

Spectrum Analysis

The key features of analyzing spectra

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Summary

This guide introduces machinery maintenance workers to condition monitoring analysis methods used to detect and analyze machine component failures. It informs the reader about common analysis methods. It intends to lay the foundation for understanding machinery analysis concepts and show the reader what is needed to perform an actual analysis on specific machinery.



Contents

- 1. Introduction 3
- 2. Common steps in a vibration monitoring program 4
- 3. Step 1 4
 - 3.1. Collect useful information 4
 - 3.2. Identify components of the machine that could cause vibration. 4
 - 3.3. Identify the running speed 4
 - 3.4. Other key considerations. 5
 - 3.5. Identify the type of measurement that produced the FFT spectrum 5
- 4. Step 2 5
 - 4.1. Analyze spectrum 5
 - 4.2. Common components of vibration spectrums 5
 - 4.3. Identify and verify suspected fault frequencies 6
 - 4.4. Determine fault severity 6
- 5. Misalignment 7
- 6. Unbalance 11
- 7. Mechanical looseness 14
- 8. Bent shaft. 15
- 9. Rolling element bearing defects 16
- 10. Gears 23
- 11. Blades and vanes. 25
- 12. Electrical problems 27
- 13. Step 3 28
 - 13.1. Multi-parameter monitoring 28
- 14. Conclusions 29
- 15. Further reading 29

1. Introduction

A vibration FFT (Fast Fourier Transform) spectrum is an incredibly useful tool for machinery vibration analysis. If a machinery problem exists, FFT spectra provide information to help determine the source and cause of the problem and, with trending, how long until the problem becomes critical.

FFT spectra allow us to analyze vibration amplitudes at various component frequencies on the FFT spectrum. In this way, we can identify and track vibration occurring at specific frequencies. Since we know that particular machinery problems generate vibration at specific frequencies, we can use this information to diagnose the cause of excessive vibration.

The key focus of this article hinges on the proper techniques regarding data collection and common types of problems diagnosable with vibration analysis techniques. This article can be used as a reference source when diagnosing vibration signatures.

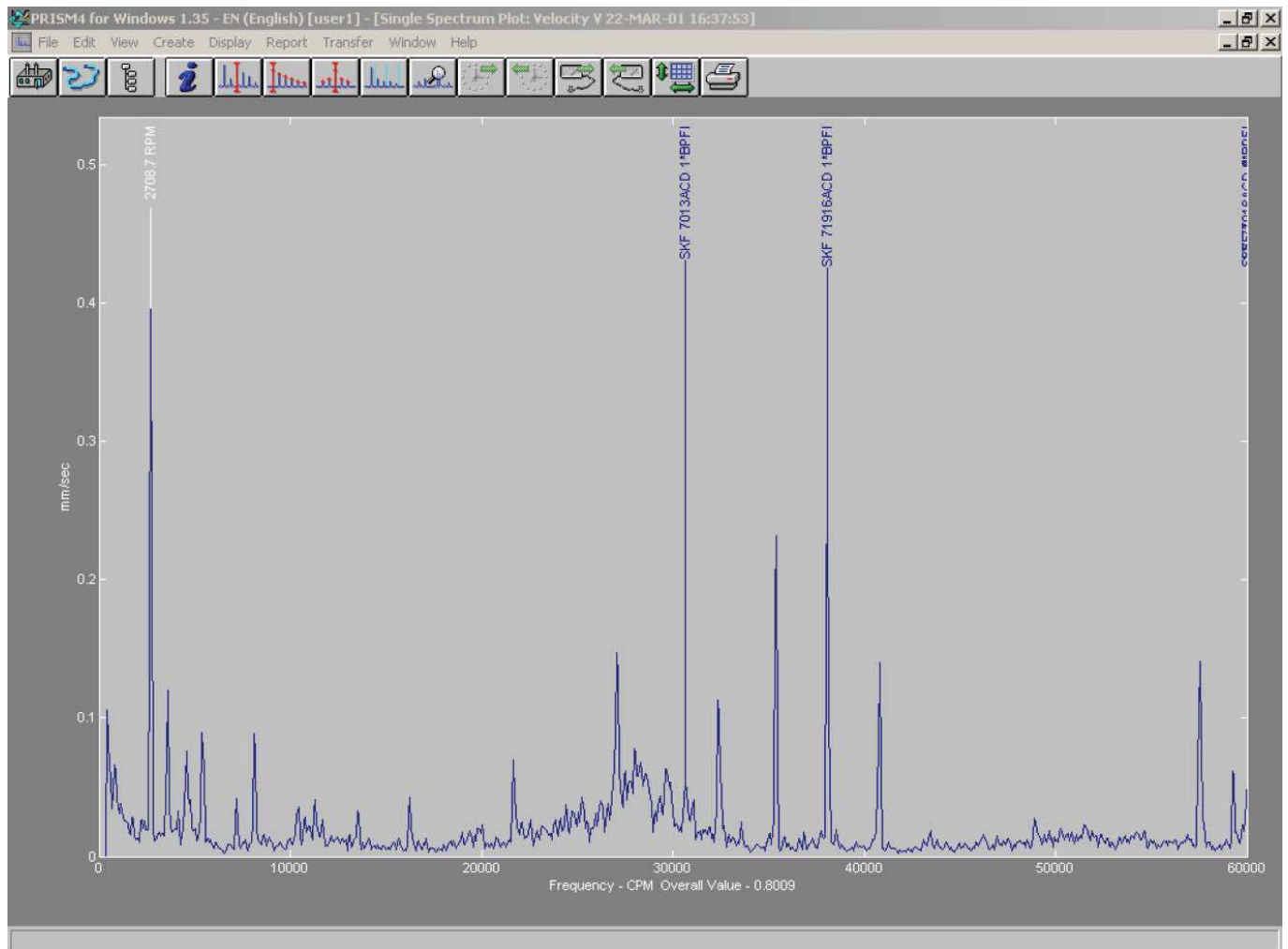


Fig 1. Example of a velocity spectrum that contains running speed (at $F = 2\,700$ RPM or 45 Hz), harmonics of running speed (at $F = 4\,500$ RPM or 75 Hz) and bearing defect frequencies (at $F = \sim 31\,000$ RPM (516 Hz) and $\sim 39\,000$ RPM (650Hz) marked with bearing overlay markers).

2. Common steps in a vibration monitoring program

There are several steps to follow as guidelines to help achieve a successful vibration monitoring program. The following is a general list of these steps:

- 1 Collect useful information – Look, listen and feel the machinery to check for resonance. Identify what measurements are needed (point and point type). Conduct additional testing if further data is required.
- 2 Analyze spectral data – Evaluate the overall values and specific frequencies corresponding to machinery anomalies. Compare overall values in different directions and current measurements with historical data.
- 3 Multi-parameter monitoring – Use additional techniques to conclude the fault type. (Analysis tools such as phase measurements, current analysis, acceleration enveloping, oil analysis and thermography can be used.)
- 4 Perform Root Cause Analysis (RCA) – In order to identify the real causes of the problem and to prevent it from occurring again.
- 5 Reporting and planning actions – Use a Computer Maintenance Management System (CMMS) to rectify the problem and take action to achieve a plan.

In this article, only steps 1 through 3 are investigated. The reader is referred to other SKF @ptitudeXchange articles on RCA and CMMS to explain these additional monitoring technologies.

3. Step 1

3.1. Collect useful information

When conducting a vibration program, certain preliminary information is needed in order to conduct an analysis. The identification of components, running speed, operating environment and types of measurements should be determined initially to assess the overall system.

3.2. Identify components of the machine that could cause vibration

Before a spectrum can be analyzed, the components that cause vibration within the machine must be identified. For example, you should be familiar with these key components:

- If the machine is connected to a fan or pump, it is important to know the number of fan blades or impellers.
- If bearings are present, know the bearing identification number or its designation.
- If the machine contains, or is coupled, to a gearbox, know the number of teeth and shaft speeds.
- If the machine is driven with belts, know the belt lengths.

The above information helps assess spectrum components and helps identify the vibration source. Determining the running speed is the initial task. There are several methods to help identify this parameter.

3.3. Identify the running speed

Knowing the machine's running speed is critical when analyzing an FFT spectrum. Running speed is related to most components within the machine and therefore, aids in assessing overall machine health. There are several ways to determine running speed:

- Read the speed from instrumentation at the machine or from instrumentation in the control room monitoring the machine.
- Look for peaks in the spectrum at 1 800 or 3 600 RPM (1 500 and 3 000 RPM for 50 Hz countries) if the machine is an induction electric motor, as electric motors usually run at these speeds. If the machine is variable speed, look for peaks in the spectrum that are close to the running speed of the machine during the time at which the data is captured.
- An FFT's running speed peak is "typically" the first significant peak in the spectrum when reading the spectrum from left to right. Look for this peak and check for peaks at two times, three times, four times, etc. Multiples of the running speed frequency can be an indication of machine health.

3.4. Other key considerations

There are many other considerations to take into account when analyzing a machine. For example:

- If the machine operates in the same vicinity as another machine, it is important to know the running speed of the adjacent machine. Occasionally, vibration from one machine can travel through the foundation or structure and affect vibration levels on an adjacent machine.
- Know if the machine is mounted horizontally or vertically. Mounting orientation affects machine response to vibration.
- Know if the machine is overhung or connected to anything that is overhung. Machine support can affect the response of the vibration sensor.

3.5. Identify the type of measurement that produced the FFT spectrum

Vibration monitoring programs use many types of measurements to determine the condition of machinery. It is important to determine which type of measurement displays the required.

- Was the measurement displacement, velocity, acceleration, acceleration enveloping, etc.? Depending upon the information needed, a particular measurement should be tailored to capture the proper results.
- How was the sensor positioned: horizontal, vertical, axial, in the load zone, etc.? Sensor response varies depending upon mounting orientation.
- Are previously recorded values, FFTs or overall trend plots available? History can help determine a machine's normal vibration level, or how quickly a machine is degrading.

4. Step 2

4.1. Analyze spectrum

Once machine vibration identification and collection is completed, the process of analyzing the spectrum can be conducted. Analysis usually follows a process of elimination: eliminate the components or issues that do not contribute to the system. From the remaining components, identify what is the contributing factor affecting the machine health.

4.2. Common components of vibration spectrums

The most common components of a vibration spectrum should be analyzed initially to determine whether or not the spectrum indicates a possible problem.

- Compare overall measurement values to prior measurements to determine if a significant increase has occurred.
- Evaluate the alarm status of a measurement point. If overall alarms are set properly, this can help indicate when a measurement needs further evaluation.
- Identify the type of measurement that indicates a problem. For example, enveloped signals can indicate bearing damage or gear tooth damage, while velocity measurements relate more to overall machine health.

Once an assessment of the measurement is conducted, specific frequencies should be identified.

4.3. Identify and verify suspected fault frequencies

Spectra may produce peaks at identified fault frequencies. These peaks may or may not represent the indicated fault. By looking for harmonics of the fault frequency, additional information can be assessed as to whether the generated frequencies are an indication of the fault. For example:

- If a peak appears at the fundamental fault frequency and another peak appears at two times (2x) the fundamental fault frequency, it is a very strong indication that the fault is real.
- If no peak appears at the fundamental fault frequency, but peaks are present at two, three and maybe four times the fundamental fault frequency, there is a strong indication the fault is valid.
- Identifying any harmonics of running speed (2x, 3x, etc.) helps determine if a fault is present.
- Identifying any bearing fault frequencies helps determine if a fault is present.
- Identifying fan or vane pass frequencies, if applicable, helps determine if a fault is present.
- Identifying the number of gear teeth and the shaft on which the gear is mounted, if applicable, helps determine if a fault is present. Moreover, this helps determine if there is a problem with a particular gear.
- Identifying pump impeller frequencies, if applicable, helps determine if a fault is present.
- As mentioned in the prior section, identifying adjacent machinery vibration, if applicable, helps determine if a fault is present.

Once the vibration source is determined, the level of severity must be assessed to evaluate whether corrective action should be taken.

4.4. Determine fault severity

Great importance should be placed upon determining the severity of a particular fault. Some components of a machine may vibrate at very high levels and still be operating within acceptable limits. Other components may be vibrating at very low levels and be outside acceptable limits. Thus, amplitude is relative, so the entire system should be evaluated, not just the amplitude.

