There's Still Value in Overall Vibration Measurements

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ABSTRACT

Computerized vibration predictive maintenance was founded on overall vibration measurements. The first data collector would only collect overall vibration measurements. Now, data collectors have so many different measurements available, it is difficult to decide which to use. Which measurements will provide the information needed to detect and analyze machine problems? Which measurements will provide the best value for the time spent collecting and analyzing the data? Unfortunately, there is no magic bullet. No single measurement, overall or spectral, will provide the information needed to detect all faults and defects.

Many papers have been written and presented in the past few years describing the use and advantages of spectral analysis, enveloped spectral analysis and time waveform analysis. Overall measurements have not been addressed very well in seminars and trade magazines for quite some time. Overall measurements can provide clues, which can point you in the right direction for effective fault analysis. With the aid of overall measurements, more time can be spent on fault analysis of equipment that is in jeopardy and less spent on routine data collection. You must have an understanding of the information provided by the different types of overall measurements and the limits of that information, though. This paper will address the characteristics, advantages and limitations of several different types of overall measurements.

INTRODUCTION

Overall vibration measurements, for the most part, have been downplayed in the vibration field for the last several years. Some feel that they are almost worthless. Afraid they might miss a call, some analysts feel that they should take nearly every measurement available to them. There are analysts taking so much data at each bearing position that their programs are not cost-effective. This paper will suggest how to use certain overall measurements, along with some spectral data, to reduce the total amount of data collected and analyzed in a program and to improve efficiency.

Overall measurements do have value. They are derived from the frequency and amplitude information that the transducer senses. Each overall measurement parameter is sensitive to a certain frequency range. The intent of this paper is to explain the information contained in several types of overall measurements and demonstrate how to apply them in a vibration program. You must learn the value and limitations of each measurement before implementing them in a vibration program.

The scope of the paper will be limited to the more common overall measures of displacement peak to peak, velocity peak, acceleration peak, Spike Energy (a high frequency filtered measure) and the less common overall measures of true-peak velocity and acceleration. Because of the vast variety of equipment and operating speeds, this paper is directed mostly at general process machinery running at medium speeds. Most of the information will apply to machines with rolling element bearings.

BACKGROUND

Before computer-based data collectors, most vibration programs consisted of recording overall velocity, displacement, and acceleration measurements on a clipboard. Those with IRD instruments had a mysterious measurement available called Spike Energy. The measurements were transferred to charts for trending. Obviously, this was very labor intensive, but it was successful. Those fortunate enough to have spectrum analyzers with plotters, would take spectra and paste them to a notebook with the overall trends. This whole process of manual storage and trending of overall and spectral data was not very cost effective, but many of those programs were successful.

The advent of computer-based data collectors and trending software made this whole trending process much more cost-effective. The first system only recorded overall measurements. Spectral data still had to be taken with a spectrum analyzer. Those programs based only on overall measurements were usually successful in identifying a machine developing a problem. Of course, the specific fault was difficult to determine with just overall measurements. An educated guess could be made by comparing overall velocity and acceleration measurements, but fault identity required an analyzer capable of providing amplitudes at specific frequencies. Fault analysis was very time-consuming but was only required on machines thought to be in jeopardy. Keep in mind that this analysis was initiated from an increase in one or more trended overall measurements.

The ability to store and trend spectral data could eliminate having to go back out in the plant for fault analysis. Many analysts were finding the value of time waveform data and were requesting that capability along with the ability to store spectral data. With these capabilities, we could just analyze problems on the computer from the data collected from periodic surveys. This demand soon brought those capabilities to us. The dynamic range and resolution of these first so-called data collector/analyzers seriously limited the value of the stored spectra. If vibration levels were high on a machine, low amplitude vibrations at bearing defect frequencies were buried in the noise floor of the spectrum. We soon became addicted to spectral information, though. Vendors started refining the instruments and software, giving us much better dynamic range and resolution. Enveloped spectrum analysis arrived on the scene and really confused most of us. Time waveform storage increased our analysis abilities considerably. More and more features have been added to our data collectors and software. Some of the data collectors now provide many of the capabilities of expensive laboratory analyzers.

With all these capabilities for different types of spectral and time waveform measurements, there seems to be less and less interest in using overall measurements. Being vibration analysts, we must analyze vibration data. The more data we have to analyze, the better, right? Yes and no. I really appreciate all the analysis features of the newer data collectors when I need them. Often though, too much time is spent collecting too much data just to try and avoid missing a call. Depending on the machine's criticality, you may need to take several different measurements to create a high level of protection, or you may only be able to justify a minimal number of measurements. The factors that determine this are budgets, manpower, costs of downtime, cost of repairs and safety. But, just because you have a large budget does not mean it should be wasted.

MAGNITUDE MEASURES

Three basic measures describing vibratory forces, referred to as magnitude or amplitude values, are used to evaluate the severity of machine vibrations. They are displacement, velocity and acceleration. In the U.S., the most commonly used measurements are displacement peak to peak, velocity peak and acceleration peak.

