Category

II

MOBIUS INSTITUTE

Vibration Training Course Book Category II

Produced by SUMICO Technologies

This manual is designed as a guide only.

In practical situations, there are many variables,

so please use this information with care.

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Chapter 1

Principles of Vibration

Objective: *Describe the relationship between the waveform and spectrum and how the waveform is generated from a machine's movement.*

This Chapter presents the following topics:

- Simple harmonic motion
- Amplitude: Peak, peak-to-peak, and RMS
- Period and Frequency
- An Introduction to phase
- Displacement, velocity, and acceleration
- Units and unit conversions
- Overall level readings and crest factor
- Complex vibration
- The FFT and the spectrum
- Orders and forcing frequencies

Introducing Vibration

A thorough understanding of the basics is required in order to become an expert in the field of vibration. A person needs to know:

- how to make a measurement
- what you are measuring
- what the signals look like
- how to interpret the final data graphically

This first section presents a brief overview of the vibration analysis process. It presents the basics of vibration including the sources of vibration. The terms **Waveform and Spectrum** are introduced along with terms used to describe it such as Frequency and other units. The goal is to become comfortable with the waveform and spectrum.

The actual measurement, signal processing, and detail diagnostic issues are covered in other modules.

What is Vibration?

Rotating machines such as fans, pumps, motors, and turbines vibrate when they are operating. The vibration can be listened to and mechanical problems or faults can often be heard. But that is only part of the story. Using special sensors and monitoring electronics, the vibration provides an early warning of a wide range of fault conditions: damaged bearings, misaligned components, out-of-balance rotors, loose foundations, and many, many other conditions. The vibration changes as the condition changes.

The forces within the machine cause vibration which is transferred to the bearings. The forces are the result of rotational and frictional forces. When vibration is measured at the bearing of a machine, it is actually the response of the bearing housing to the forces generated inside the machine.

Getting started with the basics of vibration

If you want to become an expert vibration analyst, you need to start with the basics. You need to know how to make the measurement, you need to know what you are measuring, you need to know what the signals look like, and you need to know how to interpret the final data graphically. Let's start that journey of discovery right now!

In this module we will focus on the fundamentals of vibration. You will learn about the time waveform, the vibration spectrum and the RMS overall reading. You will learn how to characterize vibration amplitudes and frequencies in various units and will see you they relate to one another. You will also learn a bit about phase and how it is used to relate one signal to another. The goal of this section is to make you comfortable with the basic characteristics of vibration, whether it is coming from a machine or a guitar string, and to teach you the terminology that describes vibration. Vibration measurement, signal processing, and machine fault detection are covered in other modules.

Introducing amplitude

The height of the vibration waveform is the "amplitude" and the amplitude is related to the severity of the vibration. When we looked at the case of the mass and the spring we said that the amplitude of the vibration was the same as how far up and down the spring moved. This is called its "displacement." We will see later that we can also talk about how fast the mass moves as it is going up and down ("velocity") or we can talk about how much is accelerates as it moves up and down ("acceleration")

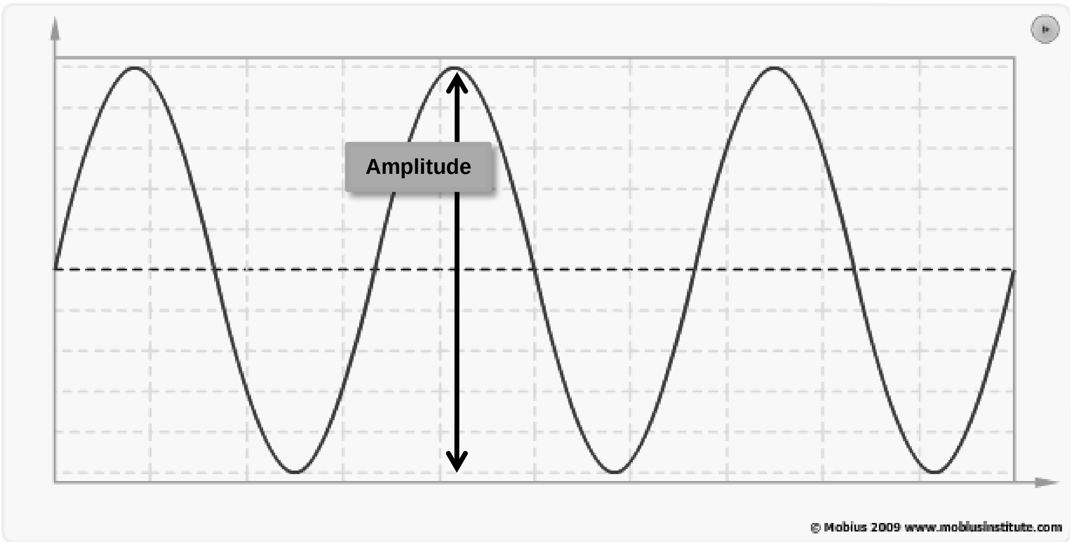


Figure 1-1 The amplitude of a sine wave

Describing vibration data

Now we will look at vibration amplitude a little more closely and we will define the important terms: RMS (Root Mean Square), Average, Peak and Peak to Peak and how these can be used to describe vibration amplitude in more detail.

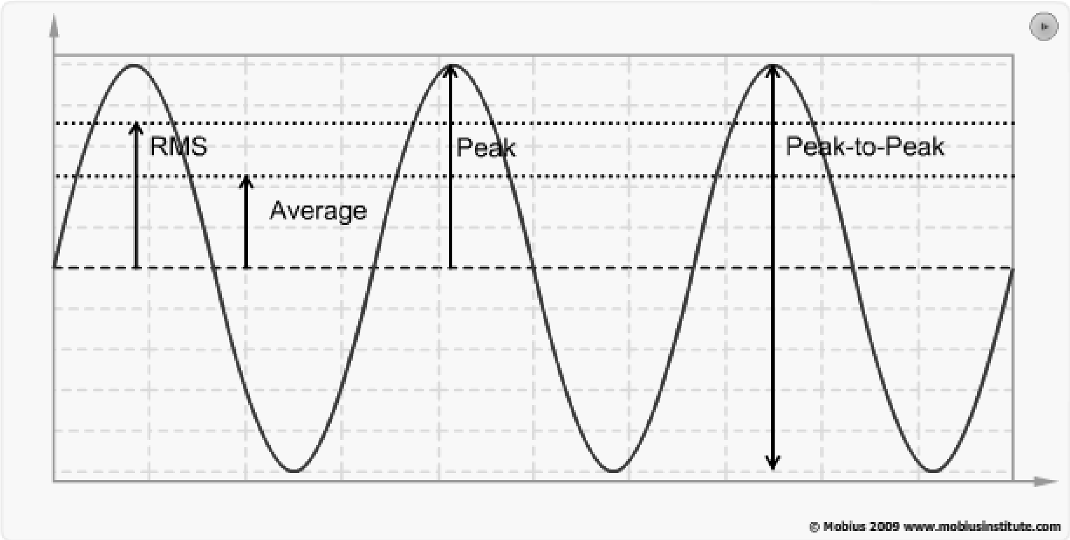


Figure 1-2 Understanding vibration amplitude

Peak and Peak to Peak amplitudes

If we consider the mass on the spring we can think about the total distance it travels all the way from the bottom to all the way at the top. This total distance is called the peak-to-peak amplitude of its vibration. This is often abbreviated "pk-pk." Looking at the figure above you can see this labeled on the waveform and it corresponds to the distance from the bottom most point on the waveform to the top most point.

The next term to define is its "Peak" amplitude. Here we are talking about the furthest distance the mass moves from its point of rest or zero on the graph. From its point of rest, how far did it move upwards or how far did it move downwards? In the case of this particular mass and spring and in the case of "simple harmonic motion" the distance it moves up is equal to the distance it moves down so we can choose either. It is also true in this case that the Peak amplitude (abbreviated "pk" or "0-pk") is equal to half the pk-pk amplitude. This will NOT be true in most cases of vibration and therefore we must understand that the peak is defined as the farthest movement from "0" in either direction up or down and the pk-pk is defined as the total movement from the lowest point to the highest point. As an example, if the mass moved down "3" and up "2" then the peak value would be "3" and the peak to peak would be "5."

RMS amplitude

RMS stands for "Root Mean Square" and in a general sense it describes the average amount of energy contained in the waveform (or in the vibration). Consider the mass on the spring bouncing up and down hour after hour and day after day, we may want to know over this span of time what was its highest movement away from zero (pk amplitude) or we may want to know how far it traveled at any time from the very top to the very bottom (pk-pk). These are both valid pieces of information to want to have.

Now let's look at it a different way. Imagine that the spring and mass is the electricity coming out of your wall outlet – this electricity is also a wave and very close to a sine wave in fact! Now consider you want to power your computer with this electricity – do you care so much that it varied in level a tiny bit during one cycle or another – or – are you more concerned about how much energy on average is coming out of the wall? If you chose the second option then what you want to measure is RMS.

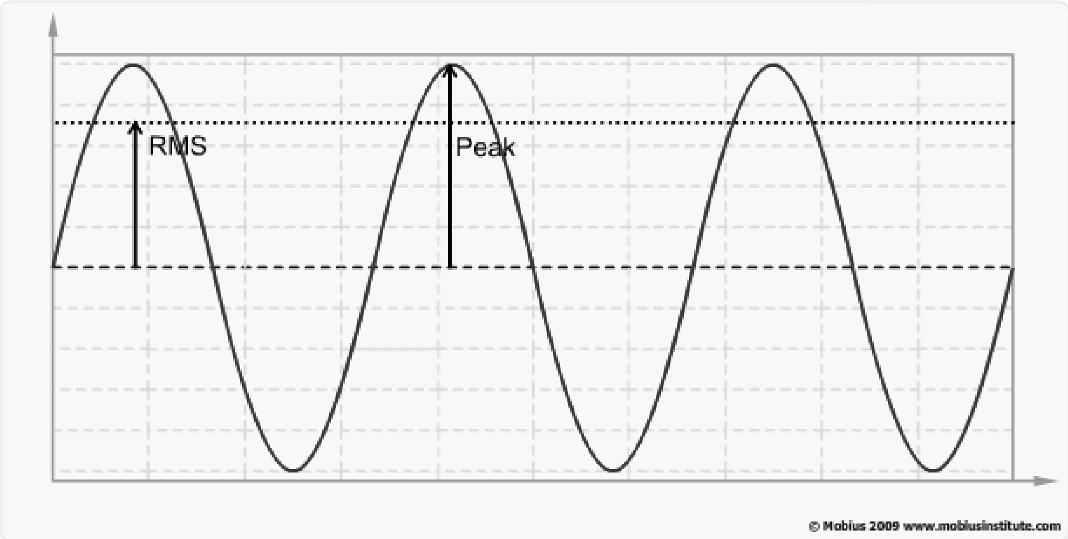


Figure 1-3 RMS and Peak amplitude of a sine wave

$$RMS = \frac{Peak}{\sqrt{2}} = 0.707xPeak \qquad 0.707 = \frac{1}{\sqrt{2}}$$

Figure 1-4 Calculating RMS for a sine wave

In a sine wave and only in a sine wave, the RMS amplitude is equal to 0.707 times the peak amplitude as per the formula above. Later we will see how to calculate the RMS amplitudes of other waveforms. For now, here is an example of how to calculate the pk, pk-pk and RMS values for a pure sine wave.

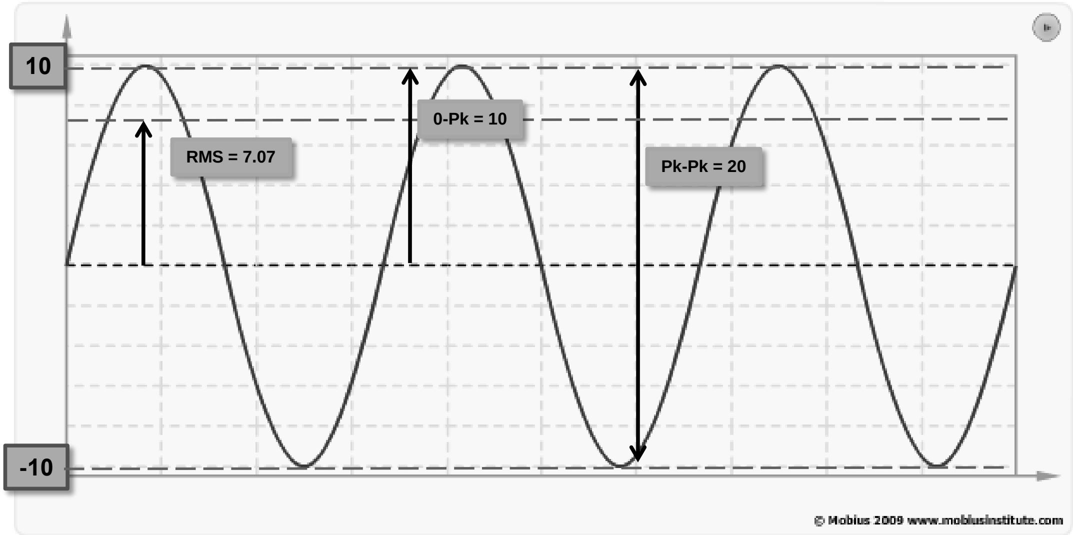


Figure 1-5 Example calculations of pk, pk-pk and RMS for a sine wave

In Figure 1-5, the amplitude of the graph is labeled 10 to -10. To calculate the pk amplitude we look for the greatest distance up or down from zero. In this case (because it is a sine wave) the distance is the same whether we go up or down and thus the peak value is 10. The pk – pk value is twice the pk value or we can measure from all the way at the bottom (-10) to all the way up top (10) and we will see that the total amount of movement is 20. Thus the pk-pk amplitude is 20. Now, because this is a sine wave, we can calculate the RMS value by multiplying the pk value (10) by 0.707. This gives us an RMS value of 7.07.

We will see later that the RMS amplitude is a measure that is often trended in unsophisticated vibration monitoring programs. Because the RMS value relates to the average amount of energy contained in the waveform, the idea is that if the RMS value goes up it means the machine is vibrating more and may have a problem. What you will learn in this course is that there are many reasons for the vibration to increase that are not related to mechanical problems and other cases where mechanical problems may not cause the RMS value to increase.

Before we move on to the next section; a quick reminder: The RMS value is only equal to 0.707 x pk for a pure sine wave. Most real life vibration is not in the form of a sine wave.

Period and frequency

The Basics – The time waveform is the electrical signal from the sensor. It is a trace of the voltage changes as the instantaneous vibration changes from moment to moment. This voltage is graphed versus time; hence the name *time* waveform. The waveform provides a view into exactly how something (like the mass on the spring) is moving or vibrating over time.

The Fan below has a coin attached to one blade to cause it to be unbalanced.

On the fan, a coin is attached to a fan blade to cause it to be out of balance. The added weight of the coin generates centrifugal force as the fan spins. The centrifugal force actually pulls on the fan, forcing it to rotate off center, causing vibration. The centrifugal force due to the coin affects the fan blade, the shaft, and is transferred to the bearings holding the shaft.

Imagine that the fan makes five complete revolutions every second. So as the shaft turns a strong pulsating vibration is felt at the bearings.

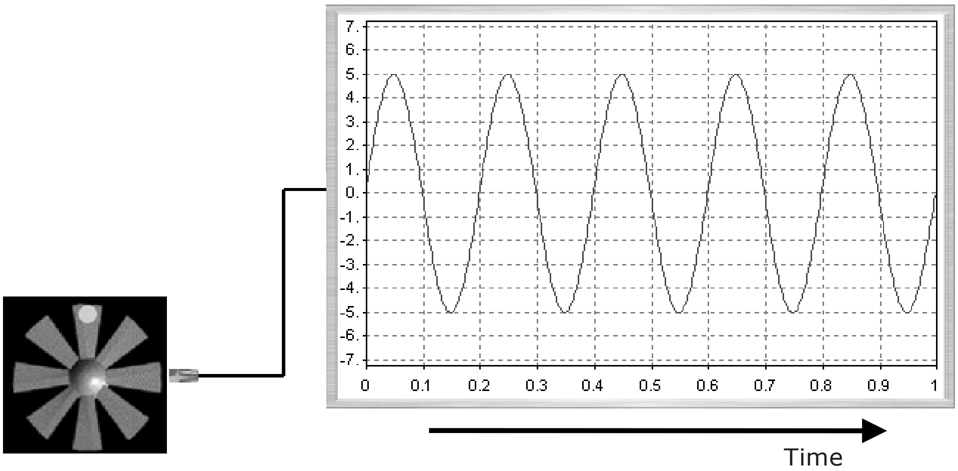


Figure 1-6 Fan with sensor mounted and resulting time waveform

The pulsation of the vibration coincides with the turning of the shaft. In fact, we get one pulsation per rotation. If a sensor is placed on the bearing, and just one second of vibration is looked at, we would see 5 pulsations or cycles in the waveform (because it rotates 5 times per second).

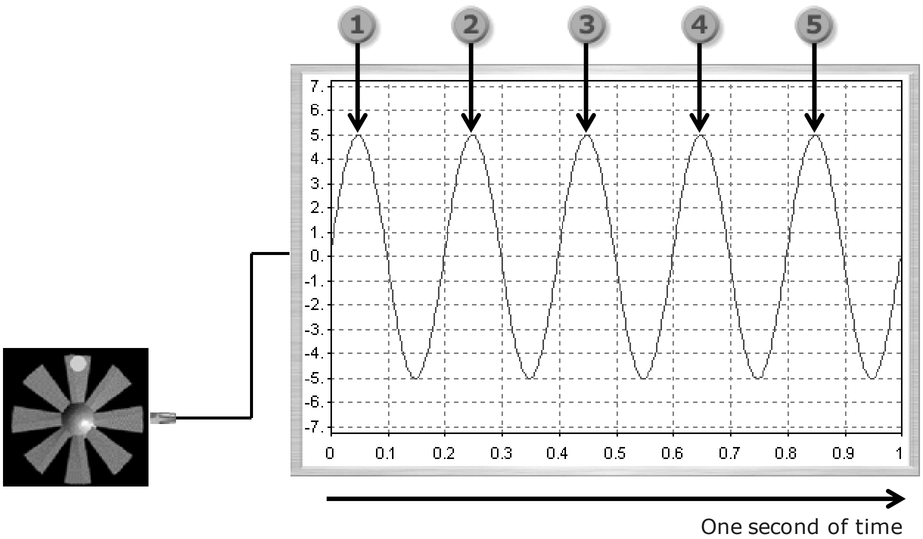


Figure 1-7 A trace of the vibration over 1 second.

As the fan turns, the coin rotates with it changing the location of the unbalance it creates.

Several snapshots are captured showing the position of the coin relative to the waveform. See Figure 1-8.

When the coin is at the top position, the waveform is also at its most upward point of travel.

