the low-pass filter is an average value which not only is dependent on the amplitude of the signal, but also on the duty cycle relative to the averaging time established by the pass band of the low-pass filter. This dependence of the amplitude on the duty cycle makes establishing amplitude levels as generic indicators of severity level very difficult when using demodulation.

To remove the dependence of the amplitude on the analysis setup (low-pass filter setting cannot exceed the Nyquist frequency or $\frac{1}{2}$ the digitization rate), CSI introduced the peak value (PeakVue) analysis method for analyzing the stress waves. In lieu of invoking a low-pass filter, only peak (absolute) values are captured over each interval of time equal to the inverse sampling rate (sampling rate equals 2.56 times user-defined analysis bandwidth). For example, if F_{\max} is set at 1000 Hz, the analyzer will actually capture 2560 samples per second ($1000 * 2.56 = 2560$ samples/sec).

The choice of which frequency bandwidth is chosen can significantly affect the resulting amplitude response in a demodulated waveform (but will typically have only minimal effect on the PeakVue waveform amplitude response). To demonstrate the effect that changing the bandwidth has when using demodulation versus PeakVue on the digitally captured time waveform, a time signal was generated and reduced using each method, employing a LabVIEW™ program.*

The time waveform used for this comparative study was generated by taking the real time waveform shown in Figure 5 and repeating it seven times (the duration of the time waveform in Figure 5 is 40 msec with 4096 data points; therefore, the time waveform used for testing includes 280 msec with 28,672 data points). The resultant time waveform is presented graphically in Figure 11 (top). This waveform was sent through a high-pass, high order filter set at 1000 Hz. The resultant signal is present in Figure 11 (middle). The final step prior to demodulation or PeakVue analysis is to full-wave rectify the signal which graphically is presented in Figure 11 (bottom).

From Figure 5, the original signal has an absolute value of approximately 120 g's. When this signal passes through the high-pass filter and is rectified, the approximate 120 g level remains (Figure 11-bottom). The next step is to employ either demodulation or PeakVue on the rectified time waveform.

The results using an analysis bandwidth of 1000 Hz, 200 Hz, and 50 Hz are presented in Figure 12A for demodulation (left column) and in Figure 12B for PeakVue (right column). The low-pass filter used for demodulation is the anti-aliasing filter specified by the analysis bandwidth. Low-pass filters are not used in PeakVue. The specification of analysis bandwidth only changes the time increments where peak values are obtained. The time increment at 50 Hz bandwidth is 20 times greater than that for 1000 Hz bandwidth. That is, if using 800 lines or 2048 time domain samples, a bandwidth of 50 Hz would require 16 seconds to obtain a time block, whereas a 1000 Hz bandwidth would require only 0.8 second.

Figure 12 clearly shows that the chosen analysis bandwidth has a very significant effect on the remaining amplitude within the time waveform when using demodulation. Notice how the demodulated waveform amplitude drops over 70% from about 28 g's to 8 g's if the bandwidth is reduced from 1000 Hz to 200 Hz. Then, the signal almost disappears entirely when a bandwidth of 50 Hz is chosen, causing the amplitude response to drop to less than 1 g as shown in the bottom plot of Figure 12A.

On the other hand, Figure 12B shows that the PeakVue amplitude response remained at approximately 110 g's when either the 50 Hz, 200 Hz or 1000 Hz bandwidth was employed. Looking at Figure 12B, note that the selection of too low a bandwidth did result in a lower definition waveform which might result in loss of one or more frequency components in the resulting PeakVue spectrum. There are also effects associated with duty cycle that could make the variations worse (or better) than in the example shown.

* LabVIEW™ is a product of National Instruments Corporation.

Even though the complete analysis would consist of computing the spectra from the time waveforms presented in Figure 12 and looking for periodicity via peaks in the spectra, the very important data (quantifying the magnitude of friction-induced and/or impact events) to be used for severity assessment is lost using the demodulation method. In most cases, PeakVue amplitudes will be significantly higher than those of demodulation due to its signal processing methods. Also, Figure 12 shows that just the choice of the high-pass filter can result in drastically different amplitudes in demodulated waveforms whereas this filter will typically usually have only minimal effect on PeakVue amplitude response.

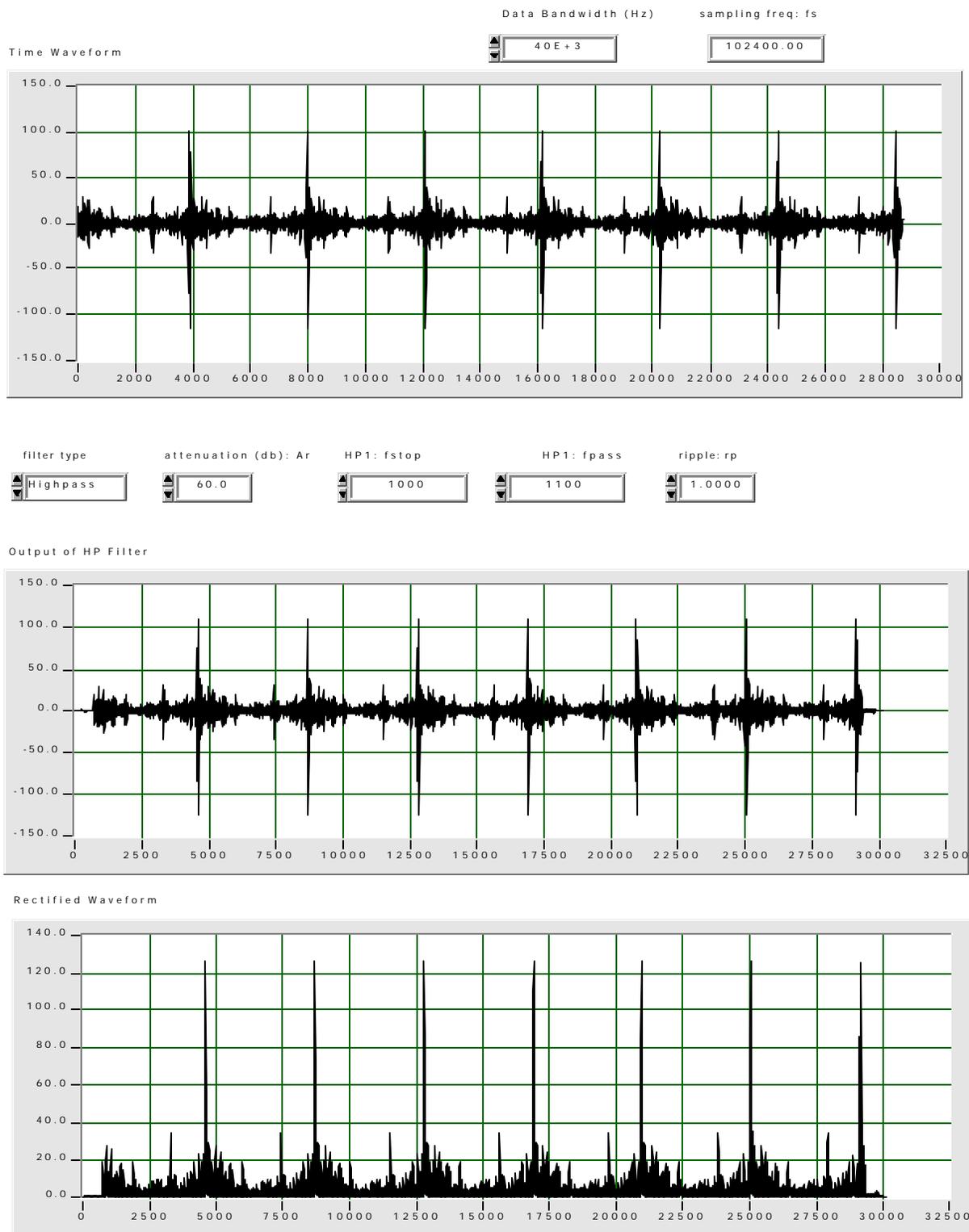


Figure 11. Graphical presentation of steps from initial time waveform (top) to passage through high-pass filter (middle) to full wave rectification (bottom) preceding demodulation of PeakVue

Fig. 12A. Demodulated Waveforms

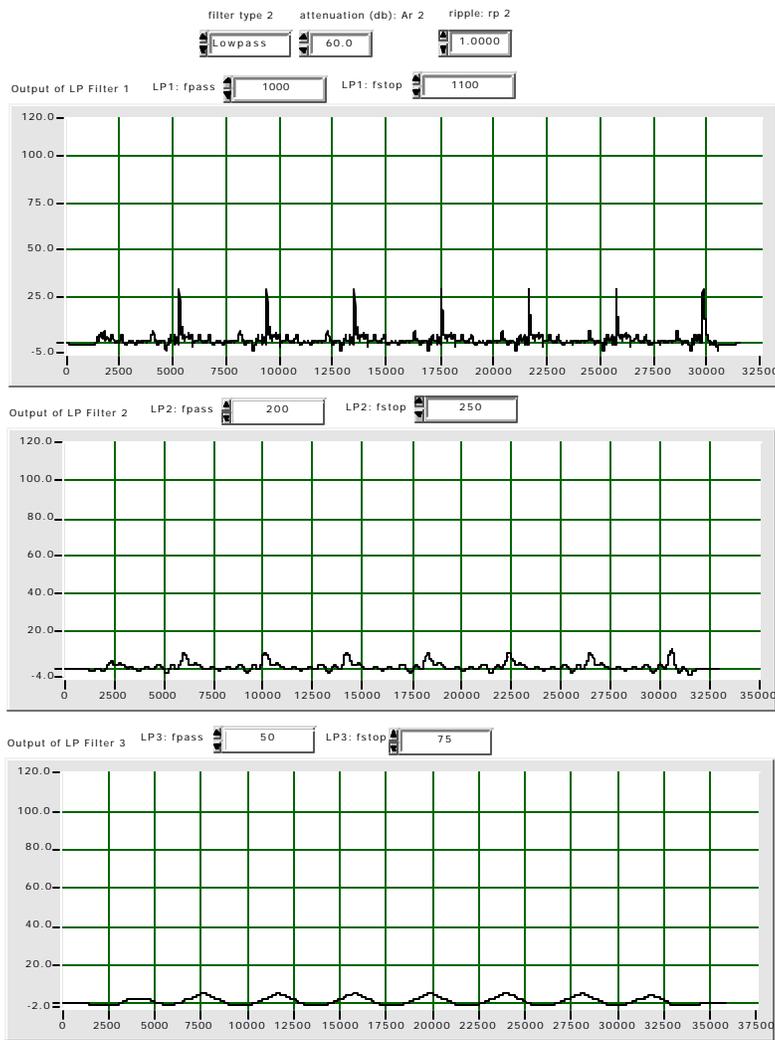


Fig. 12B. PeakVue Waveforms

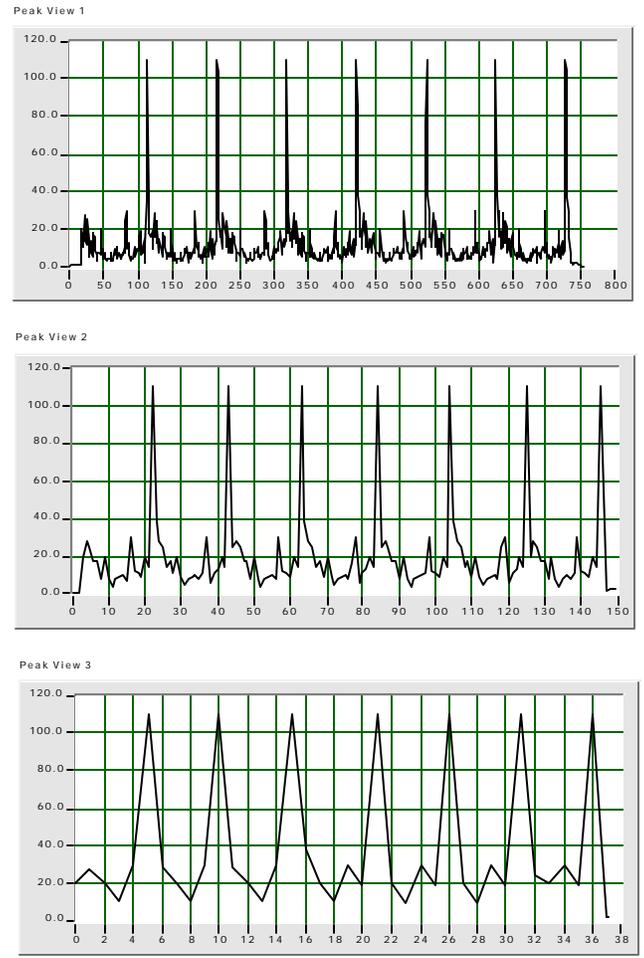


Figure 12. Time Waveforms for analysis bandwidth of 1000 Hz, 200 Hz, and 50 Hz for demodulation (left column) and PeakVue (right column)

4.0 Recommended Measurement Setup for Optimum PeakVue Measurements

4.1 Introduction

The purpose of this section is to provide proven guidelines to be used when establishing optimum PeakVue measurement setup parameters either when setting up a new PdM database or when refining one. These guidelines are based on extensive applications of PeakVue both within the field during Condition Monitoring surveys as well as in laboratory studies.

Typically, the energy from stress waves is concentrated within certain frequency ranges, depending on the type of fault present and certain operational parameters. Thus, it will be important to identify what these frequencies are so that the analyst can encompass them in his choice of the high-pass filter and analysis bandwidth (F_{max}).

When setting up for a PeakVue measurement, the analyst must determine the analysis bandwidth (F_{max}), the resolution or number of lines in spectral data, what type and how much averaging is needed for spectral data, the optimum high-pass filter to be employed, as well as the sensor (and mounting) to be used.

Given PeakVue data that has been acquired, the next action is then to alarm if a fault is present with an identification of fault location and severity. Finally, the corrective action which is recommended should be documented.

In the next subsection, the recommended setups for PeakVue data acquisition and sensor selection/mounting are presented. Table III in section 4.2.5 will provide a comprehensive tabulation of recommended PeakVue measurement setup parameters. Table V in section 5 will provide general "Alert" and "Fault" alarm levels that should be applied to various machines operating at different speeds.

In PeakVue, ALWAYS collect and store data in the sensor units (an accelerometer is recommended as the transducer of choice).

4.2 Recommended PeakVue Data Acquisition Setup Parameters

The primary objective of this section is to provide specific guidelines for recommended measurement setups needed to ensure optimum PeakVue data is captured. This will include the proper choice of high-pass filter, analysis bandwidth (F_{MAX}), number of FFT lines, number of time domain samples, number of averages, sensor selection and sensor mounting method. The choice of such parameters may be dependent on the type of fault for which one is looking (cracked gear tooth versus generalized tooth wear, for example) so this will likewise be discussed. This will be followed by selection of parameters for trending.

This section will include a comprehensive table that will tabulate optimum measurement setup parameters for machines operating at a wide range of speeds (ranging from low-speed machines at 60 RPM or less up through machines operating at speeds in excess of 10,000 RPM). This table will also be accompanied by a series of

notes that remind the reader of the importance of carefully mounting the transducer, document how measurement setup parameters must be modified to effectively evaluate gearboxes, etc.

4.2.1 Selection of High-Pass Filter

On the hardware that implements the PeakVue methodology, a finite number of band pass and high-pass filters are available from which to select. The choices currently available are presented in Table II. The high-pass (or band pass) filter selection is dependent on a) the analysis bandwidth (F_{max}); and b) on the frequency region where dominant energy is expected from the stress wave events due to potential faults that might be present.

**TABLE II.
BAND PASS AND HIGH-PASS
FILTER SELECTIONS²**

Band Pass	High-pass
20 Hz – 150 Hz	500 Hz
50 Hz – 300 Hz	1000 Hz
100 Hz – 600 Hz	2000 Hz
500 Hz – 1000 Hz	5000 Hz ¹
5000 Hz – 6500 Hz ¹	10,000 Hz ¹
	20,000 Hz ¹

NOTES:

1. Special precautions must be taking when mounting the sensor when using these filters due to the high frequencies involved (i.e., clean surface with no paint; flat rare earth magnet for 5000 to 10,000 Hz measurements; stud or adhesive mount for measurements above 10,000 Hz, etc.). Failure to take these precautions will likely result in loss of detection of all fault frequencies present at or above these frequencies in both PeakVue time waveform and spectral data.
2. Band Pass and High-pass filters given in Table II are the filter selections currently available in the CSI 2120 and CSI 2120A analyzers.

For selection of a high-pass filter, the corner frequency must be greater than or equal to the F_{max} set for that measurement point (if the user specifies a lower value, the firmware within the instrument will increase the filter setting to the next available filter). If there are multiple measurement points located on a single metallic enclosure (machine), e.g., a gearbox, then the analyst should ensure that all measurement points located on the machine use the same high-pass filter setting established for the highest analysis bandwidth (highest F_{max}). In gearboxes, if the calculation of 2.25 X Highest Gear Mesh calls for a high-pass filter falling between two of the available choices shown in Table II , the user should choose the next higher filter, not the closest filter to this calculated value (i.e., if the calculation calls for a high-pass of 1100 Hz, the user should choose 2000 Hz, not 1000 Hz, given the available high-pass filter choices in Table II).

For example, assume PeakVue measurements are to be taken on a triple reduction gearbox. This example gearbox has an input gear mesh frequency (GMF_1) of 2000 Hz; an intermediate gear mesh (GMF_2) of 1000 Hz; and an output gear mesh (GMF_3) of 500 Hz. Recall that the high-pass filter setting must be established for the highest F_{max} . Thus, the recommended high-pass filter in this case would be 2.25X GMF_1 which would call for a minimum of 4500 Hz (2.25 X 2000 Hz). Since the closest filter higher than 4500 Hz is 5000 Hz, this is the high-pass filter that should be specified for all measurements taken on each point located on this triple reduction gearbox.

Even though the same high-pass filter must be specified for all measurement locations on a gearbox, F_{max} can be changed at each point and should be optimized for each particular location using the information covered in section 4.2.2.

There are certain situations where a band pass filter should be used instead of a high-pass filter as discussed below and in Appendix A. If one of the available band pass filters listed in Table II is being used, the lower corner frequency of the high-pass filter component (that is, 20 Hz on the 20 Hz - 150 Hz band pass filter), the break frequency must be greater than or equal to the F_{max} used for that measurement point. Once again, the user should choose the band pass filter with the next higher break frequency, not the closest filter. For example, if the highest fault frequency was 200 Hz, and if a band pass filter is desired, the 500 – 1000 Hz selection should be chosen (not the 100 - 600 Hz).

In most applications, PeakVue should be set up to use high-pass filters rather than band pass filters. This would include the great majority of rolling element bearing, gear and lubrication faults for machines typically operating at 300 to 3600 RPM. Band pass filters might best be used on machines operating at either very low speeds or those with very high speeds and/or having forcing frequencies (i.e., gear mesh frequencies) located above approximately 7500 Hz. This often includes machines such as rotary screw compressors, centrifugal air compressors, etc.

Table III provides the recommended high-pass frequency for machines ranging from low speeds up to much higher speeds exceeding 10,000 RPM. It also lists many of the other measurement parameters that will be discussed below. Appendix A provides information on when and how Band Pass filters should be used in PeakVue measurements.

4.2.2 PeakVue Analysis Bandwidth (F_{max})

The bandwidth, maximum frequency, is determined by the highest possible fault frequency (also referred to as “highest forcing frequency”). In the absence of gear meshing, the inner race (BPFI) fault frequency generally is the highest frequency for rolling element bearings.. The bandwidth, F_{max} , should be set greater than 3 times BPFI (preferably 4 X BPFI).

The primary factors that influence the data acquisition parameter set, including the analysis bandwidth, are machine speed and the type of fault for which detection is

desired. As an example, consider a machine having rolling element bearings as the primary source for faults. The highest fault frequency will be the inner race. The number of rollers can cover a large range, but a large number of commonly used bearings will have less than 18 rollers. Hence the inner race fault will typically be less than 12 times running speed. It is desirable to have a minimum of three harmonics of this fault frequency within the analysis bandwidth; therefore an analysis bandwidth (F_{MAX}) of 40 orders would be a reasonable generic setup for a machine outfitted with rolling element bearings.

For gear mesh faults, the analysis bandwidth, F_{max} , should be set greater than two times gear mesh (preferably greater than three times gear mesh if 3 X GMF does not exceed 2000 Hz). If both 2.25X GMF and 3.25X GMF exceed 2000 Hz, it will be necessary to use the 5000 Hz High-Pass Filter, but special precautions pertaining to the mounting surface, mounting shape and cleanliness will demand close attention if a 5000 Hz High-Pass Filter is employed.

If there are multiple shafts within the gearbox, then a measurement point should be located on each bearing and the F_{max} set greater than twice times the highest gear mesh for the set of gears on that shaft (preferably at 2.25 X Highest GMF). However, it is important that the same high-pass filter is specified for all measurement locations at each point on a gearbox (high-pass should be set greater than or equal to 2.25 X Highest GMF); then, F_{max} can be changed at each point and should be optimized for each particular location using the information covered in this section (one F_{max} may have to be used for evaluating bearings, misalignment, eccentricity, etc., and a higher F_{max} used for evaluating the gears).

Table III provides the recommended analysis bandwidth, F_{max} , for machines running at various speeds. It likewise covers how F_{max} should be set up for both rolling element bearings and for gear sets.

4.2.3 Resolution and #Time Domain Samples

After selecting the high-pass filter and bandwidth for data acquisition, the next parameter to be selected is the frequency resolution. The resolution is set by specifying the number of lines, e.g., 400, 800, 1600, etc. The controlling criterion is to provide sufficient resolution to resolve the lowest possible fault frequency. For rolling element bearings, the lowest fault frequency is the cage frequency (FTF) which is in the proximity of 0.4 times shaft speed (i.e., the cage will complete one revolution for approximately every 2.5 revolutions of the shaft). It is important to have sufficient resolution to clearly resolve the cage frequency. This translates into having a time block of data capture 15-20 revolutions.

As a minimum, the time block of data must include six periods for the fault frequency to be resolved. Thus, to ensure that the cage frequency is displayed in the PeakVue spectrum, a minimum of 6 times 2.5 or 15 revolutions of the shaft speed must be included within the time block of data (preference is 20 revolutions of the shaft speed).

A convenient formula for computing the number of shaft revolutions contained within a time block of data is:^{*}

$$\# \text{ of Shaft Revolutions} = \frac{\text{No of lines}}{F_{\max}} \text{ (in orders)}$$

As an example, using an F_{\max} of 40 orders, then a 400 line analysis would have 10 revs within the time block of data; 800 line analysis would have 20 revs., etc.

If the maximum frequency is set at 40 orders, then the next choice is to select the number of FFT lines. If 400 lines are chosen, then the number of revs captured would equal 400/40, or 10 revolutions, which is not sufficient. If a selection of 800 lines is made, then the number of revs captured is a 800/40 or 20 rev which provides adequate resolution.

Table III includes recommendations on the number of FFT lines. Typically, for general machinery operating at speeds of 600 to 3600 RPM, 800 lines are recommended (for motors, fans, pumps and other general machinery). A resolution of 800 lines corresponds to 2048 time samples (#Samples = 2.56 X #Lines).

However, if the operating speed exceeds 4000 RPM, 1600 lines are recommended. In addition, if PeakVue data is taken on a gearbox, it is generally recommended to capture a minimum of 1600 FFT lines (corresponding to 4096 time samples).

4.2.4 Number of Averages

After selecting the bandwidth (maximum analysis frequency or F_{\max}) and resolution (number of lines), the next task is to select the number of time blocks to average. Averaging is strictly an exercise to improve signal-to-noise in the spectral data only, i.e., the time block of data is the final block used for the spectral calculation (analyzers only store the final time block captured, no matter how many averages have been requested for the spectrum unless synchronous time averaging using a trigger is invoked).

In vibration measurements, it is most always a good idea to use multiple averages in order to improve the signal-to-noise ratio in the spectrum (does not affect the waveform without synchronous time averaging). Improving the signal-to-noise ratio will enhance the appearance of true periodic frequencies while suppressing random, non-periodic components normally associated with “noise”. In general, spectral noise varies with the square root of the number of averages. That is, if the user increases the number of averages from 4 to 16 averages (4X), this should reduce spectral noise by 50%. Again, increasing the #averages will not affect the vibration waveform whatsoever since only the final time block is retained.

^{*}This very convenient formula was offered by Mr. Clyde Bridges of Mitsubishi Polyester Film, LLC in Greer, SC.

Surprisingly, in the case of PeakVue, it is not a good idea to acquire more than one time block. Hence only one average is recommended in PeakVue measurements. The primary reason for this is that the PeakVue time waveform has equal importance to the PeakVue spectrum. Therefore, it is better to spend the extra time that would be required for averaging to increasing the resolution by increasing the number of lines instead.

As a matter of fact, the authors have actually encountered times when PeakVue has been caused to actually miss detecting problems within bearings that had proven faults, simply by increasing the number of PeakVue averages. In one case, the number of averages was increased up to 6 with the result that the bearing fault frequencies (BPFO in this case) that were very pronounced in one time block data were nearly eliminated by the averaging process, primarily because it was later learned that the machine was slowly but continuously changing in speed during the averaging.

A much better result in reducing PeakVue spectral noise content can be achieved by increasing the number of FFT lines during PeakVue measurements. In fact spectral noise elimination varies directly with the number of lines. For example, if the user increases the number of lines from 800 to 1600 lines, PeakVue spectral noise should decrease by 50%. As another example, a two-block, 1600 line analysis is far better for PeakVue than is a four-block, 67% overlap averaging measurement (same amount of time for data acquisition required as that required for two-block, 1600 line).

Thus, Table III shows that only 1 average is generally recommended for typical PeakVue measurements. The user should not acquire more than 1 average unless he is assured that his machine is truly constant speed (i.e., as would be the case with a synchronous motor).

4.2.5 Tabulated Procedure for Recommended PeakVue Measurement Setups

Table III provides a comprehensive tabulation of recommended PeakVue measurement setup parameters. This is based on proven field tests in making a series of Laboratory trials and Condition Monitoring measurements when PeakVue data was acquired along with vibration data (and often during other condition monitoring data acquisition such as ultrasound, infrared thermography, oil samples, etc.).

It is important that the analyst pay close attention to the notes listed below Table III. These notes include information on how such setups should be altered when taking PeakVue data on various other machine components such as gearboxes. These notes also include recommendations that should be followed if a user hopes to obtain meaningful, repeatable PeakVue measurements.

It is also important to point out Table III is intended to be a "Guide" when establishing PeakVue measurements in a condition monitoring database. Occasionally, the user will encounter special machinery or operating conditions that will mandate setting up such measurements somewhat differently. Examples of such special measurement situations are included in Appendix A when making measurements on a low-speed component such as the felt of a paper machine or on high-speed rotary screw or centrifugal air compressors when studies to date indicate it might be better (in these cases) to employ band pass rather than high-pass filters (see Appendix A for explanation).