Displacement is a measure of the total distance traveled by a vibrating part from one extreme of travel to the other. Displacement relates to stress. It is expressed in mils peak to peak. One mill is one-thousandth of an inch (0.001-inch). For bearing cap measurements, displacement is usually used for low-frequency vibration (less than 1200 CPM) [1]. Displacement measurements on bearing caps are not usually taken unless the speed is below 600 RPM. A more common use for displacement is for measuring the relative motion between the bearing and shaft in fluid-film bearings. It is used to analyze frequencies of rotor speed and orders of running speed. Figure 1 is a chart of harmonic displacement and acceleration plotted for a vibration velocity of 0.2 IPS at various frequencies [1]. At 600 CPM (10 Hz), a velocity vibration of 0.2 IPS is 6.4 mils peak to peak. At 60,000 CPM (1000 Hz), the displacement is only 0.064 mil peak to peak. The chart in Figure 1 shows that displacement is more sensitive to low-frequency vibrations and is a poor measure for high frequency. Evaluating vibration severity with displacement at high frequencies can be difficult. At high frequencies, low amplitudes can be very severe. At these low amplitudes, noise in the signal can cause problems with data accuracy. This problem is aggravated if an accelerometer is used to take displacement measurements because the signal has to be double integrated.

Velocity is a measure of the rate of change in displacement. It takes into account displacement of the vibrating part and the frequency at which it is vibrating. Velocity relates to fatigue. This measure is sensitive to medium-frequency vibrations. The recommended frequency range for velocity measurements is from 600 CPM (10 Hz) to 60,000 CPM (1000 Hz) [1]. Velocity is often used for measurements from 300 CPM to 120,000 CPM with good success. It is sensitive to rotating speed vibrations and orders of rotating speed vibrations for the most common types of process equipment in industry. This would be equipment running from 1200 RPM to 3600 RPM. The previous statement is not meant to limit the use of velocity measurements, but to target the most common applications.

Acceleration is the rate of change of velocity. It relates to force on a machine or component. Acceleration is sensitive to high frequencies. Its use is recommended for frequencies above 60,000 CPM. In Figure 1, a 0.2 IPS vibration at 60,000 CPM (1000 Hz) is equal to 3.25 g's acceleration[1]. The acceleration for a 0.2 IPS vibration at 600 CPM (10 Hz) is only 0.03 g's. The chart in Figure 1 shows that acceleration measurements are more sensitive to high-frequency vibration and are a poor measure for low frequency.

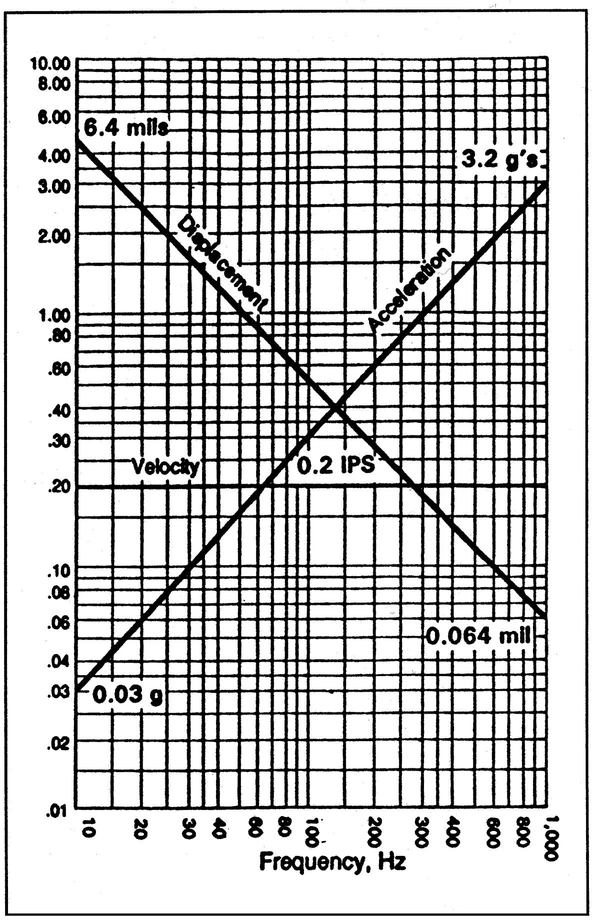

Figure 1. Sinusoidal Displacement and Acceleration Magnitude vs Frequency for a Constant Velocity of 0.2 IPS [1]. Courtesy of Dr. Ronald L. Eshleman, Vibration Institute

Spike Energy measurements (and other high-frequency filtered measurements) are a derivative of acceleration. Bearing defects generate high-frequency spike-like vibrations. Bearing and machine natural frequencies generally range from 800 Hz to 50 kHz. These natural frequencies are in the range in which the bearing defect energy is generated. Structural resonance vibrations excited by the energy from bearing defects reaching the machine surface can be detected by conventional accelerometers. The presence of these frequencies in a machine's vibration signatures is an indication of a bearing problem. For Spike Energy measurements, the signal is filtered so it only contains acceleration information between 5 kHz and 55 kHz. It is passed through a peak detector capable of detecting the short duration (high frequency) spikes of energy. An accelerometer with a mounted natural frequency in the ultrasonic range that bearing defects generate enhances the capability to detect these frequencies.[2]