- Compare the amplitude with past readings taken while operating under the same consistent conditions to determine the severity.
- Compare the amplitude of a particular reading with the same type of reading from a similar machine. A higher than normal reading on one of the machines may indicate a problem in that particular machine.
- Obtain prior history on the machine to help identify the various levels at which the machine has operated and aids in assessing machine health at its current state.
- Determine whether or not a baseline measurement (a measurement taken upon installation of a new or reconditioned machine) was taken. If so, compare the new reading to the baseline reading to help indicate the severity of vibration.

Once the information is collected and components are identified, you can begin to use the collected spectra to diagnose machinery problems. The following sections help evaluate common machinery problems and identify their associated causes and effects. In addition, examples of resulting spectra are included to use as templates when identifying these common issues. Issues such as misalignment, unbalance, looseness, bent shafts and bearing defects are discussed.

5. Misalignment

Misalignment is created when shafts, couplings and bearings are not properly aligned along their centerlines. The two types of misalignment are angular and parallel, or a combination of both.

Angular misalignment

Angular misalignment occurs when two shafts are joined at a coupling in a manner that induces a bending force on the shaft (→ **fig. 2**).

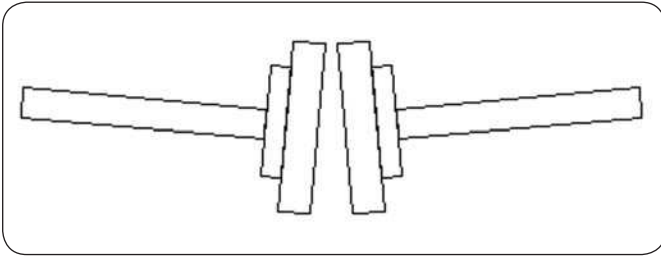


Fig. 2. Angular misalignment.

Parallel misalignment

Parallel misalignment occurs when the shaft centerlines are parallel but displaced or offset (→ **fig. 3**).

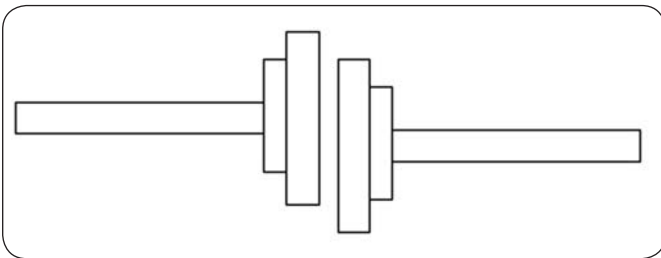


Fig. 3. Parallel misalignment.

Causes

Common causes of misalignment are:

- Thermal expansion: Expansion or growth of a component due to the heating and cooling of that component.
- Cold alignment: Most machines are aligned cold and heat as they operate. Thermal growth causes them to grow misaligned.
- Alignment of components during coupling is not correctly achieved. Therefore, misalignment is introduced into the system during installation.
- Improper alignment due to imparted forces from piping and support members.
- Misalignment due to uneven foundation, shifting in foundation or settling.

Effects

Misalignment usually causes the bearing to carry a higher load than its design specification, which may cause bearing failure due to early fatigue. Fatigue is the result of stresses applied immediately below the load carrying surfaces and is observed as spalling of surface metal. Effects on coupling in the form of damage to the coupling or excessive heat due to friction can also be seen. **Fig. 4** indicates misalignment in the system.

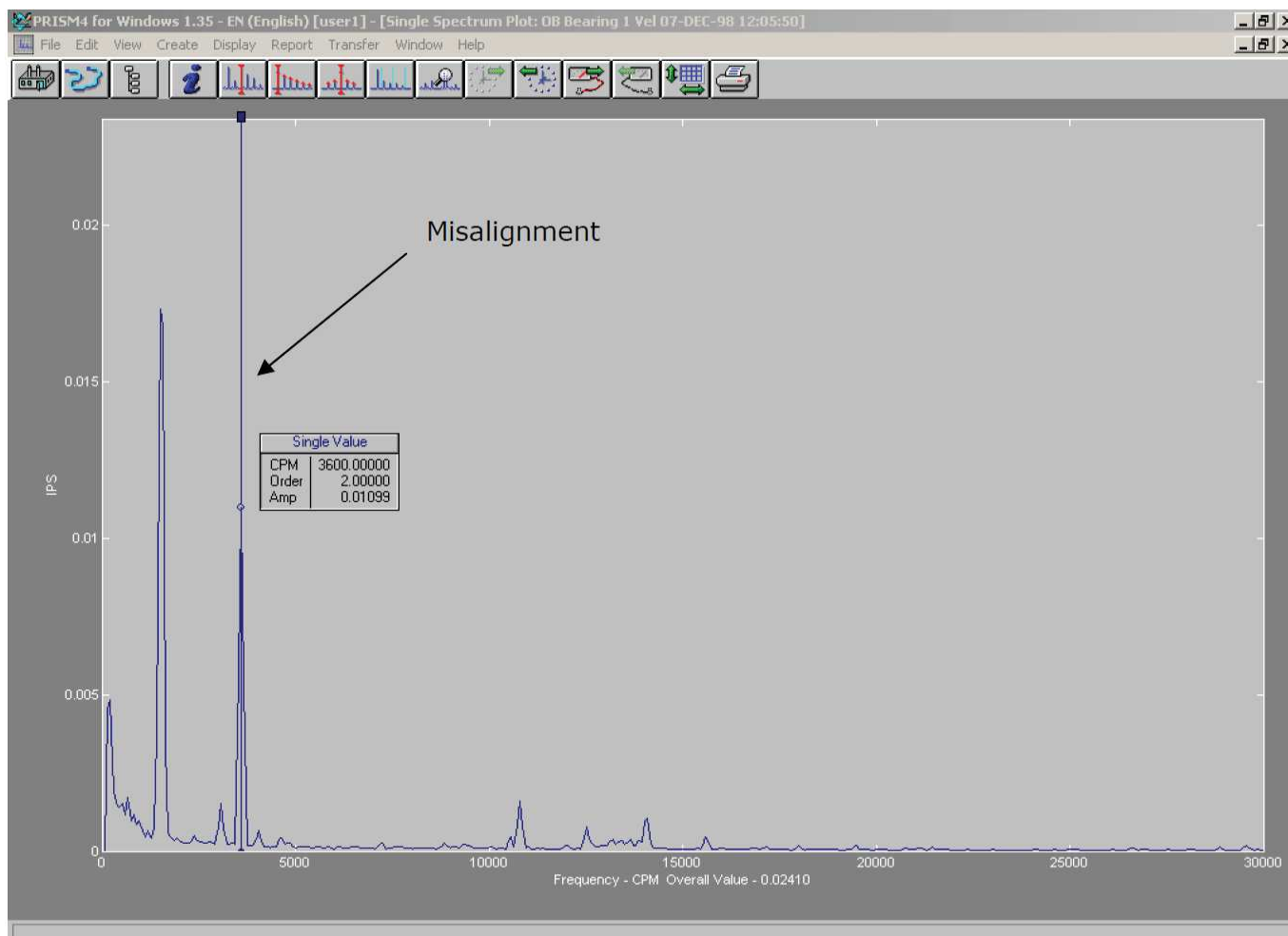


Fig. 4. High 2x running speed peak at 3 600 RPM or 60 Hz (running speed is 1 800 RPM or 30Hz) is an indication of misalignment. The first peak is most likely a belt frequency due to a worn or loose drive belt. The second peak is the running speed of the machine (1 800 RPM).
Note: 2X amplitude is not always present.

Diagnoses

The most effective analysis techniques commonly use overall vibration values and a phase measurement that helps distinguish between various types of misalignment or unbalance.

A common practice when analyzing misalignment is to look at the ratio between 1x (unbalance) and 2x (misalignment), and compare the values. When analyzing an FTT spectrum where misalignment is indicated, a higher than normal 1x amplitude divided by 2x amplitude may occur. The indication of amplitude can vary from 30% of the 1x amplitude to 100 to 200% of the 1x amplitude. An example of this is seen in **fig. 5**. The 2x amplitude (0,90 mm/sec) is almost twice that of 1x (0,45 mm/sec).

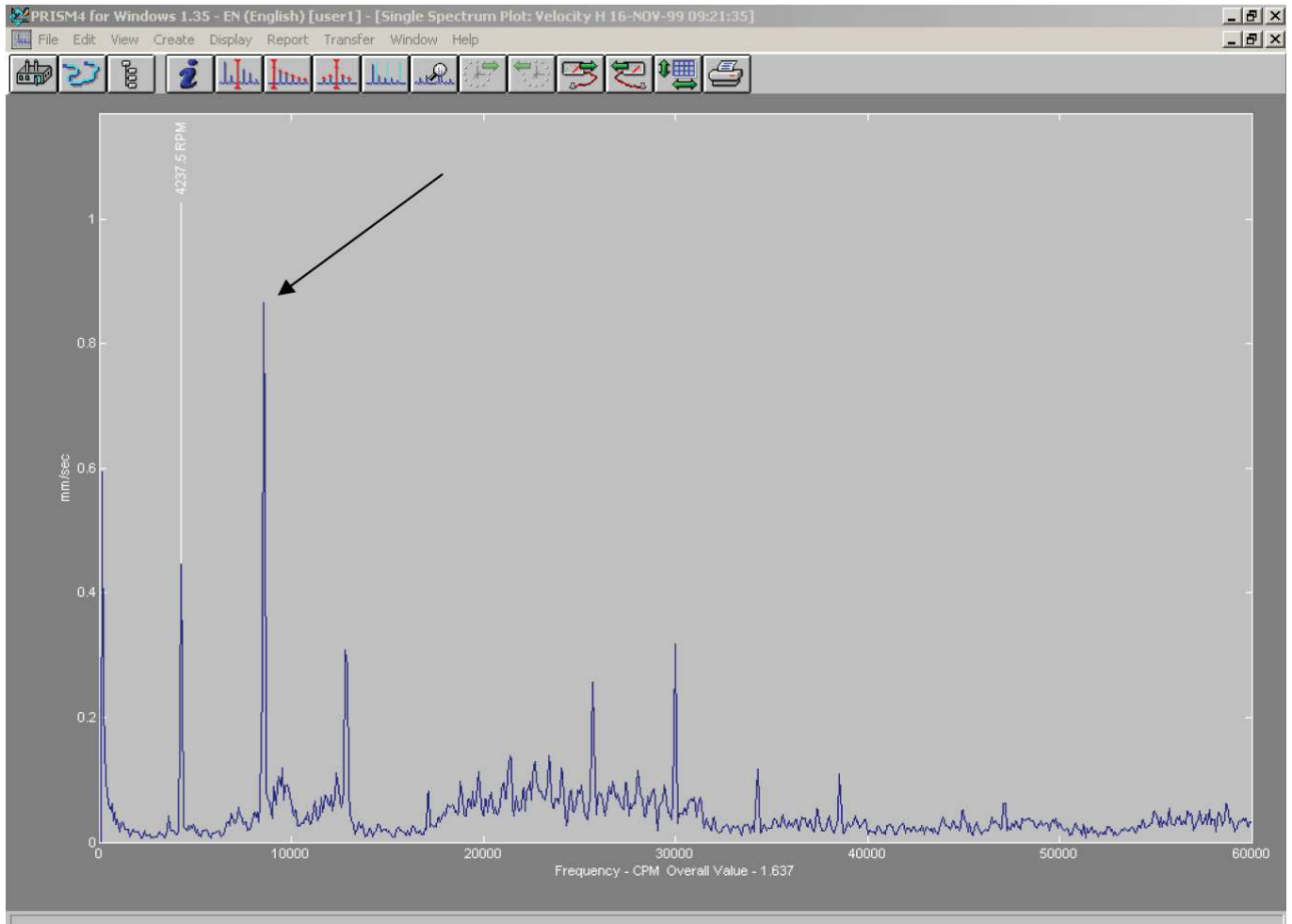


Fig. 5. FFT spectrum showing severe misalignment (the second peak in the spectrum at ~8 500 RPM (141 Hz) indicates severe misalignment, as it is almost twice the amplitude of the running speed; the peak marked with the marker is running speed (4 237.5 RPM (71Hz)).

With concern to coupling, some general rules are applied:

- Couplings with 2x amplitudes below 50% of 1x are usually acceptable and often operate for a long period of time.
- When the vibration amplitude at 2x is 50 to 150% of 1x, it is probable that coupling damage will occur.
- A machine exhibiting vibration at 2x running speed that is greater than 150% of the 1x indicates severe misalignment. The machine should be scheduled for repair as soon as possible.