TABLE III. RECOMMENDED PEAKVUE SETUP PARAMETERS FOR DETECTING ROLLING ELEMENT BEARING FAULTS ¹

RPM	HI-PASS FILTER ⁶	RECOMMENDED F_{MAX} ³		RARE EARTH MAGNET	RECOMM. #AVGS	MIN. RECOMM. LINES
		KNOWN BEARING	UNKNOWN BEARING			
0-700	500 Hz	4X BPF ²	40X RPM ²	2-Pole ⁵	1	800
701-1500	1000 Hz	4X BPF	40X RPM	2-Pole	1	800
1501-3000	2000 Hz	4X BPF	40X RPM	Flat	1	1600
3001-4000	2000 Hz	4X BPF	30X RPM	Flat	1	1600
4001-UP	5000 Hz ⁵	4X BPF	40X RPM	Flat	1	1600

NOTES:

1. This table was developed after conducting extensive research, laboratory trials and field tests (both within Condition Monitoring annual contract measurements and during diagnostic investigations). Use it as a guide when setting up databases (either in a Condition Monitoring program or on a Diagnostic project).
2. If using PeakVue measurements to detect **Gear Faults**, typically use 1600 Lines along with a High-Pass Filter exceeding about 2.25X GMF unless this frequency exceeds 2000 Hz (note that the optimum PeakVue High-Pass Filter would be specified at 3.25X GMF if this calculated frequency does not exceed 2000 Hz; if both 2.25X GMF and 3.25X GMF exceed 2000 Hz, it will be necessary to use the 5000 Hz High-Pass Filter, but special precautions pertaining to the mounting surface, mounting shape and cleanliness will demand close attention if a 5000 Hz High-Pass Filter is employed). However, if the 5000 Hz filter is chosen, the user must follow the guidelines of notes 4 and 5 below. These preparations will allow you to use a High-Pass Filter of 5000 Hz. If there are multiple shafts within the gearbox, then a measurement point should be located on each bearing and a high-pass filter used that is greater than twice times the highest gear mesh for the set of gears on that shaft (preferably at 2.25 X Highest GMF). F_{max} can be changed at various points on the gearbox (as per the F_{max} guidelines given in section 4.2.2).
3. **F_{MAX} cannot exceed the High-Pass Filter** (however, it is permissible for F_{MAX} to equal the High-Pass Frequency).
4. Paint should be cleaned off mounting surface. In all cases, mounting surfaces should be clean and free of dirt/oil/foreign particles. Surface should be smooth. If more than one layer of paint is present, the paint can significantly dampen the resulting PeakVue signal.
5. Do not use a 2-Pole Magnet when using a High-Pass Filter above 2000 Hz. Doing so will result in loss of impact response data. Use a Flat Rare-Earth magnet mounted on a flat surface and insert a thin layer of grease, silicone or wax between the magnet and the mounting surface when using a High-Pass Filter of 5000 Hz or greater. Field tests have proven that if fault frequencies are present above approximately 3000 Hz, which are detected by a flat rare earth magnet, such frequencies can be missed altogether by use of a 2-pole magnet when making PeakVue measurements. (2-pole magnets are often referred to as "dual rail" magnets).
6. In most applications, PeakVue should be set up to use high-pass filters rather than band pass filters. This would include the great majority of rolling element bearing, gear and lubrication faults for machines typically operating at 300 to 3600 RPM.

4.2.6 Trending of PeakVue Data

The primary PeakVue parameter which should be used for trending PeakVue measurements is the parameter referred to as “Pk-Pk Waveform” that is available in the PdM software. Extensive field experience within PdM programs has shown the trending of PeakVue “Pk-Pk Waveform” has proven to be a reliable indicator for detection of faults caused by impact or impulse events (bearing, gear, lubrication, cavitation and related faults).

The "Pk-Pk Waveform" parameter is not dependent on the analysis bandwidth or duty cycle of the events (note that the term “duty cycle” refers to the number of events per cycle or revolution; e.g., an inner race fault occurs more frequently than does a roller fault). This lack of dependence on analysis bandwidth or duty cycle permits generic alarm levels to be established (presented in Section 5.0).

In addition to the trending of the Pk-Pk Waveform parameters, others obtained from the PeakVue spectral data are recommended as listed below. These will be dependent on the duty cycle and hence it is difficult to establish generic levels for “Alert/Fault” alarm levels. Accordingly, Alert/Fault alarm levels should be based on amplitudes relative to baseline levels for parameters such as the digital Overall alarm level (which is referred to as the “Overall” level in the “Alarm Limit” point setup section within RBMware software as shown in Figure 17 (for example, if a baseline spectrum captured with the machine in good operating condition had an amplitude of 2 g's, it might be a good idea to set the “Alert” Alarm at approximately 4-6 g's and the “Fault” Alarm at about 8-12 g's for this digital Overall).

A sample Trend plot is shown in Figure 36 that shows the change in the “Max Peak Waveform” amplitude versus time. Note that both the “ALERT” and “FAULT” alarms were eventually exceeded on this gearbox outfitted with rolling element bearings turning at an operating speed of only 156 RPM (2.60 RPS), making it necessary to take this low speed into account when establishing alarm levels.

Following the selection of the bandwidth (maximum frequency), resolution (number of lines), and averaging mode, the next task is to define the parameters to be trended. There are five which have proven to be beneficial. They are:

1. True Peak-to-Peak level from the PeakVue time waveform (this value has proven to be the most reliable PeakVue trending value or indicator of impending problem conditions or faults for reasons discussed below);
2. The Digital Overall of the entire PeakVue spectrum after the waveform signal has passed through the high-pass filter and is submitted to the FFT algorithm (absolutely not the analog overall of PeakVue);
3. Energy in 4-10 synchronous shaft revolutions;
4. Energy in bands surrounding bearing fault frequencies of BSF, BPFO, and BPF1. If fault frequencies not known, then use two generic bands based on

probable number of rollers in bearing. Specifically, for BPFO* use a band of $[0.25 \times N \text{ to } 0.52 \times N]$ orders; for BPFI, use a band of $[0.48 \times N \text{ to } 0.75 \times N]$ orders (where N equals the number of rolling elements);

5. Energy from spectral data for sub-synchronous orders (e.g., 0.2 to 0.8 orders).

When monitoring gearboxes, it is very important to include two times gear mesh in the analysis bandwidth. This is to capture possible backlashing in addition to scuffing/scoring on the addendum and dedendum. The high-pass filter should be set higher than anticipated vibration frequencies. For certain gearing faults this could be at three times gear mesh. The problem here is it will often force a high-pass filter set at 5,000 Hz (next choice past 2,000 Hz). If impacting is occurring in a gearbox, it is between massive parts and dominant energy will be in the 1 to 5 kHz range. Additionally, the higher frequencies introduced will experience significant attenuation because of losses from gear teeth to the outer surface where the sensor is mounted. Therefore it is recommended that the high-pass filter be set slightly greater than 2 times the highest gear mesh in the gearbox.

The higher frequency components generated from lubrication faults will experience significant attenuation during their propagation to the outer surface of the gearbox. Therefore, consideration should be given to selecting a mounting scheme of a flat magnet or better (see Figure 7).

It is recommended that a measurement point be positioned at each bearing on the gearbox. The high-pass filter setting should be same for each measurement point. The resolution and analysis bandwidth will change. The key is to include up to at least two times gear mesh for any gears on the shaft being monitored and to provide sufficient frequency resolution to resolve that gear mesh being modulated (sidebanded) with either (hence lowest speed) shaft on the gearbox.

For bearings within the gearbox, the same rules apply relative to bandwidth and resolution as presented in first of this section. The only difference is the high-pass filter will remain the same, i.e., greater than two times the highest gear mesh within the gearbox.*

The parameters recommended for trending from PeakVue data from gearboxes are as follows:

1. True Peak value from the PeakVue time waveform;
2. Energy surrounding one times gear mesh and two times gear mesh. The width of the band should include a minimum of ± 3 times the highest speed shaft involved in the gear mesh.
3. Energy of synchronous harmonics of shaft speed (for each shaft speed).

*This band will include BSF for many (not all) bearings. For those not included, the sub-synchronous band will cover since BSF faults are modulated with cage.

*If this forces the high-pass filter to exceed 5 kHz, then the high-pass filter should be replaced with a band pass filter which excludes 1 and 2 times any gear mesh within the gearbox.

It is not practical to provide specific values for the recommended trend parameters. Instead, alarm levels should be based on historical generic data for the recommended trend parameters computed from the spectra. Thus, values should be specified based on data acquired when baselining (assumed baseline from healthy machine). One baseline established a reasonable multiple for baseline values is a factor of 3 for alert level and a factor of 6 for faults level for each trended parameter. These multiplication factors have proven acceptable for several cases, but should be considered for further refinement as a database evolves for specific machines.

4.2.7 Selecting the Optimum Sensor for PeakVue Measurements

Due to the dispersion 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 (relative to normal vibration monitoring) of an issue than is mounting the sensor in or near the load zone with the caution that we are monitoring “waves” and hence, must always be cautious of encountering nodal points which can occur due to multi-path transmission and in the vicinity of sharp corners, etc.

The bending stress waves introduce a ripple. Hence any sensor which is sensitive to absolute motion occurring at a high rate would suffice, providing it has sufficient frequency range and amplitude resolution capabilities. Therefore, this sensor could be an accelerometer with sufficient bandwidth, an ultrasonic sensor, a strain gauge, piezoelectric film, et al. The primary motivation behind stress wave monitoring is to acquire information for machine health monitoring which is the motivation behind vibration monitoring in a predictive maintenance program. By far, the most common sensor employed in vibration monitoring is the accelerometer; therefore the sensor of choice for stress wave monitoring is the accelerometer. The requirements for this sensor will be sufficient bandwidth, amplitude resolution and appropriate sensitivity.

The bandwidth of an accelerometer is dependent on (1) its design and (2) the manner in which the accelerometer is attached to the surface. The general effect that different mounting schemes have on the sensor bandwidth is presented in Figure 7 (sensor becomes entire system attached to the surface).

Typically, standard 100 mv/g accelerometers can be used for most all PeakVue measurements (including on low speed machines) since PeakVue information typically exceeds 500 Hz (30,000 CPM). There are special cases where either a higher sensitivity or lower sensitivity accelerometer might be needed. For some applications on very slow machinery (less than 60 RPM), the accelerometer used in making PdM route measurements will need to have higher sensitivity (500 to 1000 mv/g). These can be used for PeakVue measurements providing the sensor resonant frequency exceeds 15,000 Hz. On the other hand, for high-speed machinery (or machines that generate forcing frequencies greater than approximately 10,000 Hz), the stress wave activity may have amplitudes exceeding the linear range of the sensor to the point of over driving the sensor. In these cases, it is recommended that the smaller 10 mv/g sensors be used

for PeakVue high-speed applications. In general, a 10 mv/g sensor would be the sensor of choice for machines exceeding the 3600 RPM range.

4.2.8 Selecting the Optimum Sensor Mounting for PeakVue Measurements

Just as is the case with vibration, the transducer mounting method and the mounting surface can likewise adversely affect the PeakVue measurement (painted versus unpainted surface; flat versus curved surface, etc.). And, it will be shown that either the mounting surface or the choice of the transducer mounting method can limit how high an F_{MAX} or a high-pass filter can be chosen, thereby limiting PeakVue's ability to detect a number of faults it otherwise could reveal if required preparations of the surface and transducer mount are made that would allow higher frequency measurements to be reliably made and a greater high-pass frequency to be employed.

The variability in the response for the hand-held probe is significant. Hence, due to poor bandwidth and variability, this method of sensor mounting for stress wave detection should be avoided. Hand-held accelerometers provide relatively poor stress wave information due to the very limited frequency capabilities provided by a hand probe as shown in Figure 7 (note importantly that a hand held probe will miss most all data above approximately 2500 Hz where stress wave energies for most faults is typically concentrated).

Method 5, dual rail magnet, has been found to be useful for stress wave detection in many applications, with the precaution that the magnet must be placed on a clean¹ smooth surface (dual rail magnets are sometimes referred to as "2-pole magnets"). There should be a minimum of dual line contact made between the magnet rails and the assumed curved surface of the machine. Limitations on using this mounting scheme will be addressed later in this paper.

A flat, rare earth magnet will capture more meaningful PeakVue data than will a dual rail magnet if mounted on a flat, reasonably clean surface. This is particularly the case if either a frequency bandwidth (F_{max}) greater than approximately 3000 Hz (180,000 CPM) is taken, or if a high-pass filter greater than 2000 Hz is used. If either of these two conditions exists, actual field tests have proven that if fault frequencies are present above approximately 3000 Hz which are detected by a flat rare earth magnet, such frequencies can be missed altogether by a dual rail magnet when making PeakVue measurements.

¹ Painted surfaces should be avoided. Thick paint will totally block out stress waves exciting the sensor and clearly should be removed

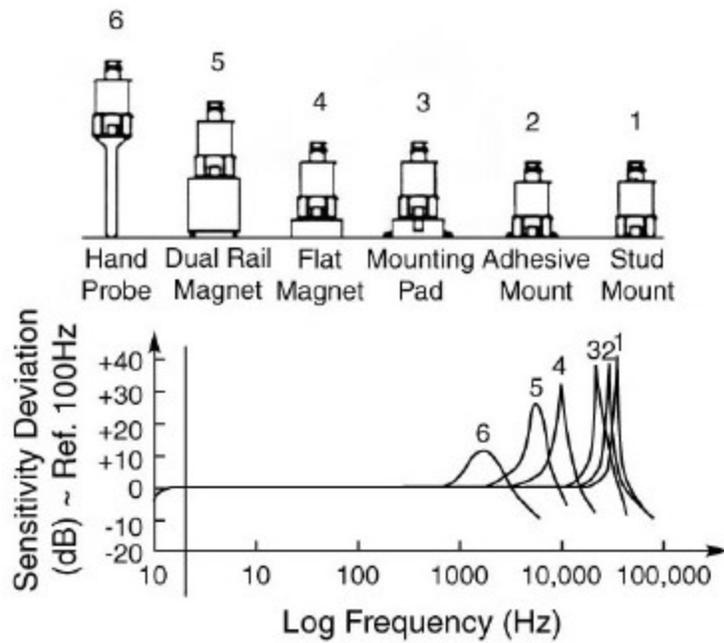


Figure 7.² Effect on frequency bandwidth of accelerometer mounting scheme to a machine surface (Figure 7 is repeated here for easy referral by the reader)

² Fig. 7 is taken from the IMI Industrial Vibration Catalog, CAT.600, page 129.

4.2.9 Setting up PeakVue Measurements within CSI's RBMware™ Software

The purpose of this section is to clearly demonstrate how PeakVue measurement points should be set up in CSI's RBMware condition monitoring software. Typical setups for both Analysis Parameter ("AP") and Alarm Limit ("AL") sets are covered and are established as per the guidelines provided in Table III within section 4.2.5.

Figures 13 thru 17 are taken directly from RBMware and show a PeakVue Analysis Parameter ("AP") setup for a machine running within the 1500-3000 RPM range (note the point naming convention which identifies this configuration is established for a gearbox input shaft being driven at a nominal speed of 1800 RPM).

Figure 13 shows the "Spectrum Parameters" tab for this PeakVue measurement point. Note that an analysis bandwidth (F_{max}) of 40 orders is specified (this lower frequency PeakVue spectrum will be used to detect bearing problems along with possible impact-induced bearing lubrication problems (as per Table V, sections A and C); another PeakVue measurement will be required on the gearbox with a much higher F_{max} set to at least 2.25 X GMF (preferably 3.25 X GMF, noting the sensor mounting precautions in Table V)) to check for possible tooth wear, cracked teeth, backlash problems, etc. as per Table V, section B). Figure 13 also shows a frequency resolution of 1600 lines is specified since this data is to be taken on a gearbox that is outfitted with rolling element bearings (see Table III, note 2, pertaining to a gearbox). Also, note that 1 average is specified and the low frequency cutoff is set at 0 Hz (to allow for proper calculation of the digital PeakVue Overall).

Periodic Analysis Parameter Set [020] - DB26

Spectrum Parameters | Signal Processing Parms | Waveform Parameters

Set Description: PeakVue 40 Orders 1800RPM 2000Hz

Spectral Frequency Setup : Order

Low Frequency Signal Conditioning Limit (CPM): 0.0

Upper Freq (CPM/ORDER) : 40.0

Lower Freq (CPM/ORDER) : 0.0

Number of Lines : 1600

Number of Averages : 1

Spectral Averaging Mode: Normal Mode

Window Type : Hanning

Spectral Weighting : None

Perform 1/3 Octave Analysis?

Number of Analysis Parameters: 1

Figure 13. "Spectrum Parameters" Tab of PeakVue Analysis Parameter Setup within RBMware Software

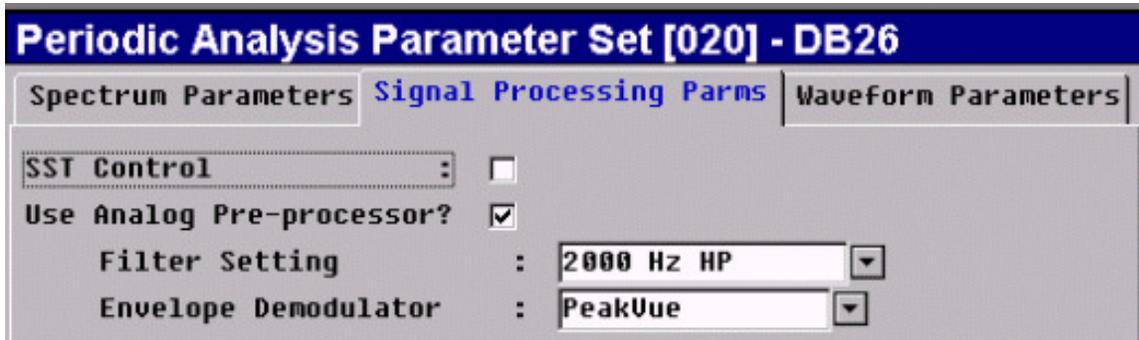


Figure 14. "Signal Processing Parameters" Tab of PeakVue Analysis Parameter Software

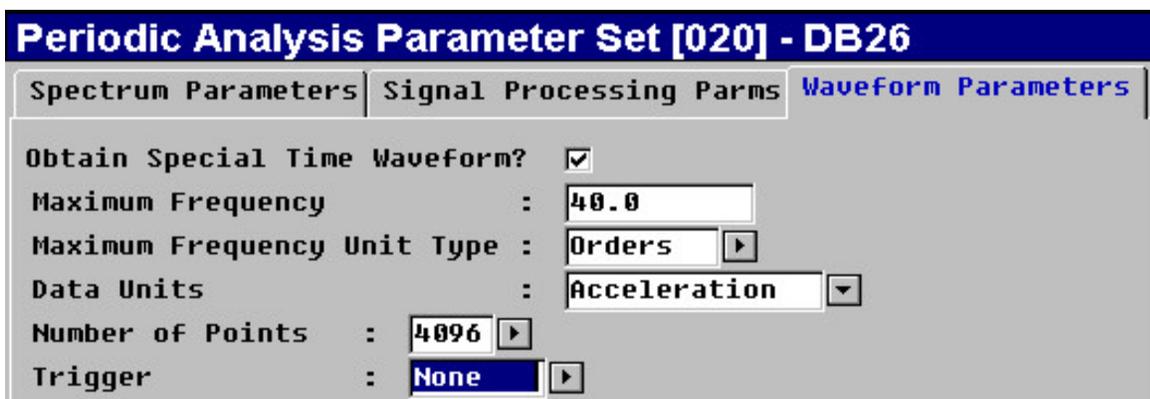


Figure 15. "Waveform Parameters" Tab of PeakVue Analysis Parameter Software

Analysis Parameter Set 20 - PeakVue 40 Orders 1800RPM 2000Hz - DB27				
Description	Parameter Units	Type of Parameter	Lower Frequency	Upper Frequency
FULL BAND	ACCEL	P-P Wave	0.0	0.0

Figure 16. "Pk-PK Waveform" Setup for True Peak-Peak Amplitude taken from PeakVue Waveform (Note "P-P Wave" Setting & Zeros in "Frequency Columns")

Figure 14 shows the recommended Signal Processing Parameters for this particular PeakVue measurement setup (again, as per Table III). Note that “Use Analog Pre-processor” must be checked. Also, the filter setting on the proper high-pass (or band pass filter must be specified (2000 Hz as per Tables II and III). Finally, the “Envelope Demodulator” should be set on “PeakVue”.

Figure 15 shows the “Waveform Parameters” setup. Here, it is sometimes necessary to obtain a special PeakVue time waveform in order to display the proper number of revs using the recommended setup information, along with the helpful equation in section 4.2.3 relating the number of Revs to the #FFT Lines and Fmax (in orders). As stated before, acceleration should be the unit of measure for PeakVue measurements. The user should then base the “Number of Points (or time domain samples)” based on the number of lines specified in Table III, depending on whether or not a gearbox, a high-speed machine, or another special machine type or component is being evaluated (recalling that $\#Points = 2.56 * \#Lines$). In this case, 4096 points are entered in order to agree with the #Lines in Figure 13 (and as per Table III for an 1800 RPM machine).

Figure 16 next shows that it is only necessary to establish one band (commonly called “FULL BAND”) when setting up the PeakVue AP band set. Here, note that it is important that “P-P Wave” is chosen for the “Type of Parameter” and that no frequencies are assigned to the “Lower and Upper Frequency” columns. This will assure the user that only the True Pk-Pk value is captured from the PeakVue waveform since the Trend Parameter section of this paper (Section 4.2.6) clearly states **“The primary PeakVue parameter which should be used for trending PeakVue measurements is the parameter referred to as “Pk-Pk Waveform”.**

Periodic Alarm Limit Set [102] - DB28						
Set Description: PEAKVUE 3G 1800 RPM						
Maximum Deviations (Bs): 5.0						
	UNITS CODE	ALARM TYPE	FAULT 'D'	ALERT 'C'	WEAK SIDE 'Lo/Hi'	BASELINE RATIO 'Br'
OVERALL	ACCEL	DU-A	2.0	1.0	0.0	5.0
PARAMETER 1	ACCEL	DU-A	6.0	3.0	0.0	5.0
PARAMETER 2	VELOC	DU-A	0.0	0.0	0.0	4.0
PARAMETER 3	VELOC	DU-A	0.0	0.0	0.0	4.0
PARAMETER 4	VELOC	DU-A	0.0	0.0	0.0	4.0
PARAMETER 5	VELOC	DU-A	0.0	0.0	0.0	4.0
PARAMETER 6	VELOC	DU-A	0.0	0.0	0.0	4.0
PARAMETER 7	HFD	DU-A	0.0	0.0	0.0	2.0
PARAMETER 8	VELOC	DU-A	0.25452	0.09898	0.0	1.6
PARAMETER 9	VELOC	DU-A	0.25452	0.09898	0.0	1.6
PARAMETER 10	VELOC	DU-A	0.25452	0.09898	0.0	1.6
PARAMETER 11	VELOC	DU-A	0.25452	0.09898	0.0	1.6
PARAMETER 12	VELOC	DU-A	0.25452	0.09898	0.0	1.6

Figure 17. PeakVue Alarm Limit Setup within RBMware Software

Figure 17 then shows what amplitudes should be entered into the PeakVue Alarm Limit Setup. Note the naming convention employed for this “AL” set, relating it to the “AP” set just established for this measurement point. **Note that it is only necessary that entries be made into the first two rows of a PeakVue AL set (only in the “OVERALL” and the “PARAMETER 1” rows).** It is important to point out that the “OVERALL” value is the Digital Overall of the entire PeakVue spectrum after the waveform signal has passed through the high-pass filter and is submitted to the FFT algorithm (thus, it includes only the RMS amplitude calculated from the spectrum). On the other hand, the “PARAMETER 1” value is derived from the True Peak-to-Peak value from the PeakVue time waveform. This will often mean that there will be a considerable difference in these two “overall” levels, with the “PARAMETER 1” value often being 3 to 5 times higher (assuming the AP set for Parameter was properly set up with “P-P Wave” chosen for the “Type of Parameter”). Because of this, “PARAMETER 1” (called “FULL BAND” in this case) is most always a more trendable and more reliable PeakVue magnitude than is the digital overall value.

5.0 Recommended PeakVue Alert and Fault Alarm Levels

This important section will show how to apply PeakVue alarm levels to various machine components and operating speeds. Studies have been conducted to determine the effect on PeakVue amplitudes of certain faults occurring on specific components and how they can generate very different amplitudes. Thus, a table will be provided to allow specification of alarms to be applied to the PeakVue waveforms based on the “Pk-Pk Waveform” value within the software (amplitudes in PeakVue spectra have not been found to be nearly as good an indicator of problem severity as has the PeakVue waveform). Thus, when an alarm is violated in the waveform, the reader will be instructed to next refer to the PeakVue spectrum to determine the cause of the alarm violation.

To set Alert/Fault levels from a theoretical perspective only for stress wave analysis would not be feasible. However, an analysis of many cases does permit generic setting of levels based on experience. These PeakVue alarm levels that are recommended in this paper should be viewed as an approximate level which may change for specific applications.

When setting Alert/Fault for normal vibration analysis, they generally are set based on the spectral data. For stress wave analysis, the variation in spectral data can vary significantly due to the duration of events and rate of occurrence which leads to large variations in duty cycle, i.e., the fraction of time, relative to the time for a few revs, occupied by the stress wave can be small. Since spectral data is an RMS value, serious problems could have relatively low spectral (RMS) values.

The parameter that can be correlated with severity levels of a fault is the “Pk-Pk” value of the impacting (PeakVue) time waveform. The qualifying parameter is the speed of the machine. For bearing faults, sufficient experience permits the setting of generic “Alert/ Fault” alarm levels. The faults (impacting) occurring on the inner race will see more attenuation than those on the outer race. Hence it is recommended that Alert/Fault levels be set up for the inner race. If the fault is identified to be the outer race, then Alert/Fault levels are increased by a factor of 2.0. For roller defect, increase inner race levels by 1.5.

For PeakVue Alert levels, the following chart (Figure 18) shows how magnitudes typically vary with speed. Note that PeakVue amplitudes are very sensitive to speed in the ranges from 10 to 900 RPM and from 3000 to 10,000 RPM, but are constant between 900 and 3000 RPM. Table IV will take these speed sensitivities into account by providing formulas that can be used to calculate PeakVue “Alert” and “Fault” Alarm levels for a wide range of speeds ranging from 10 RPM up to over 10,000 RPM.

Table IV provides guidelines for approximate PeakVue “Alert” and “Fault” alarm levels that should be assigned when building a database (of course, after at least 7 surveys have been acquired, it is recommended that statistical analysis be applied on a “per point basis” in order to provide even more meaningful alarm levels, based on actual acquired data; in this case, it is recommended that separate alarms be established by point and by direction at each location where PeakVue data has been acquired). Typically, if statistical analyses is applied after a sufficient number of surveys have been

taken, it is usually best to set the “Alert” level at the mean value (x_{ave}) plus three standard deviations (s); that is “Alert” = $x_{ave} + 3s$. Then, “Fault” = 2 X “Alert”.

This method of establishing PeakVue alarms has proven meaningful for a diverse array of machines operating at a wide range of speeds. It is important that the speed is taken into account and that the proper value (or equation) from Table IV is entered if establishing alarms for the first time. Again, after several surveys have been taken on a machine, it is a good idea to review these alarms and to use statistical analysis to refine them.