Magnitude units can be expressed in different ways: as rms (root mean square), peak and peak to peak. For simple harmonic (sinusoidal) motion, such as that represented by the single frequency waveform shown in Figure 2, the peak value in the waveform will equal the peak value in the spectrum within the measurement resolution error factors. The peak value is measured from zero to either the positive peak or negative peak. For harmonic motion, the positive peak value in the waveform is equal to the negative value. The rms value is equal to 0.707 times peak; peak to peak is equal to peak times two. Most instruments measure the rms value of the signal. The rms value is then multiplied by 1.414 to derive the peak value. Notice in Figure 2 the peak in the waveform is 0.36 IPS, and the peak in the spectrum is 0.36 IPS. The peak overall velocity reading was also 0.36 IPS. The data in Figure 2 was recorded from a shaker producing a single frequency. The RMS value times 1.414 for sinusoidal motion equals the true peak value.

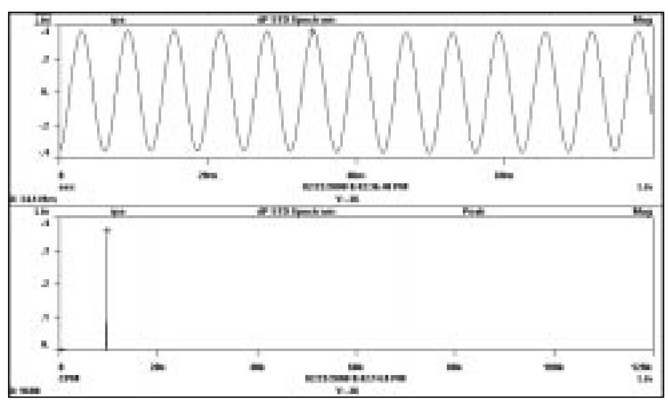

Figure 2. Harmonic (Sinusoidal) Motion.

For nonharmonic motion, such as that shown in Figure 3, the rms value cannot be converted to true peak by multiplying by 1.414. The peak in the time waveform no longer equals the peak in the spectrum. In nonharmonic motion, the value of the positive peak in the waveform rarely equals the value of the negative peak. True peak is the highest value in the waveform, either negative or positive. Notice in Figure 3 that the true peak in the waveform is 1.6 IPS. The true peak was taken from the positive side of the waveform. The highest value on the negative side of the waveform was –1.3 IPS. The highest value, positive or negative, is the true-peak value. The peaks in the spectrum shown in Figure 3 are the rotating speed vibration along with several harmonics, all of which are much lower in amplitude than the true peak or overall peak (RMS times 1.414). The overall peak (RMS time 1.414) velocity amplitude was 0.98 IPS, which is considerably less than the true-peak value. **For the rest of this paper, peak will refer to derived peak (RMS times 1.414), true peak will refer to true-peak value which is either the peak in the time waveform or a value taken from a true peak detector in an instrument.**

For bearing cap measurements, opinions vary about which is the best indicator of vibration severity, peak or true peak. Advocates of peak (RMS times 1.414) believe that it is a more accurate representation of the total energy content of the vibration signal. It is referred to as the “power under the curve.” Advocates of true peak believe that the true-peak vibration shows the real severity of a problem since the true-peak value can often be much higher than the derived-peak value.

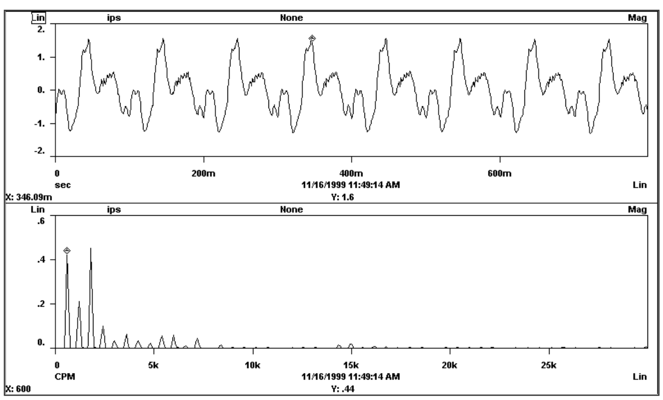

Figure 3. Nonharmonic Motion.

DATA SCREENING

Data screening is a method of collecting and looking at a minimum amount of data to determine if a fault exists on a machine. You are looking at data to screen for information that indicates you have a problem. Once you identify a problem exists, you then have to analyze the data to determine the fault and severity. Sometimes this requires further data acquisition on the machine in question, but you are only doing this on machines that you know are likely to have a problem. Using routine data collection as a screening tool, you take as little data as necessary to alert you to a problem. On most machines, high-resolution spectra, time waveforms and enveloped spectral data are not collected. Some form of high-frequency filtered measurement such as Spike Energy are collected. I place a high value on true-peak acceleration measurement.