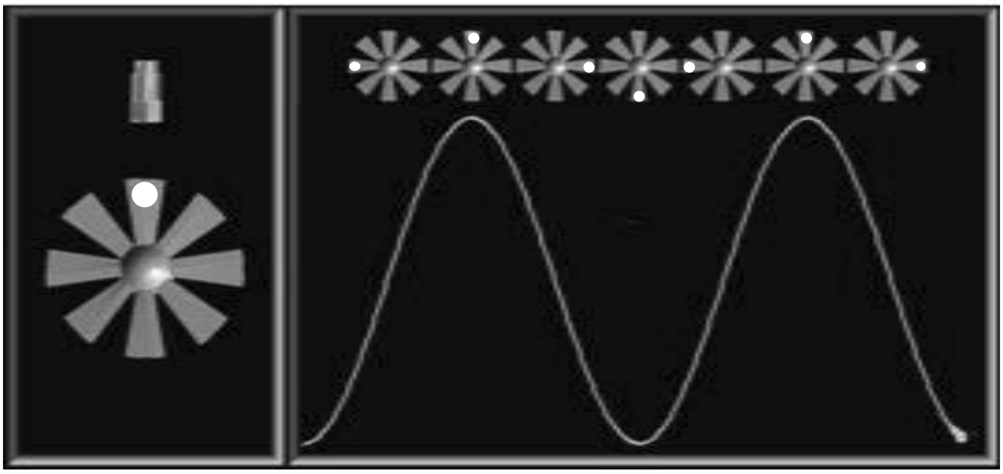


Figure 1-8 The position of the coin is shown relative to the waveform trace.

Notice also that the trough of the waveform occurs when the coin is at its most bottom position. The waveform generated by the unbalance is called a "sine wave."

This sine wave looks the same as if generated by a sensor and an analyzer.

Introducing "frequency"

We have already discussed the amplitude of vibration. It is now necessary to explain another important attribute of vibration namely: "frequency."

Frequency describes how often an event occurs in a defined period of time.

We are all familiar with the concept of frequency in other contexts. For example, how many bus departures there are per day from the bus station; how many times per day we should take our medicine; how many times per week we call our mothers etc. These are all examples of frequency.

In the vibration world, the "defined period of time" is typically either a second or a minute. When we talk about cycles per second, we use the terms "Hertz" and when we talk about cycles per minute we use the term "CPM".

The fan in the previous example was completing five complete rotations per second, or 5 hertz (5Hz).

If the fan completes 5 revolutions every second, then it is completing 300 revolutions every minute. (5 revolutions per second X 60 seconds.) It could be said then that the fan is completing at 300 revolutions per minute (300 RPM.) or 300 cycles per minute (300 CPM) in this case the two are interchangeable. Here are the formulas:

Hertz = Hz = Cycles per second = CPS

RPM = Revolutions per minute

CPM = Cycles per minute

CPM = RPM = Hz x 60

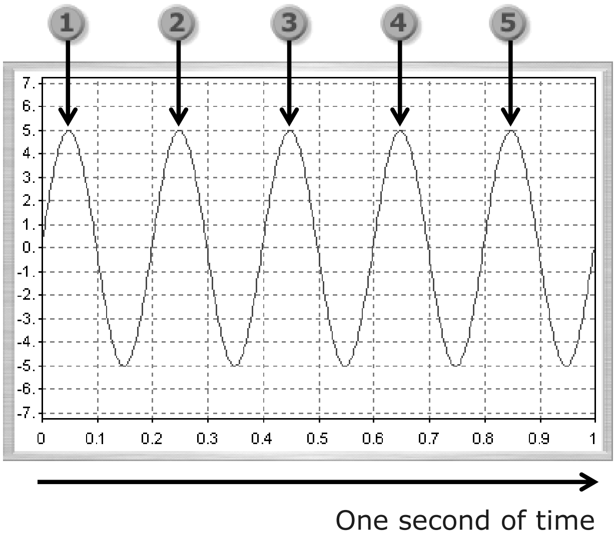


Figure 1-9

The fan rotates five times per second.
Therefore there are 5 cycles per second.
The frequency must be 5 CPS or 5 Hz.
Fan speed = 5 Hz or 300 RPM

CPM and RPM

CPM and RPM are often used interchangeably. CPM is the more general term because it can be used to describe things that are not "rotating" for example my heart might beat at 100 beats per minute and I could say my heart rate is then 100 CPM. It would not sound correct to say that my heart is 100 RPM (rotations per minute) because it is not rotating!

You may also occasionally hear "CPS" or cycles per second in place of Hertz (Hz) but Hertz is the more common term. Both terms are identical and acceptable however.

Introducing the "period"

Another important term is "period." The period is the amount of time required to complete one cycle. In the example of the mass on the spring, the period would be equal to the amount of time it takes the mass to move all the way from the bottom to the top and back to the bottom again (one complete cycle). The period can be measured in the waveform or it can be calculated from the frequency and vice versa. The formulas below show the relationship between the period and the frequency.

Period (seconds) = 1 / Frequency (Hz)

The Period is measured in units of time: **seconds** or **milliseconds.**

1 millisecond = 1 thousandth of a second.

1 ms =0.001 seconds

Calculating the Period from the Frequency : Going back to the recent example of the fan turning at 5 Hz or 5 cycles per second, we can ask: How long does it take to complete 1 cycle – or – what is its "period" in seconds? The formula is 1 / F Hz = Period (in seconds). Thus 1 /5 Hz = 0.2 seconds. So the **period** is 0.2 seconds.

Calculating the Frequency from the Period: Referring to the formulas again, the F Hz = 1/Period (in seconds). In this example the Period will be measured from the waveform.

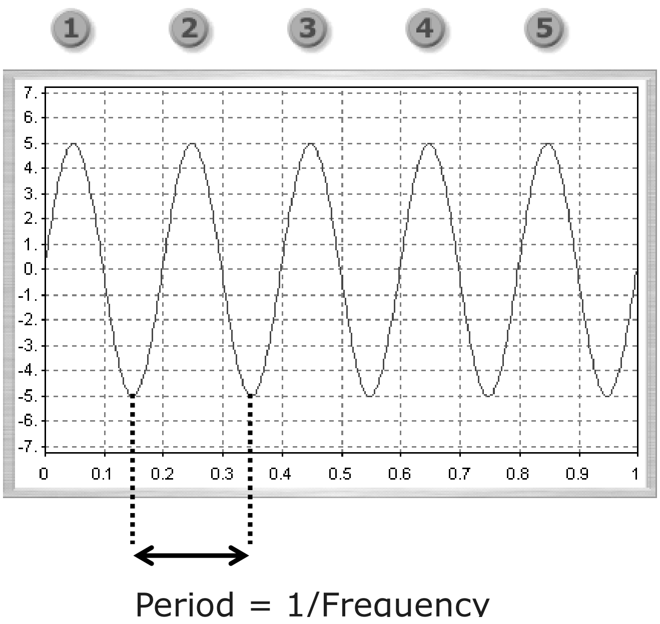


Figure 1-10 The period for one cycle is .2 seconds

The waveform in Figure 1-10 is one second long. Measuring the time for one cycle (using the bottom of the troughs as a guide,) the cycle goes from 0.15 seconds to 0.35 seconds for a duration or **period** of 0.2 seconds.

The value for the frequency can be calculated since we know the Period is 0.2 seconds. The Frequency (Hz) is 1/Period (seconds) or in this case 1 / 0.2 =5 Hz. (5 cycles per second)

The period is an important concept, as are the formulas from converting between period and frequency, because sometimes we know how long it takes for an event to occur (period) and we want to know the rate at which it occurs (its frequency) other times we will know the frequency of some event and we will want to know how long it takes for the even to occur once. Therefore we will often convert between period and frequency.

Increase the frequency

What would the waveform look like if the fan speed were doubled? There are twice as many cycles during a second due to the doubled fan speed. There are 10 pulses in one second, or 10 complete cycles in the second.

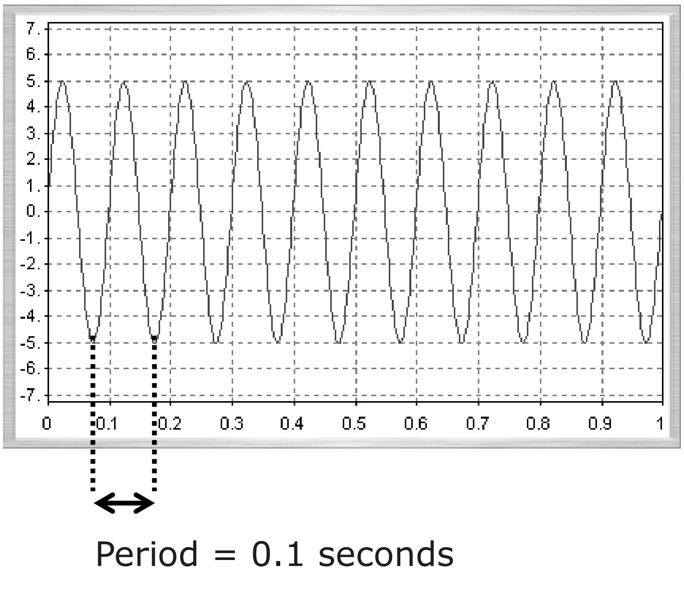


Figure 1-11 Doubling the fan speed doubles the number of cycles in one second.

The frequency now is 10 Hz (10 cycles per second) or 600 CPM (600 cycles per minute.) The fan speed is 600 RPM.

The Period is now 0.1 seconds (1÷10Hz = 0.1)

So as fan speed increases, the frequency increases, but the period decreases. Here is another example:

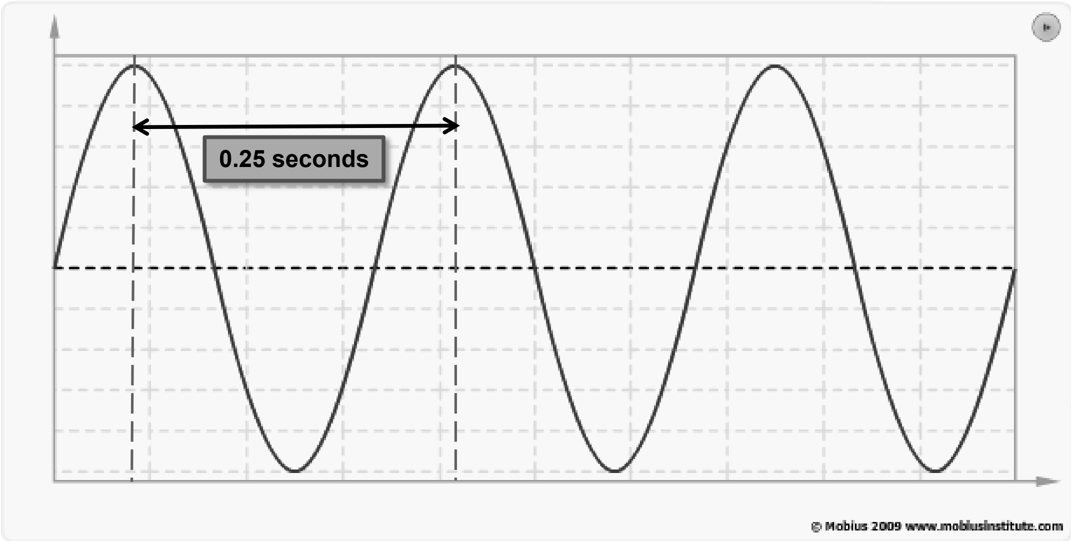


Figure 1-12 Period and frequency example.

The period is 0.25 seconds. The frequency in Hz is equal to 1 / Period in seconds or 1 / 0.25=4 Hz.

In this example, what is the frequency in CPM?

Hz x 60 = CPM
4 Hz x 60 = 240 CPM

Displacement, velocity and acceleration

When vibration amplitudes are discussed, the units must be quoted.

Amplitude can be measured in different ways, using different sensors. It can also be displayed in different ways, in other words we can switch units or convert from one unit to another.

If you consider the mass bouncing up and down on the spring; the simplest form of vibration, we can talk about its movement in three different ways.

Displacement: This describes the distance traveled by the mass, how far up and down it is moving. In the case of the vibrating shaft it is how far in and out it moves. For a child on a swing it is how high they are swinging…

Velocity: This describes how fast the mass is moving at any point, or how quickly it is covering distance. Velocity is the rate of change of displacement – how many miles one covers in an hour for example.

Acceleration: Acceleration is the rate of change of velocity. It describes how quickly the mass is speeding up or slowing down.

The various measurements could be compared to measurements made of a sports car running a slalom course, racing back and forth around the pylons. Imagine that in a large parking lot, two rows of pylons are lined up parallel to each other. A sports car starts at one row and races to the other making a U-turn around a pylon and races back to the first row. The car goes around a pylon there and races back to the 2nd row of pylons again. The car continues this pattern to the end of the rows of pylons.

Figure 1-13

There are at least 3 measurements that could be calculated on the sports car's slalom course. They are speed, the distance back and forth, and acceleration.

- **Speed:** This is describes how fast the car is moving at any instance. What is the position of the car when it is moving at its fastest speed? When it reaches the cones on either end it stops and switches directions, so its speed is zero. The maximum speed must therefore occur at the midpoint between the cones.
- **Distance back and forth:** This is not the total miles logged onto an odometer, but distance the car went back and forth from one side to the other. In other words the distance between the rows of pylons plus the turnaround space. It is how far the car travels before it has to stop and switch directions.
- **Acceleration:** When is the car experiencing the greatest acceleration? As it approaches a pylon it uses the brakes to slow down and stop, then it presses the gas to speed up in the opposite direction.

The path the sports car was traveling resembles the waveform trace from the unbalance condition of the fan. Vibration is measured in similar ways and similar units, but with different names.

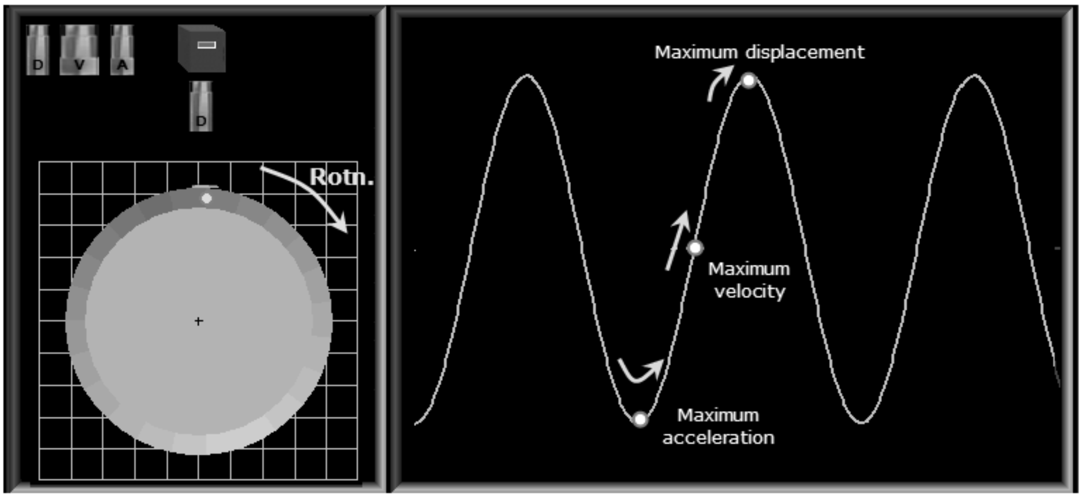


Figure 1-14 the path the car traveled resembles the vibration waveform of the fan's unbalance.

Speed is not measured in Miles or Kilometers per hour. It is measured in inches per second (in./sec., or IPS) or millimeters per second (mm/sec.) And it is not called speed but Velocity.

Distance is not measured in Miles or Kilometers, but mils or microns. And it is not called Distance, but Displacement.

The figure above describes the points of maximum displacement, velocity and acceleration. You may wish to consider the mass on the spring again; its total movement from top to bottom is its maximum displacement. When it reaches the top or the bottom, it is not moving, so its maximum velocity must occur when it is in the middle. At the top and bottom it needs to stop, change directions and speed up, so the end points are the location of maximum acceleration.

Looking at the figure more closely, what is the phase difference between two points of maximum displacement, velocity and acceleration? Notice they are 90 degrees apart from each other. This is an important point to remember. The units we choose will have an effect on our phase readings!

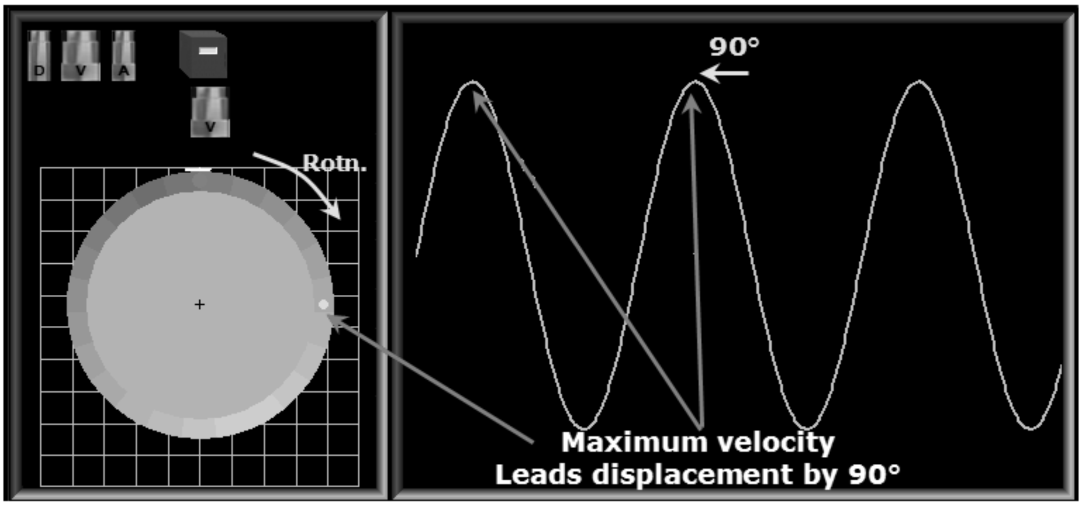


Figure 1-15 Acceleration leads velocity which leads displacement by 90 degrees

A simple way to remember the phase relationships between displacement, velocity and acceleration is this: Consider you are in an automobile. First you press the gas (acceleration) then you speed up (velocity) and then you move some distance (displacement). In vibration terms, we can say that acceleration ***Leads*** velocity by 90 degrees and velocity ***Leads*** displacement by 90 degrees. We could also say that displacement ***Lags*** velocity by 90 degrees and velocity ***Lags*** acceleration by 90 degrees.

Here is a practical example to explain why this is important. Let's say you collect vibration and absolute phase readings, using an accelerometer and a tachometer as a reference, on two motor bearings in the vertical direction. The units of vibration are in acceleration and the phase readings are 90 and 150, so we could say that the phase difference between the bearings is 150 – 90 = 60 degrees. Now, let's say the analyst converts the first measurement to units of velocity (we will see later that this is a common thing to do), this will change the phase reading at that bearing by 90 degrees. Velocity *lags* acceleration so the phase reading will now be 0 degrees at that bearing. If the analyst forgets to convert the second bearing to velocity and simply compares the phase readings at this point, he will think that the phase difference is 150 – 0 = 150 degrees! This can lead him to an incorrect diagnosis.