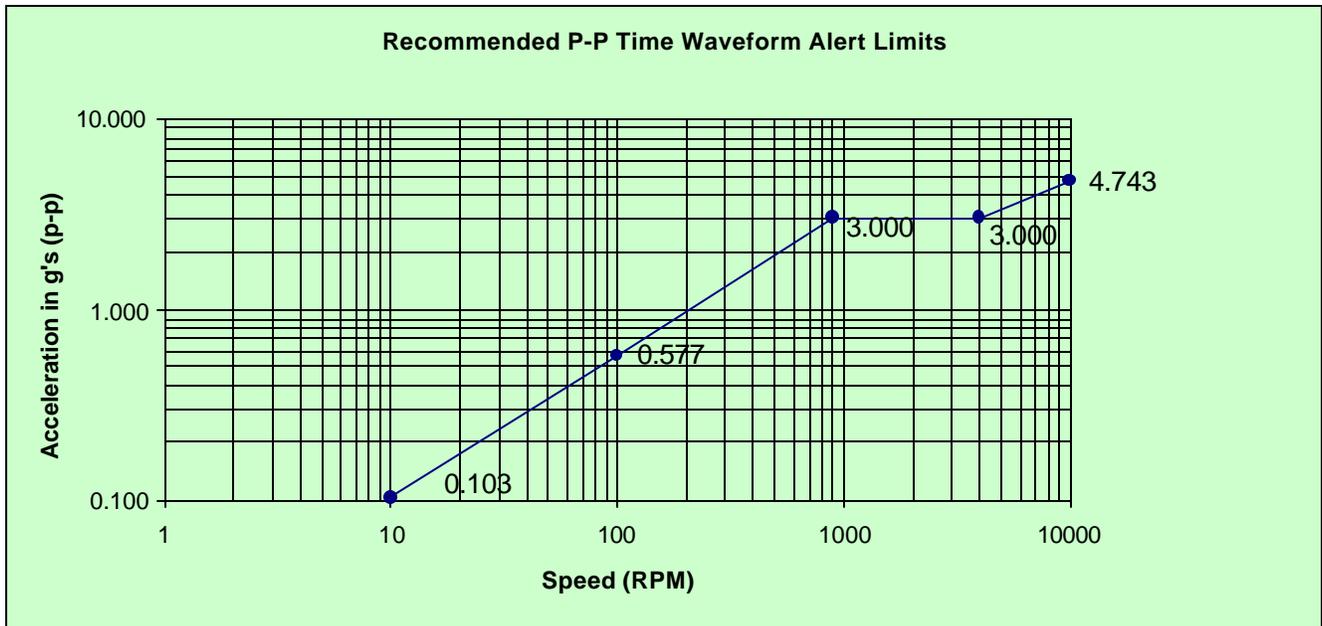


Figure 18. Recommended “Alert” values from PeakVue Time Waveforms (PK-PK value taken from time waveform including 10 Revs)

Figure 19 shows a series of examples where the Alert Alarm levels are determined for a variety of different configurations and speeds using the guidelines documented in Table IV. Notice the relationship between inner and outer race (BPFI and BPFO) alarms for cases 1, 2, 5 and 6. Also, note the difference in alarm levels of moderate and high-speed versus low-speed machines.

In addition to the absolute levels for Alert/Fault alarm levels for the PK-PK values of PeakVue Time waveform, trending of overall digital (entire analysis bandwidth), as well as energy at synchronous as well as at non-synchronous spectral frequencies are meaningful trending parameters. Values set for digital Overall alarms (Figure 17) will have to be based on reference spectral values (recommend multiply by 4-5X).

**TABLE IV. PEAKVUE "ALERT" ALARMS IN TIME WAVEFORMS
FOR BEARING AND GEAR PROBLEMS AT VARIOUS SPEEDS^{1,2}
(Peak-Peak g)**

COMPONENT RPM	R.E. BEARING FAULTS		GEAR FAULTS	
	Inner Race, Cage or Rolling Element Fault	Outer Race Fault	Worn or Scored Teeth ³	Cracked Teeth ⁴ (Fully Loaded)
0-900	$Nominal\ Speed\ Alarm\ X\ \left(\frac{Actual\ RPM}{900}\right)^{0.75}$			
901-4000 (Nominal Speed)	3g	6g	3g	6g
4001-10,000	$Nominal\ Speed\ Alarm\ X\ \left(\frac{Actual\ RPM}{4000}\right)^{0.5}$			
10,001-UP	5g	10g	5g	10g

NOTES:

1. Table V is intended to act as a Guideline providing suggested "Alert" and "Fault" Alarms to be applied to PeakVue waveforms for various faults as listed. These alarm amplitudes will likely be refined with further experience, statistical analyses and investigations.
2. Alarms are applied to the Peak-Peak Levels found in PeakVue Time Waveforms. If this waveform alarm is violated, then the analyst will refer to the PeakVue Spectrum to determine the cause of the problem (rolling element bearing, gear, lubrication, etc.).
3. Applies either to gears having numerous worn teeth around periphery or to gears having deficient lubrication causing scoring/scuffing of gear tooth surfaces.
4. Alarms given for "Cracked Teeth" assume gears are fully loaded. If gears are operated at or near no-load conditions, alarm levels should be reduced by a factor of 2. It is good practice to fully load gearing when it is being evaluated by either vibration or stress wave analysis if possible.
5. Limited experience to date on precision machinery (i.e., machine tools) suggests "Alert/Fault" alarm levels should be reduced by a factor of 2.
6. **Set Alarm Level for "PeakVue FAULT" = 2X "PeakVue ALERT" Alarm.**

**FIGURE 19. EXAMPLES APPLYING PEAKVUE ALARMS
TO A VARIETY OF FAULTS AT VARIOUS SPEEDS (from Table IV)**

1. Suspected Outer Race Bearing Fault on 1793 RPM Motor:
From Table III at 1793 RPM,
PeakVue Alert Alarm = 6.0g in Time Waveform (Look for multiple BPFO Frequencies in PeakVue Spectrum)
2. Suspected Inner Race Bearing Fault on 1793 RPM Motor:
From Table III at 1793 RPM,
PeakVue Alert Alarm = 3.0g in Time Waveform (Look for multiple BPFI Frequencies in PeakVue Spectrum)

3. Suspected Worn Teeth on an 8000 RPM High-Speed Pinion:
From Table III at 8000 RPM,

$$PeakVue\ Alert\ Alarm = \left(\frac{8000}{4000}\right)^{0.5} \times 3g = 1.414 \times 3 = 4.2g \text{ (in TWF)}$$

(Look for high amplitude at 1X GMF [and occasionally at 2X GMF and/or 3X GMF] in PeakVue Spectrum if the pinion has worn or scored teeth).

4. Suspected Broken Tooth on an 8000 RPM High-Speed Pinion:
From Table III at 8000 RPM,

$$PeakVue\ Alert\ Alarm = \left(\frac{8000}{4000}\right)^{0.5} \times 6g = 1.414 \times 6 = 8.4g \text{ (in TWF)}$$

(Look for multiple pinion running speed harmonics in PeakVue Spectrum and for 1 or 2 pronounced pulses/revolution of Pinion in PeakVue TWF).

5. Suspected Outer Race Fault on a 250 RPM Machine:
From Table III at 250 RPM,

$$PeakVue\ Alert\ Alarm = \left(\frac{250}{900}\right)^{0.75} \times 6.0g = 0.383 \times 6.0g = \underline{2.3g} \text{ (in TWF)}$$

(Look for multiple BPFO frequencies in PeakVue Spectrum).

6. Suspected Inner Race Fault on a 250 RPM Machine:
From Table III at 250 RPM,

$$PeakVue\ Alert\ Alarm = \left(\frac{250}{900}\right)^{0.75} \times 3g = 0.383 \times 3g = \underline{1.15g} \text{ (in TWF)}$$

(Look for multiple BPFI frequencies in PeakVue Spectrum).

6.0 Anticipated PeakVue Time Waveform and Spectral Patterns for a Variety of Faults and Problem Conditions

Table V is provided to give the user a good feel for the expected time waveform and spectral patterns that are anticipated to occur for a variety of faults or problem conditions that will occur with rolling element bearings, gears and lubrication. In addition, information is included on the Pk-Pk amplitudes that are expected to be seen for each of these specific fault types and conditions listed in the table.

It is important to point out Table V is offered as a Guideline listing characteristic PeakVue waveforms and spectra that typically occur for each of the faults covered. This information has been assimilated after conducting numerous tests on a variety of machines operating at a wide range of operating speeds. Such tests have been conducted in the laboratory, as well as in field diagnostic investigations and in numerous routine condition monitoring surveys. Further experience with these faults, as well as others not yet listed will, in all likelihood, result in refinements of the information contained within this table. Similarly, the amplitudes and anticipated PeakVue spectral and waveform “patterns” will likely be adjusted with additional experience.

It is hoped by the authors that Table V will be of real benefit to the users as we all attempt not only to interpret the meaningful information that PeakVue waveforms and spectra provide, but also to give useful guidance on how to evaluate the severity of such faults when they do appear. It is one thing to detect and diagnose a fault or problem condition; it is quite another thing to be able to confidently assess how severe such problems might be (and to determine if corrective actions are in fact required).

When monitoring machinery using the PeakVue method, it is most important that the user set up the measurement point parameters as described in section 4 and as tabulated in Table III.

When monitoring gearboxes, it is very important to include two times gear mesh in the analysis bandwidth. This is required to capture possible backlapping in addition to scuffing/scoring on the addendum and dedendum. The high-pass filter should be set higher than anticipated vibration frequencies. For certain gearing faults this could be at three times gear mesh. The problem here is that it will often force a high-pass filter setting at 5,000 Hz (next choice beyond 2,000 Hz as per Table II). If impacting is occurring in a gearbox, it is usually between massive parts and dominant energy will normally be in the 1 to 5 kHz range. Additionally, the higher frequencies caused by these faults will experience significant attenuation because of signal losses from the gear teeth to the outer surface where the sensor is mounted. Therefore it is recommended that the high-pass filter be set slightly greater than 2 times the highest gear mesh in the gearbox.

It is recommended that a measurement point be positioned at each bearing on the gearbox. The high-pass filter setting should be the same for every measurement point on the gearbox. However, the resolution and analysis bandwidth (F_{max}) will change and should be specified as per the recommendations of section 4 and Table III. The key is to include up to at least two times gear mesh for any gears on the shaft being monitored and to provide sufficient frequency resolution to resolve that gear mesh being modulated (sidebanded) with either (hence, lowest speed) shaft on the gearbox.

For bearings within the gearbox, the same rules apply relative to bandwidth and resolution as presented in the first part of this section. The only difference is that the high-pass filter will remain the same, i.e., greater than two times the highest gear mesh within the gearbox.*

Note the wide variance in PeakVue time waveform amplitudes for the various rolling element bearing problems listed. They range from approximately 2 to over 30 g's, depending on the particular type of bearing fault. For example, note that faults with the inner race, rolling elements or cage typically generate much smaller PeakVue amplitudes than do those associated with the outer race. This is due to greater loss as the signal is forced to pass through additional interface than does a fault originating on the outer race. Similarly, note that electrical fluting can often generate significant PeakVue amplitudes of 30 to 50 g's, or more, due to the nature of the problem.

Also, looking at Table V, note that BPFO and BPFI harmonics most always appear in PeakVue spectra when faults are present with these bearing components. On the other hand, when the rolling elements themselves develop faults, it is not unusual to see 2 X BSF exceed 1 X BSF since the rolling elements impact both the outer and inner race per revolution of the balls or rollers. And, in the case of the cage, FTF harmonics rarely appear in PeakVue spectra; instead, FTF sidebands will most often surround BSF frequencies when problems develop either with the cage, the rolling elements or with lubrication. Thus, it will be important to ensure a sufficient number of FFT lines are employed, along with an adequate number of shaft revolutions in order to be able to even see faults developing with any of the four bearing components, particularly the cage since it typically will only appear in a waveform only once every 2.5 revolutions (since FTF is typically approximately 0.4 X RPM). Thus, it is most important that the user employ the recommended measurement setup parameters provided in section 4 and Table III (for the high-pass filter, F_{max} , #Lines, etc.).

The higher frequency components generated from lubrication faults will experience significant attenuation during their propagation to the outer surface of the gearbox; therefore, consideration should be given to selecting the mounting scheme of a flat magnet or better (see Figure 7).

Referring to Table V, note that lubrication-induced faults can generate considerable amplitudes, sometimes 25 to 50 g's, or greater, particularly if the problem is friction-induced (and if it is possible to position the transducer within the load zone and in relative close proximity to the bearings). Interestingly, notice in Table V that friction-induced lubrication problems not only excite much higher frequencies than do impact-induced faults, but also generate very different looking PeakVue spectra (impact will typically show bearing fault frequencies, particularly BSF harmonics, whereas friction-induced problems generally do not result in PeakVue spectra with well defined, discrete frequencies; instead, friction will most always cause an elevated noise floor within the spectrum but have random, broadband frequency content).

* If this forces the high-pass filter to exceed 5 kHz, then the high-pass filter should be replaced with a band pass filter which excludes 1 and 2 times any gear mesh within the gearbox.

TABLE V. ANTICIPATED PeakVue TIME WAVEFORM AND SPECTRUM FOR A VARIETY OF FAULTS OR PROBLEM CONDITIONS*

FAULT OR PROBLEM CONDITION	ANTICIPATED PeakVue WAVEFORM**	ANTICIPATED PeakVue SPECTRUM
<p>A. ROLLING ELEMENT BEARINGS:</p> <p>1. Outer Race Defects</p> <p>2. Inner Race Defects</p> <p>3. Cracked Inner Race</p> <p>4. Rolling Element and Cage Problems</p> <p>5. Electrical Fluting</p>	<p>(a) 5-7g Spikes typically spaced at 1/BPFO (b) 7.5-10g (Particularly at End of Life) (c) >10g</p> <p>(a) 2.5-3g Spikes typically spaced at 1/BPFI, but seem to appear & disappear since highest spikes occur when inner race defect is impacted by rolling elements while within load zone. (b) 3.5-5g (c) >5g</p> <p>(a) 3.5-5g 1/BPFI not usually present in TWF. Instead, only 1-2 spikes occur each revolution as rolling elements impact the crack when in load zone (resembles a cracked gear tooth). (b) 5-7g (c) >7g</p> <p>(a) 2.5-3g When TWF displays approx. 15 revolutions, spikes usually occur once per 2.5 revolutions (since FTF is about 0.4 X RPM) (b) 3.5-5g (c) >5g</p> <p>(a) 5-10g Waveform patterns difficult to predict (tend to vary with severity of fluting. Typically shows high amplitude spikes, particularly when “washboard effect” is pronounced.. (b) 10-30g (c) >30g</p>	<p>(a) 1-3 BPFO harmonics in the beginning. (b) 4-6 BPFO harmonics as defects worsen. (c) 6+ BPFO harmonics, possibly accompanied by 1X RPM or FTF sidebands as faults become severe, particularly in the load zone.</p> <p>(a) 1-3 BPFI harmonics and 1X RPM sidebands in the beginning. (b) 46 BPFI harmonics with several 1X RPM sidebands, with a few 1X RPM harmonics as defects worsen. (c) 6+ BPFI harmonics with many 1X RPM sidebands, along with many 1X RPM harmonics as faults become severe.</p> <p>(a), (b) and (c) When crack is visibly present, PeakVue spectra most always show many 1 X RPM harmonics (typically do not contain BPFI frequencies that might be expected).</p> <p>(a) 1-3 BSF harmonics accompanied by FTF sidebands in the beginning. (b) 4-8 BSF harmonics accompanied by FTF sidebands as problems worsen. (c) 8+ BSF harmonics accompanied by several FTF sidebands as problems become severe (on rare occasions, may have discrete FTF harmonics, particularly if cracks or other faults begin to appear).</p> <p>(a) 1-3 BPFO and/or BPFI harmonics in the beginning. (b) 4-8 BPFO and/or BPFI harmonics, possibly 1X RPM harmonics/sidebands as fluting worsens. (c) 8+ BPFO and/or BPFI harmonics plus 1X RPM sidebands as fluting becomes severe typically causing “washboard effect” on bearing race; sometimes also accompanied by a frosted finish on one or more races.</p>

NOTES:

* Table IV is intended to act as a Guideline listing the characteristic PeakVue waveforms and spectra that are expected for various faults or problem conditions. Its contents are based on numerous field and laboratory tests. However, as the term "Guideline" infers, the contents of Table IV are expected to be refined with the results of future studies and field experience.

** These PeakVue Waveform amplitudes are approximate and are meant to serve as guidelines. In addition, they are meant to apply to general machinery components ranging in speed from 900 to 4000 RPM (see Table III). These waveform amplitudes will likewise be refined with further experience, statistical analyses, and investigations.

TABLE V. ANTICIPATED PeakVue TIME WAVEFORM AND SPECTRUM FOR A VARIETY OF FAULTS OR PROBLEM CONDITIONS*

FAULT OR PROBLEM CONDITION	ANTICIPATED PeakVue WAVEFORM**	ANTICIPATED PeakVue SPECTRUM
<p>B. GEARS:</p> <p>1. Tooth Wear or Scoring</p> <p>2. Cracked or Broken Teeth a. Subsurface Tooth Cracks</p> <p>b. Visible Cracks or Broken Teeth</p> <p>3. Gear Lubrication Problems</p> <p>4. Backlash</p>	<p>(a) 2.5-3.0g; Impacting Pattern very repeatable but typically only 1/4 to 1/2 Rev. in Waveform.</p> <p>(b) 3.0-5.0g, Impact Pattern is more Discernable and Repeatable, plus growth in Waveform Amplitude.</p> <p>(c) >5g; Impact Pattern Very Distinct and typically extends beyond 1/2 Rev. of worn gear.</p> <p>Amplitudes are difficult to predict but Waveform shows 1 or 2 regions of Pulses often 3 - 6g. TWF is best indicator of this problem.</p> <p>Amplitudes again difficult to predict but TWF will also show 1 or 2 regions of distinct Pulses often >15-20g. Very apparent in TWF.</p> <p>~3-4g; TWF amplitudes are typically low since impact energy is concentrated in very high frequencies and dissipates rapidly from gears to measurement locations.</p> <p>Often >5-10g; High amplitudes in TWF are due to impact between mating gear teeth.</p>	<p>(a) Typically 1X GMF but without 1X RPM harmonics.</p> <p>(b) Typically 1X GMF is highest but 2X GMF and/or 3X GMF may appear in Higher F_{MAX} spectrum.</p> <p>(c) High 1X GMF typically accompanied by growth in amplitude of 2X GMF and/or 3X GMF; also, 1X RPM harmonics often appear if tooth cracks propagate to the surface or root.</p> <p>Lower F_{MAX} spectrum (i.e., 20 X Pinion RPM) typically shows roughly 3-6 harmonics of 1X RPM. Higher F_{MAX} spectrum (i.e., 2.25 - 3.25 X GMF) sometimes shows elevated 1X GMF.</p> <p>Lower F_{MAX} spectrum typically shows many 1X RPM harmonics of the gear with the problem. Higher F_{MAX} spectrum sometimes shows elevated 1X GMF and harmonics.</p> <p>Lube problems normally cause elevated 1X GMF. However 2X GMF will grow if Lube problems degrade allowing Microwelding (instantaneous metal-metal contact between tooth surfaces that can cause peeling & flaking).</p> <p>Most always causes high 2X GMF in PeakVue spectra. Will often cause many sidebands around 2X GMF, spaced at 1X RPM (and also sometimes spaced at 2X RPM).</p>

NOTES:

* Table IV is intended to act as a Guideline listing the characteristic PeakVue waveforms and spectra that are expected for various faults or problem conditions. Its contents are based on numerous field and laboratory tests. However, as the term "Guideline" infers, the contents of Table IV are expected to be refined with the results of future studies and field experience.

** These PeakVue Waveform amplitudes are approximate and are meant to serve as guidelines. In addition, they are meant to apply to general machinery components ranging in speed from 900 to 4000 RPM (see Table III). These waveform amplitudes will likewise be refined with further experience, statistical analyses, and investigations.

TABLE V. ANTICIPATED PeakVue TIME WAVEFORM AND SPECTRUM FOR A VARIETY OF FAULTS OR PROBLEM CONDITIONS*

FAULT OR PROBLEM CONDITION	ANTICIPATED PeakVue WAVEFORM**	ANTICIPATED PeakVue SPECTRUM
<p>C. LUBRICATION PROBLEMS</p> <p>1. Friction-induced</p> <p>2. Impact-induced</p>	<p>Can cause excessive g levels >50g, but since friction-induced faults generate high frequencies in the range of 10,000-15,000 Hz, much of their signal rapidly dissipates with distance to transducer & at each interface. TWF is normally random with little or no periodic events.</p> <p>Impact is typically caused by metal-to-metal contact due to insufficient lubrication (and/or incorrect lubricant viscosity). If metal-to-metal contact occurs in a bearing, will typically show periodic pulses. TWF amplitudes can range to >25g, but more typically stay within 4-8g.</p>	<p>Friction-induced lubrication problems excite a wide range of high frequencies, typically ranging from just below 5000 Hz up to frequencies exceeding 15,000-20,000 Hz. Normally will likewise cause an elevated noise floor within the spectrum but have random, broadband frequency content. Often will generate FTF with a few harmonics, but will typically have poor signal-to-noise ratio.</p> <p>Metal-to-metal contact will most often generate bearing defect frequencies -- usually BPFO and/or BPFI; however, also commonly excites ball spin (BSF) frequencies accompanied by cage frequency (FTF) sidebands</p>

NOTES:

* Table IV is intended to act as a Guideline listing the characteristic PeakVue waveforms and spectra that are expected for various faults or problem conditions. Its contents are based on numerous field and laboratory tests. However, as the term "Guideline" infers, the contents of Table IV are expected to be refined with the results of future studies and field experience.

** These PeakVue Waveform amplitudes are approximate and are meant to serve as guidelines. In addition, they are meant to apply to general machinery components ranging in speed from 900 to 4000 RPM (see Table III). These waveform amplitudes will likewise be refined with further experience, statistical analyses, and investigations.

7.0 Real-World Case Studies

7.1 Introduction

In this section, case studies that demonstrate various features of stress wave analysis presented in previous sections are presented. The first case presented will be taken from two split ring pedestal bearings supporting a line shaft where one bearing was experiencing serious lubrication faults and impacting while the second bearing located near was primarily experiencing impacts. The second and third case studies illustrate detection of both a bearing and a gear fault where the decision to take corrective action was based on the PK-PK amplitude value of the PeakVue time waveform, i.e., there were no trending values available. The fourth case study is for a bearing in a gearbox where trending from PeakVue alone initiated a bearing replacement order.

7.2 Lubrication and Impact on a Split Ring Pedestal Bearing

This case study covers data acquired from two of the thirty-eight pedestal bearings supporting the primary drive shaft on a paper machine. The shaft is located on the lower level. The paper machine is located on the level above the drive shaft. Strategically located drums situated along the primary drive shaft transfer power to the paper machine through belts which introduce vertically upwards load into the bearings. The pedestal bearings along this shaft were all being replaced by split ring dual spherical bearings.

All thirty eight pedestal bearings supporting this shaft were included in portable data acquisition carried out on a scheduled basis. The sensor employed for data acquisition was an accelerometer placed on a dual rail magnet ("2-pole"). The magnet was placed on the pedestal near the top without great regard to a non-painted smooth surface. The rollers themselves were about 1" D, hence any impacting would excite primarily a frequency band in the 1 kHz to 5 kHz band (see Figure 3).

A decision was made to include these thirty-eight bearings in a permanent monitoring system. The sensors were permanently attached to the pedestal on a prepared surface that was clean and smooth. When this was done, the signals from several of the sensors were showing sudden random shifts in the bias voltage which is representative of the signal over-ranging the dynamic range of the sensor to the extreme (in this case, 100 mV/g accelerometers being used). This over-ranging never occurred in the route mode when a dual rail magnet/sensor was being used. Large amplitude, somewhat random stress wave events are typical for severe friction events.

The lubrication of the bearings was from a gravity fed oiler designed such that the oil level at bottom submerged about the roller when stationary. Power was transferred to the paper machine above the primary drive shaft through belts placed about drums located strategically along the drive shaft. Hence the load experienced by the pedestal bearings adjacent to the drums was vertical in an upwards direction.

* In route mode, dual rail magnet was attached to a rough painted surface.

The spectral and time waveform vibration data captured using a high frequency 10 mV/g sensor attached with a flat rare earth magnet on a flat smooth surface on the top of the pedestal for a 40 kHz bandwidth are presented in Figure 20. The time block of data is 40 msec which is less than ~~of~~ a revolution (speed = 696 RPM = 11.60 R PS; T = 1/11.6 = 86.2 msec/rev). Several events have been captured where most energy is in the 6 kHz to 15 kHz range. This is high for what we expect from impacting alone, but not for friction-generated events.

Note that a typical 100 mv/g accelerometer can normally withstand 50 g's (in the linear range) which explains why the permanently mounted transducers were over ranging. On the other hand, typical 10 mv/g accelerometers can withstand approximately 500 g's.

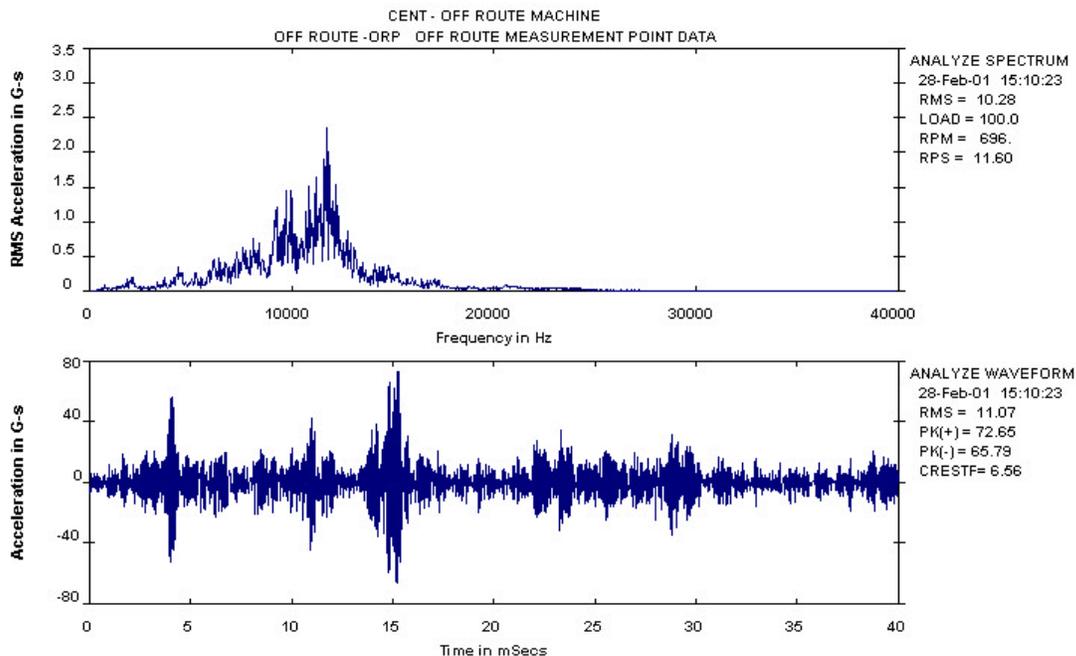


Figure 20. Spectral and time waveform for 40 kHz data on bearing pedestal adjacent to drum used to transfer via belt power from primary drive shaft to machine above power shaft.

To more carefully analyze this bearing, we need a data block which includes several revs of the shaft sampled at a high rate. This data was captured using PeakVue. The PeakVue spectral data and partial time block of data are presented in Figure 21. In the spectral data, there are indications of repetitive events occurring at 2x shaft speed with less response at 1X and 2X of BPF1.* The parameter of most concern in Figure 21 is the excessive PK-PK value of 273 g's observed in the PeakVue time waveform.** This

* A fault at BPF1 is expected on split ring bearings with modulation at twice running speed (2 X RPM) due to the split being two intentional faults located exactly 180° apart.

**The sensor is a high frequency 10 mV/g sensor attached to the top of the bearing pedestal. Since the load is upward, the metal-to-metal contact is most likely occurring directly under the sensor mounting.

type of PeakVue waveform and spectrum has classically been the result of metal-to-metal contact indicating lack of lubrication.

The autocorrelation coefficient (see Appendix B) computed from the PeakVue time waveform is presented in Figure 22. The periodic behavior at two times running speed is clearly indicated here, but the presence of BPFI is not indicated.