I started using true-peak acceleration measurements after attending an advanced vibration analysis class in 1993 taught by Nelson Baxter. Talking to Nelson during a break, I learned one of the most valuable methods of detecting defective bearings that I had ever learned before or since. He told me to look at the acceleration time waveform and identify the highest amplitude peak, either negative or positive (this is true-peak acceleration). Amplitudes over 7 g’s for ball bearings and 12 g’s for roller bearings are a strong indicator the bearing is defective. I applied this over the next few years and found it to be very accurate on machines that do not produce a lot of high frequency energy during operation. Taking time waveforms with 1024 samples, the time sample must not be over 200mS (100mS is preferred for most machines). If the time sample is longer than 200mS, the anti-aliasing filters in the instrument will filter out the higher frequencies that defective bearings produce. This can become a problem on low-speed equipment because of the short time span for the time waveform compared to the long time span between the impacts caused by the rolling elements impacting bearing race defects. I purchased two instruments in 1997 capable of taking overall true-peak measurements. Overall measurements take less storage space in the data collector and computer than waveforms and less time for data screening. Since this is an overall measurement, it can be viewed in a trend plot. I rarely take acceleration time waveform data now.

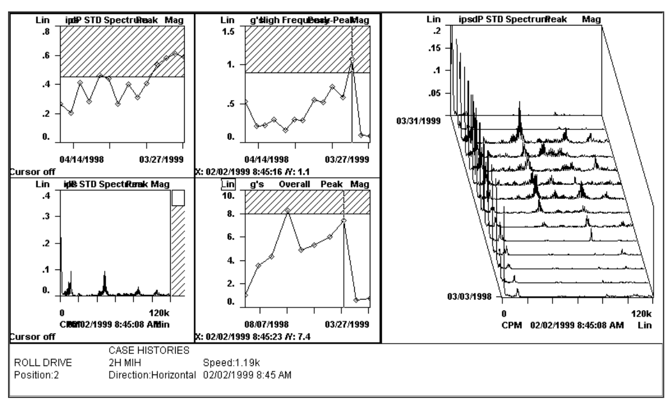

Figure 4. View for Data Screening. Roll Stand Motor in a Flourmill.

Figure 4 shows a plot display used for data screening. This data is from a roll stand motor in a flourmill. It had an outer race defect in the drive end bearing. This view allows you to see the overall velocity trend (upper left-hand display), overall Spike Energy trend (upper middle display), true-peak acceleration trend (lower middle display), and a waterfall of the velocity spectrum. This view allows you to quickly look at changes in any of the measurements taken at a given position. With a little practice, you can page down through a machine position list at a very rapid rate. When a significant change is noticed, you can look closer at the data for that machine. Other views can be brought up to get better resolution of the data. Sometimes you may have to go back out and collect more data, but keep in mind, you are only doing this on machines likely to have a problem. More time can be spent on machines in jeopardy and less time collecting and analyzing data on machines that have nothing wrong with them. The following section of this paper will show examples that will further illustrate this screening method and the application of overall measurements.

APPLICATION OF OVERALL MEASUREMENTS

Machine faults fall into different frequency ranges. Since different vibration measures are sensitive to different frequency ranges, no one single measurement will provide the information needed to detect all faults. We must select vibration measures that are responsive to the vibration frequencies that the machine will produce. Most faults in general process machines, other than bearing defects, create vibrations in the frequency range from 600 CPM (10 Hz) to 60,000 CPM (1000 Hz). The most common faults of unbalance and misalignment create vibrations at one times running speed and orders of running speed which are generally under 12,000 CPM (200 Hz). Velocity is the choice for measurements in those frequency ranges.

A lot of controversy exists over whether peak or true peak is the best measure of vibration severity. Since peak is RMS times 1.414, it represents the "energy under the curve." True peak represents the full movement of the vibrating component. Figure 5 and Figure 6 show three harmonically related sinusoidal frequencies (the fundamental and the 2^{nd} and 3^{rd} harmonics) and the resulting signal when the three frequencies are combined. The frequencies and amplitudes are identical in both displays. The only difference is the phasing of the harmonics to the fundamental. In Figure 5, the true peak is somewhat higher than in Figure 6. Since the frequencies and amplitudes are the same in both figures, the RMS value is the same for both signals resulting in identical peak values. This means that peak will represent the total energy of the harmonics in both signals. Although the true-peak value would be higher in Figure 5 than in Figure 6, the peak value would certainly alert you if the vibration condition changed from mostly a single frequency to a signal with harmonics. And, since most of us started programs using peak, it is a measure with which we are familiar. I prefer using peak velocity to measure vibration severity if the vibration signal contains mostly vibration at running speed and orders of running speed. In other words, peak velocity (RMS time 1.414) is better for lower or middle frequency vibrations. Usually, vibration signals consist of mostly running speed and orders of running speed. Peak usually represents this type of signal well. If harmonics energy increases, the peak value will increase.

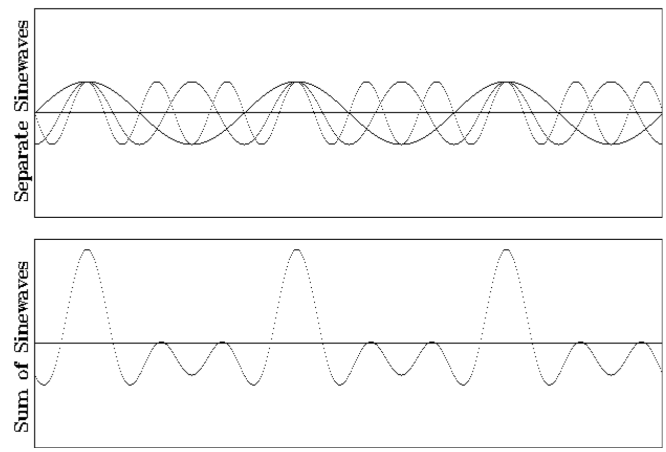

Figure 5. Three Combined Sinusoidal Signals with Identical Amplitudes. Phasing Creates High True-Peak Value.