As mentioned earlier in this section, phase readings are another factor that help determine the precise problem.

Phase analysis

Phase measurements are a very useful tool for diagnosing misalignment. If possible, measure the phase shift between axial readings on opposite ends of the machine.

Note: All phase values are $\pm 30^\circ$ because of mechanical variance.

- *Angular misalignment:* In the axial position, a phase shift of 180° exists across the coupling or machine.
- *Parallel misalignment:* In the radial direction, a phase shift of 180° exists across the coupling or machine. A 0° or 180° phase shift occurs as the sensor is moved from the horizontal to the vertical position on the same bearing.
- *Combination angular and parallel misalignment:* In the radial and axial positions, a phase shift of 180° exists across the coupling or machine.

Note: With severe misalignment, the spectrum may contain multiple harmonics from 2x to 10x of running speed. If vibration amplitude in the horizontal plane is increased two or three times, misalignment is indicated again.

Bearing cocked on a shaft

Like misalignment, a cocked bearing (aligned improperly in the housing) usually generates considerable axial vibration. However, phase measurements from the axial position help differentiate the two.



Fig. 6. Four sensor locations.

If the phase readings among the four sensor locations in **fig. 6** (12 o'clock, 3 o'clock, 6 o'clock and 9 o'clock) vary considerably, a cocked bearing is indicated.

Summary

- If there is abnormally high 2x amplitude divided by 1x amplitude, and the system contains a coupling or belt, there may be misalignment.
- If the radial 2x amplitude is abnormally high, and the system contains a coupling or belt, there may be misalignment.
- If the axial 1x amplitude is abnormally high, and the system contains a coupling or belt, there may be misalignment.

6. Unbalance

Another common indication of poor machine health is unbalance in the system. Unbalance can cause excessive forces that affect the machine. Unbalance occurs when the shaft's mass centerline does not coincide with its geometric centerline. In general, there are three types of unbalance:

- Static unbalance
- Couple unbalance
- Dynamic unbalance

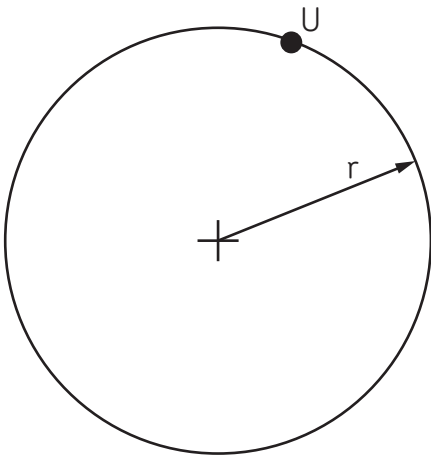


Fig. 7. Point U is the unbalance weight a distance r from the center of the rotor or disk. This type of situation in a machine is considered unbalance.

Static unbalance

With static unbalance, only one force is involved. For example, if you have a bicycle tire that has mud buildup on one area or portion of the tire, when stopped the wheel naturally settles with the clump of mud at the bottom of the wheel. Similarly, a rotor turns until the heavy spot is located at 6 o'clock (→ **fig. 8**). The term “static” implies that this type of unbalance can be observed at rest.

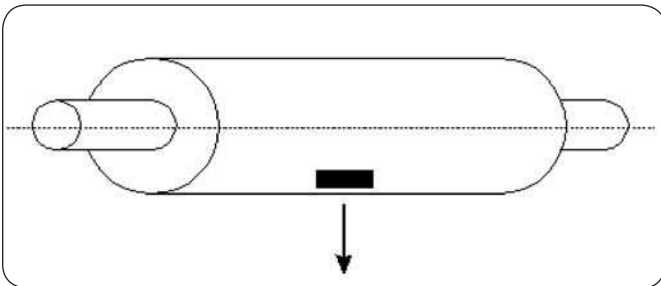


Fig. 8. Static unbalance.

Couple unbalance

Unlike static unbalance, couple unbalance cannot be measured at rest. With couple unbalance, two equal forces (weights) are 180° from each other, which causes the rotor to appear balanced at rest (\rightarrow **fig. 9**). However, when the rotor rotates, these forces move the rotor in opposite directions at their respective ends of the shaft. This causes the rotor to wobble, which produces a 180° out-of-phase reading from opposite ends of the shaft.

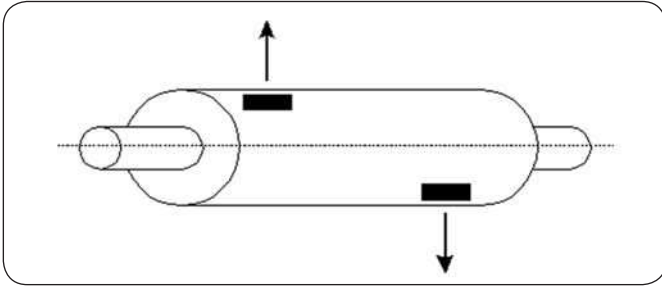


Fig. 9. Couple unbalance.

Dynamic unbalance

In reality, most unbalance is dynamic. Dynamic unbalance is the combination of static and couple unbalance. On simple machines, there is usually more static unbalance than couple unbalance. On more complex machinery, with more than one coupling or several areas on the rotor where unbalance can occur, couple unbalance is usually more prominent in the system.

When balancing a machine, always balance out static unbalance first, then couple unbalance. When balancing couple unbalance, it is important to note that balancing must occur within several planes.

Cause

Unbalance can be caused by a number of factors. Several examples include:

- Improper component manufacturing
- Uneven build up of debris on the rotors, vanes or blades
- The addition of shaft fittings without an appropriate counter balancing procedure
- Vane/Blade erosion or thrown balance weights

Key characteristics of vibration caused by unbalance:

- It is a single frequency vibration whose amplitude is the same in all-radial directions
- It is sinusoidal, occurring at a frequency of once per revolution (1x)
- The spectrum generally does not contain harmonics of 1x running speed, unless the unbalance is severe
- Amplitude increases with speed

Effects

Just like misalignment, unbalance usually causes bearings to carry a higher dynamic load than their design specifications, which causes bearing to fail from early fatigue. Fatigue, in a bearing, is the result of stresses applied immediately below the load carrying surfaces and is observed as spalling away of surface metal.

Diagnoses

The use of overall vibration, FFT spectra and phase measurements aids in diagnosing unbalance problems.

FFT spectrum analysis

Vibration caused by pure unbalance is a once per revolution sinusoidal waveform. On an FFT spectrum, this appears as a higher than normal 1x amplitude. While other faults can produce high 1x amplitude, they usually also produce harmonics. In general, if the signal has harmonics above once per revolution, the fault is not unbalance. However, harmonics can occur as unbalance increases or when horizontal and vertical support stiffness differs by a large amount.

Phase analysis

The use of phase measurements aids in the diagnosis of unbalance problems.

Note: All phase readings are $\pm 30^\circ$ due to mechanical variance.

The sensor shows a 90° phase shift between the horizontal and vertical positions when readings are compared. When the system involves predominantly static unbalance, there is usually no phase shift across the machine or coupling in the same measurement position.

Summary

Some general guidelines for phase relationships follow:

- If the radial measurement's 1x amplitude is high and harmonics (except vane passing) are less than 15% of the 1x, there may be unbalance.
- If the majority of vibration is in the radial plane, the 1x amplitude is medium to high in amplitude and the phase from the vertical and horizontal measurements differs by 90° , there may be unbalance.
- If the primary vibration plane is both axial and radial, the machine has an overhung mass and the axial phase measurements across the machine are in phase, there may be unbalance.

The spectrum on the following page (→ **fig. 10**) is an indication of unbalance in a machine.

Note: Increasing unbalance forces place increasing loads on nearby bearings. If the bearing's specified load is exceeded, damage can occur and the bearing's life is drastically reduced.

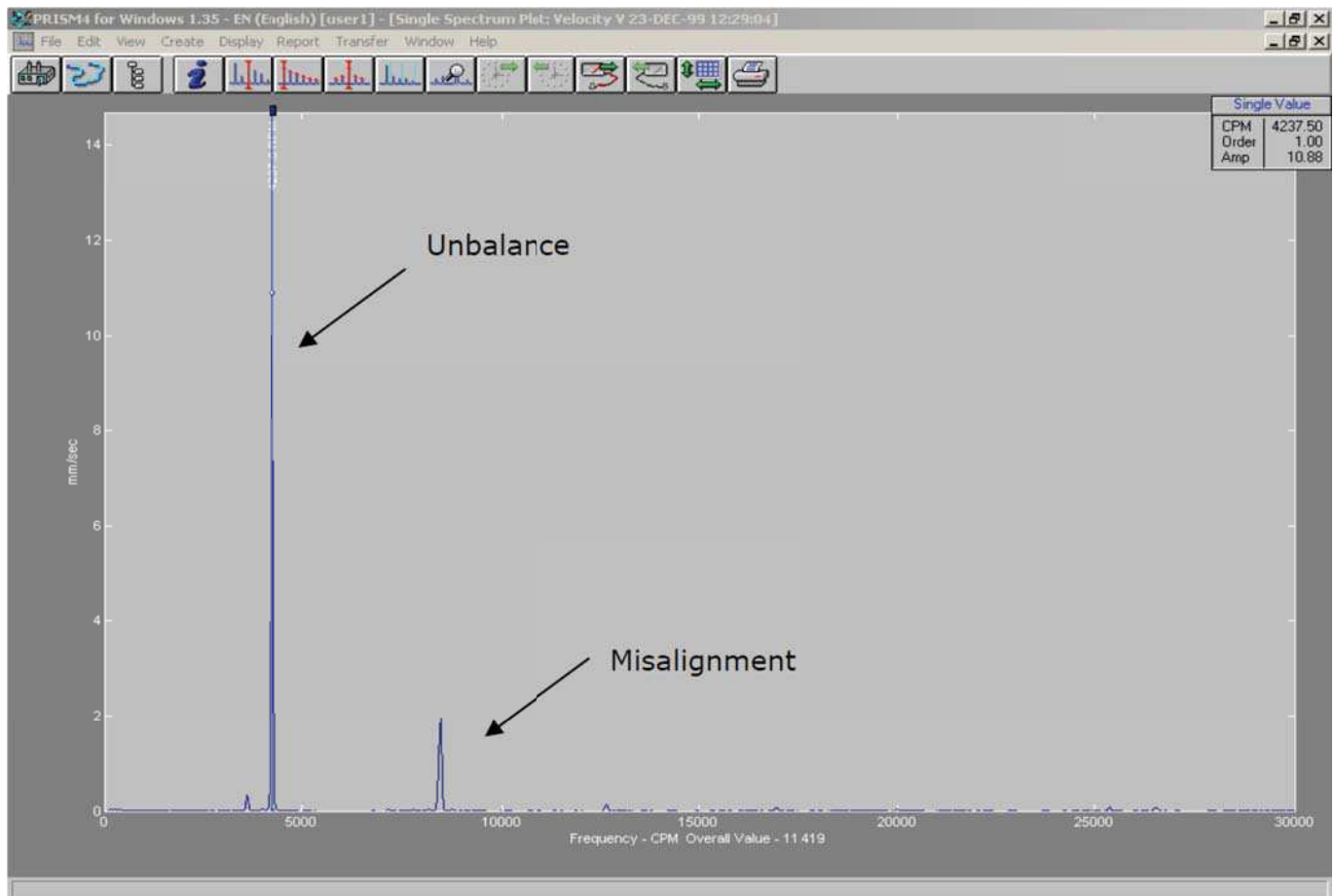


Fig. 10. FFT showing unbalance in the spectrum (at $F = 4\,237.5$ RPM or 70 Hz). Additionally, there is an indication of slight misalignment (the smaller peak) at 8 475 RPM or 141 Hz.

7. Mechanical looseness

A long string of rotating frequency harmonics or $1/2$ rotating frequency harmonics at abnormally high amplitudes generally characterizes mechanical looseness, or an improper fit between component parts (→ Fig. 11).

Note: These harmonics may be random and unorganized. For example, looseness may display peaks at 2x, 3x, 4x, 5x, 6x, etc., or at 3x, 3.5x, 4x, 5.5x, 6x, etc.