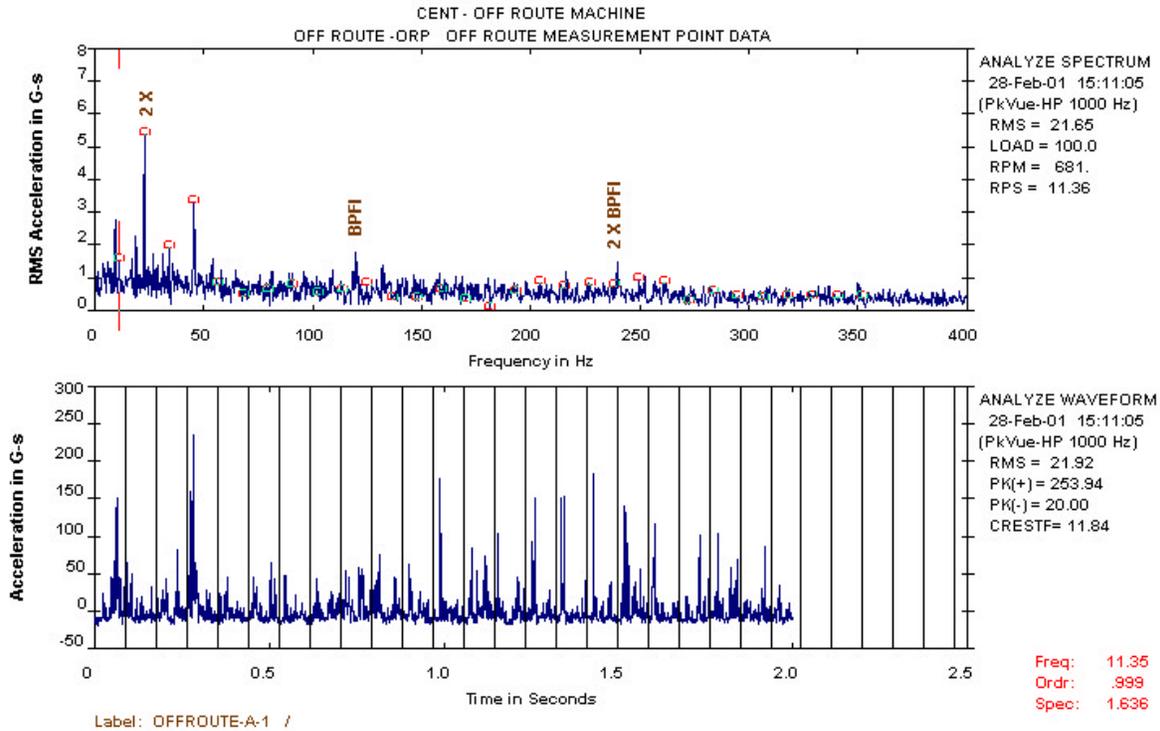


Fig 21. PeakVue data from bearing pedestal adjacent to power transfer drum

To verify metal-to-metal contact was occurring, an oil wear debris analysis was carried out on the oil taken out of the bearing. The pictorial results are presented in Figure 22. This data verified metal-to-metal contacting was occurring.

The bearing pedestal adjacent to the one presented above, but removed from the load bearing drum was not experiencing any large, randomly occurring events. Hence data was acquired for comparative analysis from this bearing. The 40 kHz bandwidth spectrum and time waveform are presented in Figure 22. Transient events that have been captured have different frequency content for different events. The spectrum shows significant energy in the 1 to 4 kHz range as well as in the 12 to 15 kHz range. The lower frequency range is consistent with what is expected for impacting and the upper range is consistent for what is expected for friction.

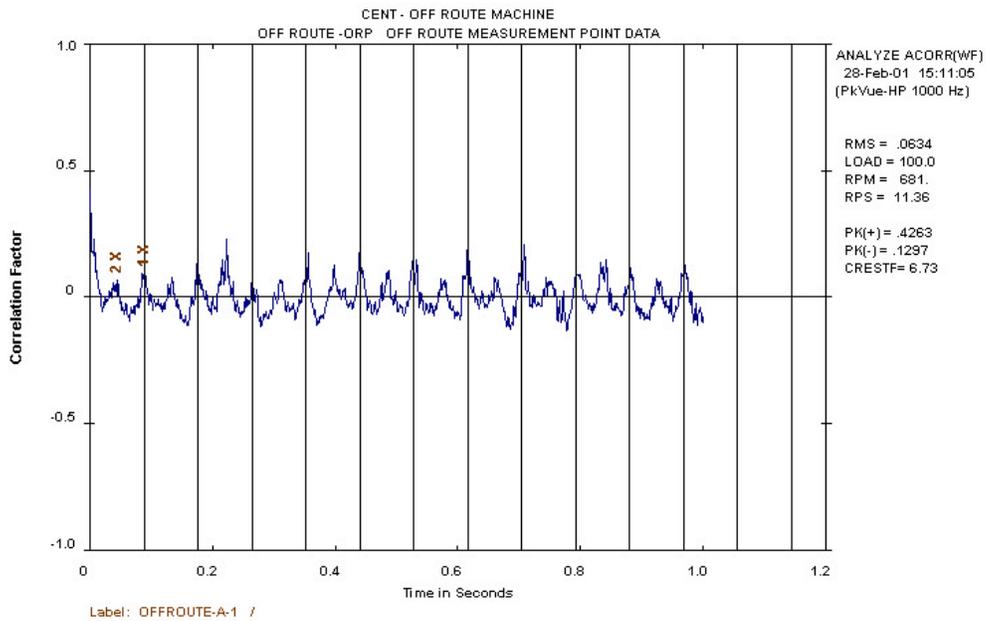
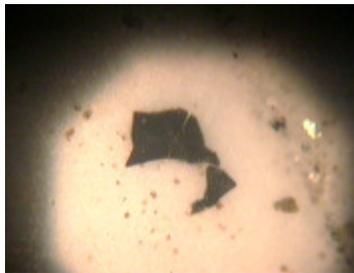
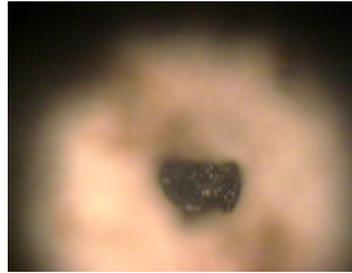


Figure 22. Auto-correlation coefficient function computed from PeakVue time waveform data of Figure 21.

To capture events over a sufficient number of shaft revolutions, PeakVue data, spectra and time waveform, was captured and are presented in Figure 25. The maximum PK-PK values were 2.4 g's (significantly lower than the 273 g's on the bearing shown in Figure 21). In the spectra data, events are clearly present at 2X shaft speed and at BPF1 (which is sidebanded with 2x shaft speed). The autocorrelation coefficient computed from the PeakVue time waveform of Figure 25 is presented in Figure 26 which clearly shows the BPF1 and 2X activity is the only correlated activity present. The second bearing was not subjected to the significant lubrication deficiency and friction.

WDA Images

Database:	Example.rbm	Meas. Point:	WDA - Wear Debris Analysis
Area:	WDA - Wear Debris Analysis	Sample No:	Bearing
Equipment:	WDA - Wear Debris Analysis	Sample Date:	3/5/01 3:56 pm



Wear debris analysis revealed a moderate distribution of metallic platelets, chunks, spheres and black oxides. All particles are typical of insufficient lubrication and metal to metal contact.

Fig. 23. Oil debris Analysis from bearing where data of Figs. 20 and 21 acquired.

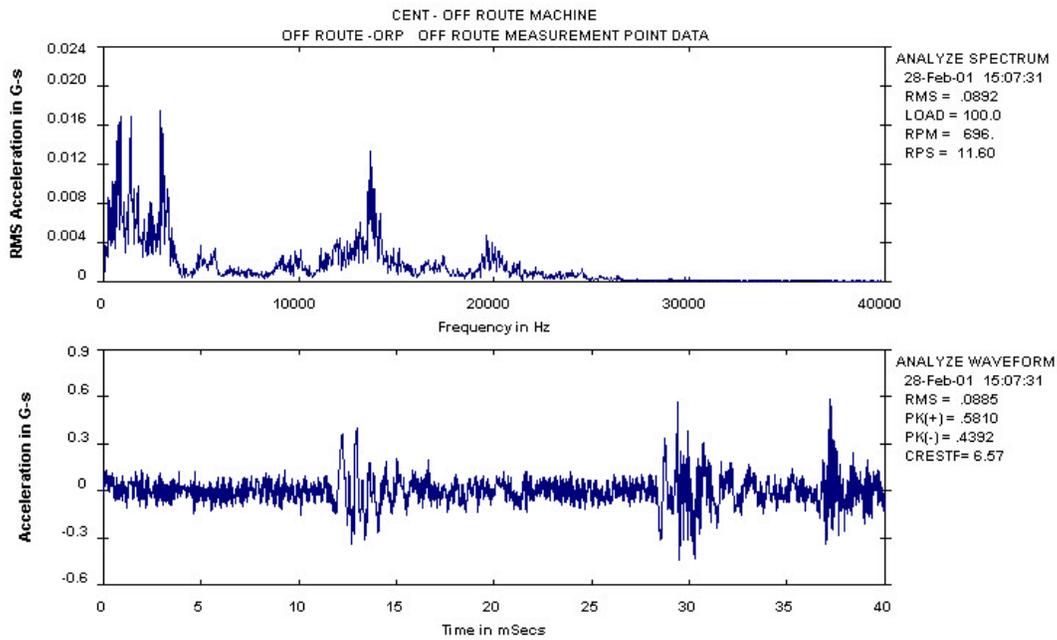


Figure 24. Spectral and time waveform for 40 kHz data on bearing pedestal adjacent to that of Figure 20, but further removed from load drum.

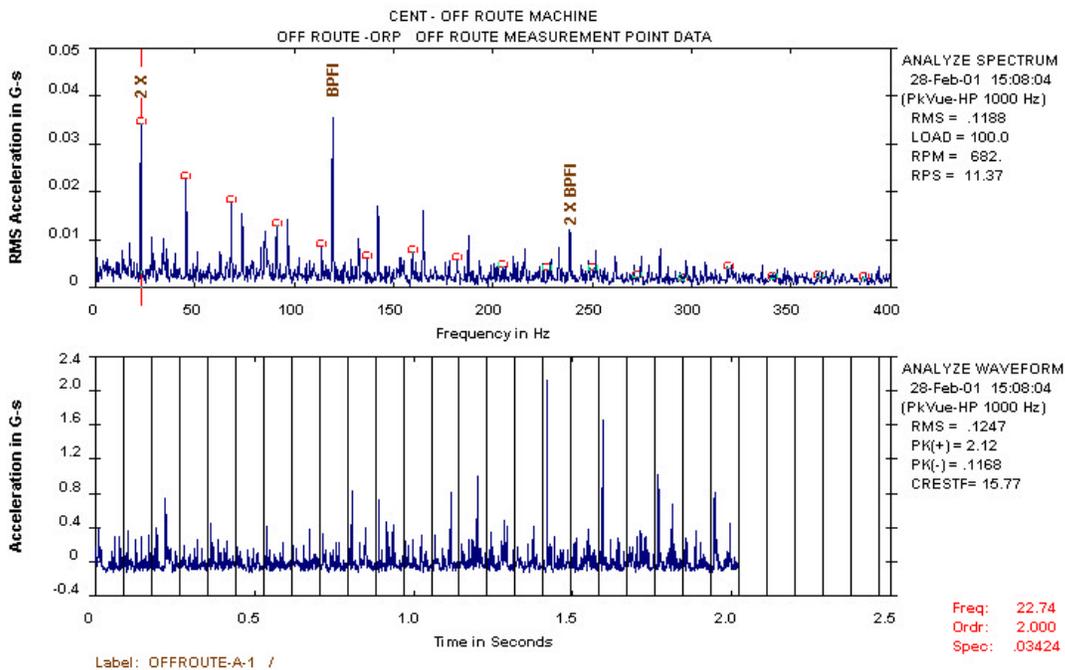


Figure 25. PeakVue Data bearing on Pedestal of Figure 24 (like that taken in Figure 21 on the bearing pedestal adjacent to power transfer drum).

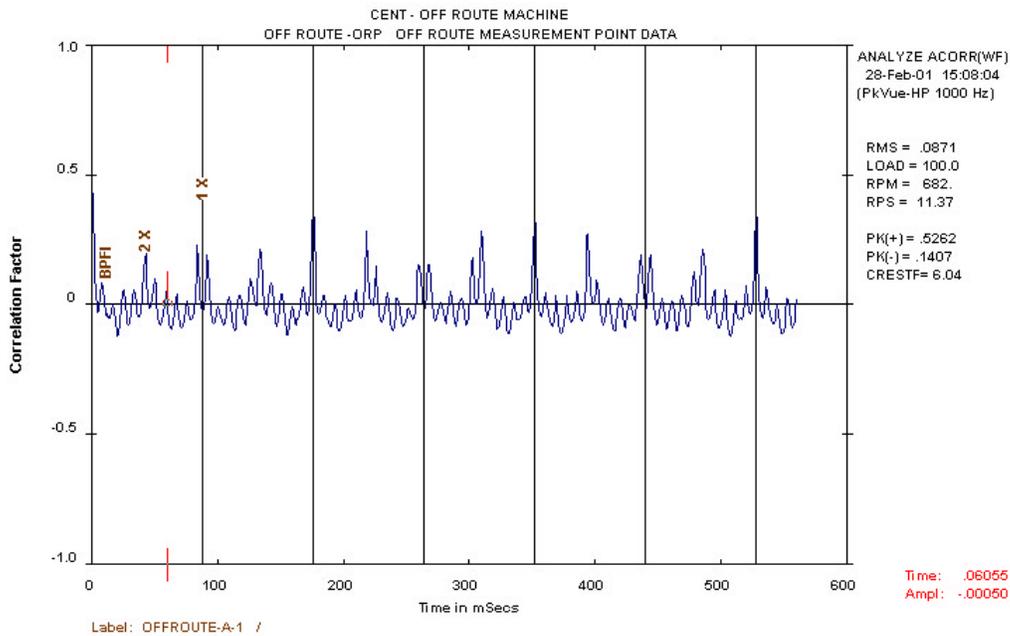


Figure 26. Auto-correlation function coefficient function computed from PeakVue time waveform data for Figure 25.

7.3 Outer Race Defect in Pinion Stand Gearbox

This pinion stand gearbox was included in a scheduled monthly condition monitoring program employing vibration analysis. The traditional vibration monitoring showed little indication of a bearing fault. In July 1997, the PeakVue methodology was introduced into the monitoring program. The PeakVue data was acquired as an add-on to a route defined point and hence was not trended.

It was obvious from the PeakVue data there was an outer race defect on the inlet shaft. The initial reading with PeakVue showed a clear outer race fault for the bearing on the input shaft with PK-PK impacts of 18 g's. The peak g readings in PeakVue continued to trend upwards (increasing to 37 g's in mid-September 1997); then began a downward trend (decreasing to 14 g's in early October). This bearing was monitored monthly on Aug. 18, Sept. 15, and Oct. 6, with both PeakVue and normal vibration. The outer race defect was persistent in PeakVue with PK-PK values of 18 g's, 24 g's, 37 g's and 13.5 g's (the normal vibration readings were showing 1.5 g's with no indication of a problem). The bearing was replaced on October 22, 1997. After replacement, the peak g-levels on the new bearing were less than 1 g.

The input shaft speed varied from 359 to 407 RPM (5.99 to 6.78 RPS). Based on the recommended "Alert/Fault" Alarm levels for BPFO presented in Sec. 4, the "Alert" level at a nominal speed of 380 RPM would be 3.15 g's and "Fault" level would be set at 6.30 g's. Clearly, the levels detected were well in excess of "Fault".

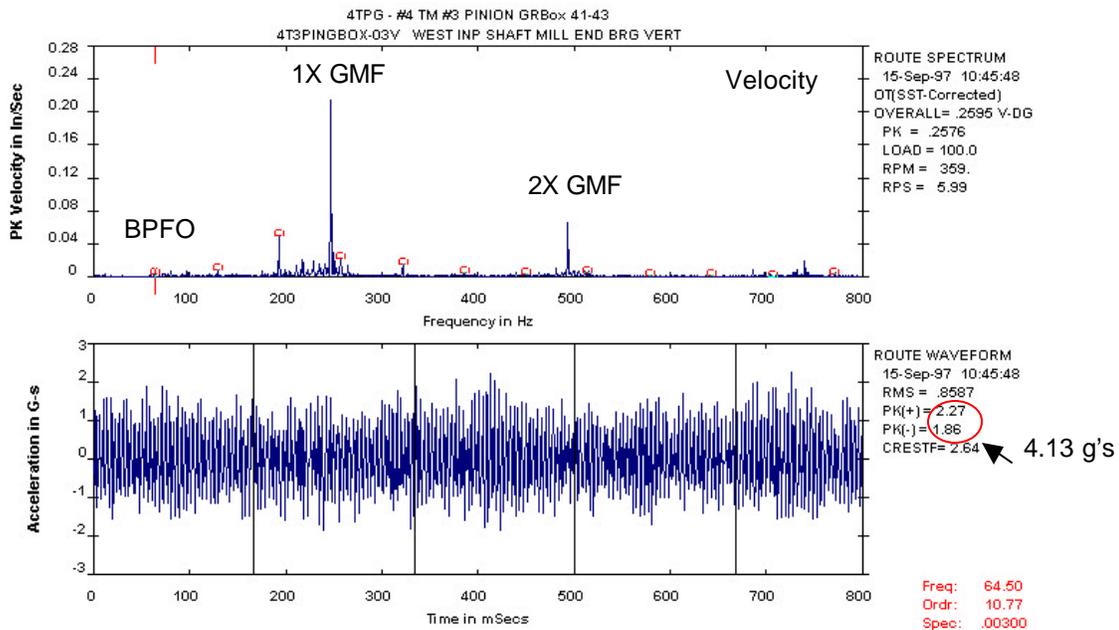


Figure 27. Velocity spectral data and acceleration time waveform taken at same measurement point and time of PeakVue data presented in Figure 28.

The normal vibration spectrum and acceleration time waveform data acquired on September 15, 1997 are presented in Figure 27. There is some indication of a possible outer race problem, but not conclusive evidence (certainly not to the extent indicated by PeakVue measurements).

The data from the PeakVue methodology also acquired on September 15, 1997 are presented in a Figure 28. The peak absolute g-levels are up to 37 g's at the operating speed of only 359 RPM (or 5.99 Hz) with a recurring rate consistent with the outer race defect frequency (BPFO).. The PK-PK levels experienced in the PeakVue time waveform are in the 25-30 g level. Impacting is at the outer race fault frequencies. For completeness, the autocorrelation coefficient function is presented in Figure 29. The correlation is high (around 50%) at time increments corresponding to the BPFO fault level.

Due to the excessive amplitudes found in the PeakVue waveform of Figure 28, and validated by the autocorrelation coefficient function in Figure 29, the bearing was pulled. The digital photo in Figure 31 shows the severe damage on the outer race showing significant spalling over a wide area. Subsequently, the bearing was replaced. Figure 30 shows the "Before" and "After" PeakVue data which showed a dramatic drop in amplitudes, with no bearing frequencies showing up in the "After" data.

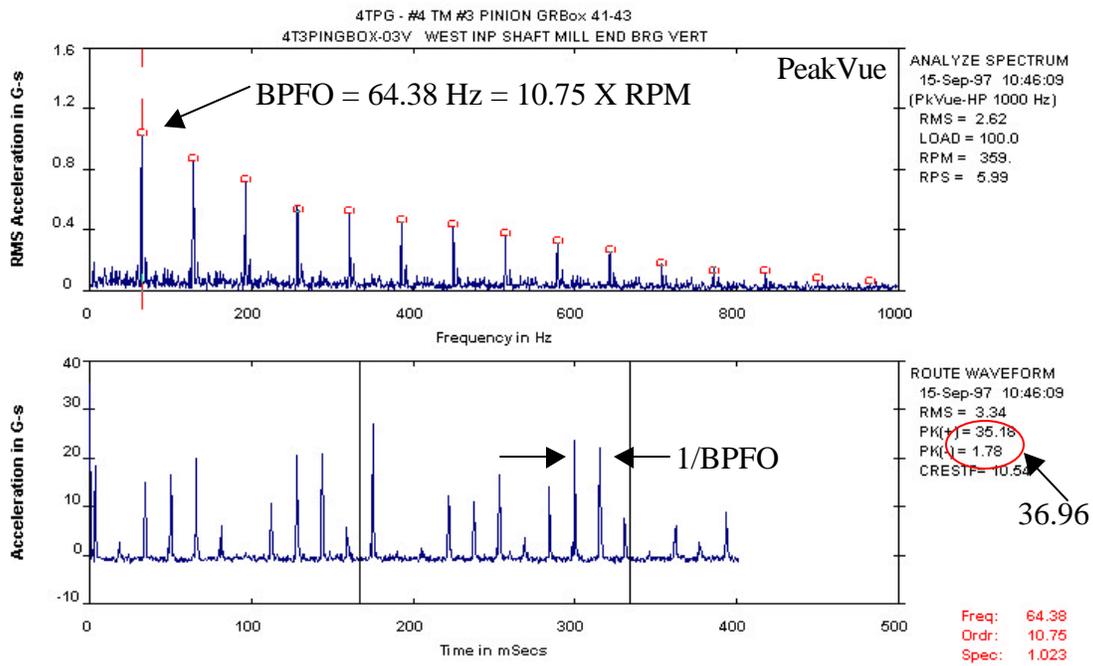


Fig. 28. PeakVue data from pinion gearbox taken at inlet on 15-Sept-97

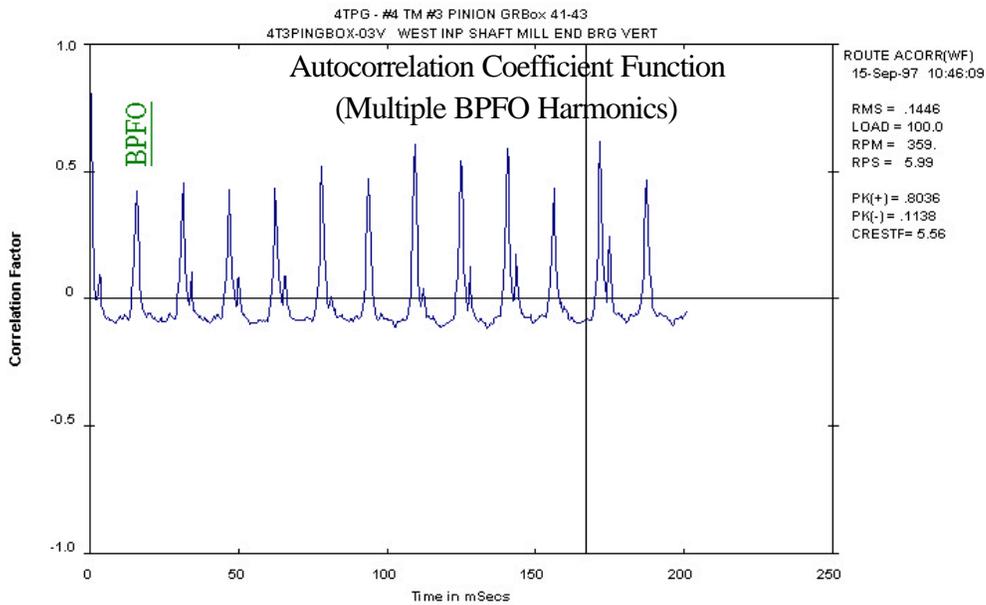


Fig. 29. Autocorrelation coefficient function computed from PeakVue time waveform of Fig. 28. (See Appendix B for explanation of Autocorrelation coefficient function)

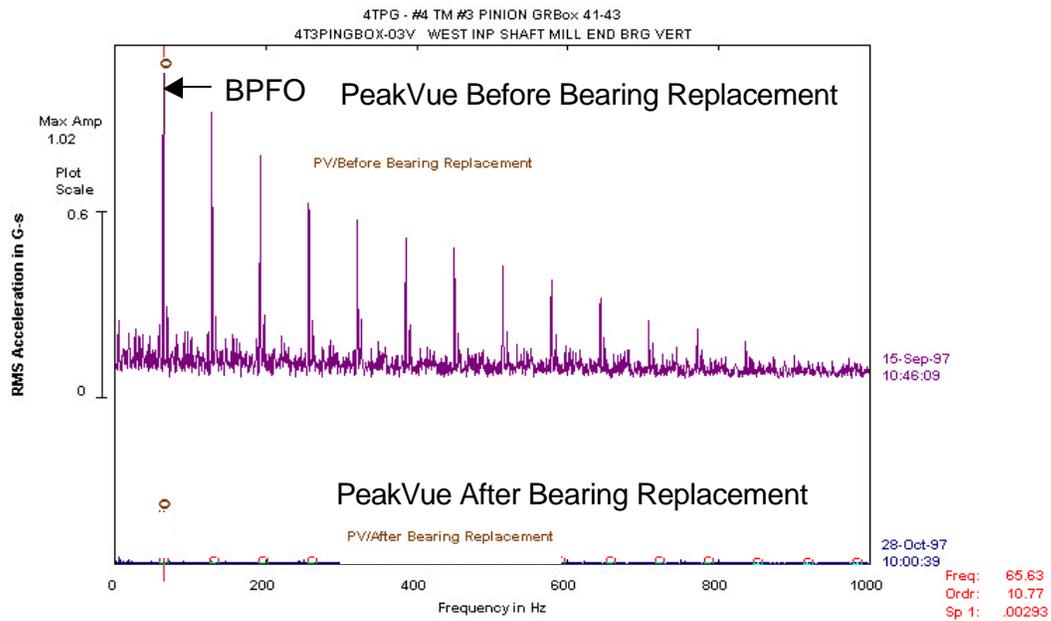


Figure. 30. PeakVue Spectral data Before and After bearing replacement.

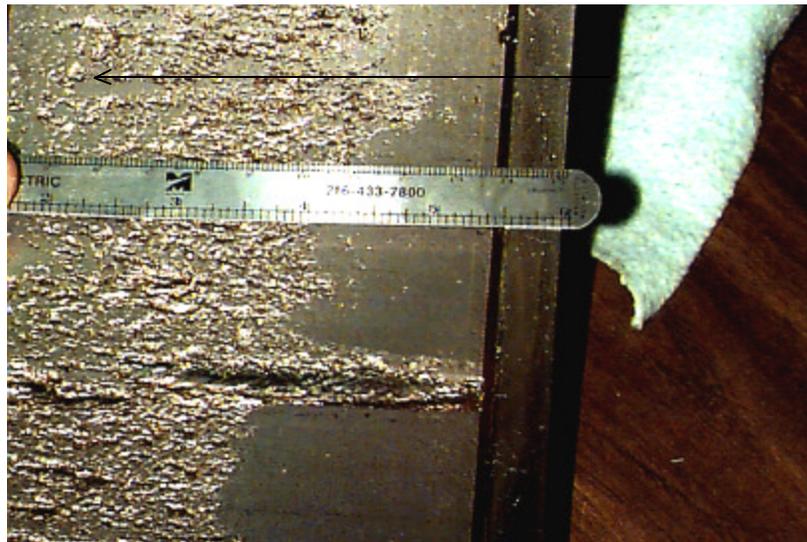


Fig. 31. Close-up picture of outer race of defective bearing removed on Oct. 22, 1997 showing significant spalling over an area of 5 x 5 square inches.

7.4 Cracked teeth in a Precision Tension Bridle Gearbox

A plan view of the Precision Tension Bridle gearbox from which data was acquired is presented in Figure 32. This gearbox was a single speed reduction gearbox with a dual shaft output. The slow speed shaft from the reduction gear set (40-tooth pinion gear driving a 158-tooth bullgear) was driving a second output shaft through a dual 90-tooth gear set. The 90 T gear on the output shaft was driving a second output shaft through a meshing 90T gear. The only accessible point on the gearbox for data acquisition using a portable system was on the inboard bearing housing over the input shaft.

This machine ran at various speeds from survey to survey. Typically, the input shaft speed was as high as 853 RPM (14.22 Hz) with an output speed of 216 RPM (3.60 Hz). At other times, the input speed would drop to about 526 RPM (8.76 Hz) causing an input gear mesh frequency (GM_1) of 350.6 Hz and an output speed of 133 RPM (2.22 Hz), generating an output gear mesh frequency (GM_2) of 201.6 Hz.