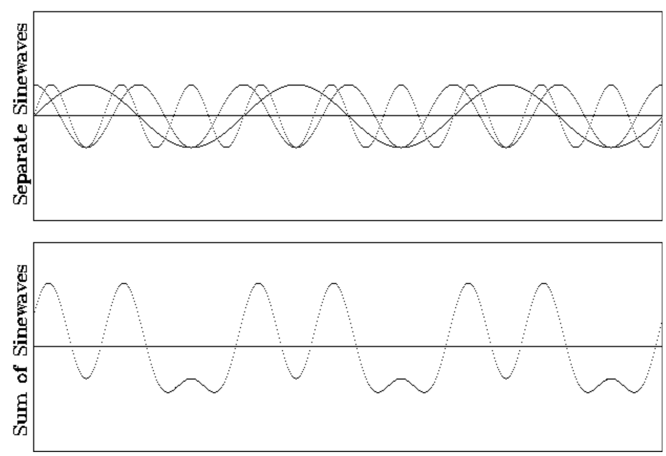

Figure 6. Three Combined Sinusoidal Signals with Identical Amplitudes. Phasing Creates Lower True-Peak Value Than Figure 5. RMS Values Are Identical for Both Figures.

I do admit that looking at peak overall velocity measurements can mislead you about vibration severity, but the velocity spectrum will usually give a clue as to whether the peak reading is representative of the problem or not. If you choose not to take spectra, then you should take true-peak velocity measurements. Even taking a lowly 400 line spectrum can provide a lot of information. When the first three harmonics are strong and the amplitude of the second and/or third harmonic(s) are high, the true-peak vibration value will usually be much higher than the peak. Although the spectrum will not give you much of a clue as to the shape of the waveform, it does give you a clue as to how different the peak will be from the true peak. Many of thousands of waveforms have been analyzed and compared to the spectrum to come to the above conclusion. Because the phasing of the harmonics can drastically change the difference between the peak and true peak, the above observation does not always hold true. Observing a spectrum like the one in Figure 3, I would definitely want to see the waveform. Between looking at the spectrum and peak overall velocity measurement however, I would conclude that the true peak would be much higher than the peak, justifying further analysis. Looking at the spectrum in Figure 3, and similar spectra from the other positions on this machine, you could assume a misaligned condition. Looking at the waveform however, leaves no doubt. The waveform in Figure 3 was taken as an "off-route measurement" after recognizing the vibration had increased considerably on this machine. This method only requires a very few waveforms to be collected and analyzed instead of one for every position in the route.

True peak is a better measure of vibration severity if the signal contains short duration spikes of energy such as would be seen in a defective bearing or gear and in some types of looseness. These are usually higher frequency vibrations. True-peak acceleration, and sometimes true-peak velocity measurements, better represents the severity of bearing and gear defects and some types of looseness.

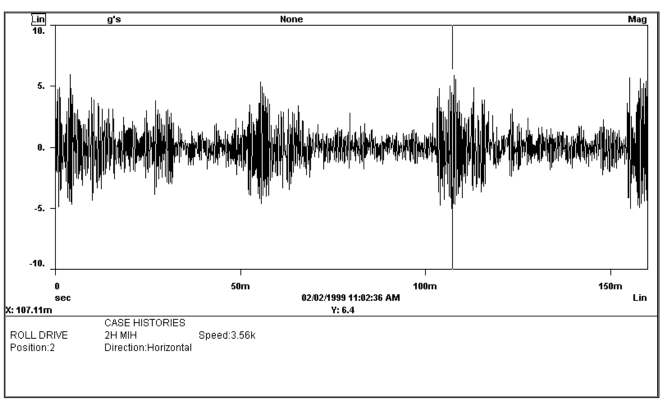

Figure 7. Acceleration Time Waveform of Outer Race Defect on Motor.

Figure 7 shows the acceleration time waveform from the motor bearing described in Figure 4. It is easy to see the spikes in this waveform and the modulations caused by the balls impacting the race defects. Because of safety considerations, measurements were not taken in the load zone. Amplitudes can be considerably lower for measurements taken outside of the load zone compared to ones taken in the load zone. Measurement location has to be considered when determining fault severity. Earlier in this paper, alarms for true-peak acceleration amplitudes for rolling element bearings were stated as 7 g's for ball bearings and 12 g's for roller bearings. These numbers work very well on most machines if the signal path to the accelerometer is relatively short and direct and in the load zone. On machines that produce high acceleration levels, such as screw compressors, positive displacement blowers and cavitating pumps, the true-peak acceleration has to be trended. Even then, the high acceleration levels produced by the normal machine operation can disguise bearing defects.