Causes

Possible causes of wear/looseness are:

- The machine came loose from its mounting
- Mounting is cracked or broken
- A machine component came loose

The bearing developed a fault, which wore down bearing elements, or the bearing seat caused excessive clearance in the bearing.

Effects

If looseness is generated from a component, there is a possibility the part will become detached and cause secondary damage.

Diagnosis

Looseness can be exhibited in varying amplitudes, both overall vibration and individual frequency amplitudes. Looseness is best diagnosed using FFT spectra and phase.

Spectrum analysis

Fig. 11 displays vibration signatures associated with components or systems. Typically, looseness is identified by abnormally high running speed amplitude followed by multiples or $1/2$ multiples. Harmonic peaks may decrease in amplitude as they increase in frequency (except at 2x, which, when measured in the vertical position, can be higher in amplitude).

Summary

- If there are a series of three or more synchronous or $1/2$ synchronous multiples of running speed (range 2x to 10x), and their magnitudes are greater than 20% of the 1x, there may be mechanical looseness.
- If the machine is rigidly connected (no coupling or belt), and the radial 2x is high, there may be mechanical looseness.

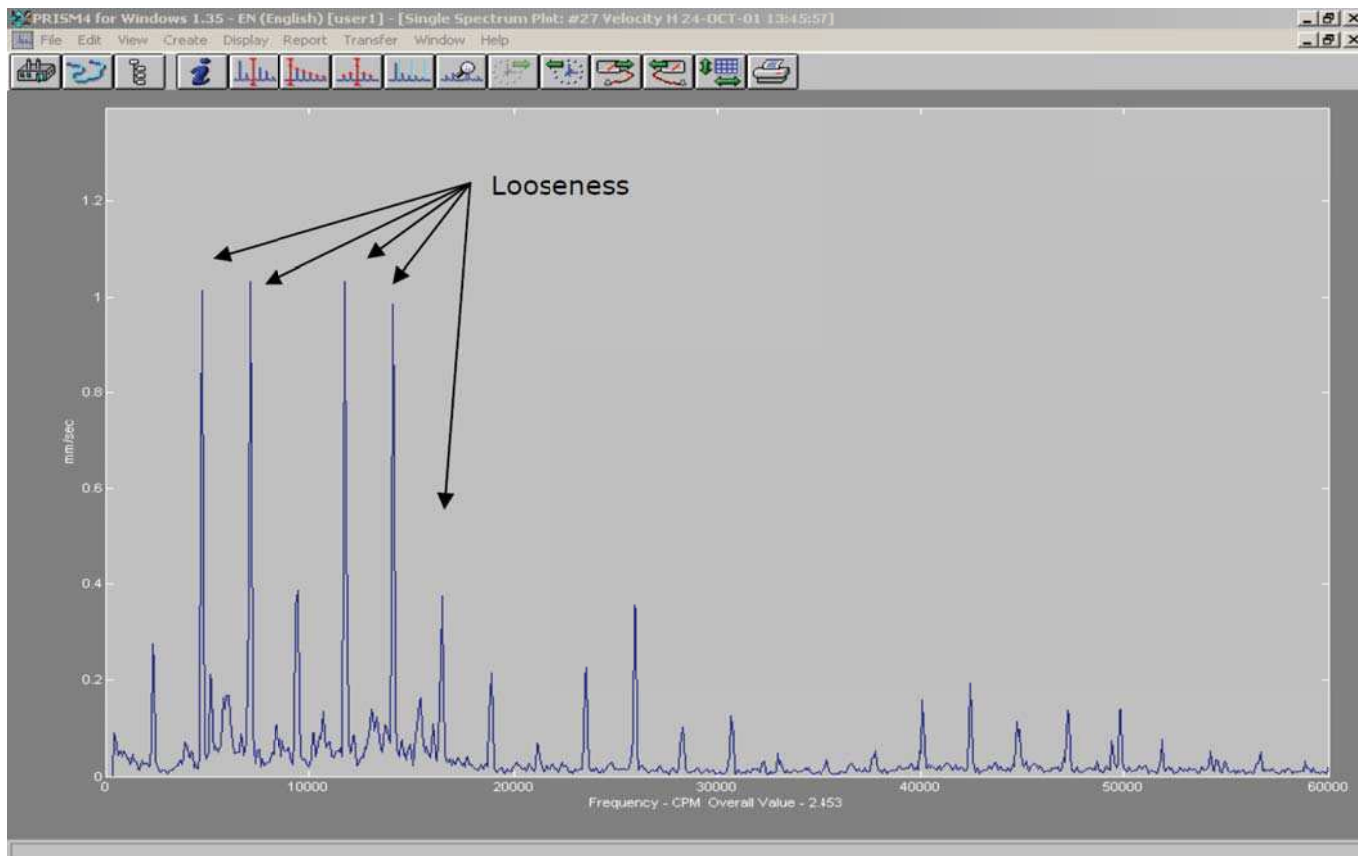


Fig. 11. FFT spectrum indicating looseness in the machine. Notice all of the repeating multiples of running speed or $1/2$ of running speed.

8. Bent shaft

With overall vibration and spectral analysis, a bent shaft problem usually emits a vibration signature that appears to be identical to a misalignment problem. The use of phase measurements is needed to distinguish between the two.

Causes

There are several causes that can result in a bent shaft:

- Cold bow: a shaft with a high length-to-width ratio can, at rest, develop a bend
- Improper handling during assembly or transportation
- High torque

Effects

As with unbalance, a bent shaft usually causes the bearing to carry a higher dynamic load than its design specification, which causes the bearing to fail.

Diagnosis

The use of overall vibration measurements, spectral analysis and phase measurements can be effective to analyze a bent shaft.

Spectrum analysis

A bent shaft typically produces spectra that have misalignment type characteristics. A higher than normal 1x divided by 2x amplitude may occur. High 2x amplitude can vary from 30% of the 1x amplitude to 100 to 200% of the 1x amplitude.

Phase analysis

Phase measurements are essential when diagnosing a bent shaft.

Note: All phase values are $\pm 30^\circ$. Radial phase measurements (vertical and horizontal) typically appear “in phase” with the shaft.

Axial phase measurements are typically 180° out of phase with the shaft. If both of the prior conditions are true, the problem is most likely a bent shaft.

Summary

If the primary vibration plane is in the axial direction, there is a dominant 1x peak, and if there is a 180° phase difference in the axial direction across the machine, there may be a bent shaft.

9. Rolling element bearing defects

Most often the bearing defect is not the source of the problem. Usually, some other machinery component or lubrication problem is causing the bearing defect. When a bearing defect is detected, you should automatically look for other root cause problems such as misalignment and unbalance. Then schedule the repair of both the defective bearing and the fault causing the bearing defect.

Bearing defects

To understand how to monitor bearings, an understanding of how a bearing defect progresses should be achieved.

Note: The following discussion relates to typical spall or crack type bearing defects on rolling element bearings.

Bearing failure may be caused by:

- Ineffective lubrication
- Contaminated lubrication
- Heavier loading than anticipated
- Improper handling or installation
- Old age (subsurface fatigue)
- Incorrect shaft or housing fits
- False brinelling due to external vibration sources while machine stands still
- Passage of current through bearing

Often, initial bearing fatigue results in shear stresses cyclically appearing immediately below the load-carrying surface. After time, these stresses cause cracks that gradually extend to the surface. As a rolling element passes over these cracks, fragments break away. This is known as spalling or flaking. The spalling progressively increases and eventually makes the bearing unusable. This type of bearing damage is a relatively long process and makes its presence known by increasing noise and vibration.



Figure 12. Spalling or flaking on the outer ring of a bearing.

Another type of bearing failure is initiated by surface distress. Surface distress causes cracks to form on the surface and grow into the material. Surface distress is usually caused by excessive load or improper lubrication. In both cases, the failing bearing produces noise and vibration signals, that if detected, give the user adequate time to correct the cause of the bearing problem or replace the bearing before complete failure.

Acceleration enveloping is an effective tool to detect and monitor the early stages of bearing failure caused by local defects. Again, this provides enough pre-warning time to possibly correct the cause of the bearing problem and potentially extend the bearing's life. Acceleration and velocity vibration measurements are also useful tools for measuring the final stages of a bearing's life. These measurements typically provide indications of imminent bearing failure (less than 10% of residual bearing life).

Velocity measurements

The prior examples and many other types of problems can cause bearings to fail. It is important to assess and understand the proper types of measurements to take and their results. One of the most common measurements used in vibration analysis is velocity. These measurements are very useful for detecting and analyzing low frequency rotational problems such as unbalance, misalignment, looseness, bent shaft, etc. The following section describes velocity measurements and provides an ISO classification to help determine severity levels.

Table 1 illustrates the ISO 2372 Standard for an overall severity of vibration. Please keep in mind that the levels are machinery and environment dependent, and have to be fine tuned in practice.

Vibration Severity	Velocity Range Limits and Machinery Classes ISO Standard 2372-1974							
in/s RMS	Small Machines		Medium Machines		Large Machines			
	Class I		Class II		Rigid Supports Class III		Flexible Supports Class III	
	Rigid	Flexible	Rigid	Flexible	Rigid	Flexible	Rigid	Flexible
0.011	Good		Good		Good		Good	
0.018								
0.028	Satisfactory		Satisfactory		Good		Good	
0.044								
0.071	Unsatisfactory		Unsatisfactory		Satisfactory		Satisfactory	
0.110								
0.177	Unacceptable		Unacceptable		Unsatisfactory		Unsatisfactory	
0.28								
0.44	Unacceptable		Unacceptable		Unacceptable		Unacceptable	
0.71								
1.10	Unacceptable		Unacceptable		Unacceptable		Unacceptable	
1.71								
2.79	Unacceptable		Unacceptable		Unacceptable		Unacceptable	

Table 1. ISO 2372 Standard.

Vibration – spectral analysis

Due to the nature of bearing defect frequencies, they occur at much higher frequencies and much lower amplitudes than frequencies related to unbalance and looseness. ISO severity charts were not developed to aid in setting parameters for detecting early bearing degradation. For bearing related issues, it is important to evaluate the bearing's FFT spectrum and its related defect frequencies.

To help determine if machine problems include a faulty bearing, bearing defect frequencies can be calculated and used as overlays to aid in diagnosis. There are several naming conventions that were adapted for use when discussing frequency analysis. The two most common conventions are listed below. The four primary bearing frequencies:

- F_{ord} – Frequency Outer Race Defect
- F_{ird} – Frequency Inner Race Defect
- F_{bd} – Frequency Ball Defect
- F_c – Frequency Cage

Or:

- BPFO – Ball Pass Frequency Outer Race
- BPFI – Ball Pass Frequency Inner Race
- BSF – Ball Spin Frequency
- FTF – Fundamental Train Frequency

When the defect frequencies (F_{ord} , F_{ird} , F_{bd} , F_c) align with peak amplitudes in the vibration spectrum, it is commonly accepted that there are defects within that particular component of the bearing. Notice that a ball defect frequency is by definition twice the ball spin frequency, as the ball defect hits the inner and outer race during one rotation.

Note: In many condition monitoring programs, the following are interchangeable. The use of one set or the other set is suggested, but do not interchange them.

- $F_{ord} = \text{BPFO}$
- $F_{ird} = \text{BPFI}$
- $F_{bd} = 2 * \text{BSF}$
- $F_c = \text{FTF}$

If bearing analysis software is not available, bearing defect frequencies should be mathematically calculated.

- $F_{ord} = (n/2) (\text{RPM}/60) (1 - (\text{Bd}/\text{Pd}) (\cos \phi))$
- $F_{ird} = (n/2) (\text{RPM}/60) (1 + (\text{Bd}/\text{Pd}) (\cos \phi))$
- $F_{bd} = 2 * (1/2) (\text{Pd}/\text{Bd}) (\text{RPM}/60) [1 - (\text{Bd}/\text{Pd})^2 \cos^2 \phi]$
- $F_c = (1/2) (\text{RPM}/60) (1 - (\text{Bd}/\text{Pd}) (\cos \phi))$

Where:

- n = number of balls
- Bd = ball diameter
- Pd = pitch diameter
- ϕ = contact angle

Fig. 13 shows a typical bearing defect in its final stages. The size and width of the hump at ~9x running speed indicates that the defect is approaching failure. In early stages, this hump may appear as non-synchronous peaks, or may not exist.