Vibration units

Now we will take a closer look at the units of vibration and how they compare to each other. Later we will see that we can view vibration data in either of the three units, acceleration, displacement and velocity by converting from one unit to another. We will also see that we can measure these quantities directly using different types of sensors.

The sensor we use and the way we display the data will depend on the application. It is important that we understand these concepts so that we can choose the correct sensor and display units for our application.

Amplitude units: Displacement

Displacement describes the movement of an object in terms of distance. In rotating machinery, proximity probes measure the distance between the sensor and the shaft.

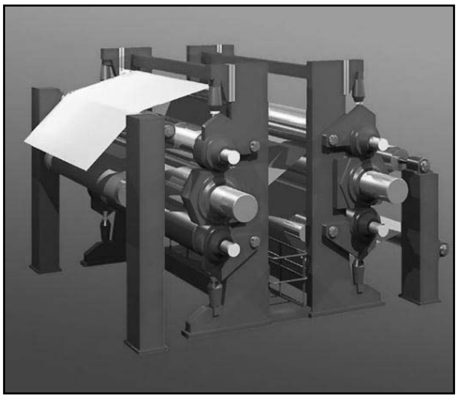


Figure 1-16

Typical units are:

- Metric: micron pk-pk
- Imperial: mils pk-pk

1 Micron = 1 μm = 1000th of a mm = 0.04 mils

1 Mil = 1000th of an inch = 25.4 microns

Notice that the waveform is measured peak-to-peak or Pk-Pk. In other words it measures the total distance the shaft travels.

Here are some important characteristics of displacement:

- At low frequencies displacement can be high (acceleration will be low).
- At high frequencies displacement will be low (acceleration will be high).
- Because displacement is more sensitive at lower frequencies it is typically the unit of choice for low speed machines (below 600 RPM.) or when you are interested in measuring low frequencies.
- Displacement is typically measured with proximity probes. These sensors are typically installed in machines with journal or sleeve bearings such as turbines and turbo compressors.
- Units of displacement are commonly used when performing balancing.

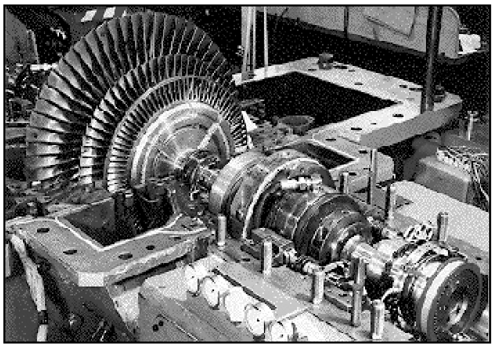


Figure 1-17

Displacement is proportional to stress. Strain is the fractional deformation produced in a body when it is subjected to a set of deforming forces. Strain produced in a body is directly proportional to stress.

To understand this concept, consider a piece of metal like a paper clip being stretched or pulled apart. As you stretch it, it will begin to get longer, and thinner and weaker until it breaks. This is how a material fails when it is subject to stress and strain.

The grid in Figure 1-18 shows the relative location of the shaft to the sensor above. Note that this is a Displacement waveform.

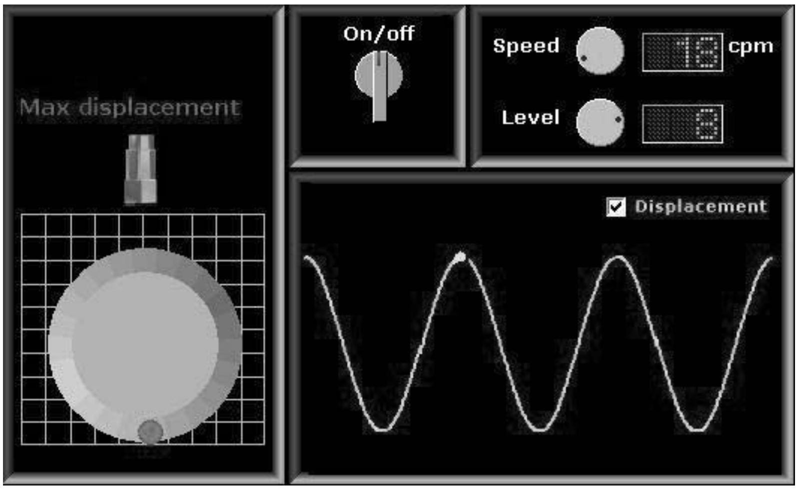


Figure 1-18 The maximum displacement in the positive direction is when the shaft is furthest from the sensor... Greatest Displacement

The highest amplitude occurs when the shaft is the greatest distance from the sensor. Then it is at the maximum displacement. In this case, since the sensor is on top, the maximum positive displacement occurs when the shaft is in the most downward position.

Amplitude units: Velocity

Velocity is a very common vibration unit. Technically, it is **the rate of change of displacement**.

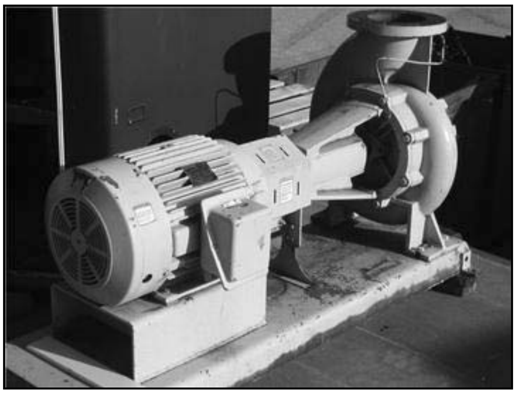


Figure 1-19 Velocity is a good unit of measure for most plant equipment.

Typical units are:

- Metric: mm/sec RMS
- Imperial: in/sec pk (or IPS pk)

1 mm = 0.04 inch

1 inch = 25.4 mm

In general velocity is a good measure of vibration across most machine speeds and frequencies of interest. For this reason, it is the most commonly used measure of vibration for general rotating machinery with rolling element bearings.

Velocity is best used to measure frequencies between 2 and 2,000 Hz (120 – 120,000 CPM)

Velocity is proportional to fatigue. Strain results from displacement and strain-cycles result in fatigue. To understand this concept, consider a paper clip. If you bend the paper clip slowly (low velocity) back and forth, it will not break, but if you bend it back and forth quickly (high velocity), it will break due to *fatigue*.

Note that Velocity is a measure of severity. When machines, machine components or structures such as piping or foundations are subject to high levels of vibration (measured in velocity), they are subject to fatigue and can be damaged by the vibration.

Earlier in this section it was pointed out that there are really three distinct waveforms, one for each of the amplitude units. Each one displays the maximum amplitude in its units as the top of the waveform. To demonstrate, the Displacement waveform will be used again.

The waveform in Figure 1-20 is the Displacement waveform from the fan with an unbalance condition. Recall that the Displacement is the distance measured from the bottom of the trough to the top of the curve or peak. Notice that the shape is the same shape as the path of the sports car on the slalom course. (Figure 1-13) The distance the car went back and forth was measured the same way.

The maximum speed of the car was achieved at the midpoint of the straightaway. The same is true for the vibration. From the bottom of the trough, the shaft of the machine increases speed toward the top or peak of the waveform. It reaches maximum speed at the midpoint and then slows the forward travel until it stops and begins returning the opposite direction.

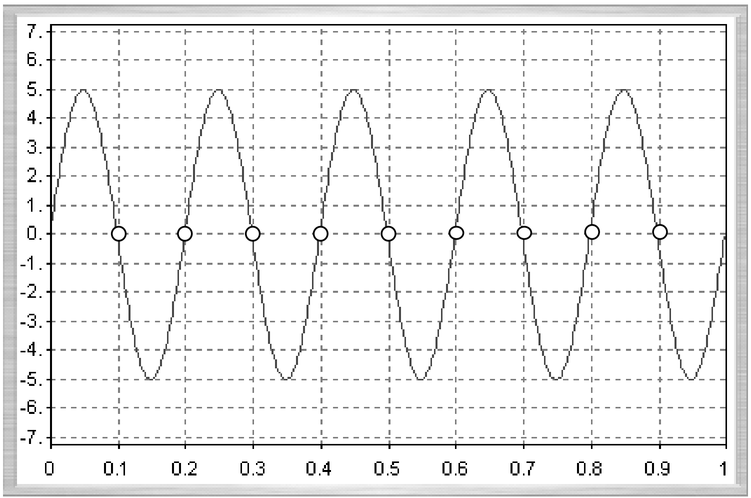


Figure 1-20 The displacement waveform. The maximum speed is at the midpoint.

The important concept to notice is that the point of the maximum speed in the positive direction is not directly in line vertically with the peak of the Displacement waveform, but to the left of it. If the graph were redrawn with this point at the top (normalized to velocity), the new peak would be offset from the peak representing the Displacement of vibration. The new Velocity peak will be to the left of the displacement peak. See Figure 1-21. Because the x axis of the graph is Time and the earliest time is on left, the velocity waveform is earlier than the displacement waveform. **It is said that velocity leads displacement**. Velocity reaches its maximum amplitude before displacement reaches its maximum amplitude.

Figure 1-21 shows the shaft's relative position to the sensor as being half way between the minimum and maximum distance (displacement) from the sensor. Although the shaft is rotating, the vibration sensor sees it as moving closer or further away. It is at this mid-point that the shaft is moving the fastest in its travel away from the sensor. It is the point of greatest speed (velocity).

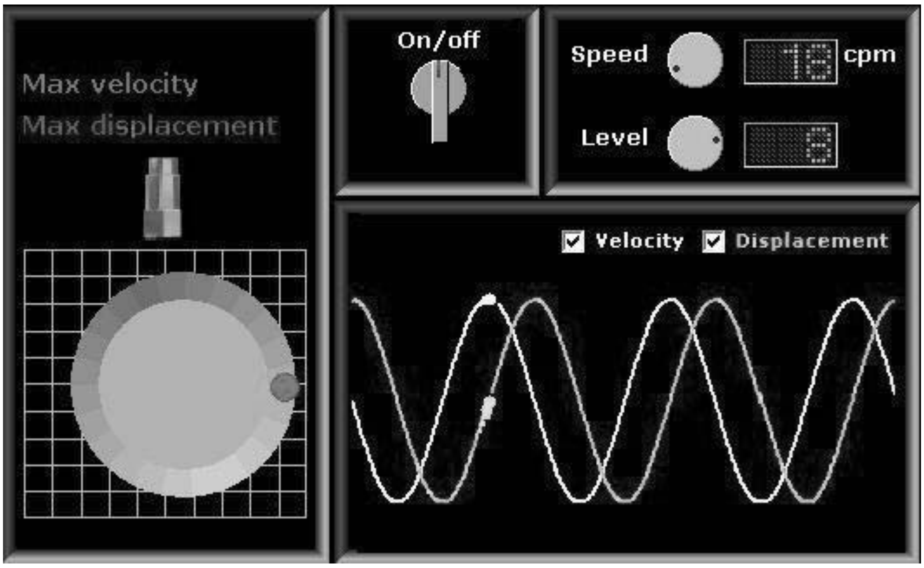


Figure 1-21 Maximum velocity occurs when the shaft is midway between the minimum and maximum displacement. The velocity waveform shows the point at the top of the waveform. The velocity waveform reaches its maximum before the Displacement waveform reaches its max

Amplitude units: Acceleration

Acceleration is becoming more popular as a preferred measurement of vibration due to the greater dynamic range available in modern data collectors.

Acceleration is the **rate of change of velocity.** It is how fast something is speeding up or slowing down.

Typical units for Acceleration are:

- Metric: g's or mm/sec^2 RMS or m/s^2 RMS
- Imperial: g's RMS, in/s^2 and AdB

Acceleration is the rate of change of velocity. Some characteristics of acceleration:

- Acceleration is most sensitive at high frequencies
- At low speeds there is little acceleration.
- Acceleration units are typically used on high-speed machines greater than 10,000 RPM. It is also used on high Frequency analysis such as bearing and gearbox analysis.

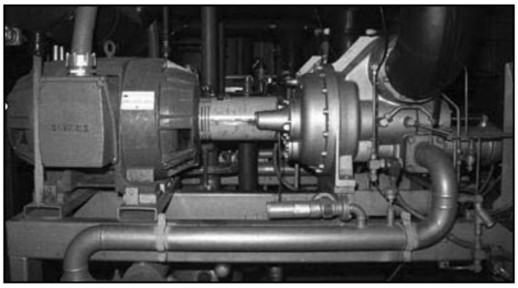


Figure 1-22 Acceleration is useful for high speed machines. It is proportional to the forces within.

Acceleration is proportional to the ***forces*** within a machine.

Returning to the example of the mass on the spring, we can consider that acceleration reaches its maximum when the mass approaches either the top or the bottom, because at this point it must decelerate and stop, change directions, the accelerate and speed up. When the mass is at the midpoint between top and bottom, its original point of rest, there is no force placed on the mass by the spring and therefore its acceleration is also zero at that point.

We can also return to the example of the sports car on the slalom course again (Figure 1-13). Recall that the greatest force exerted on the car was as it was coming around the pylons. As the car rounded the pylon at the top of the pattern, it reached its greatest speed halfway between the rows of pylons then it began to decelerate as it approached the pylon at the bottom. Its forward travel stops abruptly and accelerates in the opposite (positive) direction increasing speed toward the point of greatest speed (maximum velocity) and greatest distance (maximum displacement). Imagine the force the driver feels as he negotiates the pylon. As he comes out of the turn his body is pushed hard against the seat and he can feel his face being pulled back. There is a lot of force on the car as the tires begin to roll under in that turn.

Notice that this point of travel is at the apex of the curve (the negative peak) prior to the maximum speed and maximum distance (displacement). It is opposite the peak of the Maximum Distance and occurs before the maximum speed and maximum distance. **Acceleration is therefore said to lead velocity** by 90 degrees**.**

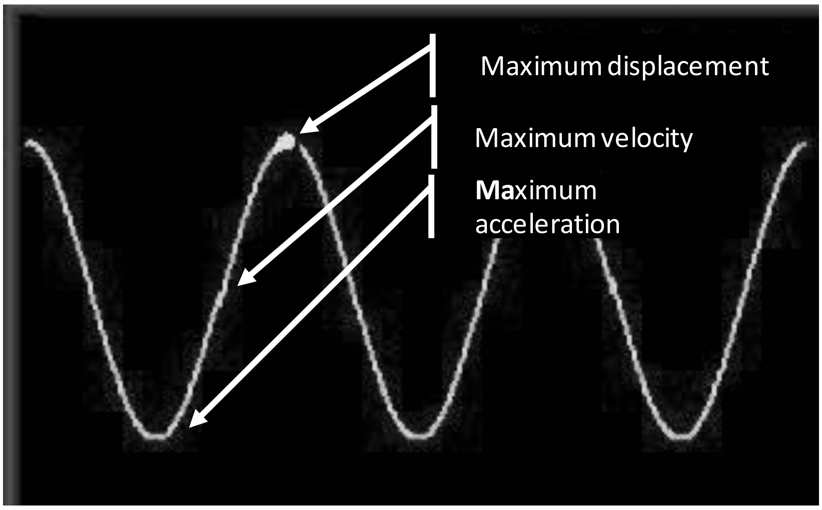


Figure 1-23 Acceleration leads Velocity which leads Displacement

Vibration is similar. The shaft moves toward the sensor, accelerating until it reaches the midpoint. It begins to slow until it "slams" against the sensor and is yanked in the opposite direction. The sensor measures the maximum force when the shaft bumps the sensor. See Figure 1-24.

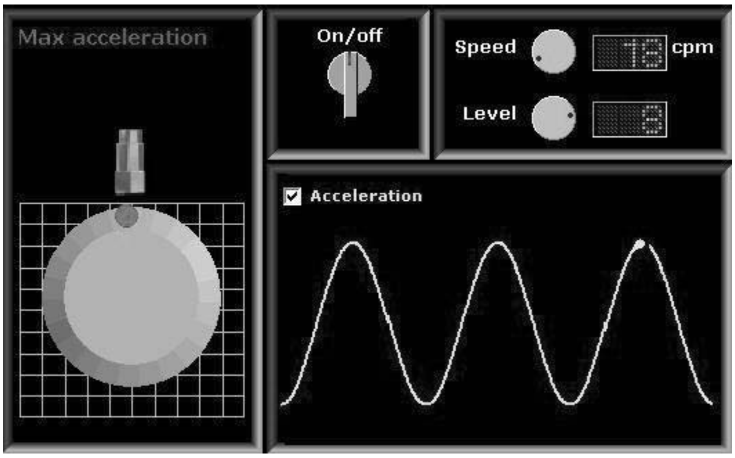


Figure 1-24 Acceleration waveform - the maximum acceleration is when the shaft is closest to the sensor. (it bumps against the sensor)

Comparing Units

As noted in prior sections, displacement is most sensitive to lower frequencies, acceleration is most sensitive to high frequencies and velocity is sensitive to most frequencies but not so great with very high or very low frequencies. To illustrate the importance of these characteristics; remembering that we can convert from one unit to another in our software. Take a look at the three graphs in Figure 1-25. These graphs are the same data displayed in the three different amplitude units.

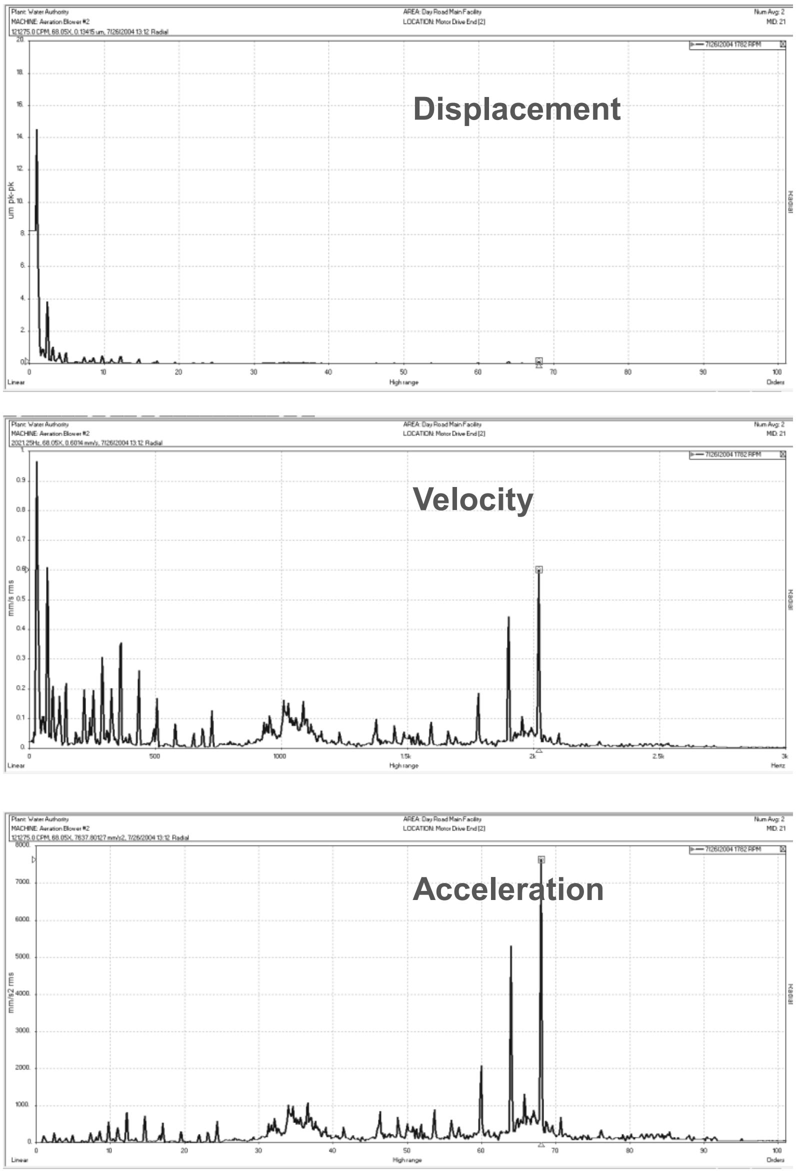


Figure 1-25 The same data in Displacement, Velocity, and Acceleration

Displacement: Notice in the top graph in Figure 1-25 that the only information in the graph is located up against the left side of the graph. These are low frequencies. As you move to the right (higher in frequency) there is no information at all.