Most examples of acceleration time waveforms that you see in seminars and papers show the modulations separated by ball pass frequency of inner or outer race, whichever the case may be. In Figure 7, the separation of the modulations is one times running speed of the motor. An enveloped spectrum or Spike Energy spectrum will show peaks at one times running speed and orders. Race defect frequencies will not show up because unbalance has become the modulator instead of race defect frequency.[3,4] When unbalance becomes the modulator, you then have a rotating load zone instead of a fixed load zone. Figure 8 shows a Spike Energy spectrum taken at the same position as the waveform in Figure 7. The peaks are at running speed and orders. Three different Spike Energy spectra were taken with three different filters with the same results.

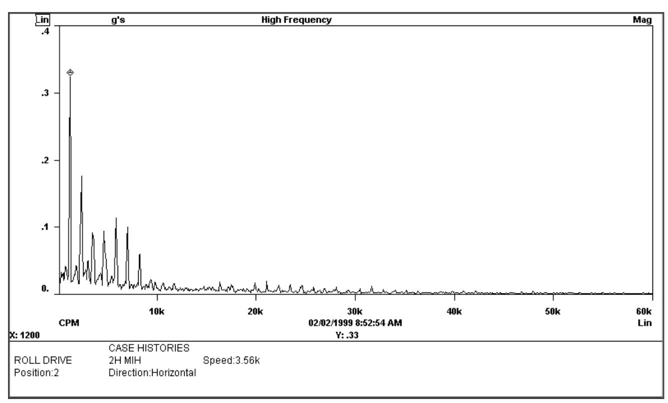

Figure 8. Spike Energy Spectrum Showing 1X and Harmonics on Motor with Outer Race Defect

The above example of one times running speed modulations in the waveform and one times running speed and orders in the Spike Energy spectrum is common in small equipment where unbalance is often the modulator.

Looking back at Figure 4, you can easily see the increase in the Spike Energy and true-peak acceleration trends. The increases in these trends should catch your eye immediately while screening data. These two high-frequency measurements immediately alert you to a possible bearing defect. Looking at the overall velocity trend, you will notice a lot of fluctuation in amplitudes. This motor is mounted on a frame hanging from the ceiling and is relatively weak. Flour collecting on the sheave, changes in belt tension and vibrations from machinery on the floor above causes the overall velocity amplitude on this motor to vary considerably. This makes it difficult to rely on the velocity measurement for much of anything. A large change in the velocity amplitude should be investigated, but the normal changes from survey to survey are ignored. The velocity waterfall definitely shows defects growing over time. So, the Spike Energy trending upward tells me there is some high-frequency vibrations going on. The true-peak acceleration, which is also a high-frequency measurement, has trended upward and is above the limit considered bad, which is 7 g’s. Knowing the measurements are not taken in the load zone, I would consider 5 or 6 g’s bad on this motor. Knowing there are no other sources of high frequency vibrations on this machine, except belts, I would be confident in declaring a bad bearing just on the Spike Energy and true-peak acceleration amplitudes and trends. Seeing the increase in high frequency activity in the velocity spectrum yields the final vote of confidence in calling for the bearing to be replaced. Time would be wasted taking high-resolution spectra because the discrete bearing defect frequencies did not show up. At the time, I did not know what bearing was in the motor anyway. Time would be wasted taking enveloped or Spike Energy spectra because only one times running speed and orders showed. If I did not have the capability of taking overall true peak acceleration measurements, it would be necessary to collect an acceleration waveform to confirm the severity of the problem. The diagnosis for this example is straightforward using minimal data resulting in a conclusion with a very high confidence level.

Figure 9 displays the data from the drive-end fan bearing on a dust collector fan in a salt plant. This fan has a chronic unbalance problem. Salt collects on the fan wheel causing an unbalance problem. Cleaning the fan helps, but corrosion requires the fan to be balanced about once every eighteen months. To aggravate this problem even more, the fan is running close to resonance causing it to be very sensitive to unbalance. Vibration normally runs from 0.7 IPS to 1.0 IPS with most of the vibration at one times fan speed. You can see there were two data points in the overall velocity trend where the vibration was below 0.3 IPS. This was after the fan had been balanced. You can see that the Spike Energy and true-peak acceleration trends (middle trends) are neither showing high amplitudes nor upward trending. The velocity waterfall does show some activity in the higher frequency area of the spectra, but there are no corresponding increases in the Spike Energy or true-peak acceleration measurements. Consequently, I am not concerned with the changes in the spectra. In fact, the shaft cover on the fan was loose and rattling. I am not concerned with a bearing defect, but the velocity amplitude is high. The waterfall is too cluttered to determine what is going on, so I would look at an individual spectrum. Figure 10 shows the spectrum. In this particular case, the overall peak amplitude is very close to the one time RPM peak in the spectrum (0.93 and 0.91 IPS respectively). This assures me that looseness is probably not the problem and that I am dealing with unbalance. Knowing this fan, I would recommend it be cleaned.