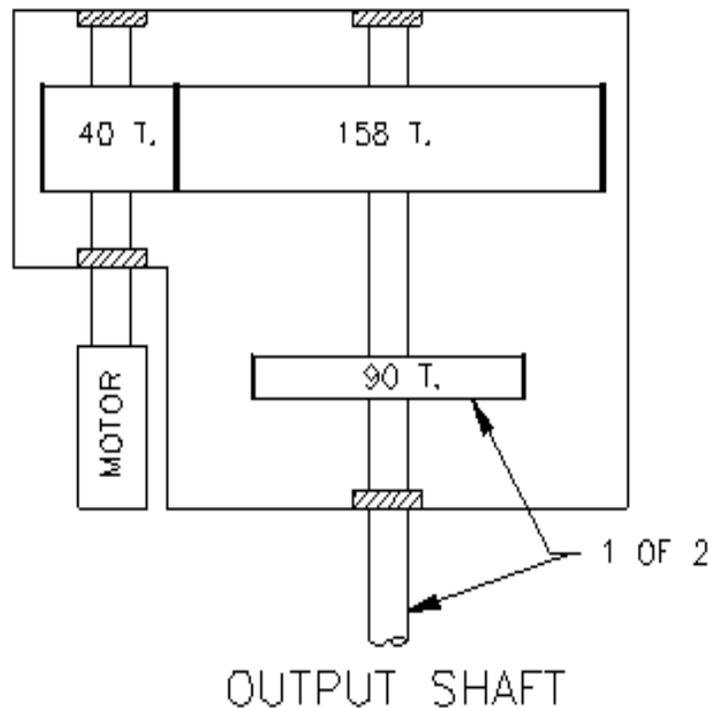


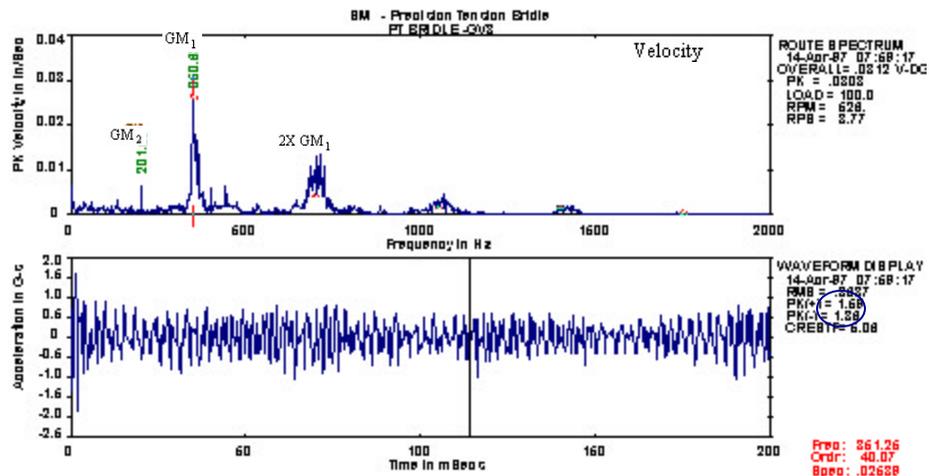
Figure 32. Plan View of precision tension bridle gearbox.

The gearbox was monitored on a scheduled basis using velocity spectral analysis. A velocity spectrum is presented in Figure 33 obtained on 11-Apr-97. Both gear meshes are highlighted on the spectral data with the input at 40 orders relative to the input shaft and the second at $(40/158) \times 90$, or 22.8 orders relative to the input shaft. The observation from historical data was that the sidebanding about the input gear mesh (GM_1) was increasing with time.

Figure 33 shows that the input speed reduction gear mesh, ($GM_1 = 350.6 \text{ Hz}$) was dominant in the spectral data and had significant sidebanding present, particularly at $2 \times GM_1$ (702 Hz). This pattern is indicative of gear wear and perhaps of some gear tooth misalignment. However, the peak velocity amplitude was only .026 in/sec at GM_1 . Similarly, the Pk-Pk acceleration data in the time waveform was less than 4 g's which was again not considered to be significant. Thus, a low priority was applied to this potential fault not requiring any corrective actions. Possible gear wear and tooth misalignment had been flagged, with the only action item recommended to initiate a visual inspection at the next convenient opportunity.

However, since this was considered to be a critical machine, it was decided to apply the PeakVue methodology on the same day (April 14, 1997). A PeakVue analysis was carried out using an analysis bandwidth of 200 Hz and a high-pass filter sitting of 500 Hz. The PeakVue spectral and waveform data are presented in Figure 33. The only activity in the spectral data is at the output shaft speed of 216 RPM (3.60 Hz) with many harmonics. The PeakVue time waveform clearly showed two broad impacting events per revolution of the output shaft. The PK-PK impacting levels exceed 40 g's. **This waveform indicated a gear with significant cracked or broken teeth (at the root) in two regions of a gear turning at the output speed of 216 RPM.**

Due to the magnitude of the impacting events, the precision of the repeatability of the impacting regions with the output shaft, and the appearance of multiple teeth (broadness) impacting, the conclusion was there were a minimum of two regions of cracked teeth on either the 158 T bullgear or on one of the two 90 T gears. The bandwidth in Figure 34 was not sufficient to encompass either gear mesh; therefore we cannot identify from this data set which gear set had the defective gear.*



Oil Temperature at 240 degrees F

Figure 33. Velocity Spectrum and Acceleration Time Waveform on the Precision Tension Bridle Gearbox (Peak Spectral Amplitude of .026 in/sec)

*The gear mesh where the defective gear is located would be present in the PeakVue spectra.

From the velocity spectrum presented in Figure 33, one would be suspicious that the defective gear would be in the high-speed mesh (GM₁) set of 40 and 158 teeth since nothing unusual was present relative to the low-speed mesh (GM₂) with the dual 90-tooth output gears. There was an additional PeakVue spectrum taken with a bandwidth of 5000 Hz. In this high frequency PeakVue spectrum, the GM₂ activity was noticeable while the GM₁ activity was absent. This led to the conclusion that the gears with the cracked or broken teeth were most likely in the GM₂ set.

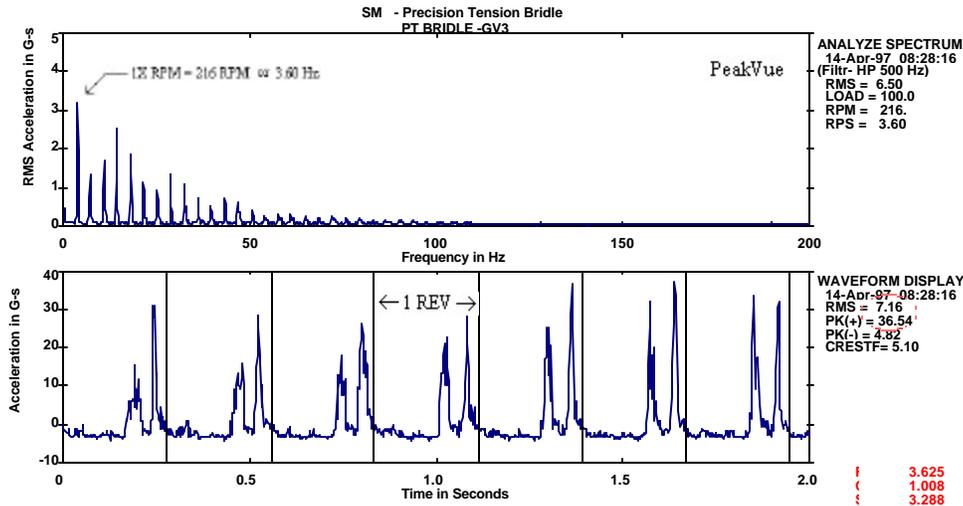


Figure 34. PeakVue Spectrum and Time Waveform on the Precision Tension Bridle Gearbox
 (Note the 2 Impacts per Revolution of the Output Shaft in the PeakVue Waveform with High Amplitudes up to 41.3 g's, Pk-Pk)

On this same date, a PeakVue spectrum with a bandwidth of 5,000 Hz and a high-pass filter setting of 5,000 Hz* was taken and are presented in Figure 35. The only frequency activity in this spectral data is at 23.5 orders of the input shaft which is the gear mesh of the 90 T gear set (GM₂). Since there are two output shafts, with each having a 90 T gear, we cannot tell which gear has the problem.

Following the acquisition of the PeakVue data, the gearbox was shortly shut down and closely inspected. One of the 90-tooth output gears in the GM₂ set was found to have two teeth about 90 degrees apart having visible cracks at the root of the teeth. This explains why 2 pronounced pulses or impacts were present per revolution of the 90-tooth output gear as shown in the PeakVue waveform in Figure 35.

*The impacting objects here are massive, therefore the energy would be predominantly below 5 kHz, but some would be present above 5 kHz.

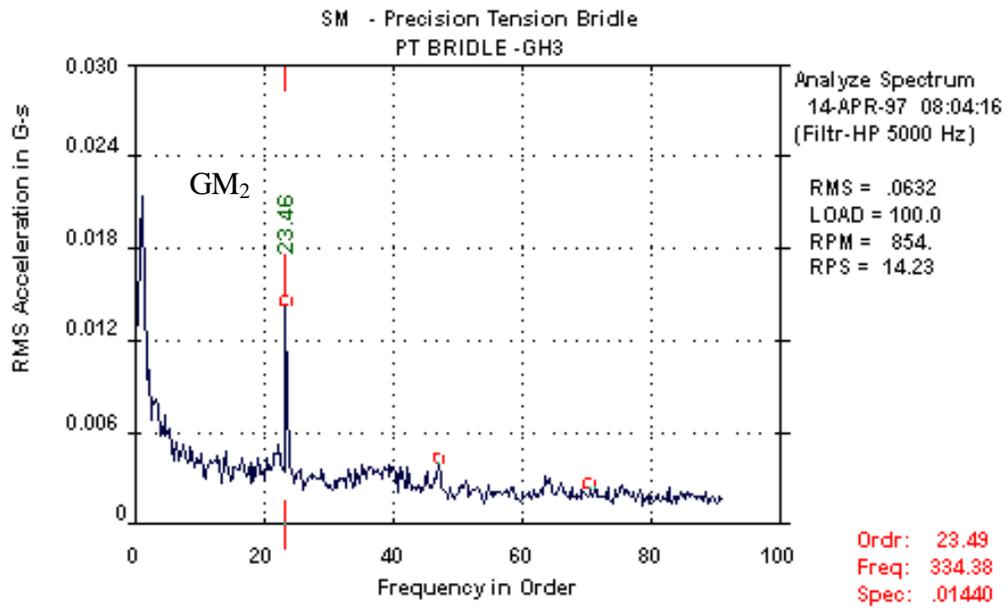


Figure 35. PeakVue data from bridle gearbox with 5000 Hz bandwidth acquired on 14-Apr-99.

7.5 Defective Gearbox Bearing Detected Through Trending of PeakVue Data

This case was from a Finish Mill Pinion Stand Gearbox. This is a two shaft gearbox with each having a 28T gear. A 8000 HP variable speed motor drives the gearbox. PeakVue measurements points were set up on this gearbox and trending began on 16-Oct-98.

One of the trend parameters captured was the peak g-level in the PeakVue time waveform (“Max Peak Waveform”) as shown in the trend plot of Figure 4. **Experience has shown the true peak-to-peak time waveform level to be a key PeakVue parameter for fault detection and severity assessment.** An upward trend for the measurement point placed on lower shaft output was noted in October 1999.

The PeakVue peak g-level trend parameter for the lower output shaft and PeakVue spectrum for the last collection date of May 25, 2000 are presented in Figure 36. Importantly, note from the Y-axis of the waveform in Figure 36 that the trend parameter being monitored was “Max. Peak Waveform in g’s”. That is, the magnitude was the True Peak g level taken directly from the raw time waveform, not a digital overall level computed from the spectrum.

The “Alert” and “Fault” Alarm levels are specified at recommended levels for this 156 RPM machine speed and fault type (using the alarm equations shown in Table IV). From the spectrum, the fault is an inner race fault that is sidebanded (amplitude modulated) at running speed which is indicative of the inner race fault going into and out of the load zone at running speed. From the PeakVue waveform magnitude in Figure 36, it is obvious that the fault exceeded the “Fault” level in November 1999 about 7 months prior to the May 25, 2000 measurements shown in Figure 36. The trend from PeakVue data continued upward and a work order was released in June 2000 to replace the bearing. The replacement occurred in mid July 2000.

The PeakVue trended data which led to the decision to replace the bearing are presented in Figure 36. The spectra clearly identify the fault as an inner race fault. The parameter trended was “maximum peak waveform” in g’s.* The “Alert/Fault” levels shown in Figure 36 were computed from the guidelines presented in Table IV within Section 5.

*The correct parameter would have been the PK-PK of the time waveform Examination of saved time waveforms indicates the recorded levels should be increased by 0.25 to 0.5 g’s.

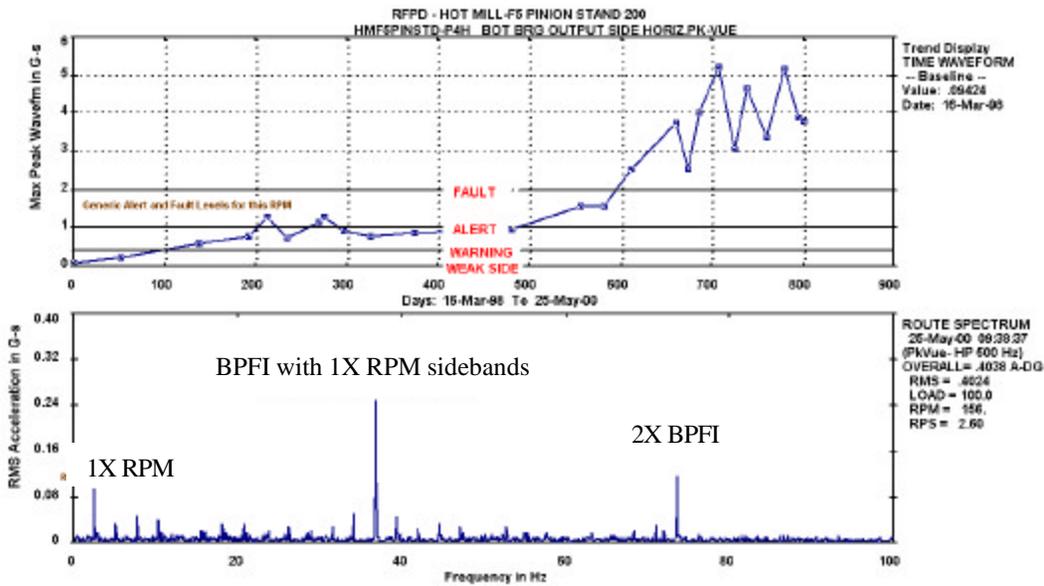


Figure 36. Maximum Peak g-level (from PeakVue) Trend from March 16, 1998 to May 25, 2000 and PeakVue Spectrum from May 25, 2000

(Note PeakVue Fault Alarm Violation in Trend)

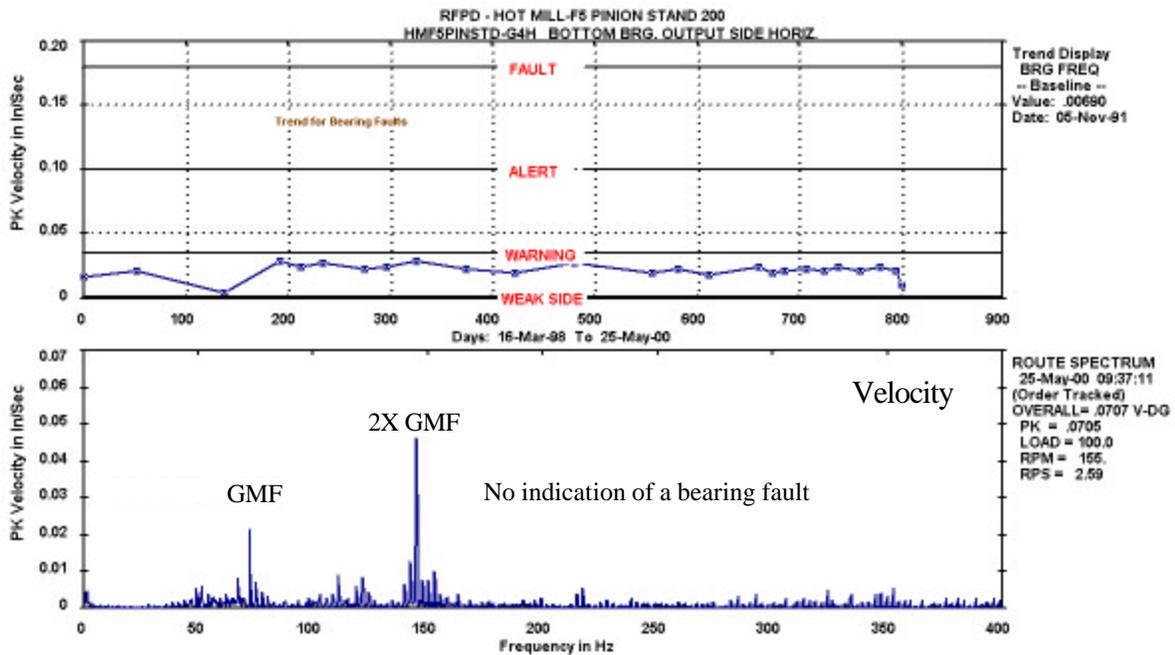


Figure 37. Trended bearing frequency energy from same data acquisition point as in Figure 30. Velocity spectral data contains no BPM indication.



Figure 38. Picture of defective bearing (Inner Race) identified to be detected by PeakValue Trend

For comparative purposes, the trend data from the same location for non-harmonic energy (covering bearing faults) from normal velocity vibration spectra are presented in Figure 37 (non-harmonic energy includes nonsynchronous data). There were no indications of a bearing fault in normal velocity spectral data.

Based on the trend values in PeakVue, a work order was released in June 2000 to replace the bearing. The bearing was replaced in July 2000. A picture of the defective bearing is presented in Figure 38. **The failure was clearly advanced and could have easily induced catastrophic failure by e.g., metal “chunks” from worn bearing components interfacing with the gear teeth during meshing.**

After bearing replacement, the trending of the PeakVue values was continued and is presented in Figure 39. In Figure 39, the parameter trended is the digital overall computed from the PeakVue spectral data. The general trend is very similar to the PK-PK values from PeakVue time data blocks, but, in itself, this parameter does not provide the analyst an indicator to judge severity other than comparison with reference values which are relative. The important point to be made from the data presented in Figure 39 is that the trended value returned to the initial reference value when the bearing was replaced.

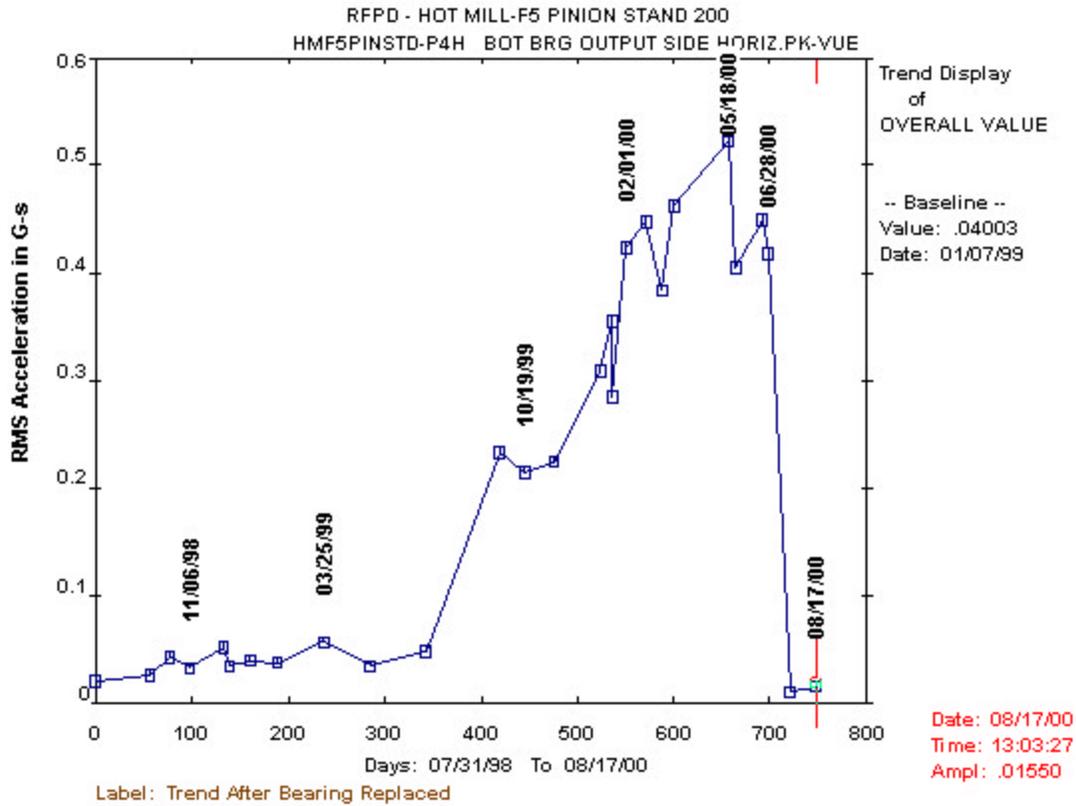


Figure 39. Trended Digital Overall computed from PeakVue spectral data Before and After the bearing replacement.

7.6 Detection of Cracked Inner Race on a Variable Frequency Drive Incinerator Fan

This case was performed on an Incinerator Fan direct coupled to a Variable Frequency Drive (VFD) motor. The plant had experienced various lubrication related problems that were identified to be misapplication of grease intended for ball bearings being used on roller bearing applications. The spherical roller bearings on the incinerator's fan shaft were among the bearings found to be using the improper lubricant. The lubricant problem had been a concern to the plant for some time and the plant was gradually replacing bearings in an attempt to remove all the improperly lubricated bearings and install bearings with a lubricant recommended for roller bearings. The plant had determined that flushing the grease out of the bearings was not feasible since the recommended replacement grease and the original grease were not compatible. This made bearing replacement the only sure way to rid the fan bearings of the misapplied lubricant.

The fan was outfitted with SKF 22222 CCK/C3W33 Double Row Spherical Roller Bearings whose bearing defect frequencies were as follows:

$$\begin{array}{ll} \text{BPFI} = 10.818 \times \text{RPM}; & \text{BPFO} = 8.182 \times \text{RPM} \text{ (19 rollers)} \\ \text{BSF} = 3.486 \times \text{RPM}; & \text{FTF} = 0.431 \times \text{RPM} \end{array}$$

Plans were made and a time scheduled for installation of new bearings with the proper lubricant. The replacement went well and the fan was re-started. Within only a few minutes, a set of vibration and PeakVue measurements was completed. Everything appeared to be normal in the vibration data except for the PeakVue measurement on the fan shaft's coupling end bearing.

Figure 40 includes a vibration velocity spectrum and acceleration waveform. This vibration data did not indicate any real problem within the bearing. Overall velocity was only .0443 in/sec with the highest amplitude being only .033 in/sec at 1 X RPM. Also, note that no bearing fault frequencies were present within the spectrum (only a few harmonics of running speed). Similarly, the acceleration level in the vibration acceleration waveform was a very nominal .427 g on this fan operating at 1268 RPM during this survey.

However, the PeakVue data taken one minute after the vibration data (and only a few minutes into the initial operation of this newly installed bearing) did suggest a problem as shown in Figure 41. Most importantly, pronounced spikes exceeding the 2.5 g Max Peak alarm level were present within the PeakVue time domain during each revolution of the fan shaft. Note that there was a constant difference of one revolution between each of the highest pulses in the waveform that reached as high as 3.09 g (in comparison with the peak acceleration level of only .236 g in the acceleration waveform shown in Figure 40). Next, looking at the PeakVue spectrum of Figure 41, note the presence of numerous harmonics of operating speed suggesting a mechanical looseness condition. Notice that no other frequencies from any other sources were present in the PeakVue data (including any bearing defect frequencies).

The PeakVue findings seemed contradictory since vibration levels appeared very smooth; bearing temperature was only 40°C (104°F); and there were no adverse sounds indicating a problem. These measurements were repeated several times as grease was added to the bearing in order to verify this was not a false alarm. Once again, the PeakVue data suggested a problem.

With PeakVue as the only indication of a problem, a decision was made to return the fan to service for a week and then to re-evaluate the bearing as well as the new grease. After the week passed, the fan was shut down and the bearing cap was removed. As grease was being wiped

away to get a better view of the bearing, a crack was noticed in the inner ring of this bearing. This immediately explained the PeakVue response captured during the previous week and its attempt to warn of the cracked race. Due to the one week running time, scoring was also noted indicating the inner race was turning on the adaptor bushing. The bearing was quickly replaced and the fan restarted. This time, the PeakVue data appeared with no distinct peaks in either the waveform or the spectrum as can be seen in Figure 42. And, of greater importance, the Pk-Pk level within the waveform dropped from 3.68 g down to .65 g. also, note that no real discernible spikes remained within the PeakVue waveform following replacement of the bearing.

Figure 43 includes a digital photo which shows the results of the bearing inspection which resulted in subsequent removal of the bearing. Note that a pronounced crack was found on the inner ring of the bearing where it fits to the journal. Also, if one looks closely at the photo, the crack is found to propagate through the inner ring, extending into the inner raceway of the bearing that was verified by removal and examination of this spherical roller bearing which revealed the cracked inner race when the rollers were placed in a cocked position. Also, looking at Figure 43, note the scoring on the inner bore which further indicated the bearing had been spinning on the shaft. The damage on the tapered bore resembled fretting-like damage although this was not due to chemically induced problems, but simply due to excessive looseness from faulty adaptor bore contact. This crack may indeed have been caused by excessive interference due to over torquing the locking nut (and/or a metallurgical fault within the bearing material).

One question remained concerning this bearing – if the bearing had a crack on its inner race like that shown in Figure 43, why weren't any BPF1 frequencies present within either vibration or PeakVue spectra? The likely reason for this was excessive looseness due to the crack. In other words, when a spall or other defect appears on an inner race, BPF1 frequencies will be generated spikes will occur in the waveform each time a rolling element impacts the flaw, and will have maximum response when a roller impacts the inner race when it is in the center of the bearing load zone. However, in the case of a fully developed crack like that shown in Figure 43, the bearing tends to lose its integrity and is no longer a tightly fitted component assembly. Thus, in the case of cracked inner races, they will sometimes generate BPF1 frequencies unless the crack has propagated through the inner ring in which case the bearing components will lose their tightness and integrity and excite only a series of running speed harmonics.

The most likely explanation for what caused the crack in the first place was that during initial bearing installation, the inner ring was expanded excessively by the adapter bushing; or that a metallurgical fault within the inner ring caused an immediate crack when the locking nut was tightened. And, unfortunately, the crack went unnoticed. Following the bearing replacement and use of the proper grease, the fan has continued to run with no further bearing or lubricant problems.