Acceleration: Now look at the acceleration graph at the bottom. Here we see prominent information towards the right end of the graph (higher frequencies) but very little towards the left end of the graph.

Velocity: Compared to the other two graphs, the velocity graph clearly contains more information. We can see vibration peaks both at the right side of the graph and also at the left side of the graph. This is the main reason that velocity is the most popular unit to use.

Why is this important? Imagine that the prominent peaks visible towards the right end of the velocity and acceleration graphs relate to a particular mechanical fault in a piece of machinery such as a motor rotor bar problem. If we were to collect or view the data in units of displacement, we would not see these peaks and we would not be able to detect the problem. Although we may be dutifully collecting the readings, they do not contain the information that is important for finding this fault. Therefore, one must consider what frequencies are of interest and then choose the correct sensor and display units to detect these frequencies.

Figure 1-26 shows the relationship among the three amplitude units.

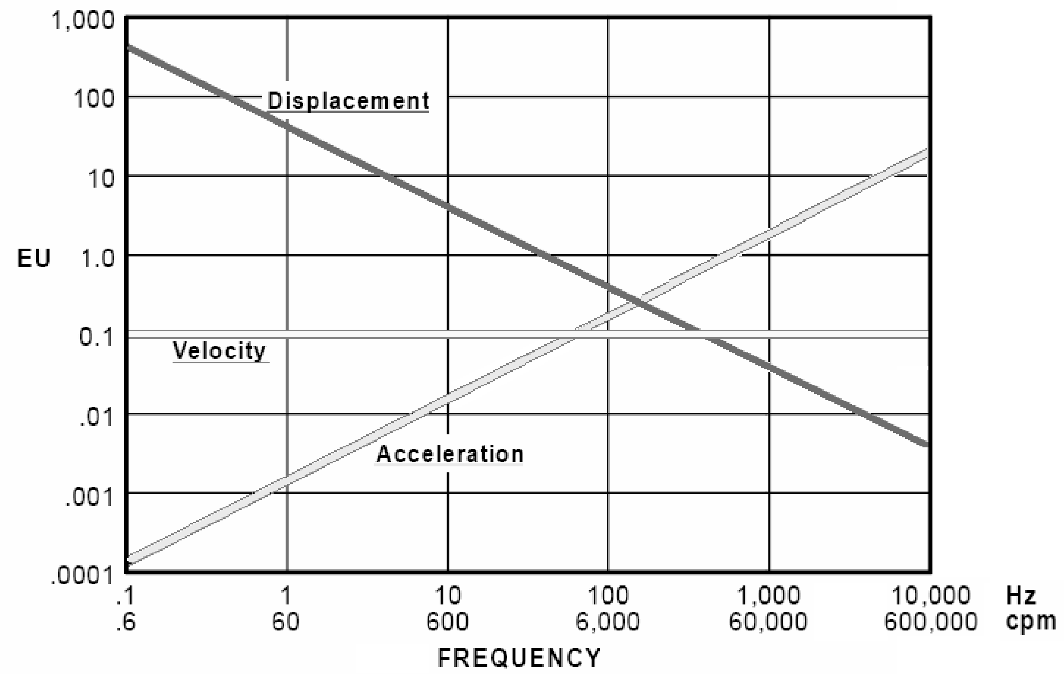


Figure 1-26 The relationship among Acceleration, Velocity, and Displacement

Which units do you use?

There are three issues related to vibration units. One is the type of sensor used, and the other is the application and the third is the frequencies we wish to measure. These are related to the speed of the machine, the type of bearing, and the failure modes of the machine.

Chapter Four discusses the different sensors and their applications. In brief:

- Proximity probes measure displacement
- Velocimeters measure velocity
- Accelerometers measure acceleration.

It should be noted that it is possible to convert between acceleration, velocity, and displacement.

Converting vibration units

Most analysts use accelerometers to measure vibration but then they convert the data to units of velocity before analyzing it. **The data collector converts from acceleration to velocity using an integrator.** Or we can say that data is "Integrated" from acceleration to velocity. Most software converts from one unit to another very easily.

Figure 1-27 The conversion software in iLearnVibration

To perform the conversion of units, the frequency must be known. Conversions are automated in software programs and data collectors.

Figure 1-27 is from the iLearn program. It converts a specific frequency and amplitude into all the units. It also displays the relationship to a Severity Chart to give some guidance into the condition. A similar units conversion program is available for use on the Mobius Institute website: www.mobiusinstitute.com.

Conversions: ISO 14694:2003(E)

Of course there are the math formulas, too. The conversions here are the ISO 14694:2003(E) standards. These are the generic ones. The following pages contain additional formulas.

These calculations are based on the following:

- Frequency is in Hertz
- D is in millimeters
- V is in mm/s
- A is in m/s^2
- G = 9.8m/s^2

$$V_{rms} = \frac{V_{peak}}{\sqrt{2}}$$

$$V_{peak} = \pi f D_{peak-to-peak}$$

$$A_{rms} = \frac{A_{peak}}{\sqrt{2}}$$

$$V_{peak} = \frac{1000 g A_{peak}}{2\pi f}$$

$$D_{rms} = \frac{D_{peak-to-peak}}{2x\sqrt{2}}$$

$$A_{peak} = \frac{2\pi f V_{peak}}{1000 g}$$

$$D_{peak-to-peak} = \frac{V_{peak}}{\pi f}$$

$$A_{peak} = \frac{2(\pi f)^2 D_{peak-to-peak}}{1000 g}$$

$$D_{peak-to-peak} = \frac{1000 g A_{peak}}{2(\pi f)^2}$$

Conversions: Imperial

- Frequency is in CPM
- D in $mils_{pk\text{-}pk}$
- V in in/s_{pk}
- A in g_{rms}
- 1 inch = 25.4 mm
- 1 mil = 25.4 micron
- 1 in/s = 25.4 mm/s
- 1 in/s pk = 17.96 mm/sec_{rms}
- 1 g = 9.8 m/s^2
- 1 Hz = 60 CPM

$$D_{pk-pk} = \frac{19098\, V_{pk}}{f_{cpm}}$$

$$V_{pk} = \frac{5217\, A_{rms}}{f_{cpm}}$$

$$D_{pk-pk} = \frac{9.958x10^7 A_{rms}}{f_{cpm}^2}$$

$$A_{rms} = \frac{f_{cpm} V_{pk}}{5217}$$

$$V_{pk} = \frac{f_{cpm} D_{pk-pk}}{19098}$$

$$A_{rms} = \frac{f_{cpm}^2 D_{pk}}{9.958x10^7}$$

Conversions: Metric

- Frequency is in CPM
- D in microns$_{pk\text{-}pk}$
- V in mm/s$_{rms}$
- A in g$_{rms}$
- 1 mm = 0.001 m = 0.04 inch
- 1 micron = 1 µm = 0.04 mil
- 1 mm/s = 0.04 in/s
- 1 g = 9.8 m/s^2
- 1 Hz = 60 CPM

$$D_{pk-pk} = \frac{27009\,V_{rms}}{f_{cpm}}$$

$$V_{rms} = \frac{93712\,A_{rms}}{f_{cpm}}$$

$$D_{pk-pk} = \frac{2.53x10^9 A_{rms}}{f_{cpm}^2}$$

$$A_{rms} = \frac{f_{cpm}V_{rms}}{93712}$$

$$V_{rms} = \frac{f_{cpm}D_{pk-pk}}{27009}$$

$$A_{rms} = \frac{f_{cpm}^2 D_{pk-pk}}{2.53x10^9}$$

Conversions: Metric

- Frequency is in Hz
- D in $micron_{pk-pk}$
- V in mm/s_{rms}
- A in g_{rms}
- 1 mm = 0.001 m = 0.04 inch
- 1 micron = 1 µm = 0.04 mil
- 1 mm/s = 0.04 in/s
- 1 g = 9.8 m/s^2
- 1 Hz = 60 CPM

$$D_{pk-pk} = \frac{450xV_{rms}}{f_{Hz}}$$

$$V_{rms} = \frac{1562\ A_{rms}}{f_{Hz}}$$

$$D_{pk-pk} = \frac{702778\ A_{rms}}{f_{Hz}^2}$$

$$A_{rms} = \frac{f_{Hz}V_{rms}}{1562}$$

$$V_{rms} = \frac{f_{Hz}D_{pk-pk}}{450}$$

$$A_{rms} = \frac{f_{Hz}^2 D_{pk-pk}}{702778}$$

Examples: Imperial

- Frequency = 1800 CPM
- Acceleration = 1 g_{rms}
- Velocity = 5217 x 1 / 1800 = 2.9 in/s_{pk}
- Displacement = 9.958x107 x 1 / 18002) = 30.7 $mils_{pk-pk}$
- Velocity = 2.9 x 25.4 x 0.707 = 52.1 mm/s_{rms}
- Displacement = 30.7 x 25.4 = 780 $micron_{pk-pk}$

$$V_{pk} = \frac{5217\ A_{rms}}{f_{cpm}}$$

$$D_{pk-pk} = \frac{9.958x10^7 A_{rms}}{f_{cpm}^2}$$

$$V_{rms} = \frac{V_{peak}}{\sqrt{2}}$$

Examples: Metric

- Frequency = 1500 CPM = 25 Hz
- Acceleration = 1 g_{rms}
- Velocity = 1562 x 1 / 25 = 62.5 mm/s_{rms}
- Displacement = 702,778 x 1 / 252) = 1124.5 $microns_{pk\text{-}pk}$
- Velocity = 62.5 x 0.04 x 1.414 = 3.5 in/sec_{pk}
- Displacement = 1124.5 x 0.04 = 44 $mils_{pk\text{-}pk}$

$$V_{rms} = \frac{1562\ A_{rms}}{f_{Hz}}$$

$$D_{pk-pk} = \frac{702778\ A_{rms}}{f_{Hz}^2}$$

$$V_{rms} = \frac{V_{peak}}{\sqrt{2}}$$

Overall level readings

Wouldn't it be nice if we could describe vibration with a single number and then trend that number or compare it to acceptance criteria? We could just sum up all of the vibration energy in a frequency band and say "this is how much it is vibrating." It would certainly simplify things!

Simple vibration measurements

Vibration can in fact be summed up into a single number that can be trended and compared to alarm limits. This number is called an "Overall level" or an "Overall RMS" reading and there are a large number of inexpensive devices available on the market that can take these readings.

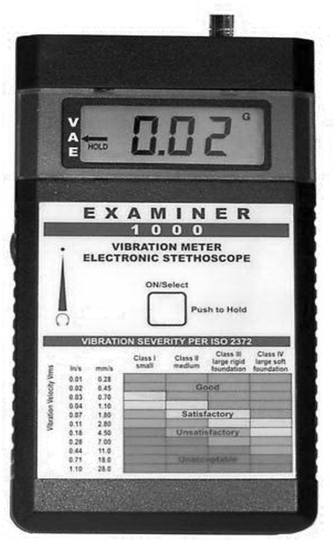


Figure 1-28

The vibration meter may give a single value of velocity (mm/s or in/sec), or may provide acceleration, velocity and displacement readings.

These reading provide an "overall level" reading using the RMS value. Over time it is expected that the level may trend upward if a fault develops.

It is common to capture and trend velocity readings and compare the readings to ISO alarm charts, the idea being that as mechanical faults develop, the vibration levels will trend upwards. If this sounds too good to be true, and if you are now wondering why you need to sit through the rest of this course if all that is required to detect mechanical faults in rotating machinery is to collect and trend a single vibration value, then you it should be noted that it is too good to be true! There are many limitations to this approach that must be considered.

Produced by SUMICO Technologies

Understand the history

Twenty or thirty years ago, before digital computers became ubiquitous, RMS overall readings were the only type of measurement most people could take in the field. There were high priced and complex instruments that could take more sophisticated readings but they were only used in critical situations where the cost of the instrument and the personnel to run it were justified. Even today as the prices of high end digital data collectors keep dropping, these simple instruments remain an order of magnitude less expensive.

Figure 1-29 Low priced overall vibration meters

Because these simple instruments were in wide use for a long time, numerous guidelines from ISO and others were produced to help people make sense of the readings. If you ask some people about vibration, this is how they understand it, as one value "how much is that machine vibrating?" "It's vibrating a lot!" But this gives us no indication of what the problem might be. It is like asking someone how they liked the symphony and them answering "It was loud." It tells you nothing about the quality of the sound, that the violin was a bit out of tune or the flute squeaked.

Consider the example of your automobile. You can tell when something is wrong by the sounds it makes, but it can make all sorts of sounds that mean different things and not all of them, even if they are loud, are problems. There are limits in just measuring the overall "loudness" or overall vibration energy and trying to make heads or tails of what it means in terms of the mechanical condition of the machine. Here are some more warnings.

Warning I: Limited frequency range

According to the ISO standard for RMS overall readings, they are only taken in the limited frequency range between 3 and 1,000 Hz. Later in this course we will learn about the relationship between vibration frequencies and mechanical components and mechanical faults in machines. For now you will simply need to accept the fact that many mechanical problems produce frequencies outside of this range. These include bearing vibration, gears, friction caused by a lack of lubrication, cavitation etc.

Warning II: Depending on a single value

As noted in the analogy of the symphony, a single value does not give us any indication as to the source of the vibration. In other words, we cannot differentiate between unbalance, misalignment, looseness, resonance, bearing wear etc. All of which are common machine faults. Even if we do get some indication that the condition of the machine is changing based on the change in the overall value, it still does not provide us with enough information to resolve the problem.

If we collect and trend overall values in all three units; displacement, velocity and acceleration (Figure 1-30) we can get a bit more of an idea of what frequencies are dominant in the vibration, remembering that the three units accentuate low, mid and high frequencies respectively. Although an improvement from the use of a single value, it is still a very crude method compared to the technology available to today.

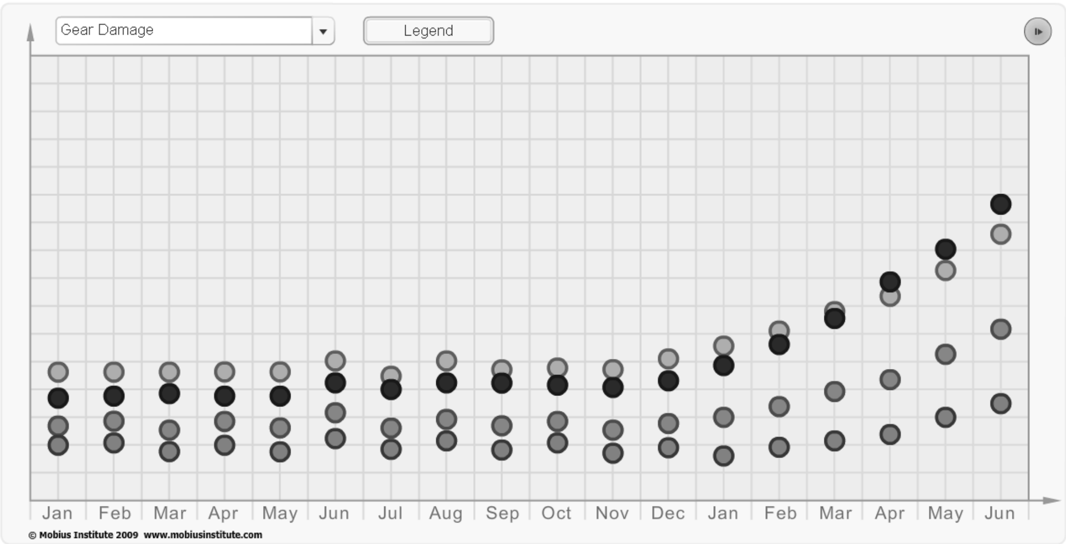


Figure 1-30 Trending overall values in displacement, velocity and acceleration

Warning III: An "overall" is not universal

If you remember, we noted that the RMS levels are NOT equal to 0.707 times the peak value for most signals and we mentioned that there are more complicated means of calculating the RMS values for these signals. Unfortunately, different calculations and techniques are used in different instruments so it may not be valid to compare the values collected with different instruments. They can however still be trended.

We also mentioned that the ISO overall value is taken within a specific frequency range of 3 – 1,000 Hz. Some vibration meters offer other frequency ranges, so you need to be careful that you know what you are collecting. Different meters will also use different filter types to remove the unwanted frequencies that can also result in variations in readings between different meters.

If you intend to use the ISO 10816 chart, ensure you know how the velocity reading was derived. Next we will explain some of the ways to calculate RMS overall readings.

RMS: Analog method (True RMS)

Recall from the previous section that for a pure sine wave, the RMS value is 0.707 times the Peak value but that this is NOT true for any other waveform. Most vibration is NOT sinusoidal, so we will need to use other means to calculate the RMS level.

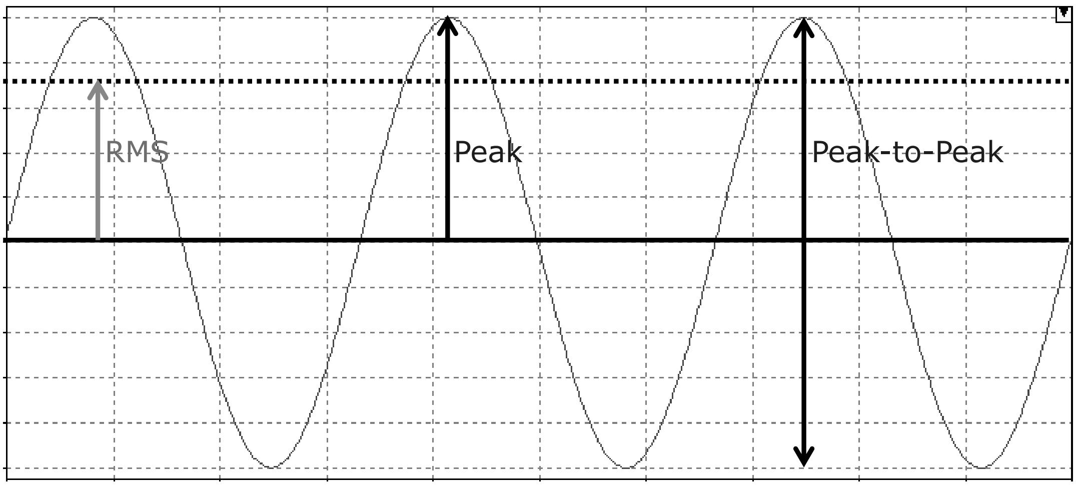


Figure 1-31

There are two ways the RMS is derived based on whether it computed digitally or is from an analog signal.