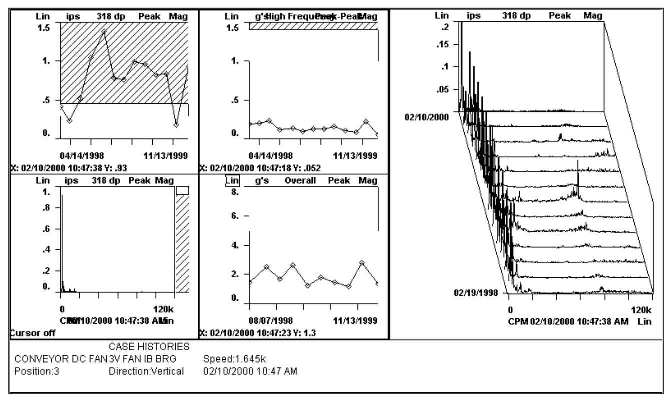

Figure 9. Data Display for Dust Collector Fan Bearing in a Salt Plant. Fan Has Chronic Unbalance Problem.

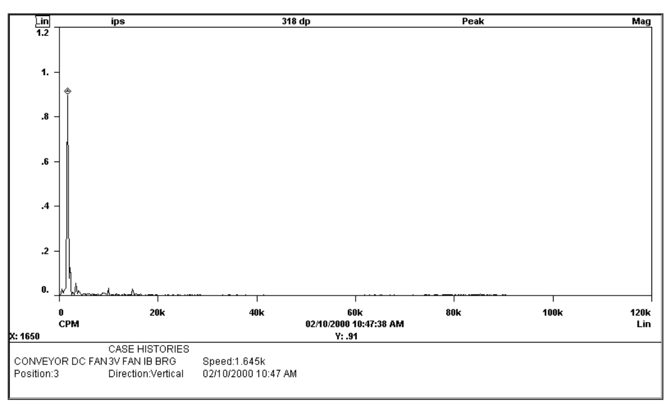

Figure 10. Spectrum from a Dust Collector Fan in a Salt Plant Showing High 1X RPM.

While running a routine vibration survey, if I see a significant change in the overall velocity vibration levels, I will often take velocity time waveforms just to be sure that I get a complete picture of the problem. Figure 11 shows the velocity time waveform from the fan and position described in Figures 9 and 10. Notice there is some distortion, but not a lot. This indicates unbalance, but the near resonant condition helps contribute to the near single frequency waveform (near sinusoidal). There is some modulation in the waveform. The short time span displayed in Figure 11 does not show the modulation very well, but the peaks in this waveform range from 0.66 IPS to 1.2 IPS. This fan sits on spring "isolators", which are amplifiers in this instance. I have recommended the isolators be removed and the fan balanced, but as of this writing, they have not been removed.

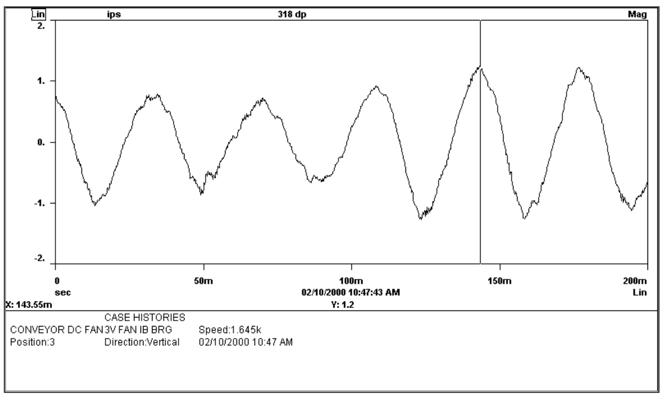

Figure 11. Velocity Waveform from Dust Collector Fan in Figures 9 and 10.

This next example is a motor that drives a refrigeration screw compressor at a chemical plant. The motor has journal bearings and is instrumented with proximity probes. This is not a good example for using overall measurements, but it does show the screening method used for monitoring bearings with displacement probes. Waveform data is an absolute must in determining faults in these types of bearings. High-frequency measurements cannot be taken from proximity probes. Therefore, the only overall measurement to take is displacement. Figure 12 shows data for the outboard motor bearing "Y" position. You can see a significant increase in the overall displacement measurement. The displacement spectrum waterfall clearly shows the increase in the one times RPM vibration. The harmonics are very low amplitude indicating that unbalance is probably the problem with possibly excessive bearing clearance. This display also lets you get a look at the waveform, which is most important. Looking at the waveform at only one position is usually not enough to determine the fault. The waveform of the "X" position of the same bearing is shown in Figure 13. At first glance it looks almost sinusoidal, but a closer look shows the cycle halves are not symmetric. In fact, the shape of the waveform has a distinctively shorter fall time than rise time, which will cause the FFT to create several harmonics. This is a relatively new installation. The foundation is suspected of settling, causing some distortion in the motor. The point of this example is to show how the overall displacement measurement gives a quick visual alert to a possible problem. Further analysis and investigation is certainly needed and warranted in this case.