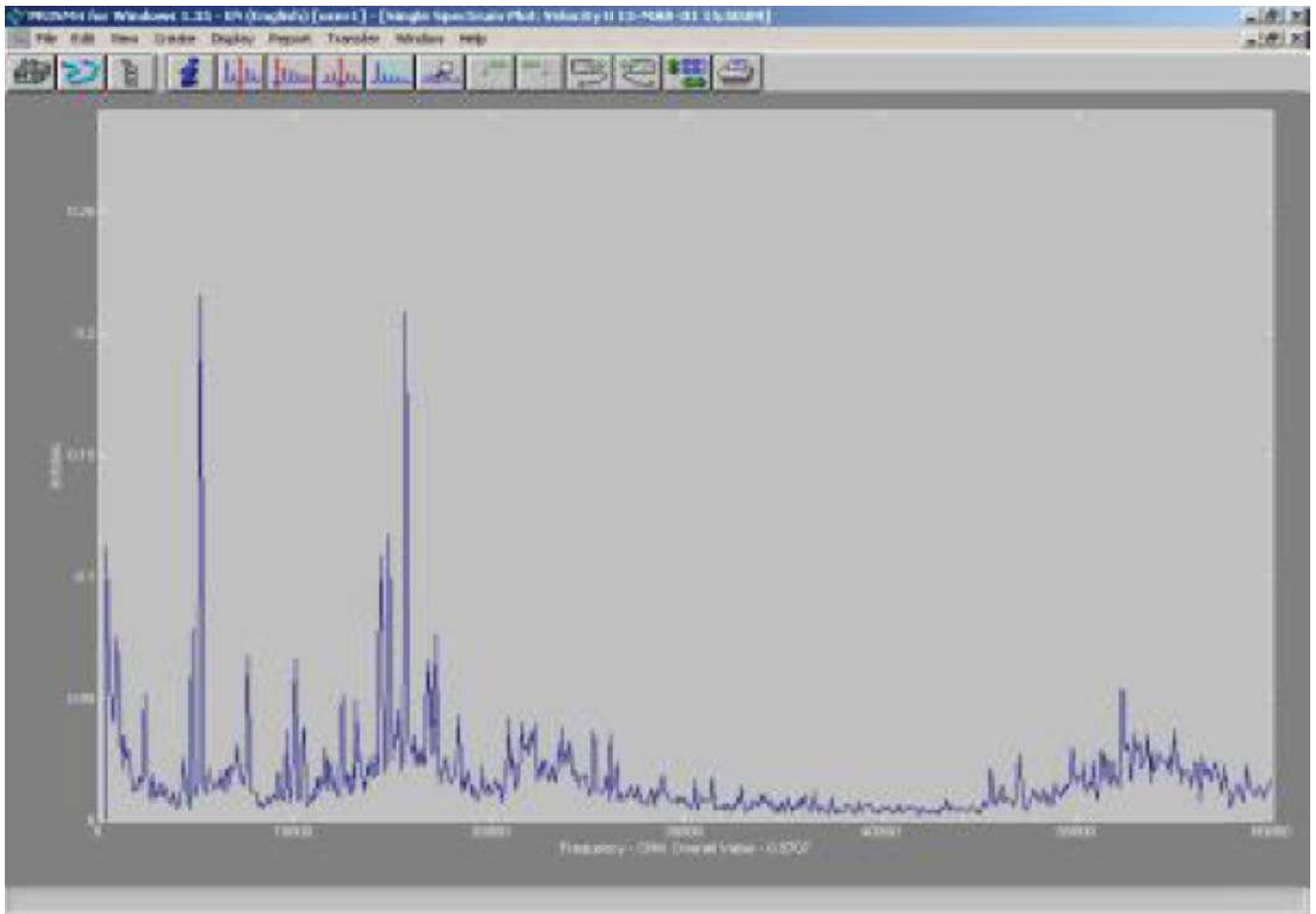


Fig. 13. Velocity measurement with typical bearing frequencies indicated as a "hump" in the spectrum at approximately 9x running speed. The other peaks to the left side of the spectrum are unbalance, misalignment and some looseness due to the loss of loading properties.

Acceleration enveloping spectral analysis

In the early stages of degradation, a bearing defect may not be detectable on normal acceleration or velocity vibration spectra. This is due in part to:

- The vibration that is present in the bearing frequency range may not be shown by the FFT.
- The vibration's amplitude is so small that low frequency rotational vibrations mask it.

Acceleration enveloping measurements monitor bearing frequency ranges at which the defect's repetitive impacts occur and filter out all non-repetitive impact signals (i.e., low frequency rotational events). The repetitive impact signals are enhanced and appear as peaks at the defect's frequency. To assist in determining if a machine's problems include a faulty bearing, bearing defect frequencies can be calculated and overlaid on the vibration spectra.

The enveloped time domain of an acceleration measurement and spectra for an inner ring defect are shown in **fig. 14**. When collecting acceleration enveloping readings, it is important to also collect time domain data. Time domain data can be very useful in the diagnosis of vibration problems in components such as gears and bearings. **Figs. 14** through **19** show examples of spectrum and time waveform data. All of the illustrations contain captions to describe each figure and its data.

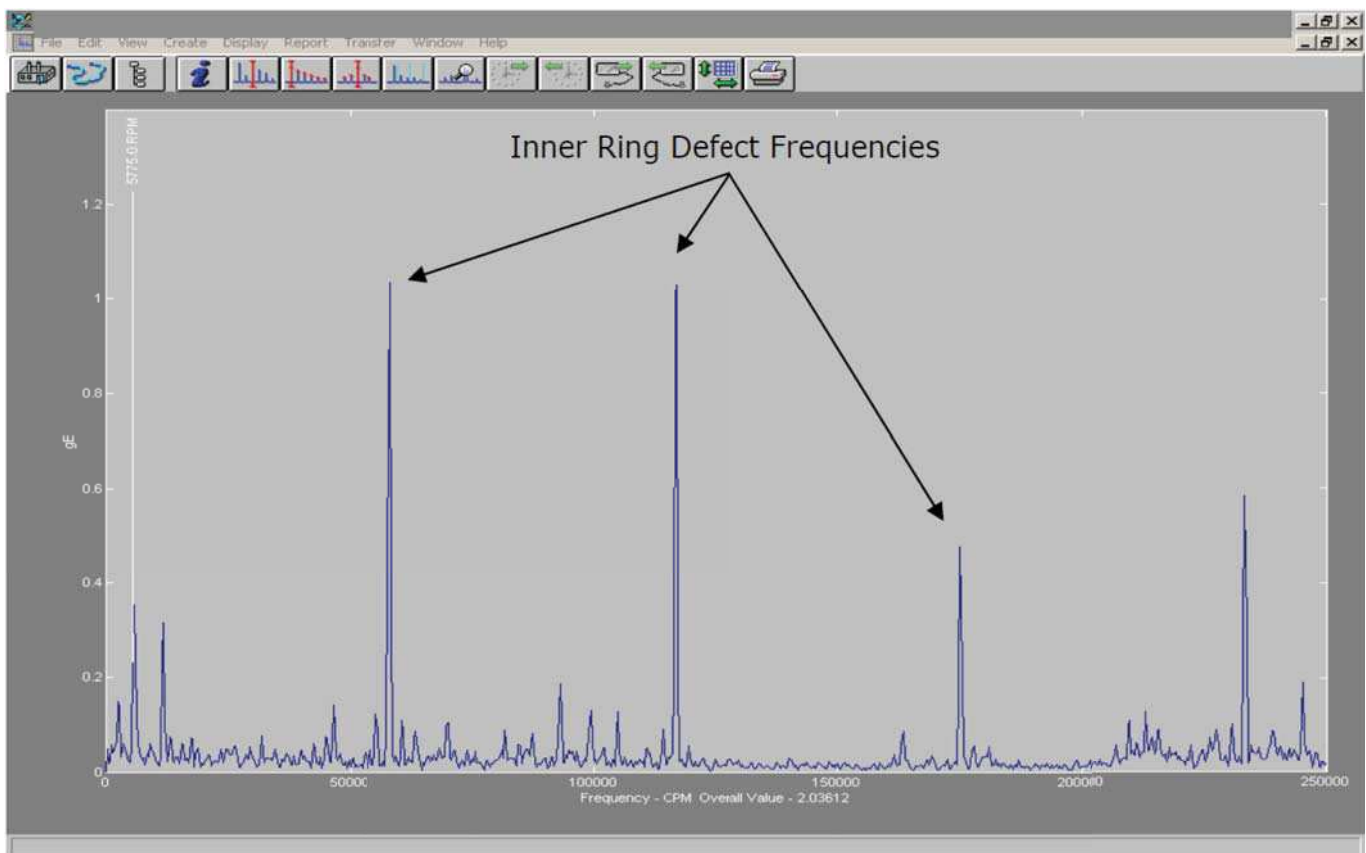


Fig. 14. Inner ring defect frequencies displayed in an enveloped spectrum. The first peak, from left to right, is running speed (5 775 RPM). The large peaks at ~51 000, 115 000 RPM, etc., are peaks in the spectrum related to the defect frequency of the inner ring of the bearing. These peaks indicate a possible defect on the bearing's inner ring.

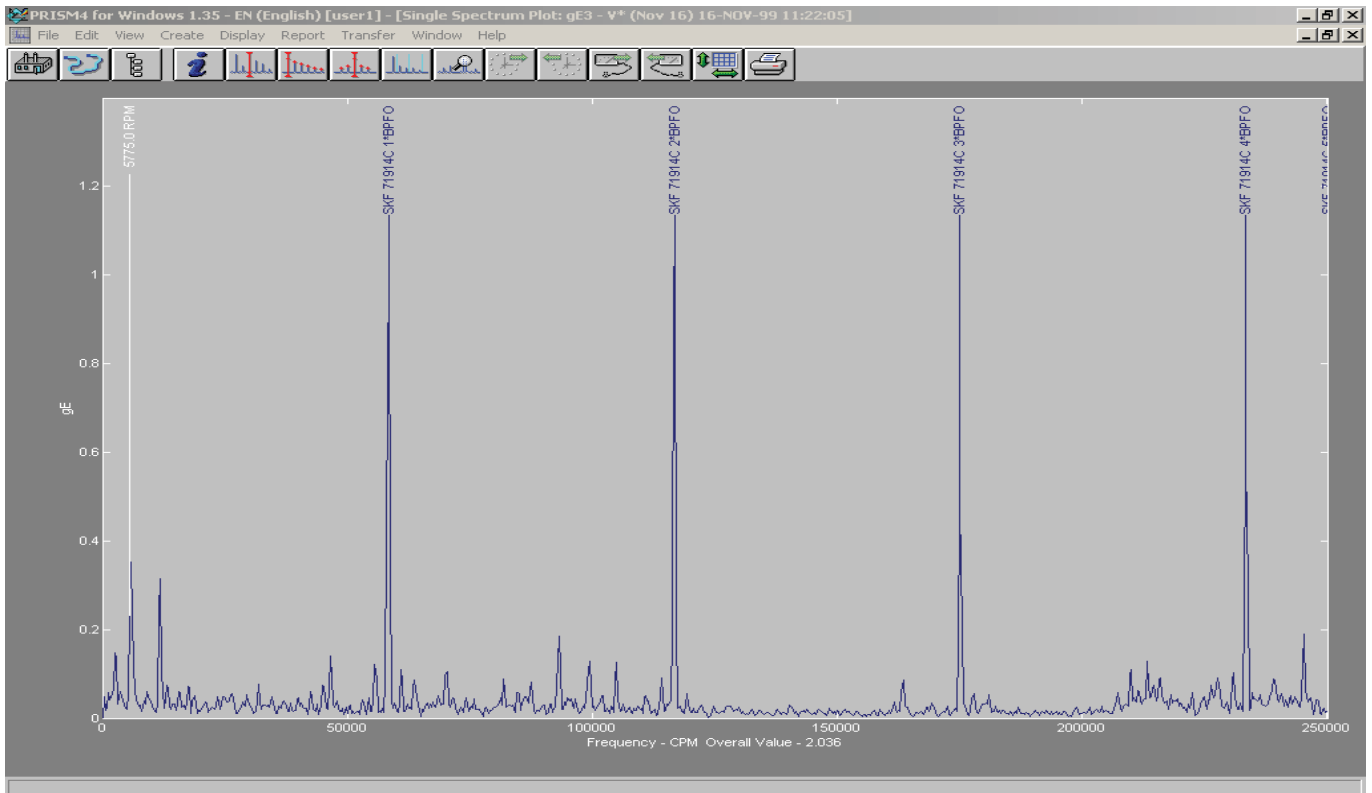


Fig. 15. Enveloped spectrum with outer race defect and bearing frequency overlays. This spectrum indicates a defect is present on the bearing's outer race.