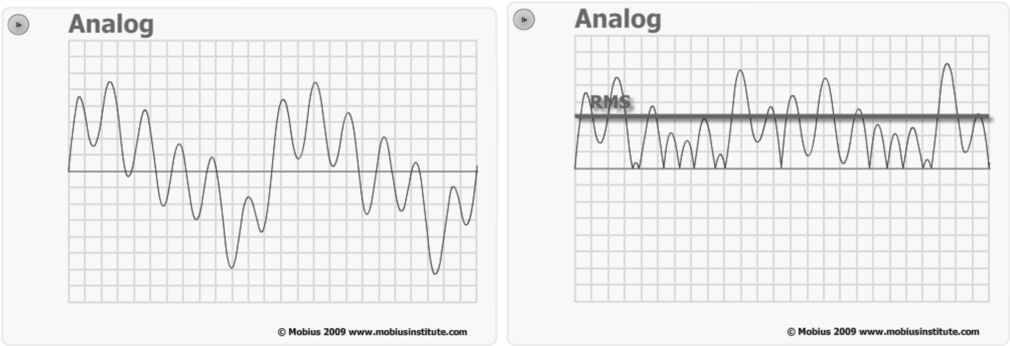


Figure 1-32 The True RMS calculated from an Analog signal. The squaring makes it positive.

$$\sqrt{\frac{1}{T}\int_0^T v(t)^2\,dt}$$

Recall that RMS is calculated as the reverse of its name. First the values are squared so that they are positive. Then the average value is calculated of the values under the curve. And finally the square root is computed of that averaged value. Figure 1-32 shows the process for the Analog signal.

RMS: Digital method

The process is not quite the same for a digital signal.

In this case, the analog signal is ***digitized*** or broken up into a number of discrete samples "n." Each sample "n" is squared. The sum of the squares is then divided by the number of samples. The square root of this value is computed resulting in the true RMS value as per the formula below.

$$\sqrt{N_1^{\,2} + N_2^{\,2} + N_3^{\,2} + \ldots}$$

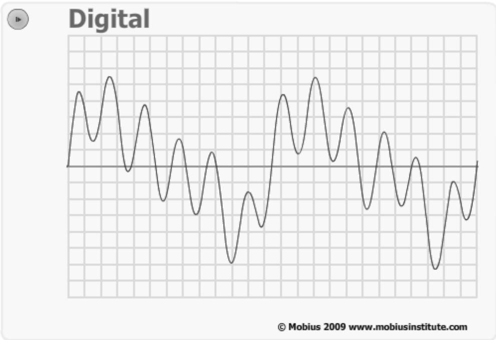


Figure 1-33 True RMS computed from a digital signal

RMS: From the spectrum

The RMS overall values can also be computed from the vibration spectrum. One benefit of this approach is that the RMS value can be calculated for any frequency range such that it captures the particular frequencies that are relevant for this application. The down side of that is that you have to remember that this overall RMS value is NOT the same as the ISO overall value, so you cannot just compare the value collected to an alarm chart. Additionally, if you want to trend these readings you will have to be sure to not change the settings on the data collector from test to test for this point.

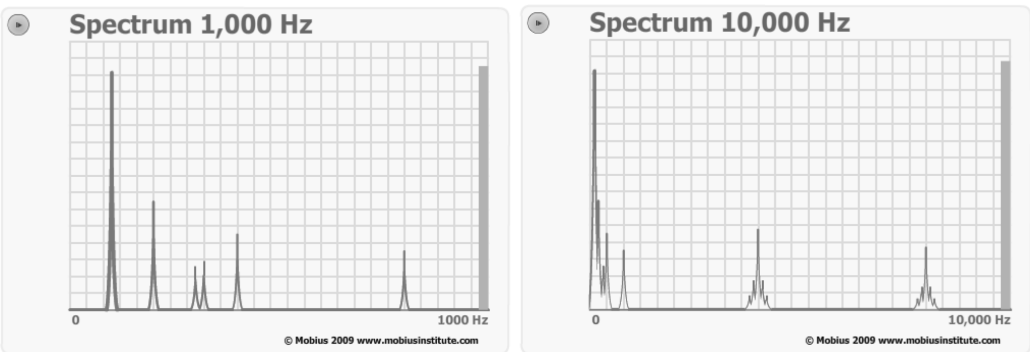


Figure 1-34 The RMS value can be calculated from the spectrum

Vibration Severity and ISO 10816.

The ISO 10816 defines vibration severity as the RMS level of vibration velocity measured over a frequency range **of 3 Hz to 1000 Hz**.

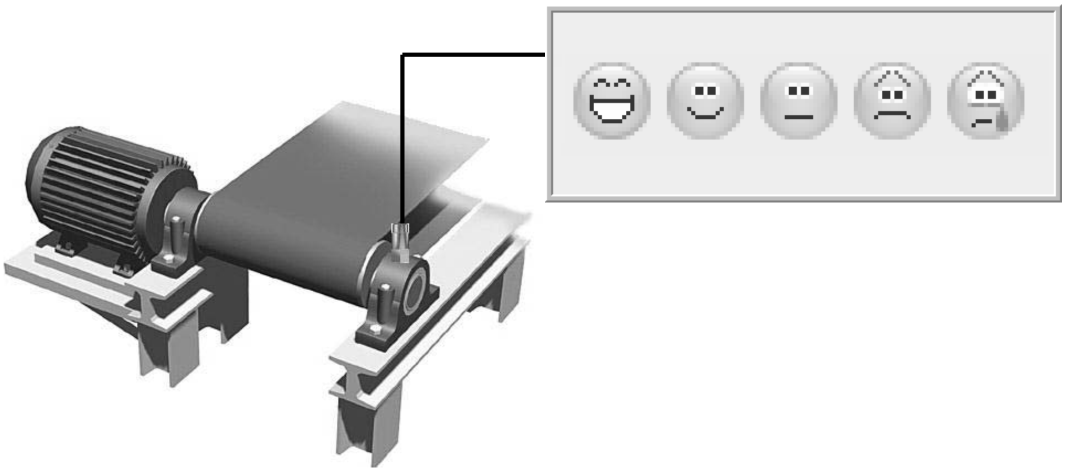


Figure 1-35

Instead of measuring the amplitude of a transient at a single high frequency, the vibration severity reading represents an average of all vibration components within a wide and comparatively low frequency range.

Vibration severity is directly related to the energy level of machine vibration, and thus a good indicator of the destructive forces acting on the machine.

The ISO alarms are discussed in another chapter.

Overall measurements and trending can provide good results and be a valuable time saving tool. The analysis and alarm limits sections provide additional options for alerting to problems.

Crest factor

The crest factor of a waveform is the ratio of the peak value to the RMS value. It is sometimes called the peak-to-RMS-ratio. The ratio between the peak and the RMS value gives us an idea of how much impacting there is in a waveform. A signal from an out-of-balance machine will be sinusoidal and thus it will be close to 1.4. However machines with a bearing fault will have a more "spiky" waveform so the crest factor will be much higher.

$$Crest\ factor = \frac{Peak}{RMS}$$

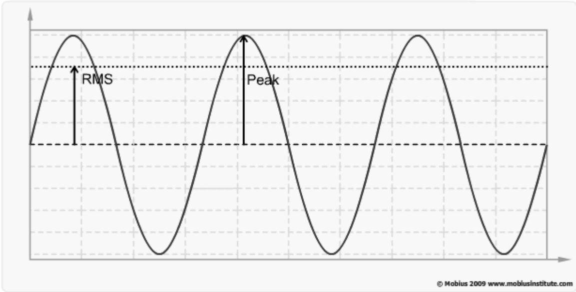


Figure 1-36 The crest factor is 1.41 for a pure sine wave

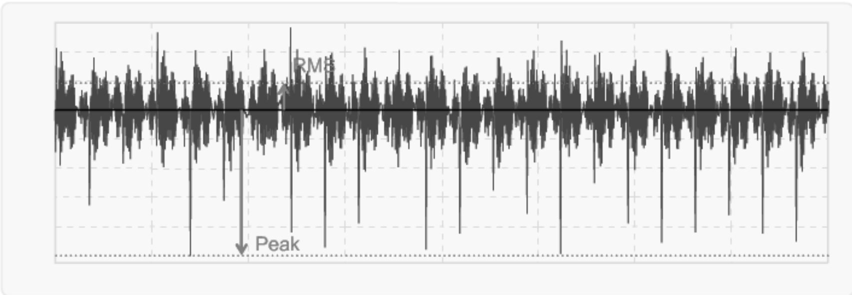


Figure 1-37 The crest factor for this waveform is high due to impacting

The crest factor adds some more information to a simple reading such as an RMS overall reading by giving us some indication of the amount of impacting in the waveform. This can help us determine the root cause of the vibration and the type of fault it may indicate.

Complex vibration

Thus far we have discussed simple vibration. We used an example of a mass on a spring bouncing up and down. We explained that the vibration had the characteristics of frequency (Hz, CPM or RPM) and amplitude (pk, pk-pk or RMS) and that the motion could be described in terms of displacement, velocity or acceleration. We also noted that we could relate one vibration signal to another in terms of phase.

Now we will ask the question: What happens if we have two masses on springs?

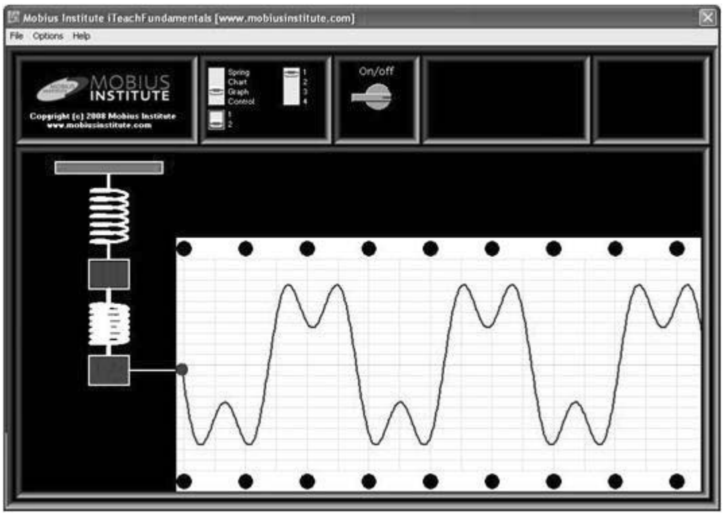


Figure 1-38 What happens when we have two masses and springs?

Remember that each mass and spring creates a single sine wave with an amplitude and frequency. If we look at Figure 1-38 we can see that the sum of the two simple waveforms is already quite complex. In fact, if we did not know this was created by two masses and two springs, we would really have no way of knowing what caused this waveform.

How this relates to a machine

In a machine, things get even more complicated. The machine has a rotor that is spinning, pulleys, gears, bearings, fan blades and many other sources of vibration. Each of these components are vibrating in their own way, at their own frequencies and amplitudes and all of this adds together to create a complex waveform.

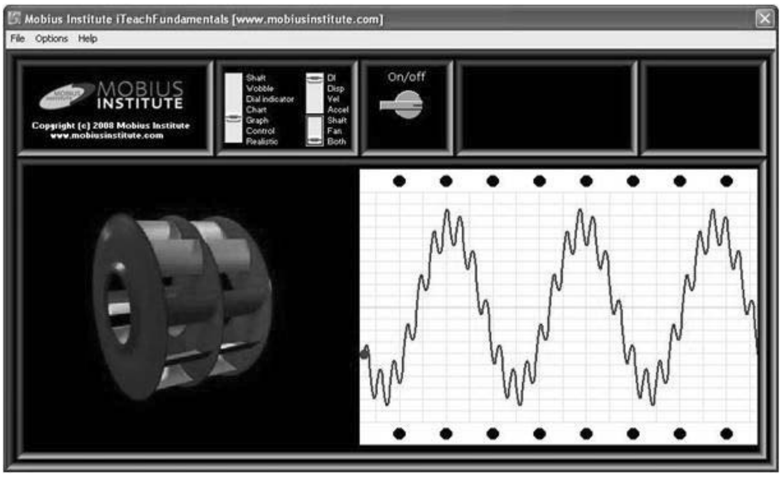


Figure 1-39 Two sources of vibration in a fan

Consider the vibration due to a fan

In Figure 1-40 we see a simple squirrel cage fan and the corresponding waveform. In this example we can consider two sources of vibration, one from the shaft rotating and the second from the fan blades. In order to visualize this, imagine the fan has a weight on it like we did in an earlier example, as the weight spins around and passes a sensor, we get a waveform related to the shaft rotational rate frequency. This is because the weight passes the sensor one time per revolution.

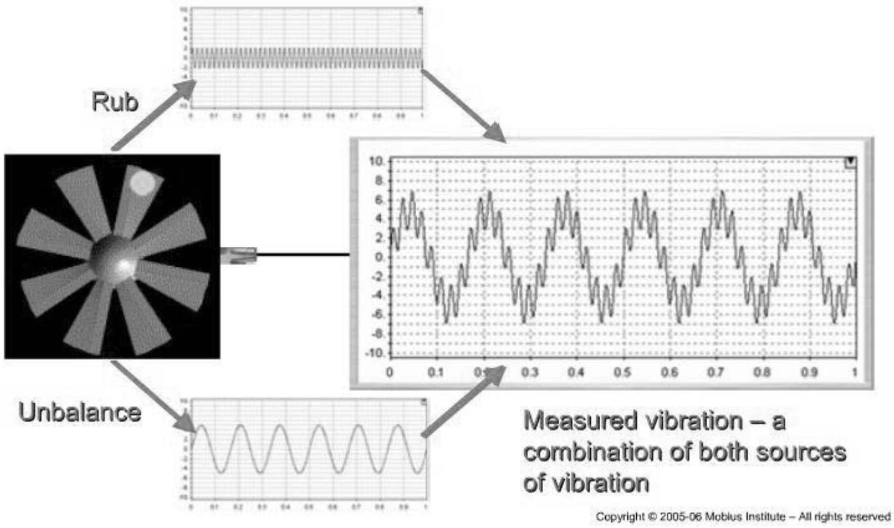


Figure 1-40 The individual sources of vibration combine to produce one complex waveform.

To visualize the vibration coming from the fan blades (Figure 1-40), imagine you hold a piece of paper so that the blades hit it as they pass. How many hits will the paper receive? In one rotation of the shaft, the paper will get hit once by each blade. Therefore the rate at which it gets hit is equal to the number of fan blades times the shaft rate. This is another source of vibration in the fan and it will have its own amplitude and frequency.

In Figure 1-40 we can see that when we add these two sources of vibration together, the result is a complex waveform. If we just had that waveform and did not know it came from the fan shaft and fan blades, we would have no idea of how to interpret it.

How to deal with complex vibration

As we have noted, when added together, even two simple waveforms can result in a very complex waveform. Yet, what we wish to do is somehow separate the vibration coming from the shaft from the vibration coming from the fan blades so that we can analyze them separately. Our point here is that we cannot do this by analyzing the time waveform.

This does not mean that the time waveform is not useful, in fact we will return to it and discuss ways to analyze it in both the category II and category III classes.

In the next section we will introduce a new way to look at vibration data and it will help us deal with the problem of waveforms adding up and getting complicated in the time domain.

Introducing the spectrum

The vibration spectrum allows us to separate components that overlap in the waveform and display them by frequency. As we noted at the end of the last section, even when we have only two simple sources of vibration added together, the waveform becomes quite complicated and it is impossible to separate it into its components – i.e. to know it was created by these two simple sources. The vibration spectrum allows us to do just that; it allows us to separate the vibration waveform into its components.

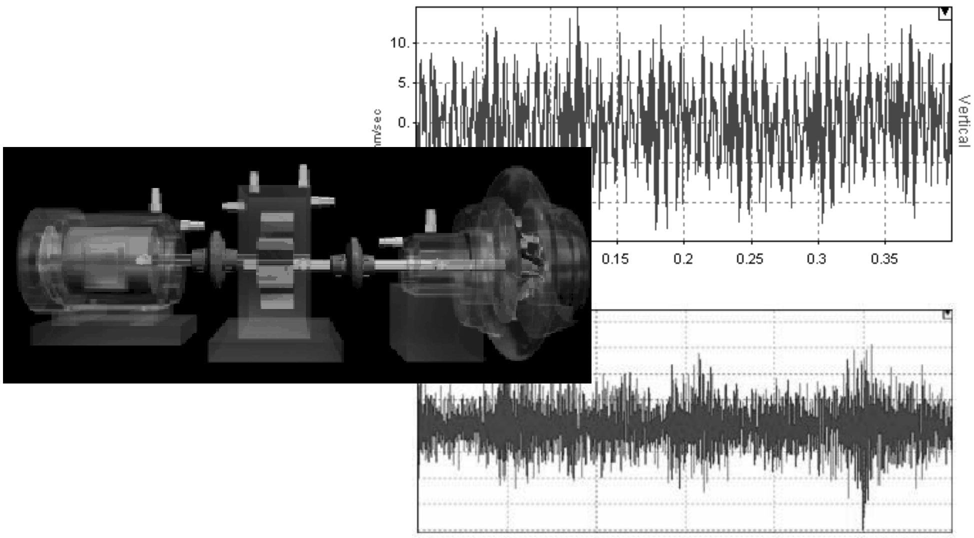


Figure 1-41 The waveform from a machine is complex

The spectrum is derived from the waveform through a process called the Fast Fourier Transform, or FFT.