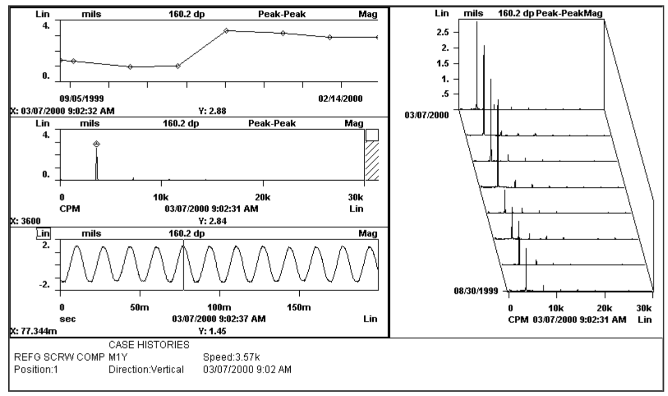

Figure 12. Refrigeration Screw Compressor Outboard Motor Bearing "Y" Position. Proximity Probe.

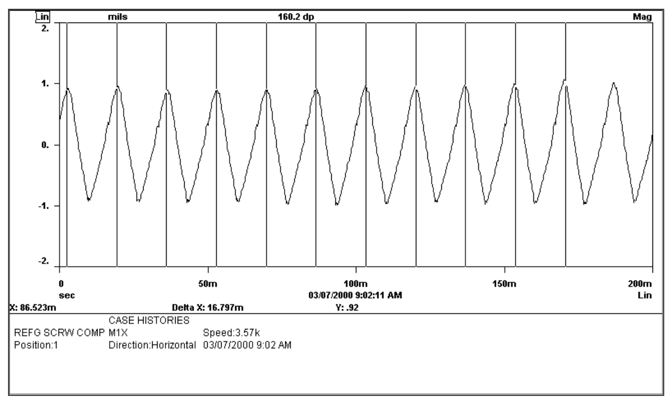

Figure 13. Refrigeration Screw Compressor Outboard Motor Bearing "X" Position. Proximity Probe.

The data from Figure 14 is from a positive displacement blower in a flourmill. These blowers are often called "pumps" or "fluidizers" because when the material is placed in the air stream created by the blowers, it acts like a liquid in the duct or piping. All of the overall measurements, as well as the spectral data, eventually indicated a problem. The overall velocity measurement and the velocity spectrum did not show a significant change until the fault was very severe. At that point, all you needed to detect a problem was the ability to hear or feel. Positive displacement blowers created a lot of high frequency energy during the process of compressing air. The alarm amplitude of 7 g's true peak for ball bearing and 12 g's true peak for roller bearing no longer applies. In fact, the normal amplitudes of true-peak acceleration ranged between 7.9 and 9.6 g's on this blower. The first indication of a fault was when the true-peak acceleration increased to 12.6 g's. The Spike Energy measurements were ranging from 0.9 g/SE to 1.6 g/SE before the fault. When the true-peak acceleration increased to 12.6 g's, the Spike Energy only increased to 2.1 g/SE (probably not enough of an increase to indicate a problem). In the next survey, there was no change in the true-peak acceleration. The Spike Energy measurement increased to 5.0 g/SE. The true-peak acceleration increased to 40.7 g's and the Spike Energy to 16.7 g/SE on the final survey before the blower was replaced. The velocity measurement increased from 3.8 IPS to 1.0 IPS. The velocity spectrum shows an increase in all frequencies indicating a lot of impacting. The bearings were loose and the rotors were rubbing. This example shows that the true-peak acceleration and Spike Energy measurements need to be trended on machines that produce a lot of high frequency energy during their normal operation.

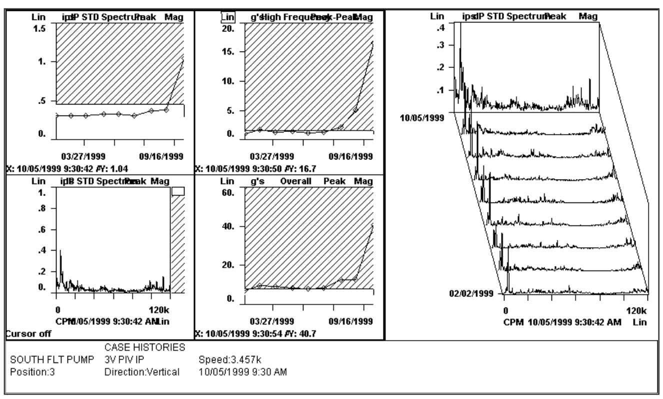

Figure 14. Positive Displacement Blower with Defective Bearings Causing the Rotors to Rub.

CONCLUSION

The methods and techniques described in this paper are not meant as a shortcut to avoid proper vibration analysis. In fact, using these methods without proper knowledge of vibration analysis techniques will create a lot of incorrect diagnoses. The intent of this paper is to provide a method of using overall measurements to quickly identify that a problem exists. Overall measurements provide clues which can point you in the right direction for effective fault analysis. With the aid of overall measurements, more time can be spent on fault analysis of equipment that is in jeopardy and less time spent on routine data collection. You must have an understanding of the information provided by the different types of overall measures and the limits of that information. Knowledge of what each overall measure represents is paramount.

There are numerous factors that determine the level of protection that should be provided for a machine. These factors are budgets, available manpower, costs of downtime, costs of repairs and safety. Using overall measurements to screen data for problem machines can make better use of resources in many plants.

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