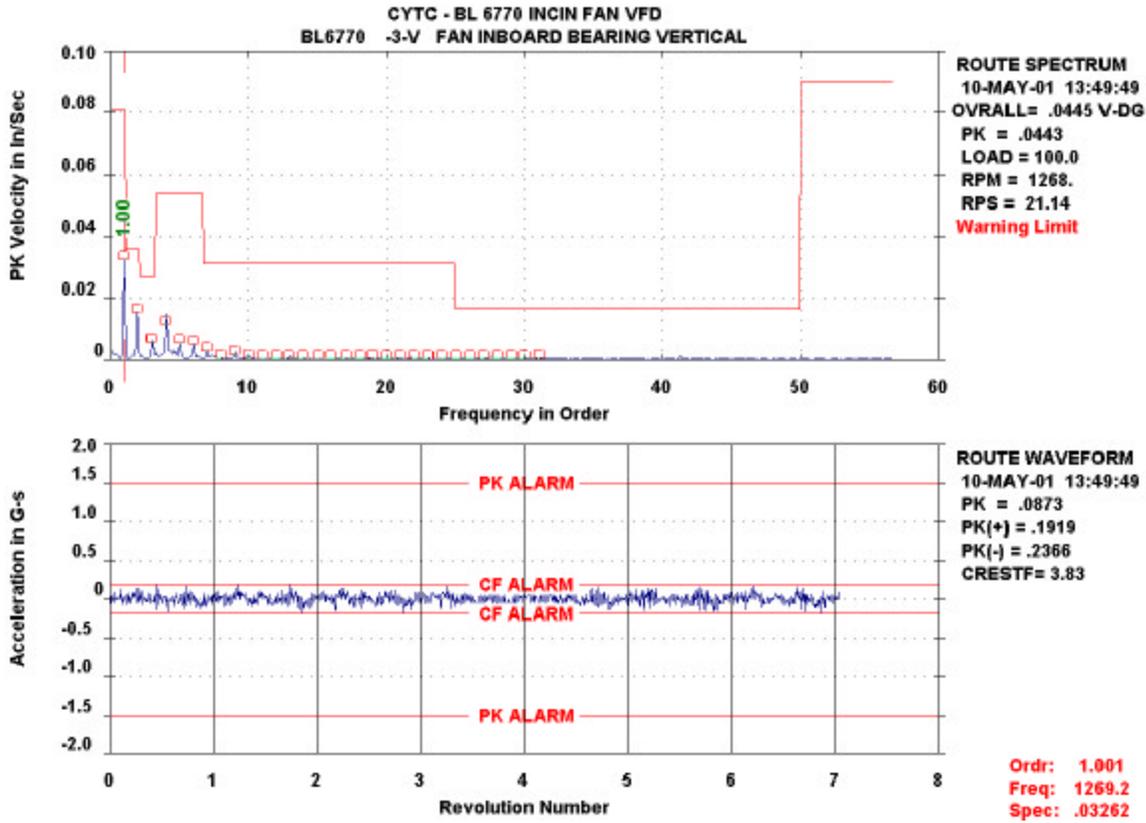


Figure 40. Vibration Data when Inner Race Crack was found to be Present

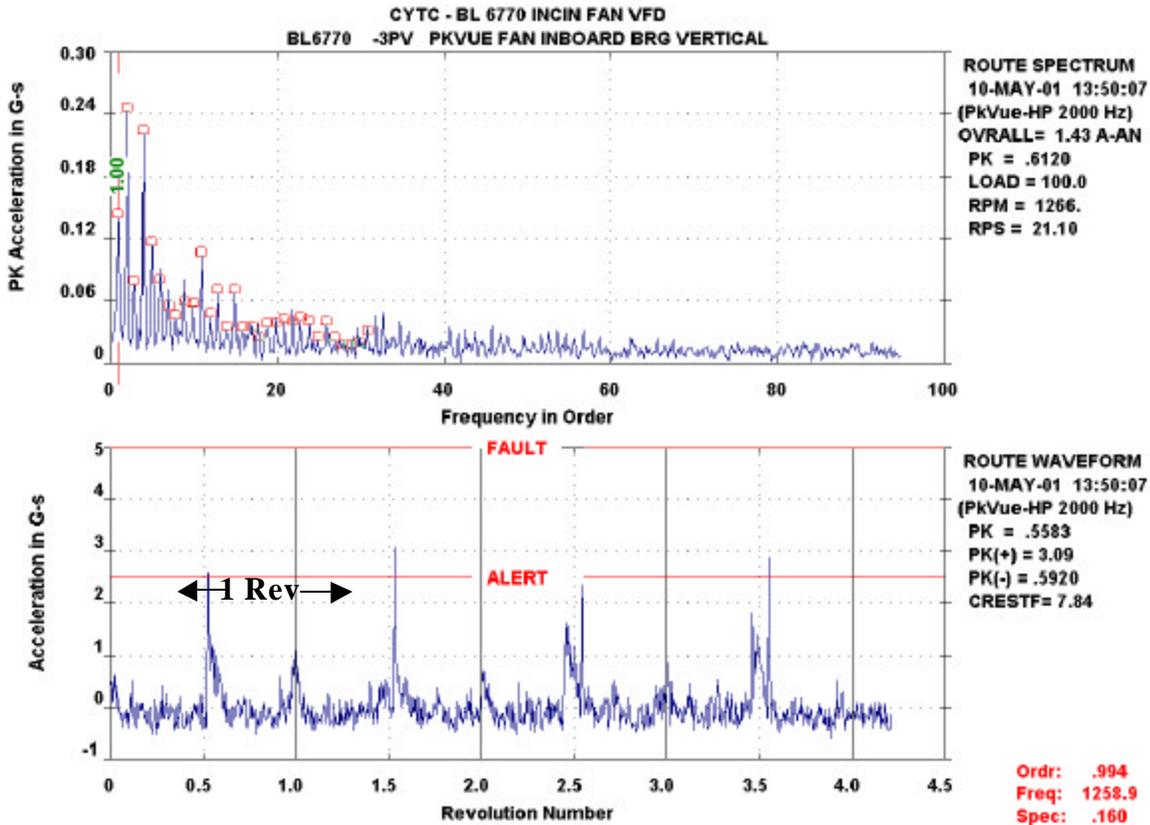


Figure 41. PeakVue Data when Inner Race Crack was Present (Note 1/Rev Pulses)

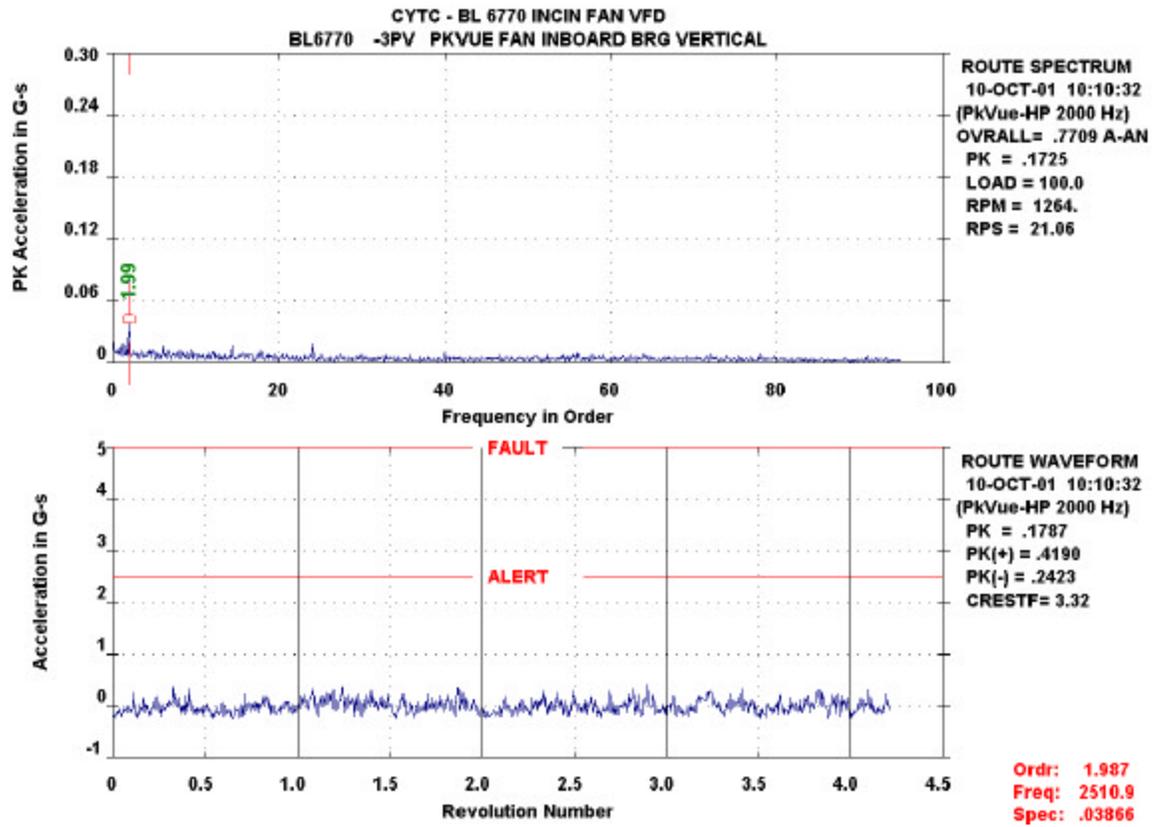


Figure 42. PeakVue Data After Bearing and Lubricant were Replaced

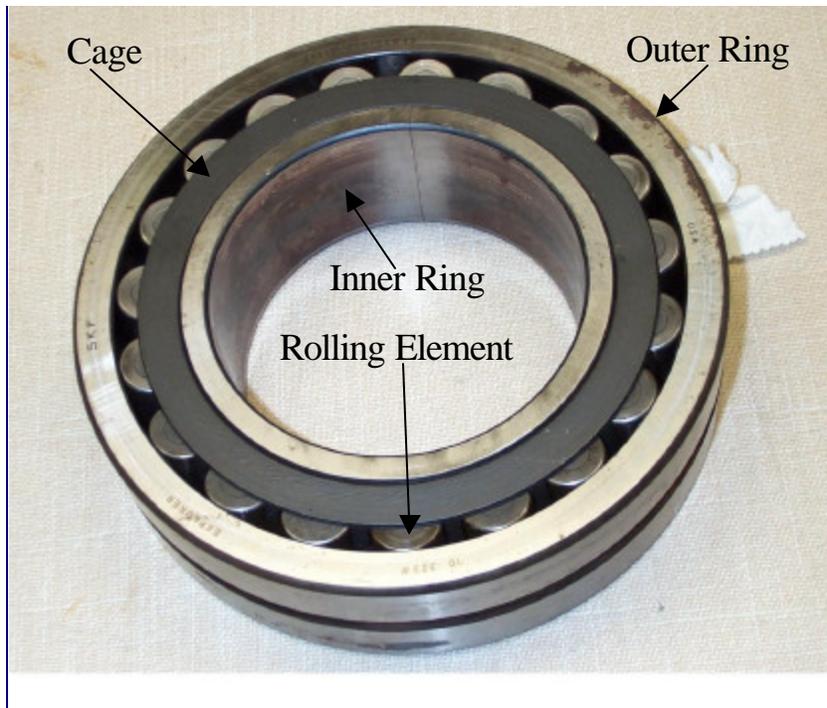
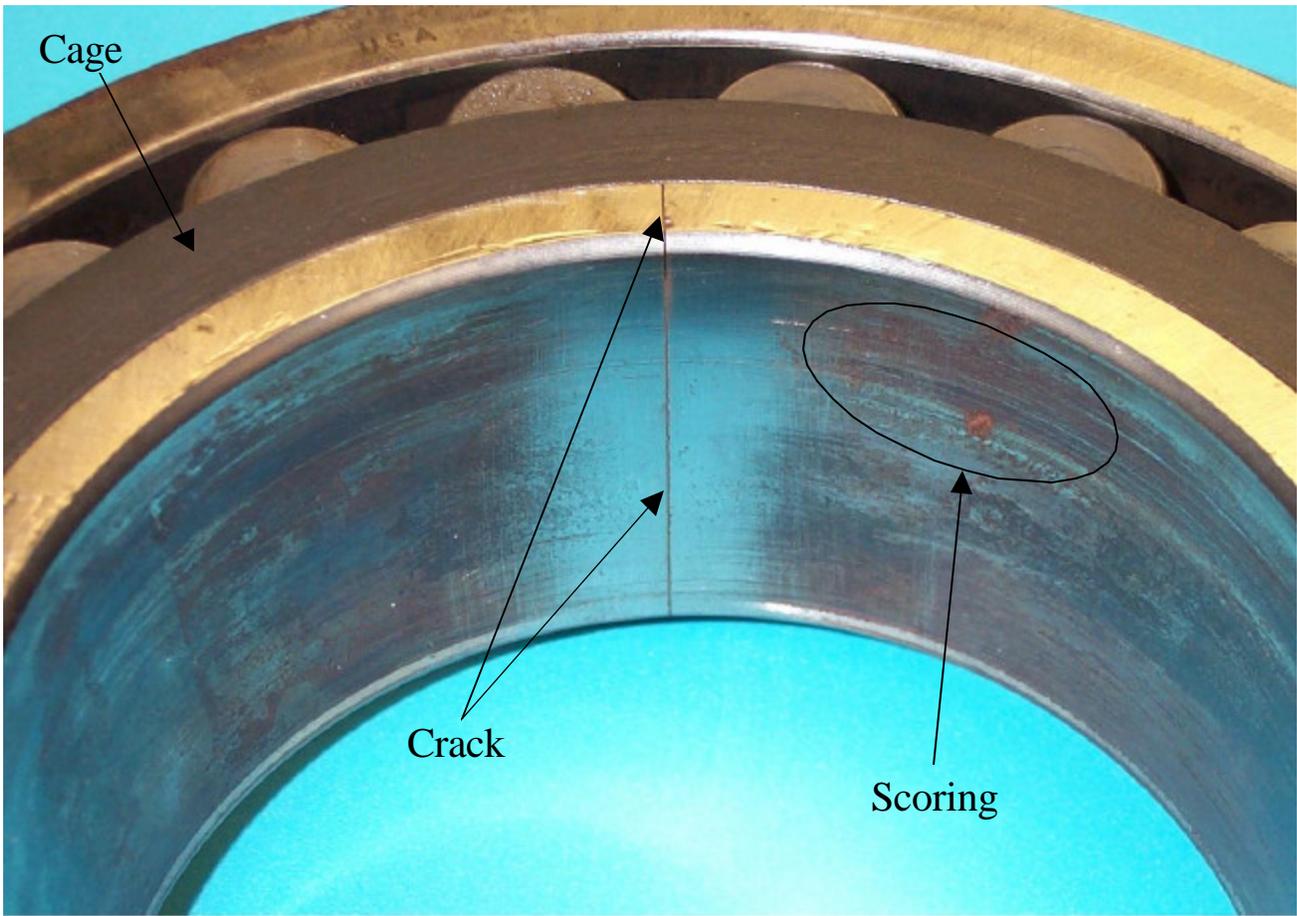


Figure 43. Cracked Inner Race and Scoring of Inner Ring on Bearing found by Removal of Bearing

7.7 Detection and Correction of a Lubrication Problem on an AC Induction Motor Driving a Belt-Driven Fan

Measurements on the motor belt driving a fan indicated increasing PeakVue “Max. Peak Waveform” levels as evidenced by the trend plot shown in Figure 44. A PeakVue spectrum suggested faults in the outer race of the inboard (drive-side) bearing as seen in Figure 45 which was captured on June 6, 2001. Note the presence of 5 harmonics of the outer race frequency (BPFO). The waveform likewise showed roughly 3 spikes per revolution (that corresponded to a BPFO of 3.56 X RPM as shown below). Initial recommendations were made to lubricate the motor bearings and evaluate the effect, particularly on PeakVue spectral and time waveform amplitudes. Vibration measurements taken on the same day and shown in Figure 46 did not sense bearing fault frequencies and the acceleration time domain showed only nominal Pk-Pk levels of 0.98 g for the motor operating at about 1796 RPM. This motor was outfitted with an NTN 6207 bearing that acted as the fixed bearing having to withstand thrust. Likewise, this same bearing also saw the greatest forces due to belt tension imposed upon it. Bearing fault frequencies for this NTN 6207 bearing were as follows:

$$\begin{array}{ll} \text{BPFI} = 5.435 \text{ X RPM}; & \text{BPFO} = 3.565 \text{ X RPM (9 rollers)} \\ \text{BSF} = 2.303 \text{ X RPM}; & \text{FTF} = 0.396 \text{ X RPM} \end{array}$$

During the next survey taken on August 9, 2001, a high-pitched, screeching noise was heard when taking data on this particular machine. Subsequent vibration velocity and acceleration measurements did not indicate a problem as shown in Figures 47 and 48. About the only frequency appearing in the vibration velocity spectrum was 1 X RPM, with very small amplitudes of running speed harmonics (but no bearing frequencies appeared). And, in the vibration acceleration spectrum in Figure 48, there was some low amplitude activity in the region of 40,000 to 60,000 CPM (roughly 700 – 1000 Hz) but the highest amplitude in this region was only .025 g while the waveform Pk-Pk level was only a moderate 0.85 g.

However, the PeakVue data taken on August 9 showed an increasing trend in the plot of Figure 44; and the PeakVue data in Figure 49 showed increasing numbers of BPFO harmonics. Thus, a decision was made to grease the bearing while simultaneously looking at PeakVue spectral and time waveform displays in the portable analyzer as grease was added to this drive-side motor bearing. Technicians continued carefully adding grease and watched as PeakVue amplitudes dropped. However, they stopped adding grease when PeakVue amplitudes began to climb. Figure 44 shows the significant drop in the PeakVue “Max Peak Waveform” level from 3.85 g to 0.49 g after introduction of new grease was added to the bearing. Likewise, Figures 49 and 50 compare PeakVue spectral and waveform data “Before” and “After” grease was added. PeakVue taken after lubricating the bearing with grease showed that each of the spikes within the time domain disappeared and there were no BPFO frequencies whatsoever still present within the PeakVue spectrum in Figure 50.

In this case, vibration acceleration data was again taken immediately before and after grease was added on August 9 as shown in Figures 51 and 52, but showed only 1X, 2X and 3X RPM components, along with some activity again in the 40,000 to 60,000 CPM region (however surprisingly, note there was no real change in this higher frequency region when the new grease was added as shown in Figure 52). Unlike the PeakVue data, note that there was no real indication in the vibration data of a lubrication problem either before or after grease insertion. The vibration acceleration waveform showed no real change whatsoever (in fact, the crest factor after lubrication

was slightly higher than that before lubrication; higher crest factors show less sinusoidal activity which indicates more impact/impulse response).

In summary, this case study showed that PeakVue can be a reliable indicator of lubrication problems, indicating anomalies either due to inadequate lubrication, over-lubrication in the case of grease, or employment of the incorrect lubricant for the application.

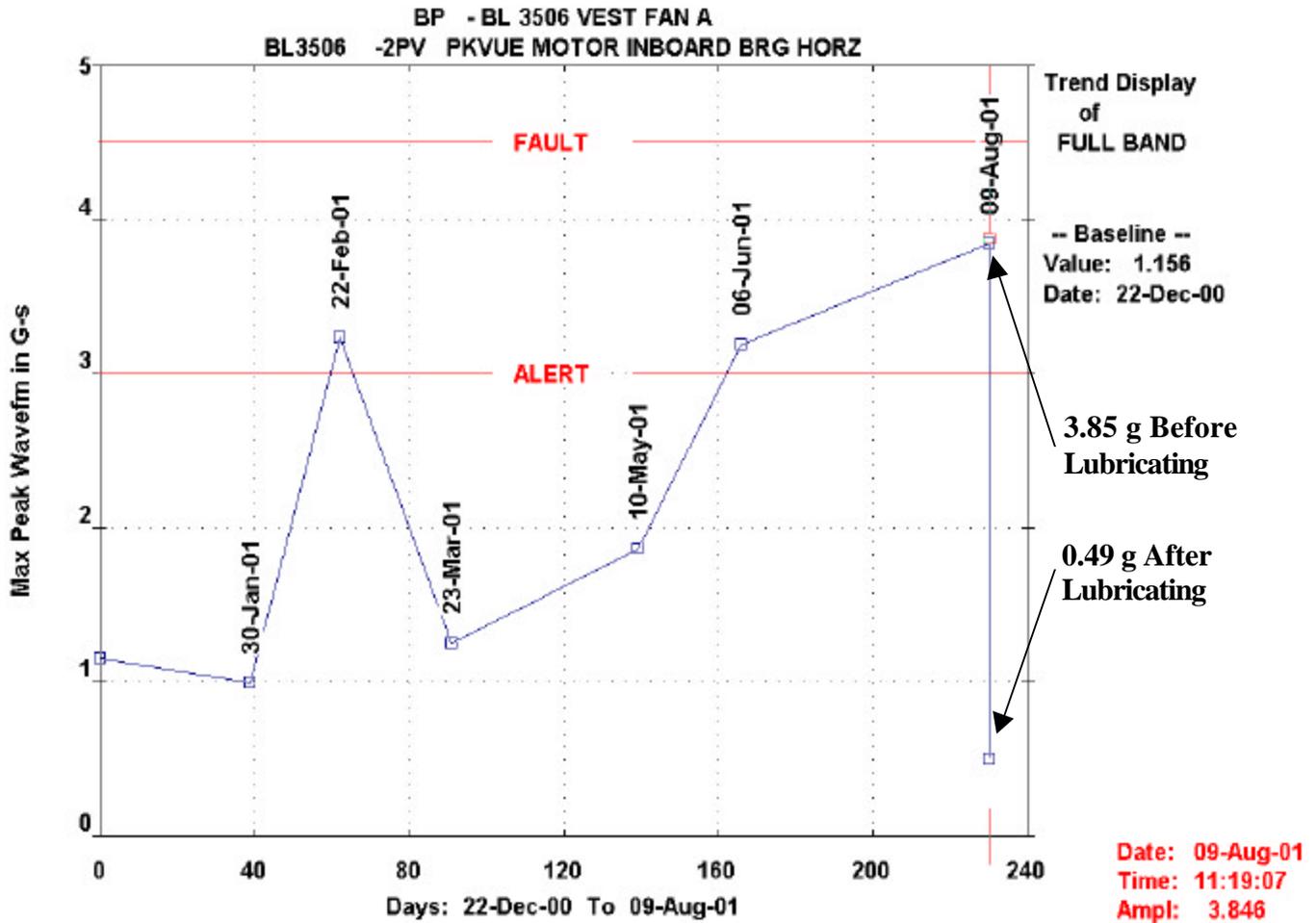


Figure 44. Trend of PeakVue Max Peak Waveform Level Before and After Problem Resolution (Note the drop from 3.85 g to 0.49 g when new grease was added on 9-Aug-01)

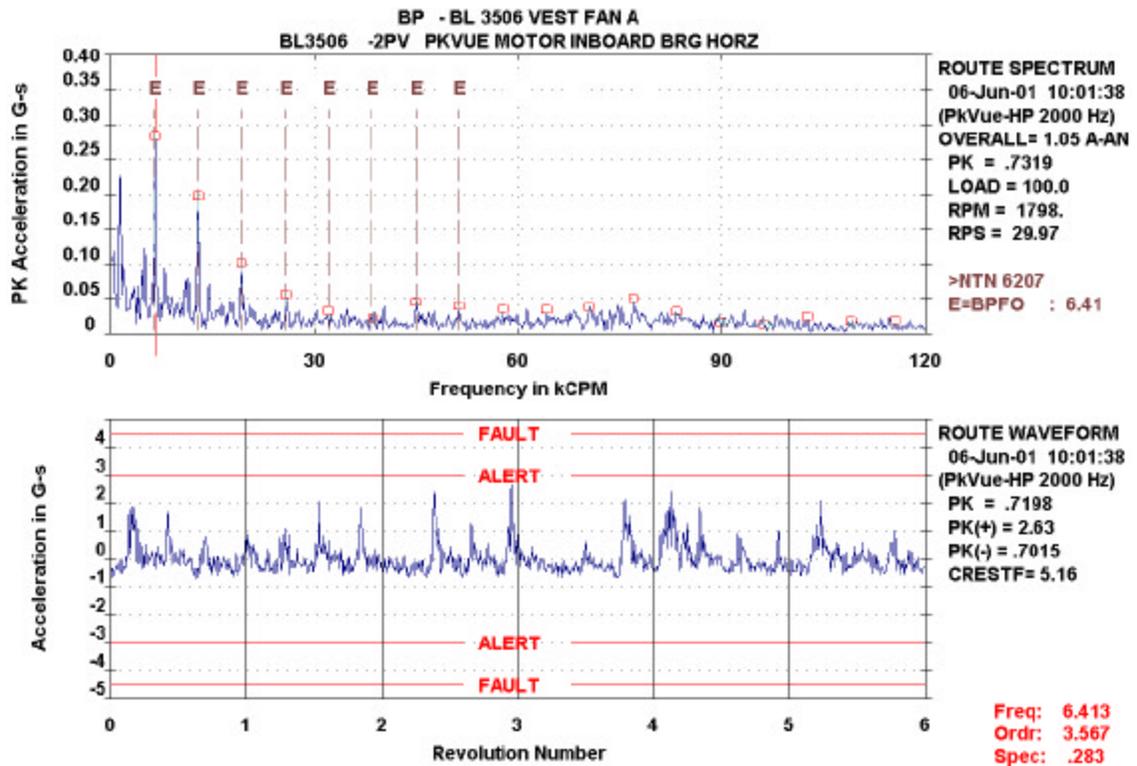


Figure 45. PeakVue Data Showing Many BPFO Harmonics (June 6, 2001)