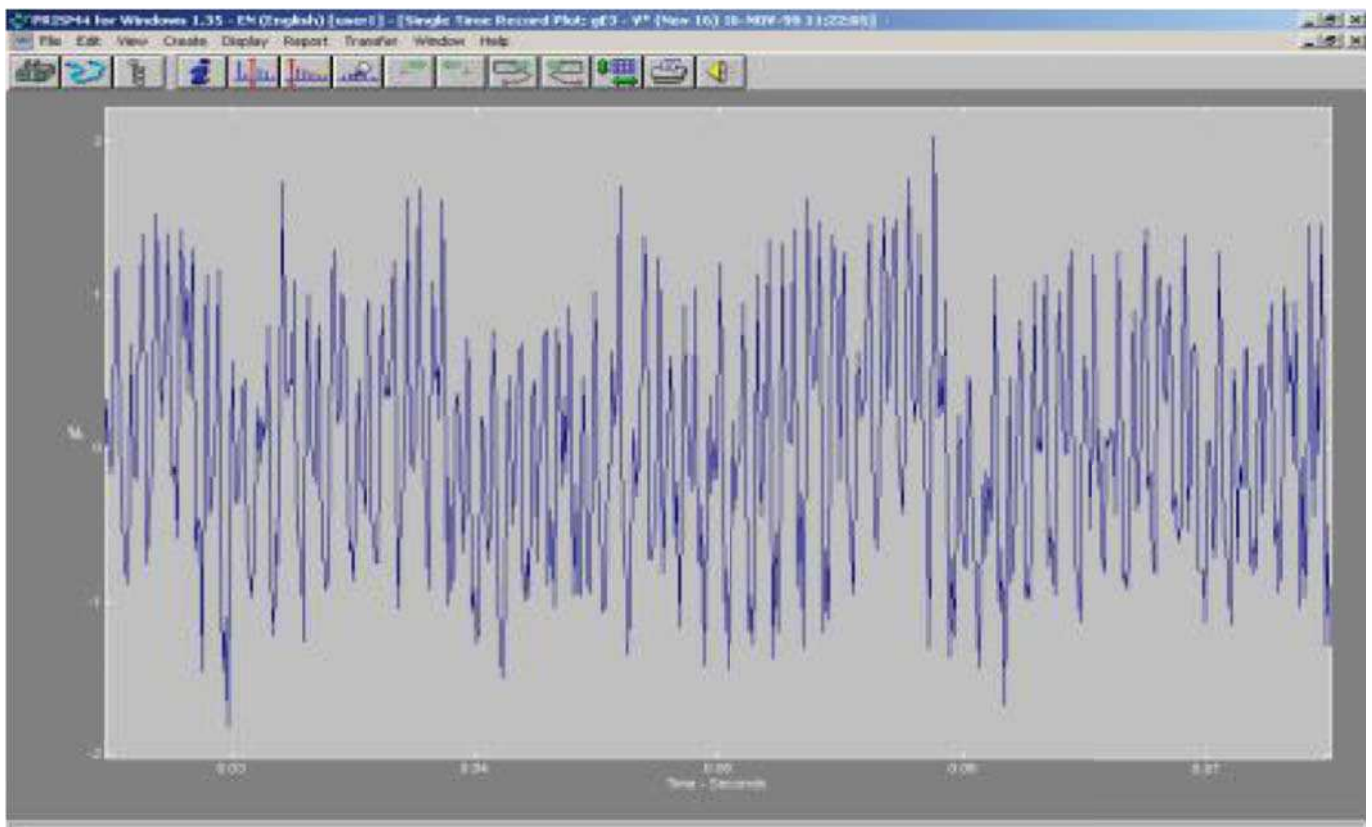


Fig. 16. Enveloped time waveform (defect outer race). The defect is indicated by the modulation of this signal. The expansion and contraction of the peaks from a high amplitude (2 gE) then toward the center (0) indicate that energy is being generated as the rolling element over-rolls the defect.

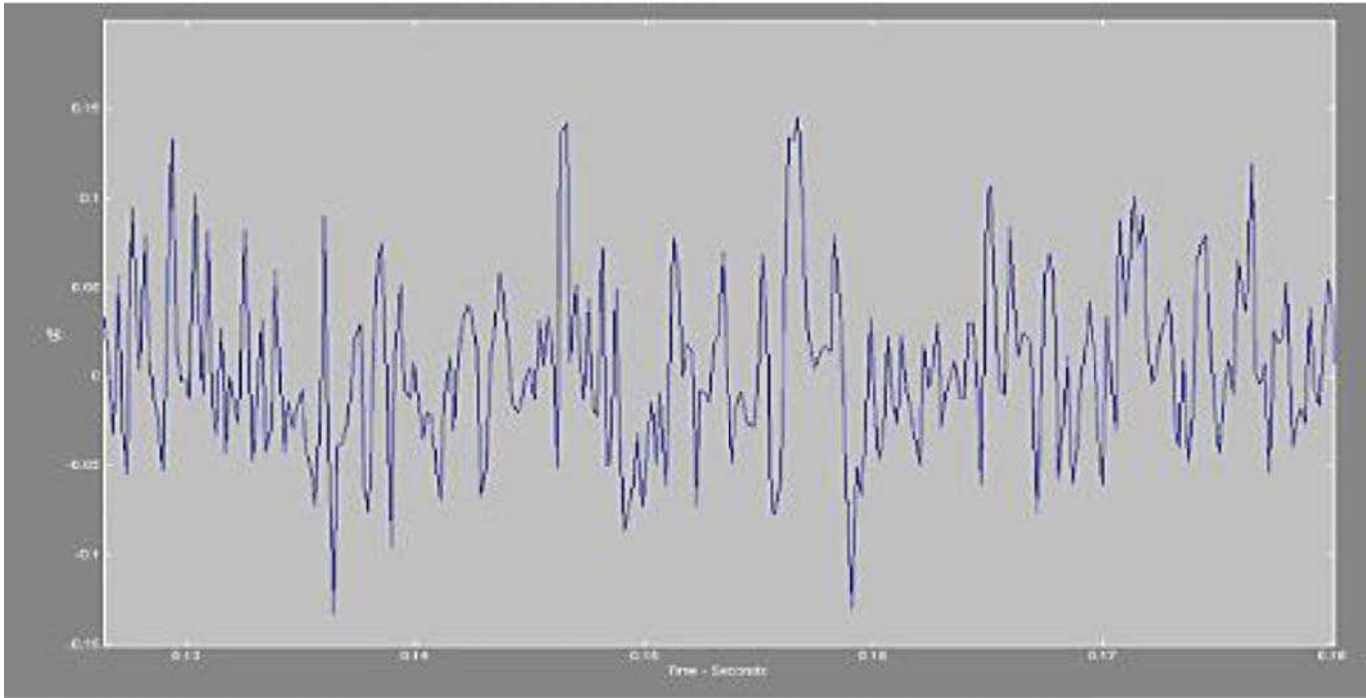


Fig. 17. Enveloped time waveform (defect inner race). The defect is indicated by the modulation of this signal. The expansion and contraction of the peaks from a high amplitude (0,2 gE) then toward the center (0) indicate that energy is being generated as the rolling element over-rolls the defect.

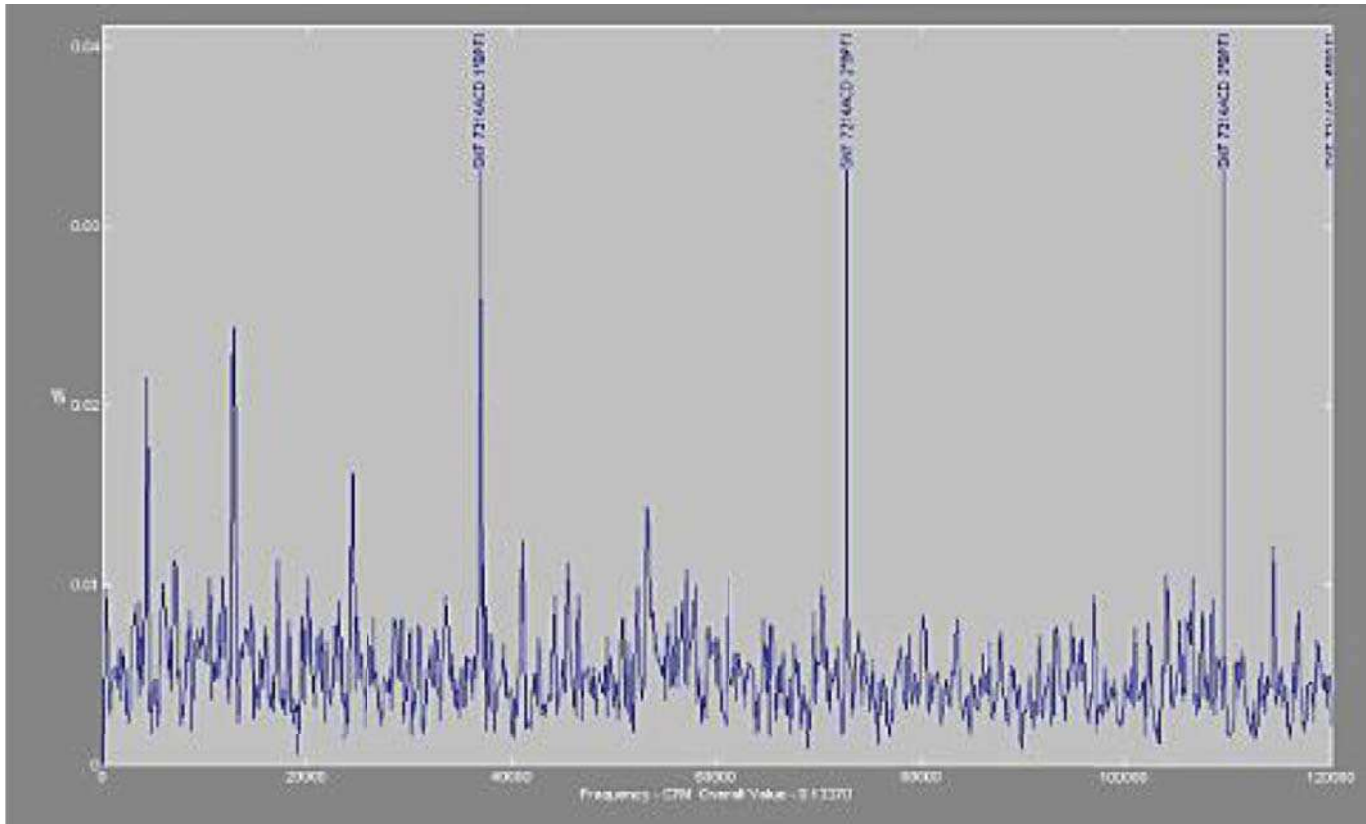


Fig. 18. Enveloped spectrum with inner race defect and bearing frequency overlays. This spectrum indicates a defect is present on the bearing's inner race.

Journal or plain bearings

Journal, plain and tilted pad bearings have infinite life, provided there is adequate lubrication to support the oil wedge on which the shaft rotates. The shaft performance is normally monitored by two orthogonally positioned eddy current probe transducers that measures the relative displacement between shaft and bearing block. Displacement warning alarms will indicate the need for possible shaft balancing procedures. The bearing condition is best assessed by sump oil analysis, which confirms excessive metal wear.

SEE measurements have been applied to trend lubrication conditions of rotating element bearings, but it has not been reported as being applied to journal bearings. SEE transducers are based on acoustic emissions concepts, which provide indications of metal-to-metal contacts. Acceleration enveloping is normally only applied where impulsive defect forces are measurable, such as in rotating element bearings.

Summary

Figs. 14 through 18 are advanced examples of data from an FFT Analyzer. Time waveform and spectrum analysis are difficult subjects to explain thoroughly in an article that overviews the key features of spectrum analysis. There are extensive training courses on analyzing vibration data. Some key issues to consider when using vibration analysis as a method of determining machinery health are:

- Collect both spectrum and time waveform data to complete a thorough data analysis.
- Develop skills around condition monitoring through training and application.
- Build a knowledge bank of machinery responses and problems. This helps you apply previously gained knowledge and minimizes repeat mistakes.