Remember that a sine wave has a frequency and an amplitude. What we are doing with the spectrum is simply graphing this information in a different way, on a graph of frequency versus amplitude instead of a graph of time versus amplitude. The FFT separates the various waveforms from the complex waveform and displays them on a graph according to frequencies. See Figure 1-42

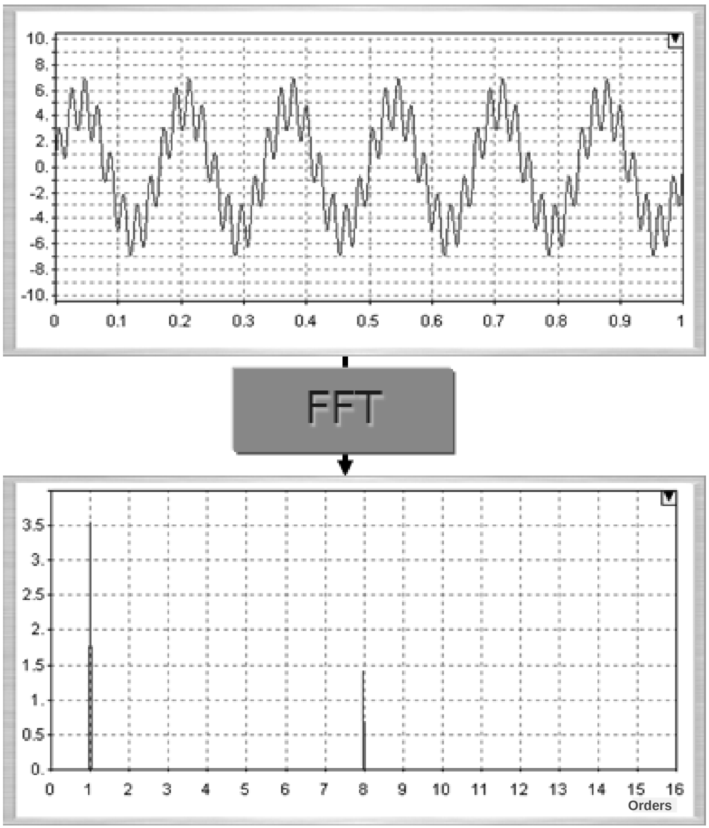


Figure 1-42 The FFT process separates the individual waveforms and displays them according to frequency.

To keep it simple, the fan example (with thecoin on a blade) will be used again. Assuming that it is rotating five times per second, it might produce a spectrum like the one shown at the top of Figure 1-43. If the speed of the fan is doubled, which also increases the vibration amplitude, (shown at the bottom of Figure 1-43) the height of the peak will increase, and the peak will move to the right – in fact, it has moved twice as far along the x-axis, since the frequency of the peak has doubled.

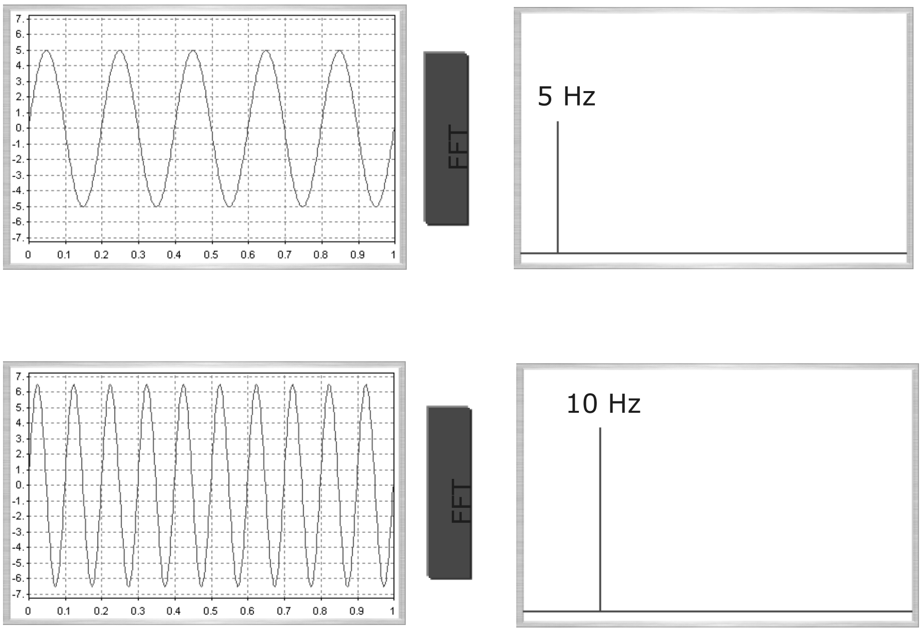


Figure 1-43 The top waveform of a 5 Hz cycle produces a spectrum with the 5 Hz peak. The bottom waveform is a result of doubling the speed to 10Hz. It produces a spectrum with a peak at 10 Hz. Notice the height of the peaks reflects the amplitude of the waveform.

This is just one frequency extracted from a simple sinusoidal waveform. Both waveforms generated a spectrum with one peak. The process works similarly when the waveform is more complex.

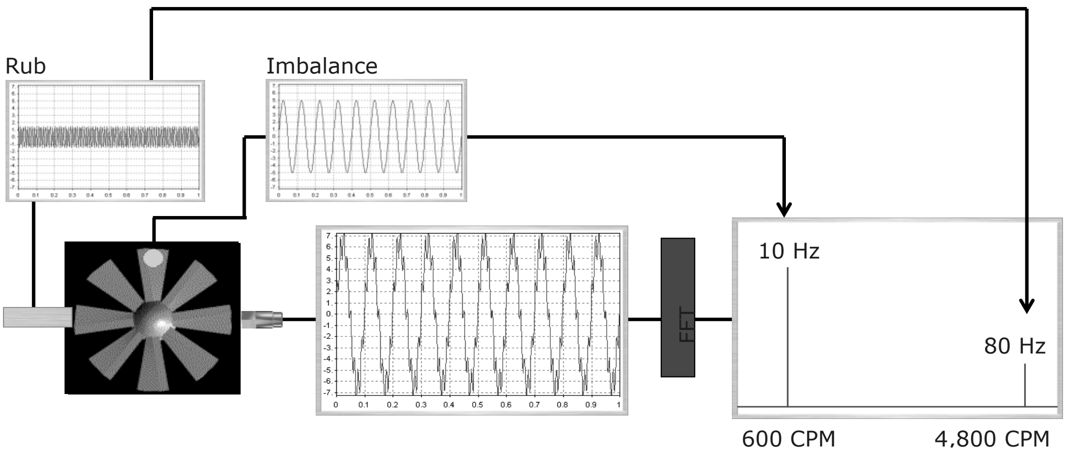


Figure 1-44 The FFT process separates the complex waveform into the individual waveforms and displays them in the spectrum.

Similar to the earlier fan example, a metal block is inserted to rub against the fan blades. This produces a second peak on the spectrum. Recall that the vibration due to the rub occurred at a higher frequency (there are 8 blades, so there are 8 pulsations for every rotation). If the fan rotates at 10 revolutions per second, or 600 revolutions per minute (RPM), the frequency of the peak due to the rub will be 8 times that value: 80 cps or 4800 CPM.

Building the spectrum

Figure 1-45 shows a machine with 3 sources of vibration, motor speed, a bearing, and fan blades. The waveform below the machine is the composite of the three. To the right of the machine is a box with the individual frequencies overlaid on each other.

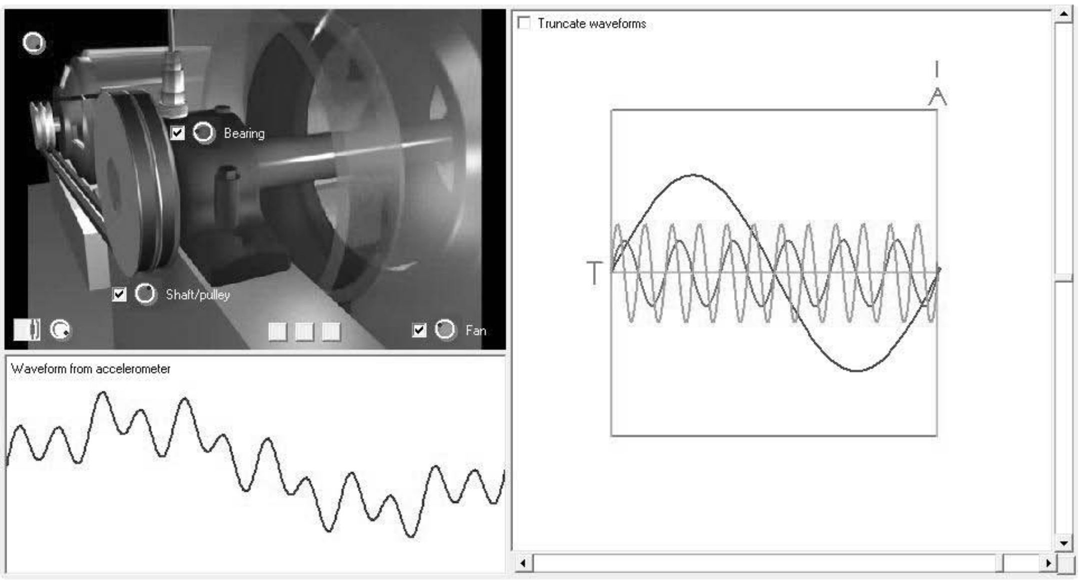


Figure 1-45 Three sources of vibration are combined in the composite waveform (left). They are individually overlaid in the box to the right of the machine.

The FFT process separates the individual sine waves from the composite waveform and displays them according to their frequencies.

The spectrum is as if we are looking at those separated waveforms from the end. Notice how in Figure 1-46 the individual waveforms are in a 3 dimensional box that is being rotated. Figure 1-47 shows the fully rotated box and the end view of each of the sine waves.

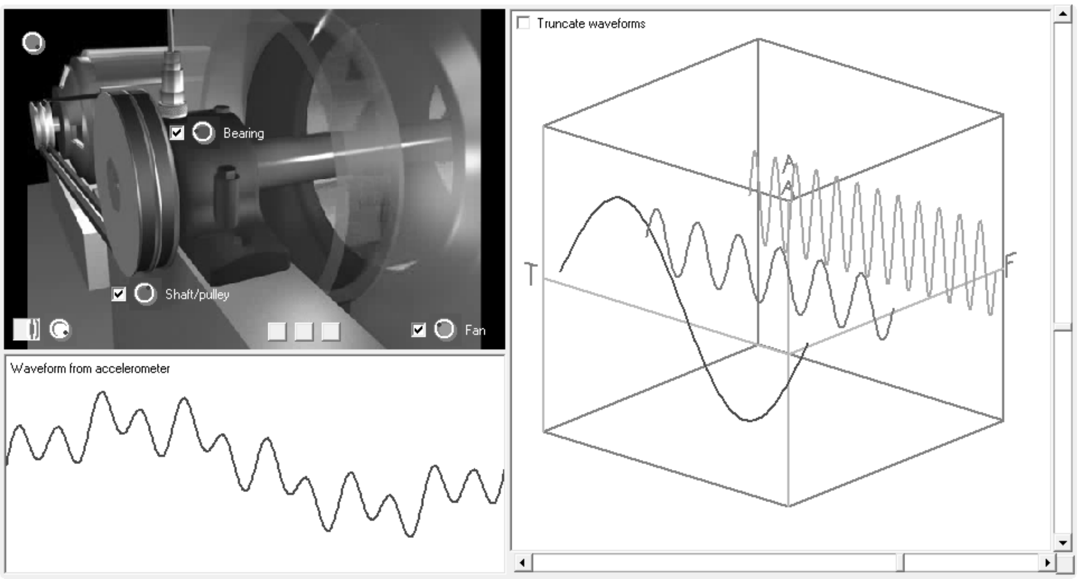


Figure 1-46 The individual waveforms are shown in a 3 dimensional box that is partially rotated.

Figure 1-47 shows the 3 dimensional box rotated 90 degrees. The X axis of this view is Frequency. The individual lines or peaks shown are the end view of each of the waveforms. They are spaced apart according to their individual frequencies.

The tall peak on the left is the running speed of the pulley. The next peak is from the fan bearing. The peak on the right is from the fan blades.

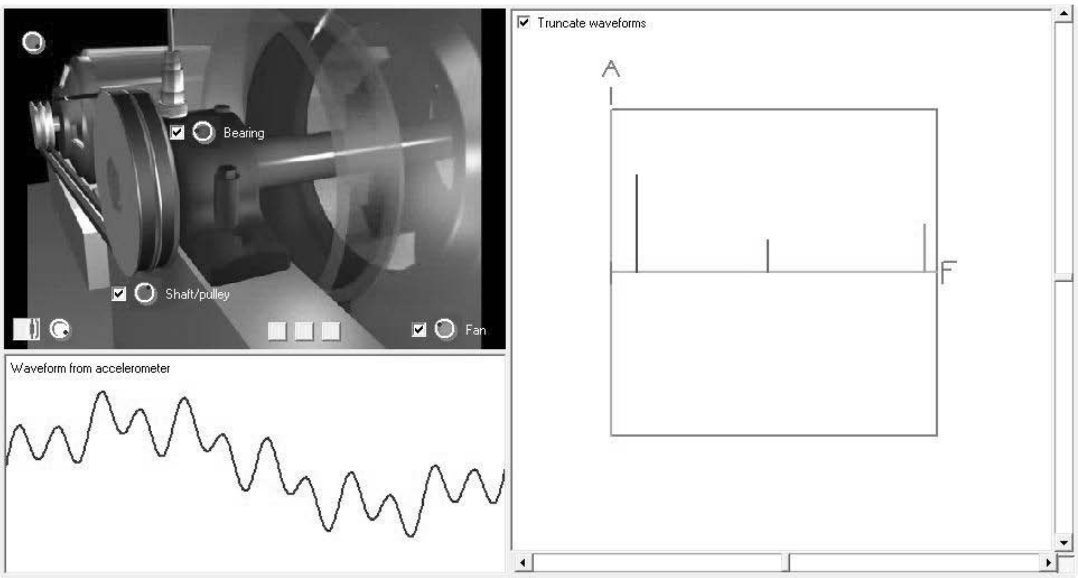


Figure 1-47 The 3 dimensional box is shown rotated 90 degrees revealing the end-view of the waveforms. They have been truncated so that nothing is shown below the zero line. The X axis is Frequency.

Peaks relate to parts of the machine

This new view of the vibration called the spectrum is the key to seeing the condition of machinery. The frequency tells the **source** of the vibration and the amplitude tells us about the **severity** of the vibration.

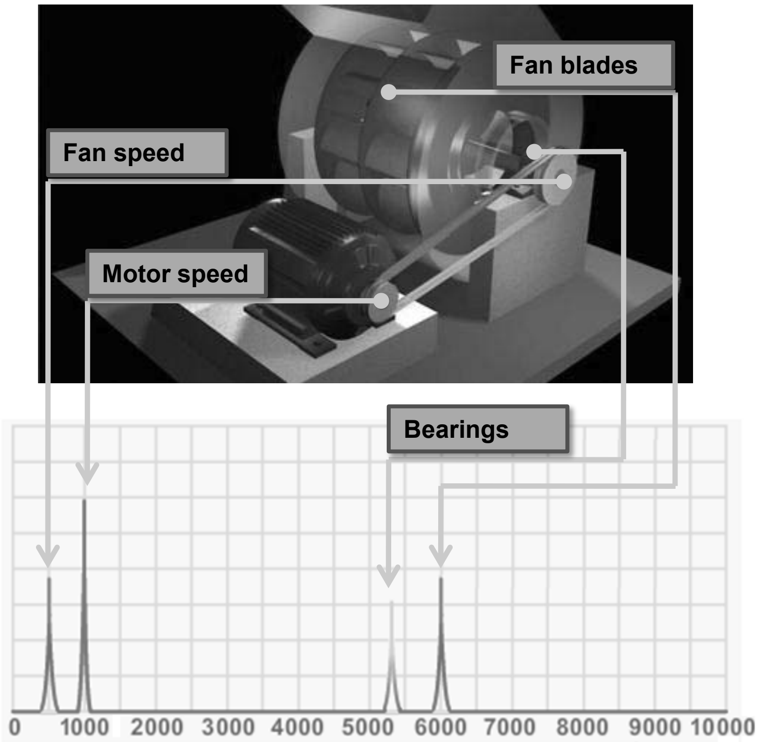


Figure 1-48 Relating vibration frequencies to machine components

The concept described in Figure 1-48 is incredibly important in understanding vibration in machines. What we have demonstrated here is that we can relate a particular machine component to a particular frequency in the vibration spectrum. This is how we will know that we have a bearing problem as opposed to an unbalance or misalignment problem. We will talk more about this in a moment.

The X axis of the graph in Figure 1-48 is in "frequency." Frequency can be expressed in 3 different units. We have already discussed two of them, cycles per second or hertz (Hz), and cycles or rotations per minute (CPM or RPM). The third unit is Orders.

Understanding Orders

Most of the discussions in this chapter that have been regarding the source of some vibration have explained the frequency in terms of running speed or its multiples rather than CPM, Hz, or cps.

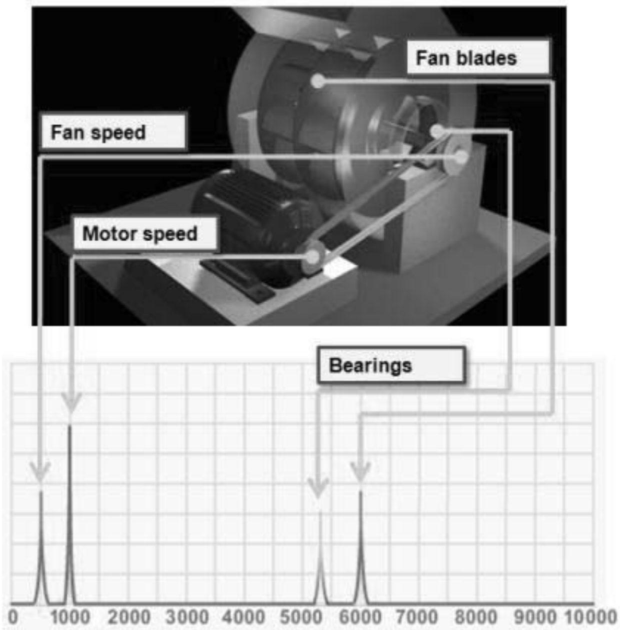


Figure 1-49 2 Frequency units are CPM and Hz.

Going back to the example of the fan from the last section; the coin on the fan blade produced a vibration once every time the shaft turned. It didn't matter how fast the shaft turned. The vibration occurred at the same frequency as the shaft. Therefore instead of saying that the coin went around at 100 CPM or 50 Hz we could just say that the coin went around "Once per revolution" or "At the shaft rate frequency." These statements would hold true no matter what the turning rate of the shaft.

Similarly, the "rub" on the fan was at 8 times the running speed. 8 fan blades hit the paper in one revolution of the shaft or the paper was getting hit at a rate of 8 x the shaft rate. It doesn't matter if the running speed was 5 Hz, 6Hz or 10 Hz. The rub still occurred at 8 times the shaft rate.

In order to simply things, we could simply say that the shaft turns at "1x" which means "1 x the shaft rate" and the fan blades pass at "8x" or "8 x the shaft rate."

Notice that what we have done is simply relate everything to the rotational rate of the shaft. If I can identify the shaft rate then I can immediately identify the fan blade rate (by multiplying it by 8 in this case). This greatly simplifies things!

It is very useful to refer to the occurrence of something in terms of multiples of running speed rather than absolute terms of Hz or CPM. It is good to know the specific frequency, but it is generally far more useful to know the frequency relative to turning speed.