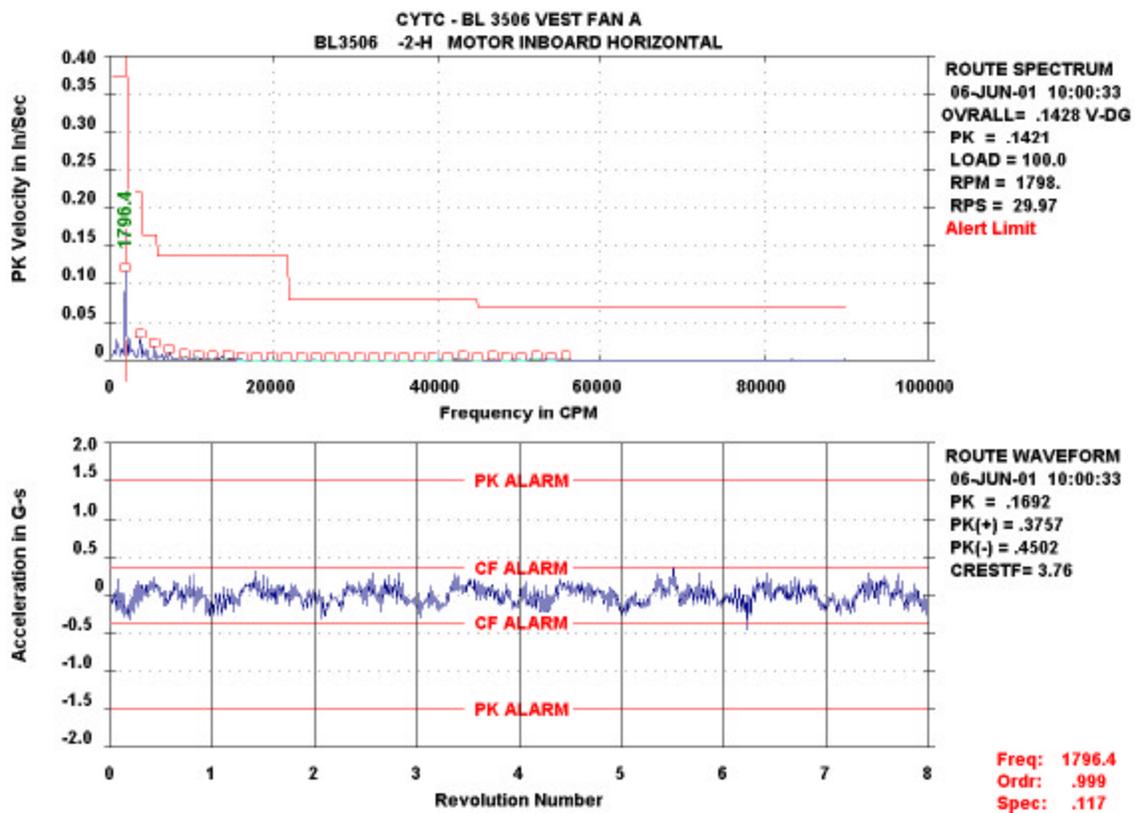


Figure 46. Vibration Velocity Data Also Taken on June 6, 2001

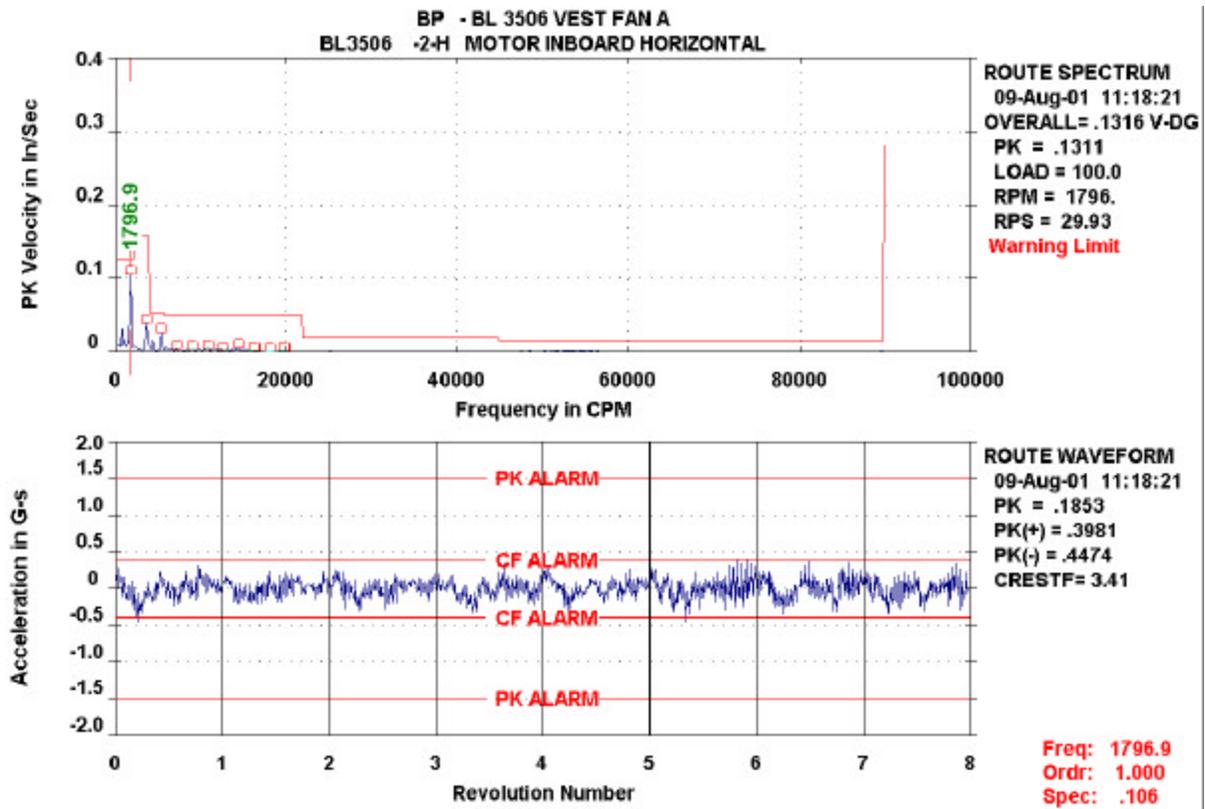


Figure 47. Vibration Velocity Taken when High-Pitched Sound was Heard (Aug. 9, 2001)

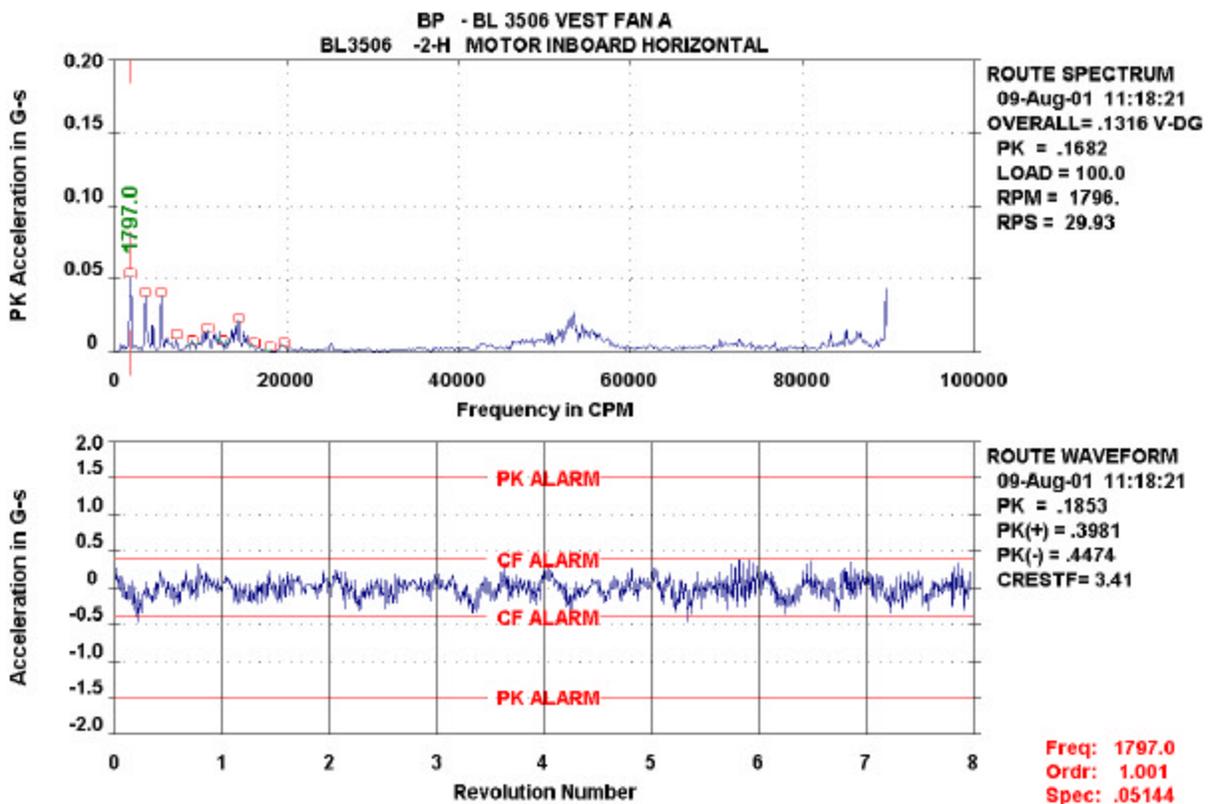


Figure 48. Vibration Acceleration when High-Pitched Sound was Heard (Aug. 9, 2001)

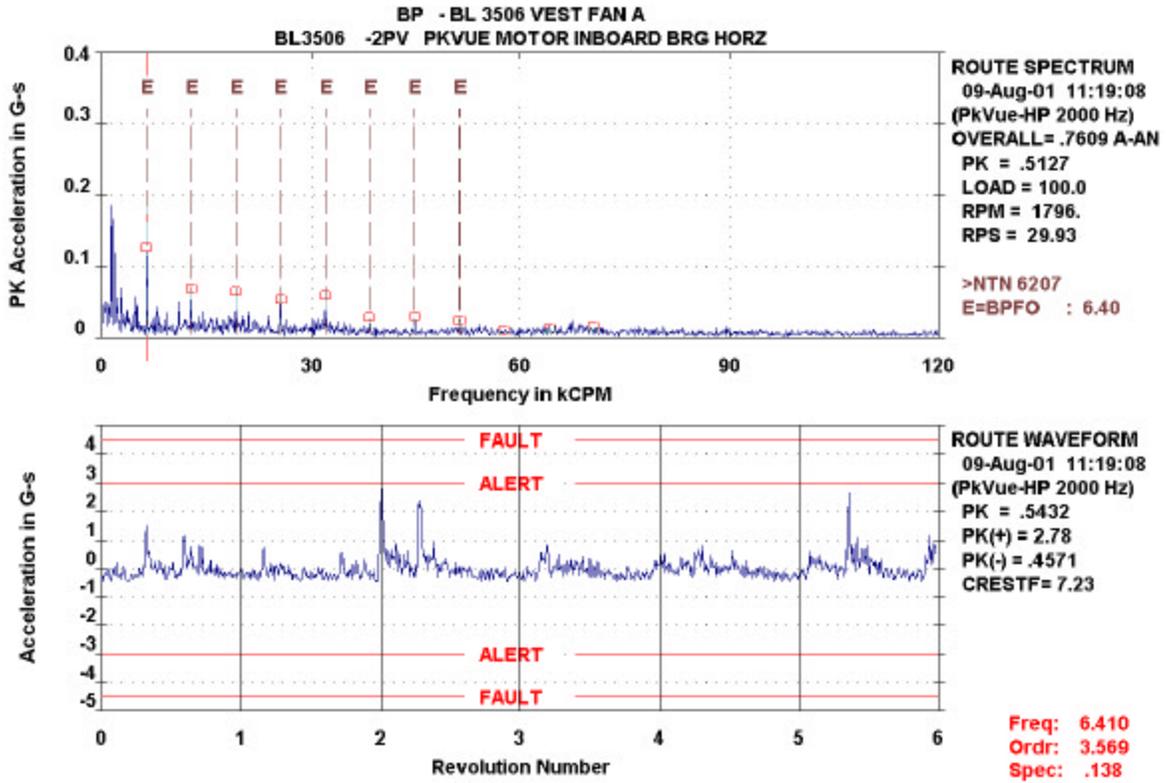


Figure 49. PeakVue Taken Just Before Adding Grease (Aug. 9, 2001)

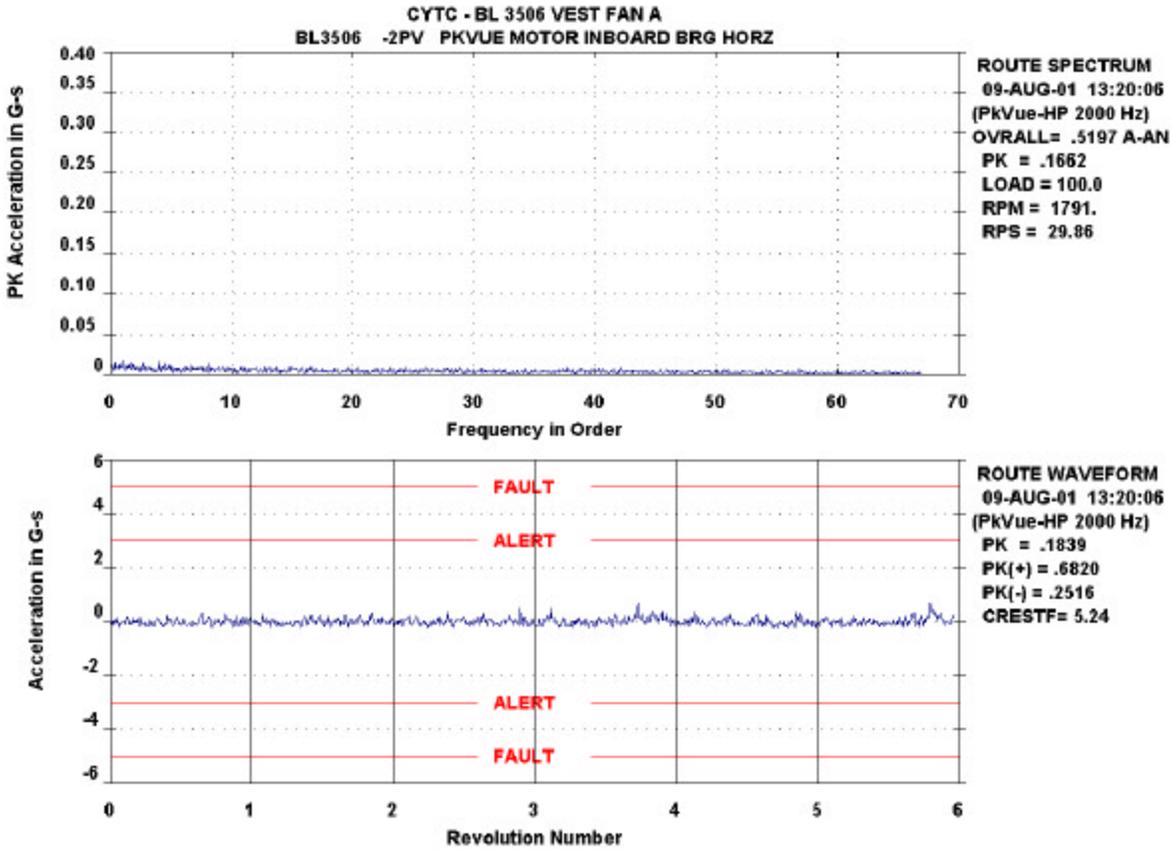


Figure 50. PeakVue Taken Just After Adding Grease (Aug. 9, 2001)

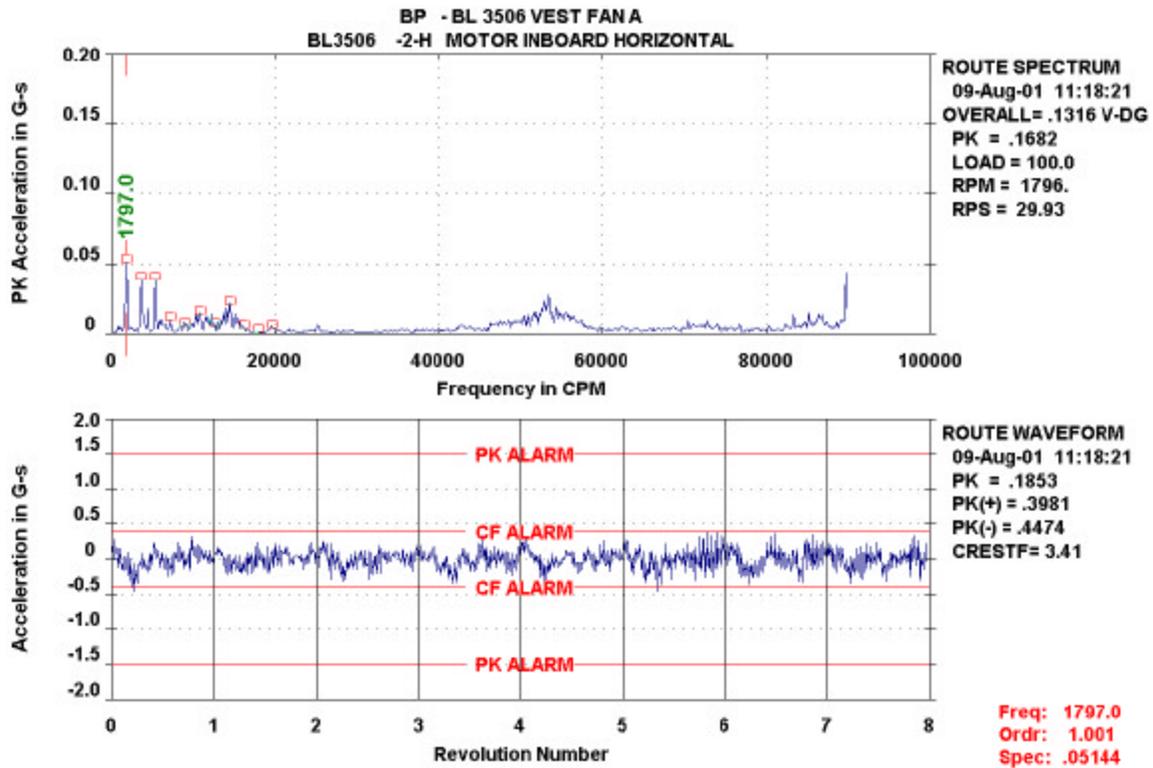


Figure 51. Vibration Acceleration Taken Just Before Adding Grease (Aug. 9, 2001)

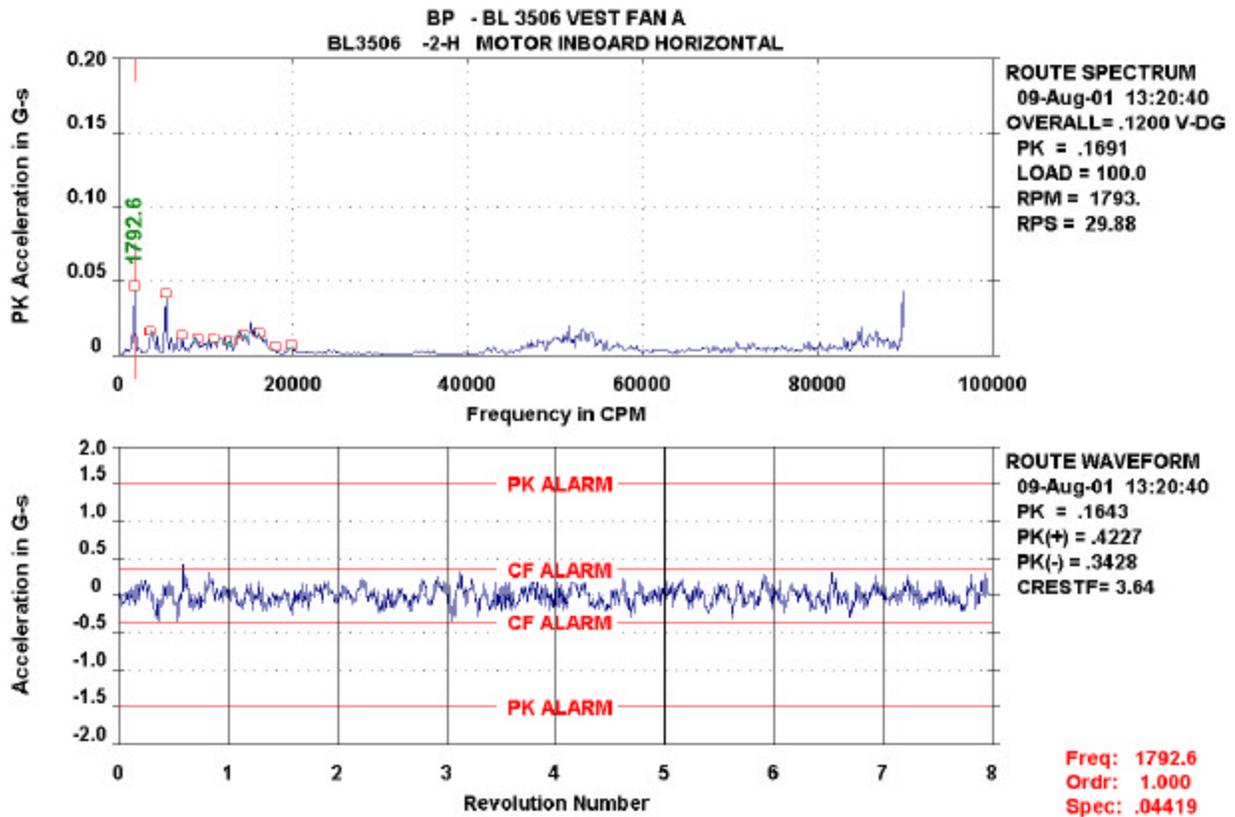


Figure 52. Vibration Acceleration Taken Just After Adding Grease (Aug. 9, 2001)

8.0 Summary, Conclusions and Recommendations

8.1 Summary

This paper began with a summary of nondestructive testing employing stress waves. In some cases, the stress waves were introduced into the elastic medium by a) high frequency impacting through electrical excitation of piezoelectric crystals or b) impacting the structure through dropping various size metallic balls of sizes less than 0.5 inch Diameter onto the surface. For acoustic emission testing, cracks within the vessel walls are introduced as a result of stressing the vessel at higher levels than normally experienced by the vessel. The sudden release of stress through cracking introduces movement of material on a microscopic level and subsequent emission of stress waves.

Stress waves can be present as both longitudinal and bending waves. The bending waves are dominant for activities present in a rotating machining. A brief quantitative framework based on the Hertz theory was presented for the generation of stress waves impacting events. Even though broad frequency bands are excited when impacting occurs, it was found (theoretically and experimentally) that the preferred frequency band lies in the 1 to 50 kHz range with the preferred frequency inversely related to the size (mass) of the impacting element. The key parameter, which determines the preferred frequency band, is the contact time between the two bodies during the impact.

Spalling and fatigue cracking also are sources for stress wave emission, but the equivalent contact times are generally short relative to impacting (about the same for balls less than 0.5 inch diameter).

Once the stress waves are generated, they propagate away from the initiation site experiencing dispersion due to the fact that the propagation speed of bending waves is proportional to the square root of the frequency within the wave packets. In addition to dispersion, the attenuation of the waves is inversely proportional to the frequency of the wave, which is magnified when crossing a boundary between media.

Given that a stress wave packet reaches an area where they can be detected, attention must be given to the sensor to be employed for detection. If it is an accelerometer, attention must be given to the sensor bandwidth and its mounting. If it is an ultrasonic (which generally is a narrowband) sensor, then attention should be given to the most probable frequency where energy will be found from the stress wave generation and propagation to the sensor location.

Fault identification generally is made possible through spectral analysis of captured stress wave events. It was demonstrated that capture in such a manner as to preserve the peak levels of the captured stress waves provides the analyst with a valuable parameter for trending and severity analysis.

The claims of peak value methodology for stress wave capture and analysis to detect and isolate faults, provide meaningful trending capability, and assist in severity assessment was demonstrated through several case studies. These studies included impacting and spalling. One case clearly demonstrated the expectation that, for a roller

in the 1 inch diameter range, impacting is accompanied with lower frequency excitation than for the same roller experiencing spalling.

8.2 Conclusions

The capture and analysis of stress waves, which accompany many faults within rotating machinery, provides the analyst with very useful information regarding condition monitoring of rotating machinery. The classical spectral analysis identifies a fault is present within the machinery and generally, the specific component involved. The capture of the true peak value found within the time waveform associated with each impacting event further provides the analyst the ability to carry out severity assessment using both the "Max. Peak Waveform" amplitude trend and also the distribution (or "pattern") of frequencies within stress wave spectra.

For most faults, which generate stress waves, the dominant frequency band for the resultant stress wave packets will lie in the broad frequency band of 1 to 50 kHz. For larger ($>1.0D$) rollers impacting, the dominant energy will be in the 1 to 10 kHz range; whereas for the same rollers experiencing spalling, the dominant energy will typically be found in the 5-15 kHz range.

Even though the dominant energy band may lie outside the frequency band, the sensor responds to the total band of energy excited from stress wave activity. This is generally broad and hence may be detected (perhaps not optimally) with many sensor selections (sensor plus mounting). However, when monitoring lower diameter rollers (less than $0.5D$), a sensor system responsive to a minimum of 10-15 kHz should be employed, i.e., the commonly used accelerometer on a 2-rail magnet attached to the surface may be unresponsive to important stress wave activity (a flat, rare earth magnet mounted on a clean, flat surface may be required to detect stress wave energy concentrated in frequency ranges at or above approximately 5000 Hz).

The detection of the peak values associated with impacting events has proven to provide meaningful data for fault severity assessment. The spectra data computed from the time waveforms consisting of sequential peak values incorporating 10 or more revolutions of the machinery monitored provide fault identification. In addition to spectral analysis, the autocorrelation function coefficient computed from same peak value time waveform provides further diagnostic assistance.

8.3 Recommendations

Stress wave capture and analysis, where emphasis has been placed on capturing the peak values associated with each event, has proven to be a very useful methodology for fault detection, fault progression, and severity assessment. Therefore, the following recommendations are presented.

1. Specify measurement points for peak value (PeakVue) analysis at each bearing on each machine. When establishing such stress wave points, use the PeakVue measurement setup criteria given in Table III within section 4.2.5.
2. Extract and trend the "Max. Peak" value from the PeakVue time waveform. This true peak value should be extracted from the peak value time block covering a minimum of six revolutions (preferably 10+ revolutions) of the machine.

3. For rolling element bearings, it is recommended that a minimum of 16 revs be included within the peak value time waveform (providing adequate resolution for cage fault resolution).
4. The bandwidth of the spectrum computed from the peak value time block should include the maximum of:
 - a. A minimum of three harmonics of the BPFI fault frequency for rolling element bearings
 - b. Two times gear mesh for gearing systems. If this requires two measurement sets (one for gear faults and one for bearing faults), then proceed with two measurements.
5. The high-pass filter setting should always be equal to or greater than the maximum bandwidth measurement used for the peak value analysis on the machine for all peak value measurement sets on a specific machine.*
6. The parameters, recommended for trending are:
 - a. Pk-Pk value (in sensor units) of the peak value time waveform extracted from a time block inclusive of 6 or more revolutions of the shaft measurement point assigned to.
 - b. Digital overall computed from the entire broadband spectrum.
 - c. Energy of 4-10 synchronous shaft revolutions.
 - d. Energy of non-synchronous energy covering BSF, BSFO, and BSFI.
 - e. Energy of subsynchronous energy covering cage frequency.
 - f. Energy surrounding one and two times gear mesh.

*By machine, the definition is based on that segment covered by a continuous metallic housing.

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APPENDICES

Appendix A. Application of Band Pass Filters during PeakVue Measurements on Special Machinery Types or Operating Conditions

Appendix B. Explanation of the Autocorrelation Coefficient Function

Appendix A

Application of Band Pass Filters during PeakVue Measurements on Special Machinery Types or Operating Conditions

A.1 Introduction

In the body of this report, the emphasis has been placed on stress waves generated between two metallic objects. These events lead to stress wave packets consisting primarily of higher frequencies (greater than 1 kHz). Thus, the emphasis was placed on using high-pass filters in the conditioning path of the signal. There are times when it is more appropriate to select band pass filters. One such event occurs on paper machines (as well as on press machines). This occurs when a felt develops certain classes of flaws which cause the felt to impact the rolls. Felt is constructed of a soft material. Thus, when a felt impacts a hard material, it excites much lower frequencies than does impact of hard material on hard material.

In the next section, a case study of a defective felt on a paper machine detected using PeakVue is presented. The last section will be a discussion on when a band pass filter selection in conjunction with PeakVue is the appropriate choice.

A.2 Case Study: Defective Felt on a Paper Machine

The velocity (ips) spectral data acquired on the 4th press roller on 16-Feb-00 are presented in Figure A.1. The activity in the vicinity of 50-60 Hz was noted to be greater than it had been. The velocity time waveform of Figure A.1 does not indicate any problem. The activity in the spectrum, especially in the 50-60 Hz range, does suggest periodic activity. The autocorrelation coefficient was calculated and is presented in Figure A.2. Here, there are possibly two periodic events occurring. The highest frequency event, the minimum lag time, is at 32 Hz (about 30% correlation) which is the dominant peak in the spectrum. The second has a period of 1.3 sec which corresponds to one event per 5+ revolutions of the roller. This longer period event could correspond to once per revolution of the felt.

A decision was made to apply PeakVue at this measurement point. A felt impacting will most likely excite a structural resonance frequency. For press sections, this has been observed to typically be in the 50-150 Hz range. Thus a band pass filter was selected, which incorporates the suspected structural resonance. The filters available are presented in Table II. Both the 20-150 Hz and the 50-300 Hz band pass filters were employed. The PeakVue data for the 20-150 Hz band pass filter are presented in Figure A.3. The only activity of note in the spectral data is the activity at 0.193 order (which is the felt turning speed) with many harmonics (where "first order" refers to 1 X Roll speed). The PeakVue time waveform does confirm the repetitive pattern of 0.193 orders but the autocorrelation coefficient function in Figure A.4 leaves no doubt of the impacting at the felt turning speed (note that $0.193 \times \text{RPM} = 0.787 \text{ Hz} = 47.2 \text{ CPM} = 1 \text{ X Felt RPM}$). Note the clear impacts occurring at the rate of once per 5+ roll revolutions in the Autocorrelation coefficient in Figure A.4.

The obvious conclusion was that the felt had a minimum of one defective region. This was confirmed and the felt was replaced.