10. Gears

Gears are used to transmit power from one system to another. It is important to understand how gears work and what symptoms to look for when performing an analysis. Moreover, you should fully understand the two key elements to consider:

- Gear mesh frequency (GMF)
- Sidebands of GMF

By monitoring these two elements, you can establish how the gear affects the system and the significance of the problem.

Gear mesh frequency

Gear mesh frequency equals the number of teeth on the gear multiplied by the speed of the shaft to which the gear is attached.

- $GMF = (\text{\# of teeth on the gear}) (\text{speed of the shaft to which the gear is attached})$

Example:

- $GMF = (50 \text{ teeth}) (1\,180 \text{ RPM})$
- $GMF = 59\,000 \text{ CPM or } 983,3 \text{ Hz}$

In addition to evaluating GMF, it is important to use the proper span (F_{\max}) regarding frequency range to observe the GMF at higher frequencies in the same vibration signature. To achieve this span, GMF should be multiplied by a factor of 3,25.

Example: Using the above GMF:

- $F_{\max} = 3,25 \times GMF$
- $F_{\max} = (3,25) (59\,000 \text{ CPM})$
- $F_{\max} = 191\,750 \text{ CPM or } 3\,195,8 \text{ Hz}$

If the GMF is not known, use:

- $F_{\max} = 200 \times \text{shaft running speed}$
- $F_{\max} = 200 \times 1\,180 \text{ RPM}$
- $F_{\max} = 236\,000 \text{ CPM or } 3\,933,3 \text{ Hz}$

The factor of 3,25 relates to a gear characteristic that wear problems do not necessarily occur at fundamental gear mesh frequency (1x GMF), but may occur at 2x or 3x GMF. In fact, one of the most common frequencies at which gear mesh is detected is 3x GMF. This is attributed to the three motions of gear interaction: engaged sliding, rolling and disengaged sliding. Hence, three pulses per revolution. The consideration of this factor should be evaluated when collecting gear mesh data.

Gear mesh frequency sidebands

Gear mesh frequency sidebands can be more significant than GMF. The sidebands are spaced around the GMF relative to the RMP of each mating gear. When the amplitude of the sidebands increases and the number of sidebands present increases, there is likely a problem with the gearbox components.

Additionally, if one or both interfacing gears have worn teeth, the spectrum also exhibits sidebands around GMF. These sidebands are spaced at a distance equal to the shaft speed.

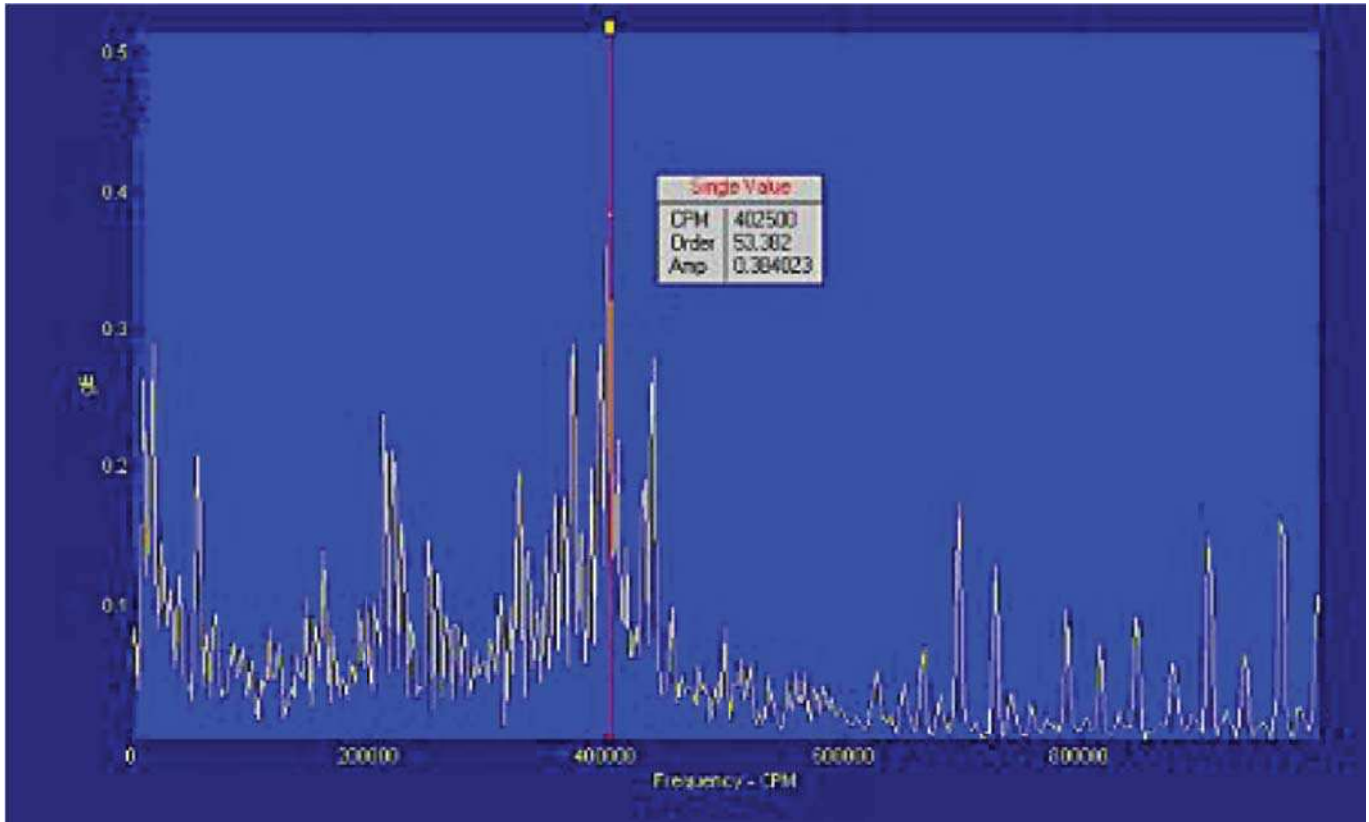


Fig. 19. Spectrum with gear mesh frequency at 402 500 RPM, marked with the overlay. The shaft is turning at 7 545 RPM with 53 teeth on the gear; therefore, $7\,595 \times 53 = 402\,500$ RPM.

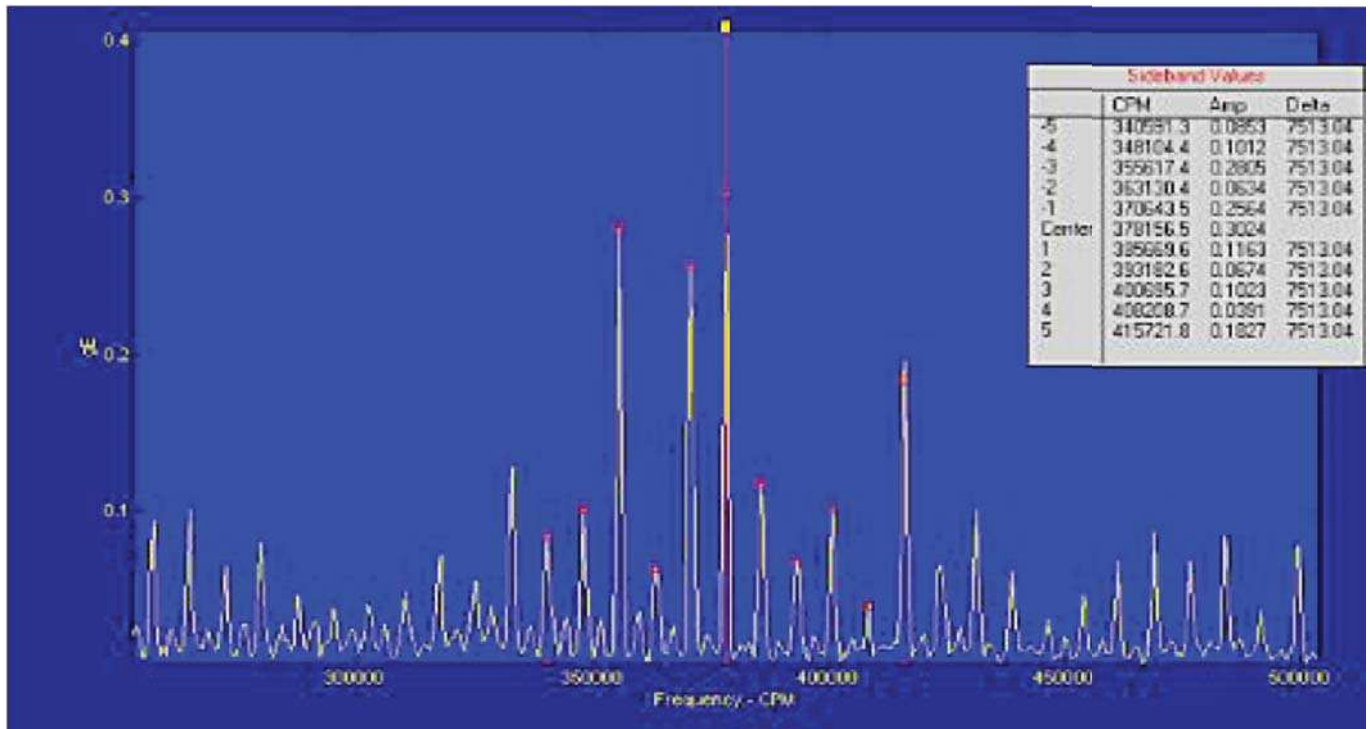


Fig. 20. Spectrum containing gear mesh frequency at 378 157 RPM, marked with sideband markers. Sidebands are spaced at 7 513 RPM, which is the nominal speed of the shaft on which the gear is riding.

11. Blades and vanes

Unlike some other types of machine condition vibration, flow-induced vibration can be very dependent on operating conditions. In other words, depending upon the machine's work, or the induced load, the machine can exhibit varying conditions. Flow-induced vibration conditions are as follows:

- Hydraulic or aerodynamic forces
- Cavitation or starvation
- Recirculation
- Turbulence
- Surging or choking

Pumps, blowers, turbines, etc., inherently produce hydraulic or aerodynamic forces as their impellers impart work into the fluid they are handling. Under normal conditions, such forces are handled rather easily. A problem arises when these forces excite resonant frequencies and cause problems such as cavitation or excessively high vibration. The most commonly generated signal related to hydraulic or aerodynamic forces is Blade Pass Frequency (BPF):

- $BPF = \# \text{ of vanes} \times \text{impeller RPM}$

Example:

- $BPF = (6) (3\,600 \text{ RPM})$
- $BPF = 21\,600 \text{ CPM or } 360 \text{ Hz}$

These forces are generated by a pressure variation or pulse each time a blade loads or unloads as it passes nearby stationary components. From a vibration signal standpoint, it is common to look for BPF and harmonics of BPF. This occurs if the impeller is not properly aligned with the diffusers and centrally located within the housing. Another common blade frequency is Blade Rate Frequency (BRF):

- $BRF = (\# \text{ impeller vanes}) (\# \text{ diffusers}) (\text{RPM}) / K$
- $BRF = [(18 \text{ impeller vanes}) (24 \text{ diffusers}) (\text{RPM})] / 6$
- $BRF = 72 \times \text{RPM}$

Where:

- K = Highest common factor of impeller vanes and number of diffuser vanes

Thus, BRF (72 × RPM) is 4x higher than BPF (18 × RPM) in this case. And, as was pointed out above, this machine would likely suffer much higher pulsation, as more than one set of impeller and diffuser vanes would line up with one another (in this case, six impeller vanes would simultaneously be directly opposite diffuser vanes at angles of 0°, 60°, 120°, 180°, 240° and 300°), which results in pronounced pulsation at BRF. If there were either 17 impeller vanes or 25 diffuser vanes, at no instance in time would more than one set of impeller and diffuser vanes line up with one another. Therefore, high vibration would be unlikely.