If someone were to cite that the 1X peak was high, then most practicing analysts would know that they were talking about the peak at the running speed (because that frequency is 1 times the speed of the machine). In our example, we would refer to the 8X peak as the "blade pass" peak at eight times the running speed frequency.

Analysts say the frequency in different ways. The 1X peak may be called the "one times peak", or it may be called the "one X peak". It is the same.

The term for expressing frequency in multiples of running speed is **Orders.**

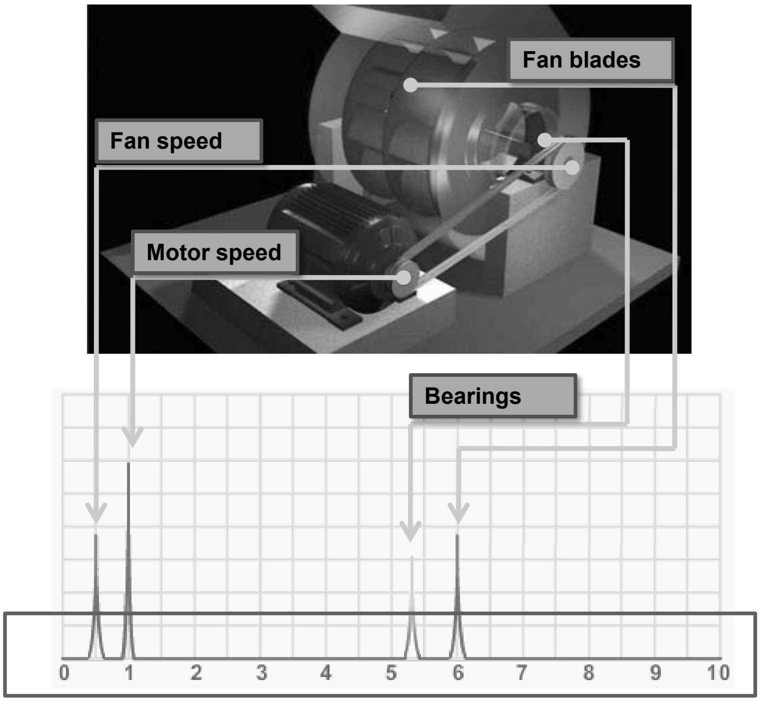


Figure 1-50 The motor fan vibration displayed in orders

It is common for analysts to speak of frequency in terms of orders. For example, "there is a peak at 12 orders." "There is a group of peaks between 15 to 20 orders." The term is used interchangeably with "12 times running speed," or "12 X," etc. This is much easier than multiplying 12 fan blades by 3578 CPM (the actual running speed)

In Figure 1-50 we can see the squirrel cage fan vibration displayed in orders where "1" on the graph corresponds to "1x" or 1 x the motor shaft rate. In relation to the motor shaft, the fan blades pass at a rate of "6x" or 6 x the motor shaft rate and the fan shaft rotates at 1/2x or ½ the motor shaft rate.

Another example illustrates the differences that occur when using Orders. It is using our fan example again at 2 different speeds.

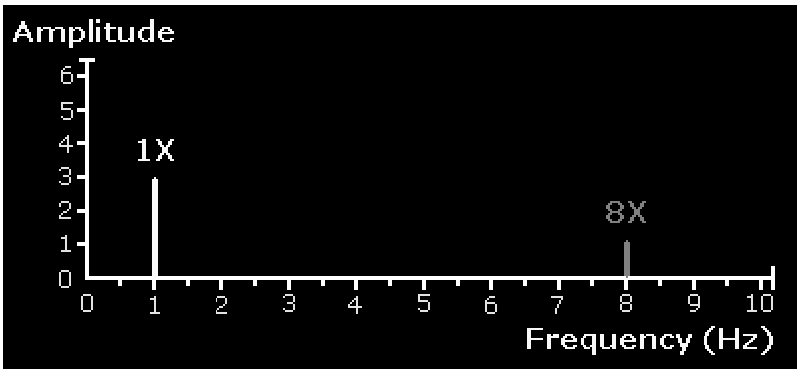


Figure 1-51 1X is at 1 Hz or 60 CPM. Blade pass of 8X is at 8 Hz or 480 CPM.

The first speed is 1Hz. So 1X is 1 Hz or 60 CPM. This one is very simple since the machine is turning at 1 Hz. Blade pass is at 8 Hz.

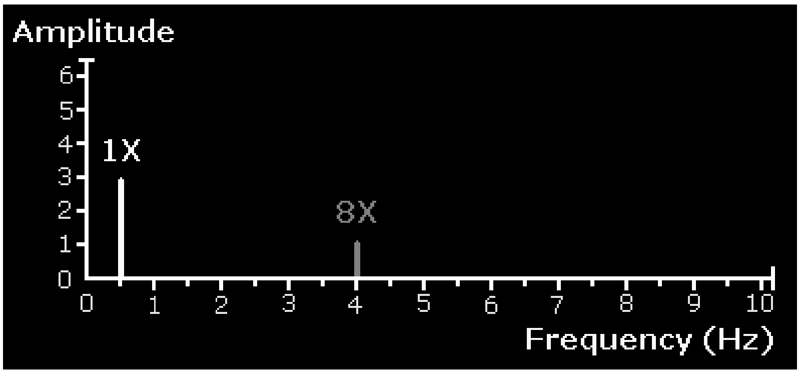


Figure 1-52 1X reduced to 0.5 Hz or 30 CPM. Blade pass of 8X is at 4 Hz or 240 CPM.

Now, reduce the running speed to 0.5 Hz. So 1X is 0.5 Hz or 30 CPM.

Blade pass of 8X is at 4 Hz or 240 CPM.

Whether the fan is turning at 0.5 Hz or 20 Hz, the frequency component due to the blades rubbing the block will always be 8 times that frequency.

Rather than displaying the spectrum graph with the horizontal axis (x-axis) in units of Hz or CPM, it can be changed to Orders. All the peaks can then be easily stated in terms of Orders.

Working in orders is also important because it allows us to overlay and compare graphs that are "normalized." If a machine changes speed from 1755 to 1735 CPM the peaks will not overlay in a graph. But if the graphs are "normalized", 1X will appear in the same place, 10X will appear in the same place, and so on. We will discuss this further in the Vibration Analysis section.

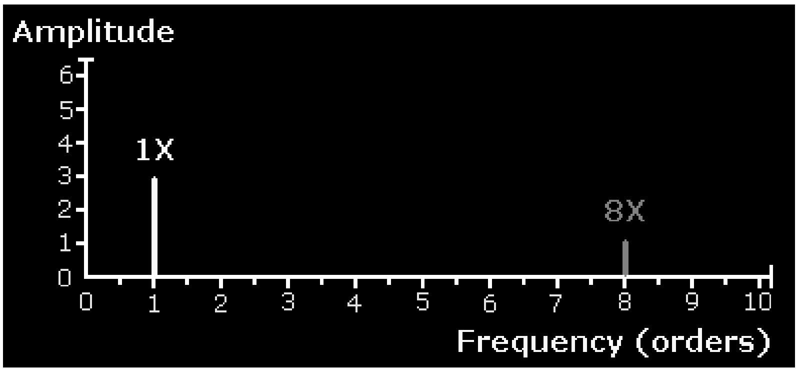


Figure 1-53 Spectrum with frequency in Orders.

Units of Orders is so useful in trying to find the source of peaks in a spectrum. When a peak is present at 5 orders it is easier to relate it to a physical occurrence such as vane pass on a pump. A peak at 38 orders could be rotor bars in a motor. A peak at 3X could relate to a 3 jaw coupling.

Of course there are peaks that are not integer (whole number) multiples of turning speed. You will soon learn that there is a HUGE difference between 6X and 6.1X – fan blades, pump vanes, etc. can generate "integer" orders (e.g. 5X, 6X, 10X), and only bearings and external shafts can generate vibration at non-integer orders (e.g. 3.09X, 6.1X, 7.3X)

All the energy in a spectrum can be grouped into one of three categories. The three categories all relate to the concept of Orders.

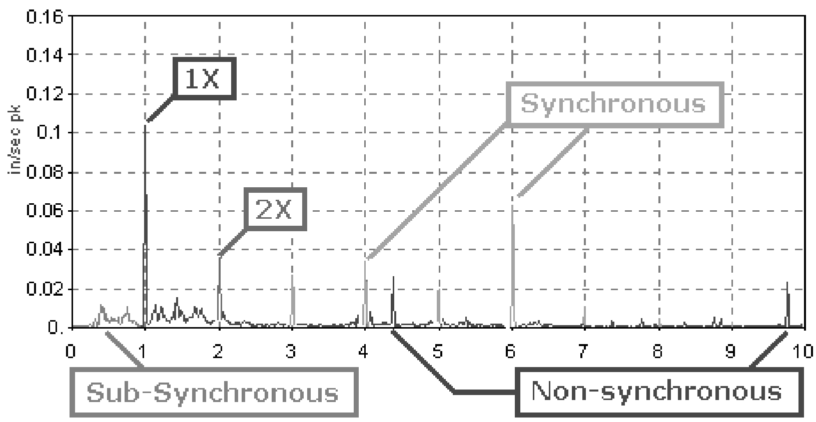


Figure 1-54 Spectral energy can be categorized into one of three groups.

Synchronous Energy- Energy that is an integer (whole number) multiple of running speed.

Non-synchronous Energy – Energy that is a fractional multiple of running speed.

Sub-synchronous Energy – Energy that is below running speed.

We will define these terms again later in the course.

How peaks relate to each other

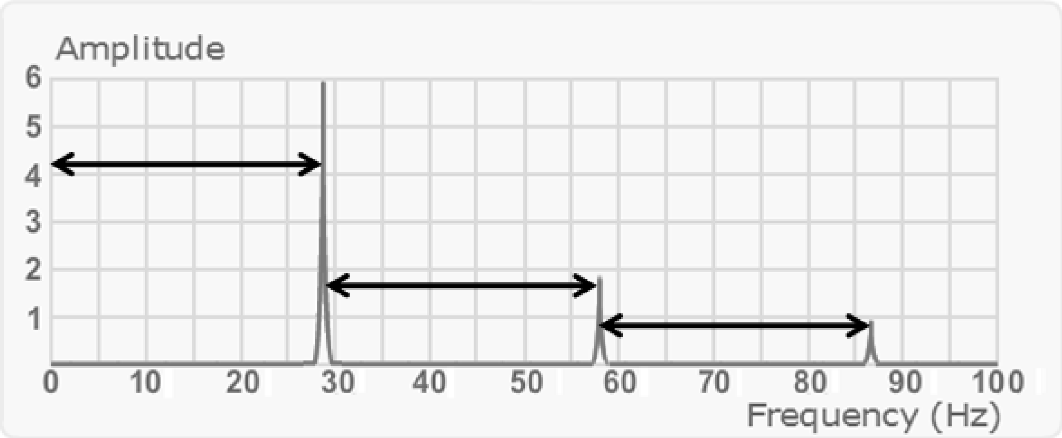


Figure 1-55 These peaks are multiples of the first one

Not only do we want to relate every peak in the spectrum to the motor shaft rate (1x), we also want to be able to quickly determine how peaks in the spectrum relate to one another. For example in Figure 1-55 we can see that these three peaks are multiples of the one on the left. It is easy to see this if we measure the distance between them (with the arrows), but if we just look at the number on the graph scale at the bottom (in Hz) it is not immediately obvious that the three are related.

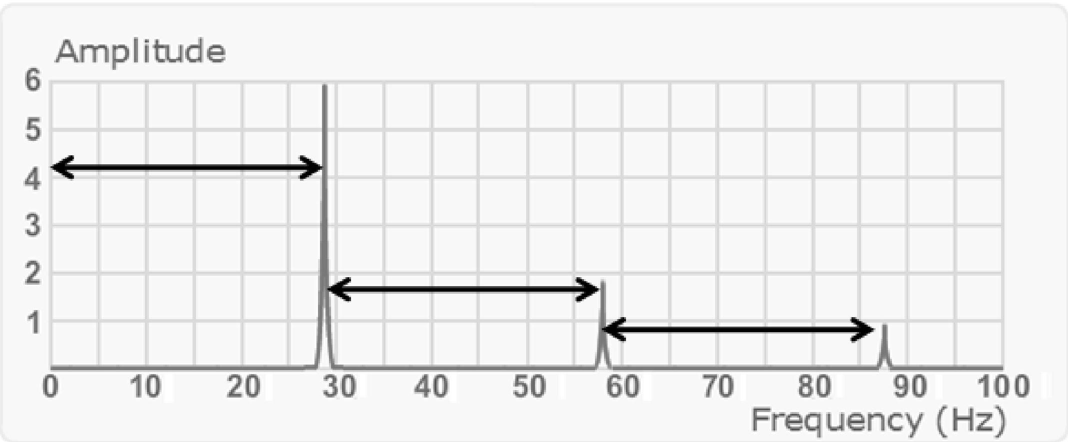


Figure 1-56 The peak on the far right is not a multiple

If we did not have the black arrows displayed on the plot in Figure 1-56 it would be difficult to see that the peak on the far right is not a multiple. We will see later that the difference between these two plots is very significant in terms of how they relate to potential mechanical problems in machines.

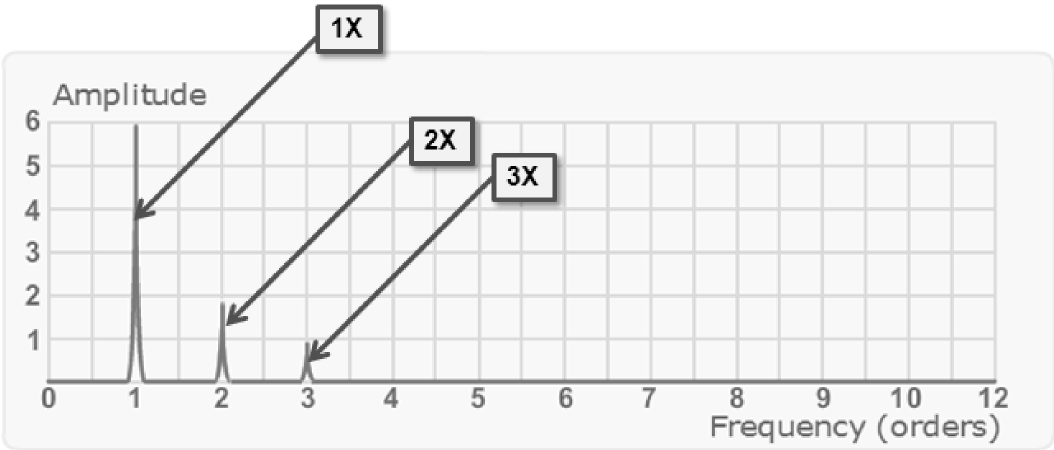


Figure 1-57 The same plot in orders

By changing the x axis on the graph from Hz to "orders" and by defining the first peak as 1x, it is now obvious that the three peaks are related (Figure 1-57) and in Figure 1-58 it is also obvious that the peak on the far right is not related.

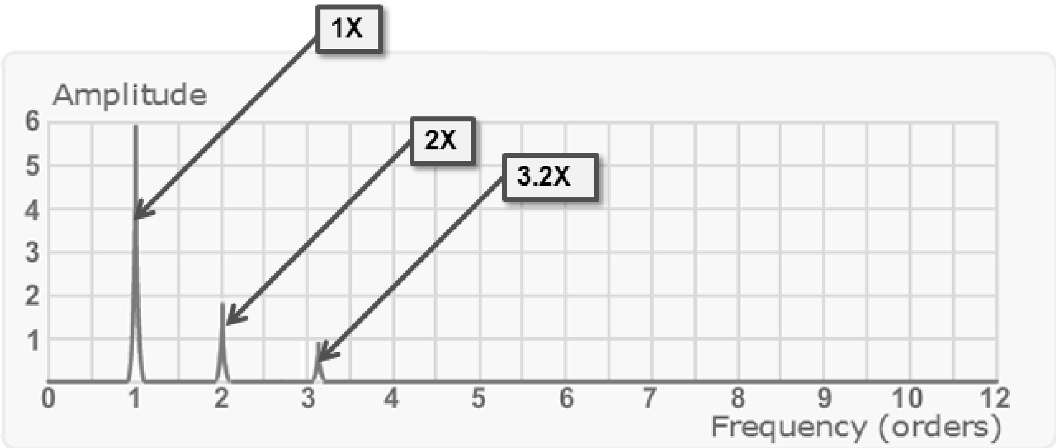


Figure 1-58 The peak on the right is clearly not a multiple

The frequency unit "orders" helps us in three ways

- It makes forcing frequency calculations easier (next section)
- It allows us to compare two spectra collected at slightly different speeds
    - This allows us to trend data on our machines
- It makes it easy to see if peaks in the spectrum are related to each other or not

Forcing Frequencies

We have already spent quite a bit of time talking about forcing frequencies without having properly defined them. Different components in a machine produce forces at particular frequencies. We used a fan with the weight on a blade to show how an unbalance force generated forces at the shaft rate frequency and we held a piece of paper up against the fan blades to show how they generated forces (hits) at a different frequency. These are both examples of forcing frequencies.

Introducing "forcing frequencies"

In a general sense we can say that forcing frequencies allow us to relate specific peaks in the spectrum (specific frequencies) to particular machine components (fan blades, shaft, gear teeth etc) and also to particular mechanical faults (bearing wear, unbalance, misalignment etc.)

Converting a waveform to a spectrum allowed us to separate the waveform into its components or sources of vibration. Forcing frequencies now allows us to relate those individual sources to machine components and mechanical faults. The concept of forcing frequencies is therefore central to the practice of using vibration for machinery fault diagnostics or condition monitoring.

Forcing frequencies may also be referred to as "**fault frequencies**" or "**defect frequencies**".

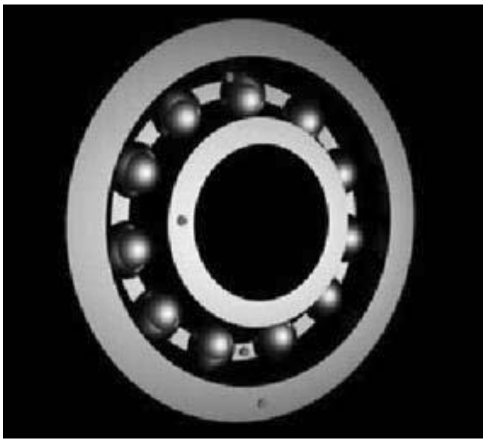


Figure 1-59 A rolling element bearing creates forcing frequencies related to inner and outer race faults as well as ball faults

Examples of Forcing Frequencies include:

- Blade passing rate
- Vane passing rate
- Bearing frequencies
- Ball spin
- Cage rate
- Ball pass inner race
- Ball pass outer race
- Belt Frequency
- Gearmesh
- Rotor-bar passing rate

Calculating forcing frequencies

Common forcing frequencies are calculated by multiplying the number of components times the shaft rate. For example:

- Blade pass rate = # Blades x RPM
- Vane pass rate = # Vanes x RPM

If I have a motor that runs at 3000 CPM and it has a cooling fan mounted on its shaft with 6 blades, then the blade pass rate = # blades x shaft rate or 6 x 3600 CPM = 21600 CPM.