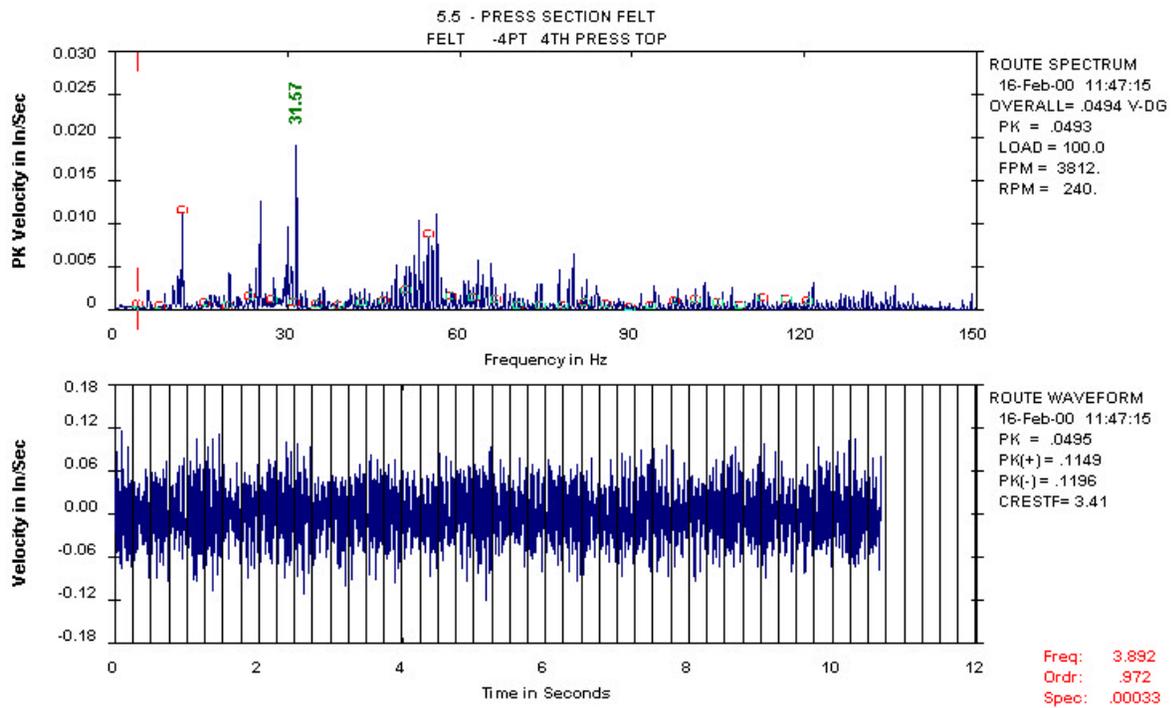


Figure A.1. Velocity spectral and time waveform from Sensor Press Roll in paper machine collected on 16-Feb-00.

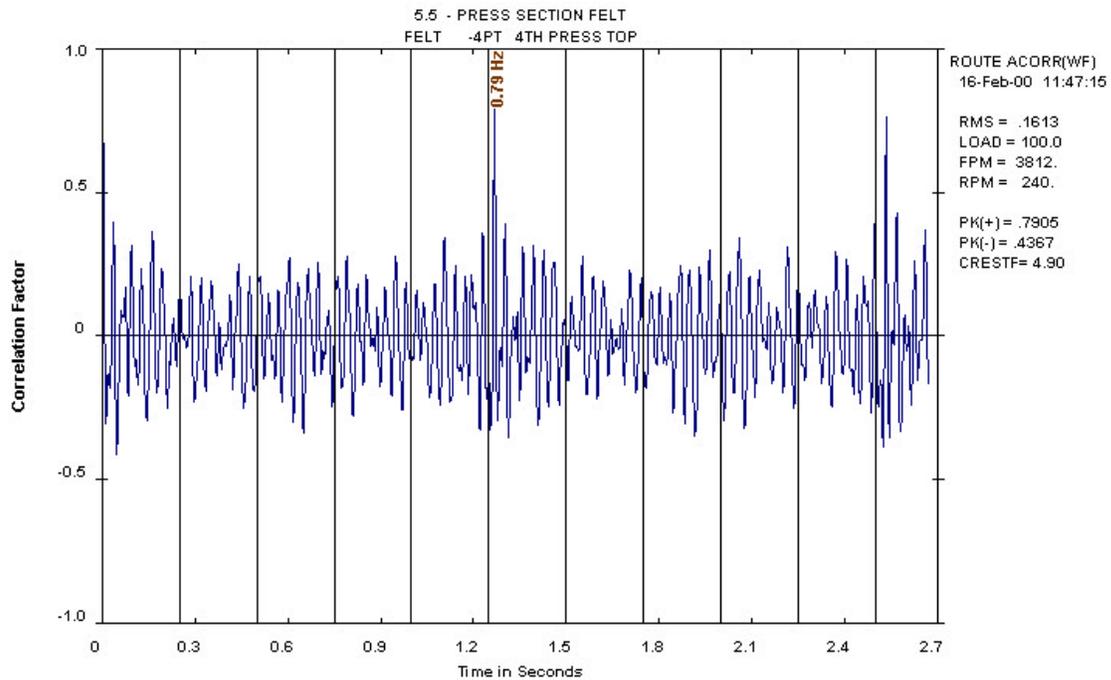


Figure A.2. Autocorrelation coefficient computed from the velocity time waveform in Figure A.1.

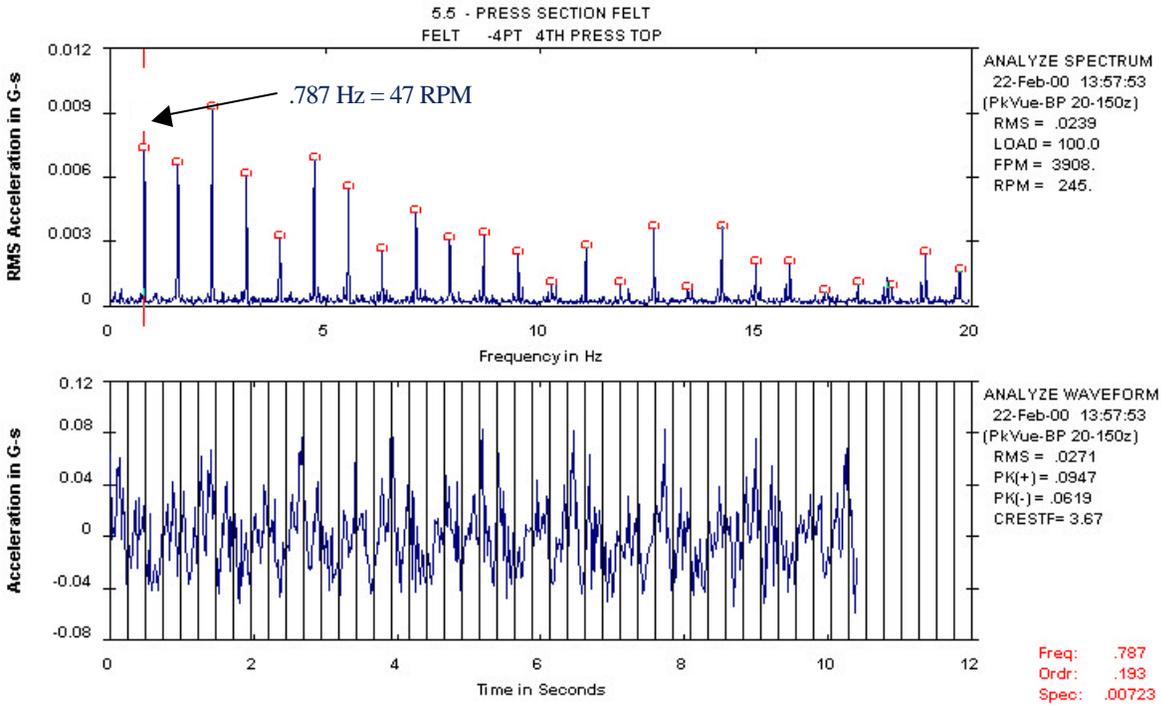


Figure A.3. PeakVue spectrum and time waveform from same measurement point as that of Figure A.1 in sensor units collected on 22-Feb-00.

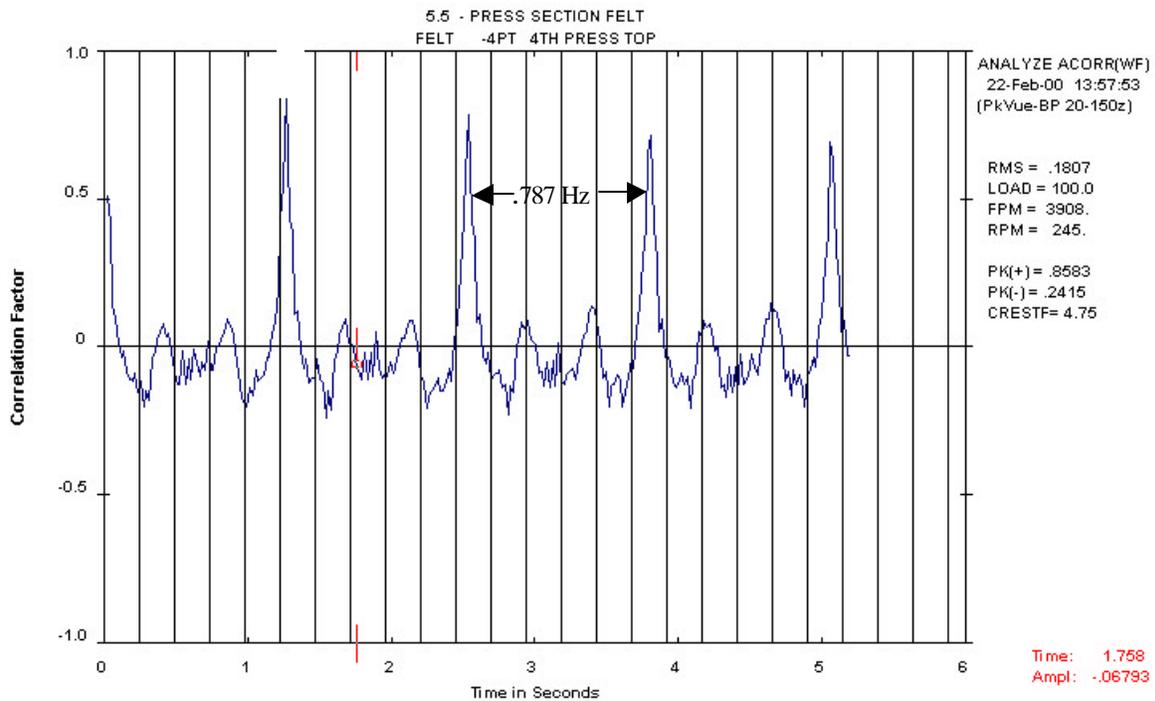


Figure A.4. Autocorrelation coefficient computed from PeakVue Time waveform from Figure A.3.

A.3 Consideration for Selecting a Band Pass Filter in Lieu of a High-pass Filter

An obvious application for selecting a band pass filter over a high-pass filter is when structural resonances (or other system natural frequencies) could possibly be excited by an impacting event which occurs at a slow rate but is periodic (a defective felt is a text book example). A less obvious case is when monitoring for bearing faults on a gearbox that has rolling elements of reasonable size (greater than 0.5"D), along with gear mesh frequencies within the system.

To illustrate, consider a certain gearbox driven by a gas turbine with the objective of generating power. The input gear mesh was about 10 kHz. A lower gear mesh in the gearbox was about 3.7 kHz. The objective was to detect a certain bearing with faults. If we follow the rules in Table III within section 4 regarding selection of the high-pass filter, and refer to the available filters shown in Table II within section 4, we would select the 20 kHz high-pass filter. The problem is we would be attempting to detect possible impacts from gears having significantly attenuated energy at frequencies greater than 20 kHz.

The solution is to select a band pass filter which is sensitive to energy in a frequency band excluding gear mesh and two times gear mesh. The approximate 3.7 kHz gear mesh is the one closest to the region we expect most energy from impacting rollers. Thus a band pass filter was selected with a bandwidth of 5 kHz to 6.5 kHz (see Table II).

Other special applications may benefit from different band pass filters.

Appendix B

Explanation of the Autocorrelation Coefficient Function

B.1 Introduction

In this appendix, the objective is to introduce the autocorrelation coefficient function from a mathematical perspective followed by an example to assist the user in developing a “feel” for its properties and how it can assist the analyst in an overall diagnostic effort. In the next section, the mathematical definitions are presented (extracted from Ref. 9) with discussions relative to their use in machine condition monitoring. In the last section, an example using the LABview program is presented. Here, noise will be introduced to assist in developing a “feel” for how the autocorrelation coefficient function responds when a signal contains periodic events mixed in with random noise.

B.2 Basic Discussion of Autocorrelation Coefficient Function

The basic mathematical definition of the autocorrelation function (from Ref. 9) is:

$$R_x(\tau) = \lim_{T \rightarrow \infty} \frac{1}{T} \int_0^T x(t) x(t+\tau) dt \quad (B.1)$$

The various quantities in Equation (B.1) are:

1. $x(t)$ represents the value of the signal x , at the time t .
2. $R_x(\tau)$ represent the value of the autocorrelation function at the time (referred to as delay time τ), derived from and continuous with time signal $x(t)$.
3. T is the total time for which the integration defined in Equation (B.1) is carried out. The “ $\rightarrow\infty$ ” represents that the integration is carried out for a long period of time.

The spectral data we normally rely upon in our normal vibration analysis is compiled by first computing the PSD (power spectral density) function using the FFT algorithms for Fourier transformation and then, with appropriate normalization, taking the square root of the normalized PSD function at each frequency point. Therefore, we should be able to compute the autocorrelation function by first contracting the PSD versus frequency for each frequency point from the spectral data followed by inverse Fourier transformation (which is accomplished through the same FFT algorithms). If we pursue this method, it is difficult to generate the physical understanding we are seeking. Thus we proceed directly from the defining equation, Equation (B.1).

In Equation (B.1), it is assumed that the function (signal) is continuous and has no limit. The signals analyzed are converted from continuous (analog) to discrete (digital) through an A/D converter. We limit the bandwidth of the analog signal by passing its output signal or data through a low pass (anti-aliasing) filter to eliminate any aliasing.

In the digital domain, the continuous analog signal is converted to discrete values at sequential discrete times established by the sampling rate. Typically, the discrete

transform of X values will consist of a block size of 2^n (n is an integer) points which are the correct size for further processing using the FFT algorithms. The number of points more commonly used are 1024 for 400 line resolution; 2048 for 800 line resolution, etc.

Assuming a block size of 1024 samples (or “points”), i.e., we have digital values for $x(t)$ at constant Δt intervals (inverse of sampling rate) from the first interval through the 1024th interval. Represent this set of numbers by

$$\{x_j\} = x_1, x_2, x_3 \dots, x_{1024},$$

by the product of [“l” times Δt] and Δt is the inverse of the sampling interval. Normalize the set $\{x_j\}$ so that the mean value is zero.

Referring to Equation (B.1), we need the product of x at a specific time $t, i\Delta t$, and the value of x at a specific time $(t + \tau)$, where τ is the lag time. Let the lag time be represented by $(j\Delta t)$. Then the integrand $x(t) x(t + \tau)$ can be represented at a specific time $(i\Delta t)$ by $(x_i x_{i+j})$. Using this nomenclature, the equation defining the autocorrelation function becomes:

$$R = \frac{1}{M} \sum_{i=1}^M x_i x_{i+j}, \tag{B.2}$$

where M is equal to or less than $N/2$ number of points in the $\{x_j\}$. The set will contain both positive and negative numbers. In the summation of the $x_i x_{i+j}$ overall i values, some of the products will be positive and some will be negative for all j_s except $j = 1$. For $j = 1$, Equation B.2 becomes the mean square for the set $\{x_j\}$. For the set of numbers making up the autocorrelation function, R_j , the first mean square will always be the largest or, as a minimum, equal to the largest value.

If the set $\{x_j\}$ represents a random set of numbers, all values of R_j , with the exception of the first sample, will be zero (providing the set is sufficiently long). This is so because the set of numbers in the string of $x_i x_{i+j}$ with $j > 1$ will have equal numbers of + and – values. Accordingly, noise contributions to the set $\{x_j\}$ will tend to cancel out leaving components that are periodic exceeding zero when the lag time (t or j) corresponds to the period of the periodic events.

Given the properties that the first component in the autocorrelation set $\{R_j\}$ is the largest and the fact that noise tends to disappear in the set $\{R_j\}$, a new function, the autocorrelation coefficient function (C_j) is defined as:

$$C_j = \frac{1}{R_1} [R_1, R_2, \dots, R_M] \tag{B.3}$$

The C_j values will range between ± 1 . The noise components in the original signal set $\{x_j\}$ average out to zero. Therefore, the autocorrelation coefficient function, C_j , is a measure of the degree of correlation at each value of j (τ) with highly correlated values approaching ± 1 . This is one of the properties of this function that makes it a very useful

tool for the condition monitoring analysis. A second property is the property that the concept of “harmonics” commonly encountered in spectral analysis do not exist in the autocorrelation domain, i.e., all the energy in the many harmonics sometimes encountered in spectral analysis does not exist in the autocorrelation domain, i.e., all the energy in the many harmonics sometimes encountered in the frequency domain will manifest themselves at the delay time corresponding to the fundamental frequency in the spectral domain.*

In Equation (B.1), there was a time, T , over which the integration (summation) is to be carried out. The only qualifier was that T is large. Obviously, T must be defined prior to the collection of the data set $\{x_i\}$. Additionally, the sampling rate which defines Δt also must be defined. Fortunately, the same rule which specifies the bandwidth and number of revolutions in the frequency domain is applicable in the autocorrelation domain.

For example, assume we wish to have a maximum frequency (bandwidth) which captures the highest fault frequency having 4 or more harmonics. Once F_{max} is chosen, the sampling rate, which defines Δt , is set at $2.56 * F_{max}$. Additionally, we need resolution appropriate for resolving the lowest expected fault frequency (generally the case). To resolve cage frequency (FTF), a minimum of 15 shaft revolutions should be captured within each block of data. The conclusion is that the block of time data used to compute spectral data can also be use to compute the autocorrelation coefficient function.

B.3 Example of Autocorrelation Coefficient Function

The example presented in this section is computed using the time waveform data presented in the top graph of Figure 11 in section 3.3. This data was the result of taking a data set which had been captured from a bearing experiencing lubrication problems and then repeated seven times (only 6 repetitions will be used here). Clearly the resultant time signal has periodicity since the exact same signal was repeated six times.

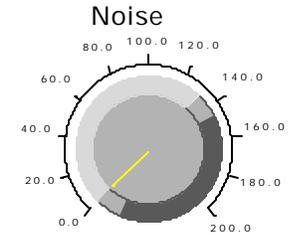
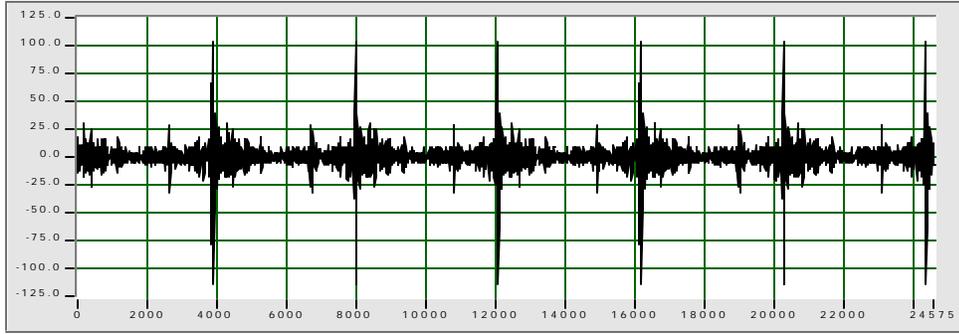
The autocorrelation as well as the autocorrelation coefficient functions were computed from the referenced signal using the LabView program, The results are presented in Figure B.1. The top graph is the time waveform which contains six repetitions of the beginning signal. The graph in the middle is the autocorrelation function. In both the autocorrelation and the autocorrelation coefficient functions, the total delay time is ~~of~~ the total time included in the top graph which is the maximum delay time that can be used. The frequency of the periodic component in the initial time waveform is the inverse of the delay time of the first large peak in either of the two autocorrelation functions. The degree of periodicity is seen to be 100% in the third or lower graph of Figure B.1 (note the +1 value at the delay time corresponding to the period of the periodic component which corresponds to a 100% degree of periodicity).

* Other components show up which on first glance appear to be harmonics but are not. This will be highlighted in the example to follow.

In the third graph of Figure B.1, we noted that the first large peak occurs at a delay time equal to the period of the periodic component. At a delay time of twice that for the first peak, the magnitude of the peak in the correlation coefficient again is 1.0. The inverse of this delay time would be a period of ~~the~~ the period for the known periodic component; it is not a second “harmonic” which would be the case in the frequency domain. This high degree of correlation at a period double that of the fundamental is readily apparent in the time data (top graph of Figure B.1). Basically, the correlation at a delay time of twice the basic periodic events comes from the correlation of every other event.

To illustrate the effect of random noise mixed with the periodic components, noise was added to the time block presented in the top graph of Figure B.1 and the analysis was repeated. The results are presented in Figure B.2.. From the top graph of Figure B.2, the signal-to-noise ratio exceeds 2, but the periodic components are readily apparent. The noise signal component is mostly gone in either of the autocorrelation functions (shown in the middle and lower graphs of Figure B.2). The level of correlation is slightly less (third graph of Figure B.2 showing about a 0.95 to 0.98 value) for this case where noise was added to the signal in comparison with the previous example where no additional external noise was introduced. In this second case, the degree of correlation would decrease somewhat. Even so, use of the autocorrelation coefficient function still clearly identified the periodic components and noticeably increased the signal-to-noise ratio far above 2 which existed in the original signal (upper graph of Figure B.2).

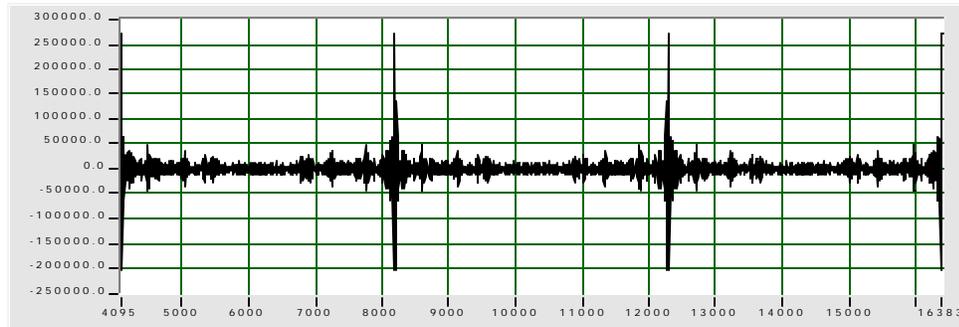
7 copies Time Waveform



Array Size

24576

Cross Correlation 1:6



Max Value

271707

Normalized Cross Correlation 1:6

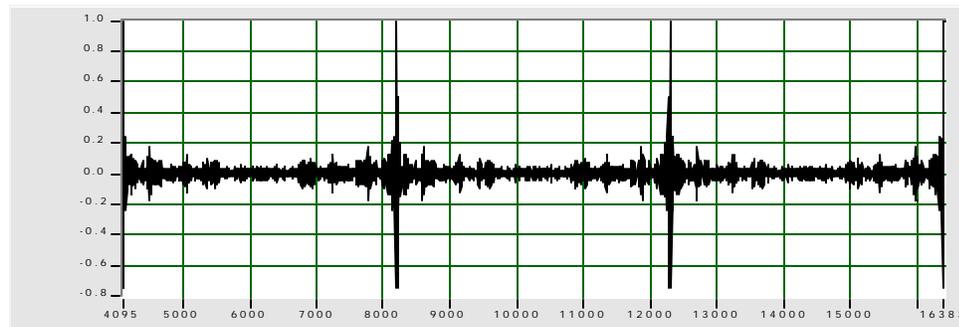
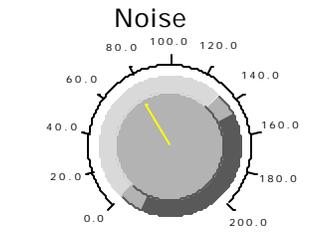
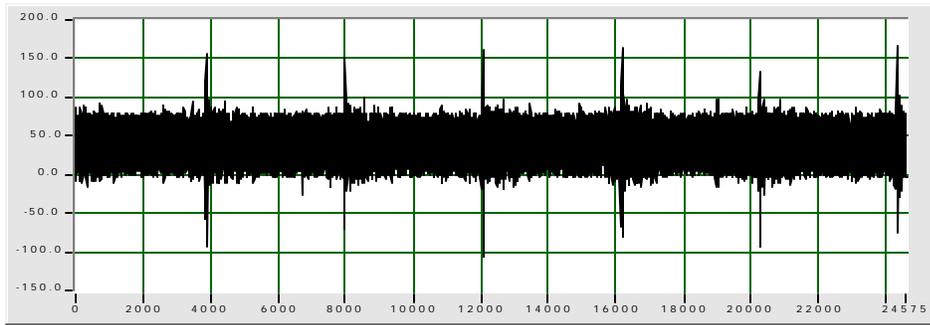


Figure B.1. Illustration of autocorrelation (middle graph) function and autocorrelation coefficient function (lower graph) computed from time waveform without any noise added (upper graph).

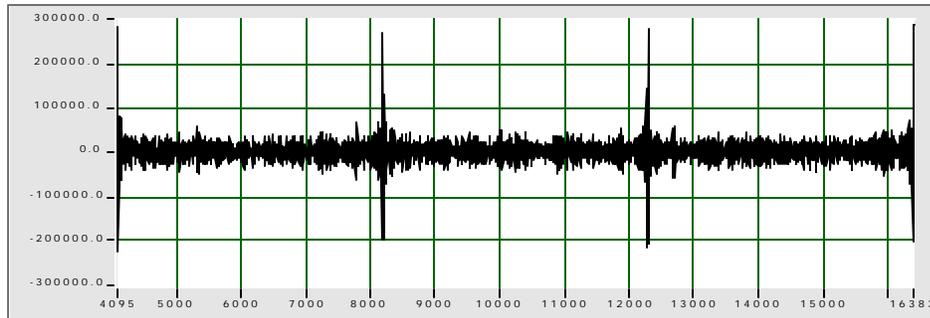
7 copies Time Waveform



Array Size

24576

Cross Correlation 1:6



Max Value

285302

Normalized Cross Correlation 1:6

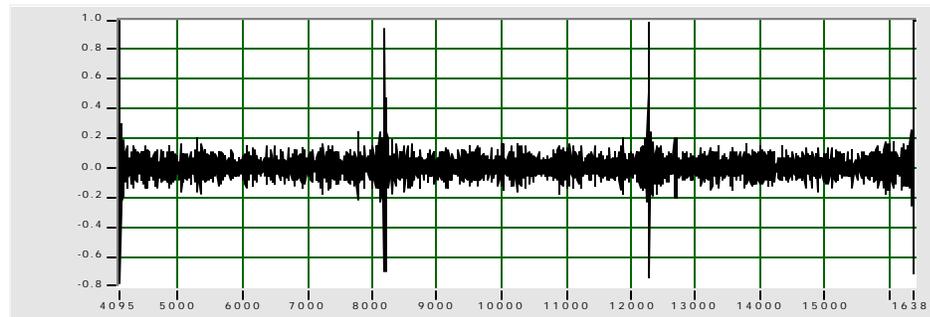


Figure B.2. Illustration of autocorrelation (middle graph) and Autocorrelation coefficient function (lower graph) Computed from Time Waveform with considerable noise added (upper graph).

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