Cavitation is a common centrifugal pump problem and can be quite destructive to internal pump components. Cavitation most commonly occurs when a pump is operating with excess capacity or low suction pressure. Since the pump is actually being starved, the fluid is being pulled apart as it tries to fill the cavity. This process causes pockets of vacuum that collapse or implode quickly, which creates impact that excite natural frequencies of the impeller and nearby components. The most common characteristics of cavitation are:

- Random, broadband energy between 20k and 120k CPM, which can cause excessive system, wear
- The sound of sand or gravel being pumped through system
- Starvation is the counterpart to cavitation and also involves insufficient airflow

As it relates to a pump, *recirculation* is the opposite of cavitation. It can occur when the pump is operating at a lower capacity than required or a high suction pressure. Recirculation causes fluid to move in more than one direction at the same time, which causes excessive noise and vibration. The most common characteristic of recirculation is:

- Random, broadband energy between 20k and 120k CPM

Flow turbulence occurs when something interferes with normal system flow. The most common characteristics of flow turbulence are:

- Low frequency, random vibration below 1x RPM that is commonly in the range of 50 CPM to 2 000 CPM
- Erratic, widely pulsating amplitudes

Choking or “stone walling” usually occurs when the discharge pressure is too low. This causes the velocities to increase in the diffuser section. Common characteristics of choking are:

- Increases in BPF and harmonics of BPF
- The overall noise floor rises across the entire frequency band

Surging usually occurs when the discharge pressure is too high or the volumetric pressure is too low based upon machinery design conditions. The most common characteristics of surging are:

- Increases in BPF and harmonics of BPF
- The overall noise floor rises across the entire frequency band

12. Electrical problems

Monitoring components other than mechanical systems can also be beneficial to an analysis program. Electrical problems can be evaluated using vibration technology. Electrical problems can be detected from the generation of magnetic fields in machinery. These fields create flux, which induces electromagnetic forces that impart forces mechanically and ultimately affect the bearings.

2x line frequency

Many issues associated with electrical problems are detected at 2x line frequency. In North America, line frequency is 60 Hz (3 600 CPM), and in Europe it is commonly 50 Hz (3 000 CPM). Therefore, be aware of these frequencies:

- 120 Hz (7 200 CPM)
- 100 Hz (6 000 CPM)

There is a lengthy discussion as to why 2x line frequency is a key feature in monitoring electrical equipment. The following example helps explain this aspect.

With every motor revolution in a two-pole motor in Europe, rotating at 3 000 RPM (50 Hz) causes a magnetic pull toward the closest pole 2x per revolution (i.e., the magnetic pull goes from zero to maximum twice per revolution). This rotation causes the electrical signal to fluctuate from 0 to 100 Hz or 6 000 CPM every revolution.

Therefore, when a rotor is not centered within the stator, it causes a variable air gap between the rotor and the stator, which affects the 2x line frequency.

Stator problems

Stator problems are detectable with vibration analysis:

- Stator eccentricity (stationary differential air gap)
- Shorted lamination (insulation problems)
- Loose irons (loose or weak stator)

These problems exhibit high 2x line frequency and may or may not generate pole pass frequency sidebands, as they are generated in the stator and are not modulated by either running speed or slip frequency. Formulas for electrical motors include:

- N_s (Synchronous Speed) = $(120 F_L)/P$
- F_s (Slip Frequency) = $N_s - \text{RPM}$
- F_p (Pole Pass Frequency) = $F_s (P)$
- RBPF = (# of Rotor Bars) (RPM)

Where:

- F_L = Electrical Line Frequency
- RPM = Rotor Speed
- N_s = Synchronous Speed
- F_s = Slip Frequency
- F_p = Pole Pass Frequency
- P = Number of Poles
- RBPF = Rotor Bar Pass Frequency

Rotor problems

Rotor problems are detectable with vibration analyses:

- Broken or cracked rotor bars
- Bad high resistance joints between rotor bars and shortening rings
- Shorted rotor lamination
- Loose/Open rotor bars that make bad contact with end rings

The most likely area of concern for broken or cracked rotor bars is the presences of pole pass frequency sidebands around 1x RPM and running speed harmonics.

Broken or cracked rotor bars and/or high resistance joints can produce pole pass sidebands around higher running speed harmonics up to and including the 2nd, 3rd, 4th and 5th running speed harmonics.

Loose or open rotor bars are indicated by vibration and harmonics at Rotor Bar Passing Frequency (RBPF). In addition to RBPF, the signature may contain sidebands around RBPF spaced at 2x line frequency and may have a higher amplitude than the RBPF frequency. The RBPF vibration signal is at a high frequency and is calculated using the following formula:

- $RBPF = (\# \text{ of Rotor Bars}) (RPM)$
- $RPM = \text{Rotor Speed}$

13. Step 3

13.1. Multi-parameter monitoring

When conducting any type of condition monitoring program, it is valuable to evaluate the system with several different analysis parameters. A multi-parameter approach gives the greatest amount of resulting data to help determine the problem's root cause.

A multi-parameter approach to condition monitoring uses several types of measurement technologies to detect and diagnose bearing and machinery problems. This allows for early detection and provides more ways to measure deviations from normal signals. Multi-parameter monitoring is very effective for bearing monitoring.

For example, if a rolling element bearing contains a defect on its outer race, each roller strikes the defect and cause a small, repetitive vibration signal. However, this vibration signal is of such low amplitude that overall vibration monitoring does not detect it. Therefore, a multi-parameter monitoring approach is most effective.

There are several categories of monitoring to consider. Each category is developed for a specific reason. The following section explains each type of technique.

- *Overall vibration*: Monitors low frequency machine vibrations and detects rotational and structural problems such as unbalance, misalignment, shaft bow and mechanical looseness. It is also used to detect bearing problems in their later stages.
- *Acceleration enveloping*: Filters out low frequency vibration noise and enhances high frequency, repetitive bearing and gear mesh vibration signals. This method proves very effective for early detection and diagnoses of bearing problems. With bearings, acceleration enveloping technologies provide ample pre-warning time, which allows a maintenance person to take early corrective action to effectively extend bearing life.
- *Acceleration*: This measurement is primarily an indication of how quickly the system is changing. Acceleration is very important for dynamic mechanics, as acceleration relates to system force and mass. Additionally, the higher the frequency, the higher the acceleration – even at the same velocity level.
- *Velocity*: Allows us to monitor the rate at which the system is increasing. Velocity is a good indication of individual component speed. Velocity is used as a monitoring technique to distinguish between component problems and indicate bearing issues in late stages of degradation.
- *Displacement*: Describes the distance between two points. Today, displacement is rarely used in condition monitoring as a standard measurement, as it is primarily an indication of roundness. However, it is used at very low frequencies where responses from other types of measurement techniques give poor results.

14. Conclusions

This guide intends to aid in the understanding of condition monitoring. Reference this material for program development to be aware that a multiple-parameter monitoring program gives the greatest amount of certainty.

15. Further reading

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 - Paper Mills Gaining from Condition Monitoring, 1999/4
 - Paper Mill Gains from Condition Monitoring, 2000/3
 - High Tech keeps Mine competitive, 2001/2
 - Fault Detection for Mining and Mineral Processing Equipment, 2001/3
- Technical Associates of Charlotte (diagnostic charts, background articles and books): <http://www.technicalassociates.net/>
- The SKF Reliability Maintenance Institute (RMI) offering of hands-on training courses. Contact RMI: <http://www.skfusa.com/rmi>
- Vibration Institute: <http://www.vibinst.org/>
- Vibration Resources: <http://vibrate.net>

SKF – the knowledge engineering company

From the company that invented the self-aligning ball bearing more than 100 years ago, SKF has evolved into a knowledge engineering company that is able to draw on five technology platforms to create unique solutions for its customers. These platforms include bearings, bearing units and seals, of course, but extend to other areas including: lubricants and lubrication systems, critical for long bearing life in many applications; mechatronics that combine mechanical and electronics knowledge into systems for more effective linear motion and sensorized solutions; and a full range of services, from design and logistics support to condition monitoring and reliability systems.

Though the scope has broadened, SKF continues to maintain the world's leadership in the design, manufacture and marketing of rolling bearings, as well as complementary products such as radial seals. SKF also holds an increasingly important position in the market for linear motion products, high-precision aerospace bearings, machine tool spindles and plant maintenance services.

The SKF Group is globally certified to ISO 14001, the international standard for environmental management, as well as OHSAS 18001, the health and safety management standard. Individual divisions have been approved for quality certification in accordance with ISO 9001 and other customer specific requirements.

With over 120 manufacturing sites worldwide and sales companies in 70 countries, SKF is a truly international corporation. In addition, our distributors and dealers in some 15 000 locations around the world, an e-business marketplace and a global distribution system put SKF close to customers for the supply of both products and services. In essence, SKF solutions are available wherever and whenever customers need them. Overall, the SKF brand and the corporation are stronger than ever. As the knowledge engineering company, we stand ready to serve you with world-class product competencies, intellectual resources, and the vision to help you succeed.

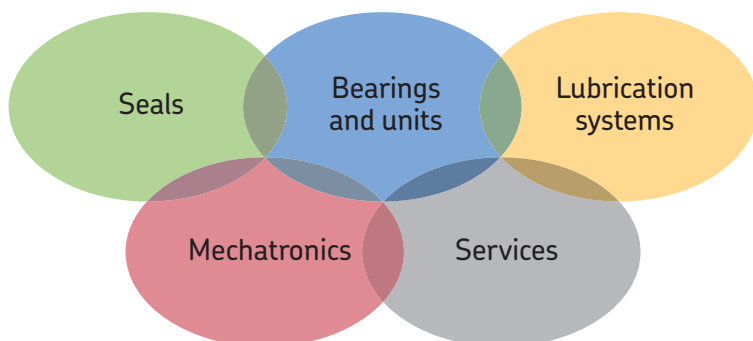


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Evolving by-wire technology

SKF has a unique expertise in the fast-growing by-wire technology, from fly-by-wire, to drive-by-wire, to work-by-wire. SKF pioneered practical fly-by-wire technology and is a close working partner with all aerospace industry leaders. As an example, virtually all aircraft of the Airbus design use SKF by-wire systems for cockpit flight control.

SKF is also a leader in automotive by-wire technology, and has partnered with automotive engineers to develop two concept cars, which employ SKF mechatronics for steering and braking. Further by-wire development has led SKF to produce an all-electric forklift truck, which uses mechatronics rather than hydraulics for all controls.





Harnessing wind power

The growing industry of wind-generated electric power provides a source of clean, green electricity. SKF is working closely with global industry leaders to develop efficient and trouble-free turbines, providing a wide range of large, highly specialized bearings and condition monitoring systems to extend equipment life of wind farms located in even the most remote and inhospitable environments.



Working in extreme environments

In frigid winters, especially in northern countries, extreme sub-zero temperatures can cause bearings in railway axleboxes to seize due to lubrication starvation. SKF created a new family of synthetic lubricants formulated to retain their lubrication viscosity even at these extreme temperatures. SKF knowledge enables manufacturers and end user customers to overcome the performance issues resulting from extreme temperatures, whether hot or cold. For example, SKF products are at work in diverse environments such as baking ovens and instant freezing in food processing plants.



Developing a cleaner cleaner

The electric motor and its bearings are the heart of many household appliances. SKF works closely with appliance manufacturers to improve their products' performance, cut costs, reduce weight, and reduce energy consumption. A recent example of this cooperation is a new generation of vacuum cleaners with substantially more suction. SKF knowledge in the area of small bearing technology is also applied to manufacturers of power tools and office equipment.



Maintaining a 350 km/h R&D lab

In addition to SKF's renowned research and development facilities in Europe and the United States, Formula One car racing provides a unique environment for SKF to push the limits of bearing technology. For over 60 years, SKF products, engineering and knowledge have helped make Scuderia Ferrari a formidable force in F1 racing. (The average racing Ferrari utilizes around 150 SKF components.) Lessons learned here are applied to the products we provide to automakers and the aftermarket worldwide.



Delivering Asset Efficiency Optimization

Through SKF Reliability Systems, SKF provides a comprehensive range of asset efficiency products and services, from condition monitoring hardware and software to maintenance strategies, engineering assistance and machine reliability programmes. To optimize efficiency and boost productivity, some industrial facilities opt for an Integrated Maintenance Solution, in which SKF delivers all services under one fixed-fee, performance-based contract.



Planning for sustainable growth

By their very nature, bearings make a positive contribution to the natural environment, enabling machinery to operate more efficiently, consume less power, and require less lubrication. By raising the performance bar for our own products, SKF is enabling a new generation of high-efficiency products and equipment. With an eye to the future and the world we will leave to our children, the SKF Group policy on environment, health and safety, as well as the manufacturing techniques, are planned and implemented to help protect and preserve the earth's limited natural resources. We remain committed to sustainable, environmentally responsible growth.

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