The same calculation can be made in Hz. If the motor is running at 50 Hz and the cooling fan has 6 blades the blade pass rate will be 6 x 60 Hz = 360 Hz.

To be clear, it is the number of components x the shaft rate ***of the shaft the component is mounted on***! Therefore, if we have a multi shaft machine, not only will we need to know how many fan blades, pump vanes, compressor lobes we have, we will also need to calculate the speeds of each shaft.

Forcing frequencies: Belt drive machine

Here is how one determines the shaft rates for a belt driven machine:

Output speed = Input speed x (Input sheave diameter / output sheave diameter)

This could also be written:

$$S2 = S1 (D1 / D2)$$

Where

- S2 = Output shaft rate
- S1 = input shaft rate
- D1 = input sheave diameter
- D2 = output sheave diameter

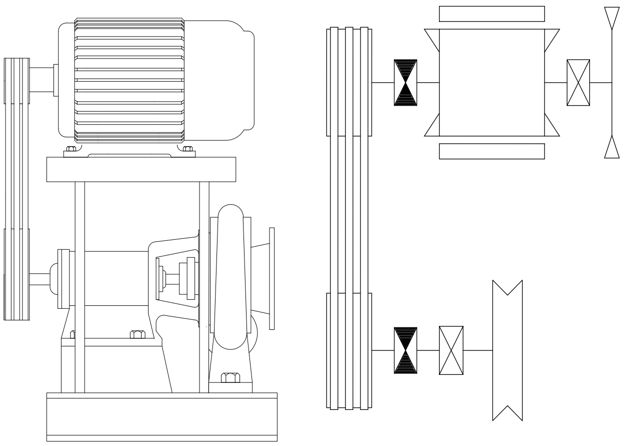


Figure 1-60 S2 = S1 x D1 / D2

Example:

A motor running at 3000 RPM with a pulley diameter of 60” drives a pump with a pulley diameter of 20”. What is the shaft rate of the pump in RPM? The pump has 6 vanes on its impeller, what is the impeller pass rate in CPM?

Calculations:

S1 = 3000, D1=60, D2 = 20, Pump vanes (PV) = 6

Solve for S2 (the pump shaft rate). S2 = S1 (D1 / D2)

S2 = 3000 (60 / 20) = **9000 RPM**

What about vane pass rate? 6 vanes x the pump shaft rate (9000 RPM) = **54,000 RPM**

Don’t forget it’s the number of elements (such as pump vanes) times the rate of the shaft that they are attached to, not necessarily the motor shaft rate!

Forcing frequencies: Calculating the belt rate

In a belt driven machine, the belt also spins around at its own frequency. This is another forcing frequency we need to calculate. Although there is a formula to calculate the belt rate, it is often easier to use a strobe light to measure the belt speed in relation to the shaft speeds. It should also be noted that the reason we are calculating these frequencies is so we can identify them in the vibration spectrum, because they will appear in the spectrum, we can sometimes use the spectrum itself and make some educated guesses about which peaks relate to the belt.

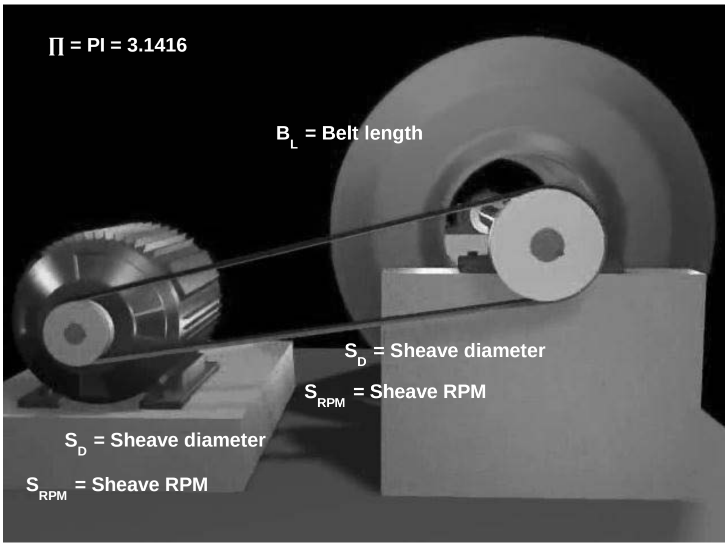


Figure 1-61 Belt rate: BR = ∏ x SD x SRPM / BL

The belt rate formula is as follows:

$$\mathbf{(BR) = (\pi \times S_D \times S_{RPM}) / B_L}$$

Where

- BR = Belt Rate
- π = 3.14
- S_D = Shaft diameter
- S_{RPM} = Shaft RPM
- B_L = Belt length

Note that the sheave diameter and shaft RPM are taken from the same shaft. The sheave diameter and belt length need to be in the same units of length.

Forcing frequencies: Gear driven machines

In a gear driven machine it is necessary to calculate the speeds of each shaft as well as the gear mesh frequency. The gear mesh frequency is the rate at which the teeth of the gears mesh with (or hit) each other.

Gear mesh (GM) = # teeth x shaft rate

The number of teeth and the shaft rate are for the shaft that the gear is mounted on. In other words if the motor shaft is turning at 100 CPM and it has a gear mounted on it with 20 teeth, the gear mesh frequency is 100 CPM x 20 teeth = 2,000 CPM.

In order to calculate the output shaft rate we use the following formula:

$$S2 = S1 \times (T1 / T2)$$

Where

- S1 = The input shaft rate
- S2 = The output shaft rate
- T1 = Teeth on the input gear
- T2 = Teeth on the output gear

You may notice that the formula is similar to the belt driven machine formula except sheave diameters are replaced with gear teeth.

Example calculation:

A motor running at 20 Hz drives a pump via a gearbox with 90 teeth on the input shaft and 30 on the output shaft. What is the pump shaft rate in Hz?

S1 = 20, T1 = 90, T2 = 30

S2 = 20 (90 / 30) = **60 Hz**

Forcing frequencies: Practice calculations

[1] If the following compressor ran at 1785 CPM, and there are 8 vanes on the impeller, calculate the compressor vane-pass rate in CPM, Hz and orders:

- 8 vanes x 1785 CPM = 14,280 CPM
- 14,280 CPM / 60 = 238 Hz
- 8 vanes x 1x = 8x

Which of these was easiest to calculate? What would happen if the compressor speed changed to 1773 RPM when tested the next time; which of these calculations would NOT change? The answer is orders – and this is why orders are the preferred unit of frequency.

[2] If the following fan had 12 blades, and the motor RPM was 1800, calculate the fan blade-pass forcing frequency in orders, Hz and CPM:

- 12 blades x 1x = 12x
- 12 blades x 1800 CPM = 21,600 CPM
- 12 blades x (1800 / 60) = 360 Hz

[3] If there were 8 vanes on the following compressor, and the compressor vane rate was 28,560 CPM, calculate the RPM of the compressor:

- 28560 / 8 = 3,570 CPM

Forcing frequencies: Recap

We have just taken a brief look at what forcing frequencies are and we have given some examples and formulas for how to calculate some of them. There are other forcing frequencies we have not covered yet. At this point, the important thing is to understand that in order to do vibration analysis we need to begin by understanding the machine, how fast the shafts turn, how many fan blades, pump vanes and gear teeth there are, and then we need to calculate the forcing frequencies. This is the process for relating peaks in the spectrum to particular machine components and then to particular mechanical faults.

Earlier we gave an example of the inadequacy of the RMS overall level to describe how a symphony sounded. All the overall RMS value could tell us was how loud or soft it was. Now that we understand a bit about the vibration spectrum, frequency analysis and forcing frequencies we should now understand that these tools will allow us to say much more about the symphony and what in particular does not sound right.

Understanding forcing frequencies is an important part of vibration analysis!

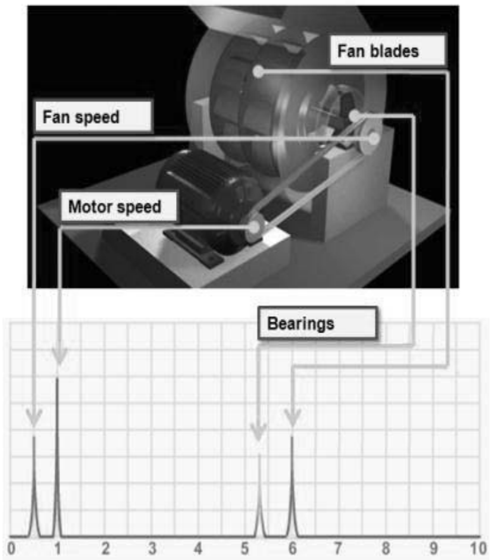


Figure 1-62 Understanding forcing frequencies is an important part of vibration analysis

An introduction to phase

In this section we will provide an introduction to the concept of phase. In later chapters we will discuss applications for phase measurements.

In prior sections we discussed some characteristics of vibration such as amplitude and frequency. Consider two children on swings, swinging next to each other. Let us say they move at the same frequency and the same amplitude. These two characteristics; amplitude and frequency, give us a great deal of information to help us visualize the two children on the swings and how they are moving. What they do not provide however is any indication of how the children are swinging *relative* to each another. Are they moving forward and back at exactly the same time? Is one swinging forward while the other one is swinging back? Is one of them closely following the other, arriving at the top in front just after the first one arrives there and then arriving at the top in back just after as well?

These are the types of questions that phase readings will answer for us.

At this point, the important concept to understand is that phase is a relationship between two things.

Before we continue, please remember that one rotation of the shaft or one cycle of vibration is equal to 360 degrees.

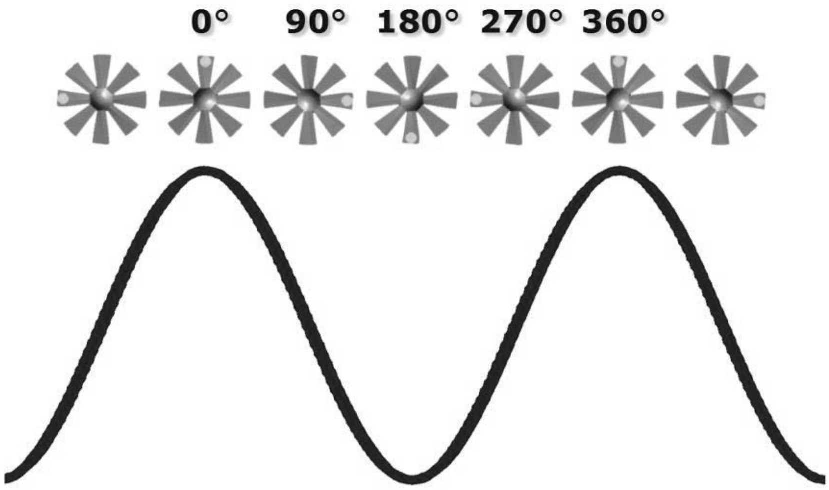


Figure 1-63 One cycle is equal to 360 degrees

Figure 1-64 shows two fans that are rotating together. They are said to be in-phase with each other. Notice how the sine waves reach their peaks at the same time? They are in time. They are synchronized. They are in-phase.

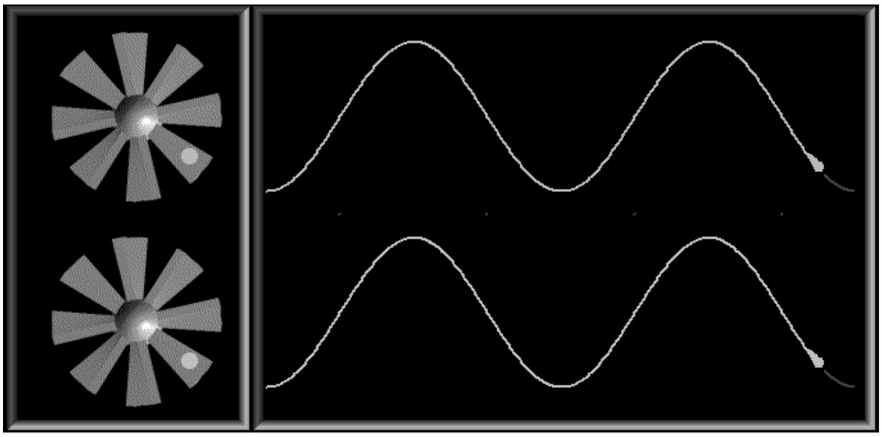


Figure 1-64 Two machines in phase with each other

In this case the two fans are also vibrating at the same amplitude. Please note that amplitude and phase are not related. The phase relationship will not change if the amplitudes of either waveform changed. The phase is related to the timing; when do each of them pass the maximum in the vertical direction? If they arrive there at the same time then they are "in phase."

Also notice that the position of the blades with the coins is the same for both fans.

Introduction to phase: Out-of-phase

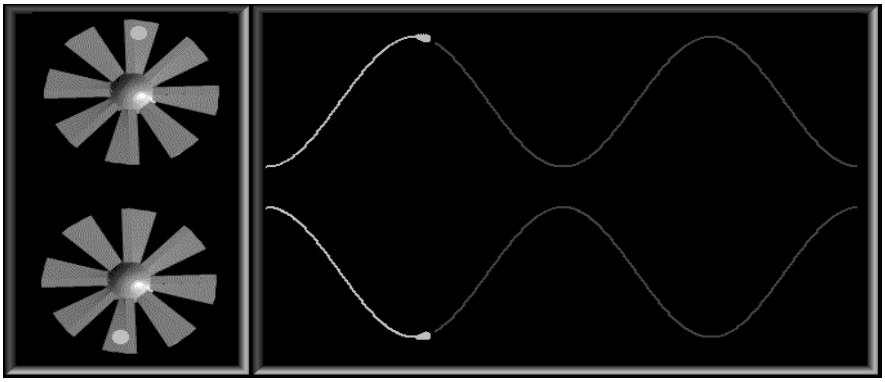


Figure 1-65 The fans are out of phase

Figure 1-65 shows the waveform of the two fans moving in opposite directions from each other. The peak of the top fan occurs when the bottom fan is at the bottom of its travel. They are “out of phase” with each other. If this referred to the children on the swings they would be swinging exactly opposite to each other – when one is in front the other is in back.

There is a way to measure how much out of phase the two fans are. Using the top waveform as the reference waveform, one cycle can be divided into 360 equal intervals. Each interval is actually a degree of rotation and one rotation is 360 degrees.

The peak of the lower fan reaches its peak 180 intervals after the top wave reaches its peak. It is 180 degrees different from the top waveform. It is said therefore that it is 180 degrees out of phase with the top fan. This is also ½ the period.

Phase is measured in degrees. The fans in Figure 1-66 are 90 degrees out of phase.

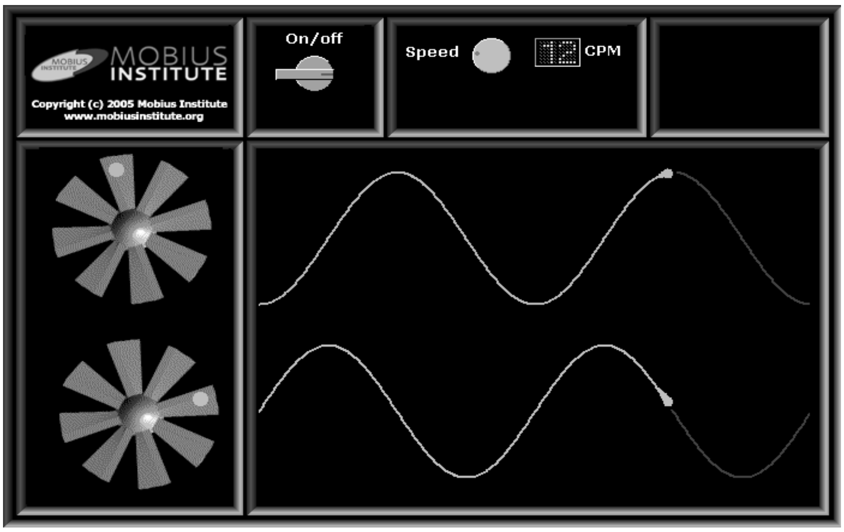


Figure 1-66 The fans are 90 degrees out of phase with each other.

Produced by SUMICO Technologies

Where does phase come from?

We'll discuss phase measurements in more detail in the Data Acquisition section but there are a few important points we should cover now. As we mentioned, phase is the relationship between two things; it could be the movement of the two fans in relation to their unbalance forces or it could be the relationship of the movement between children on swings. In either case, when measuring phase one needs to collect two things in order to understand the phase relationship between them.

When measuring phase with a single channel data collector, one typically uses a tachometer as a phase reference. This is called **Absolute Phase.** In this case it will show the phase relationship between the tachometer and the vibration sensor at the shaft rate frequency.

When measuring with a two channel data collector it is often possible to measure the phase relationship between the two vibration sensors directly. This is termed **Relative Phase**.

Using a tachometer reference

In Figure 1-67below we can see a typical configuration for collecting phase using a tachometer and a vibration sensor. A piece of reflective tape (white in the image) is placed on the shaft. When the reflective tape passes by the beam emitted from the tachometer (labeled "2") it creates a pulse. We can see the pulse in the top right graph labeled "2." It should be noted that the tape passes by the tachometer one time per shaft revolution, so the frequency of it passing is the same as the shaft rate.

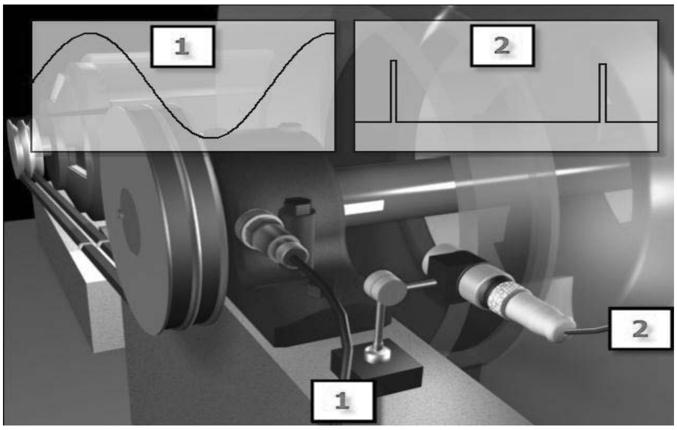


Figure 1-67 Measuring phase with a tachometer

The vibration sensor (labeled "1") is collecting a vibration reading from the bearing housing. We can see this in the graph labeled "1." The vibration waveform has an amplitude and a frequency, as we know from our earlier understanding of vibration. Now the question is how do these two items relate to each other? This is where we measure phase.

Let's overlay the two graphs, the one from the tachometer and the one from the vibration sensor, in time. You can see this in Figure 1-68. The next step is to measure the difference, in degrees, between the tachometer pulse and the high point of the vibration wave, remembering that one revolution is equal to 360 degrees.

Looking a little more closely at Figure 1-68 it appears that the phase difference is about 80 degrees or slightly less than ¼ cycle.

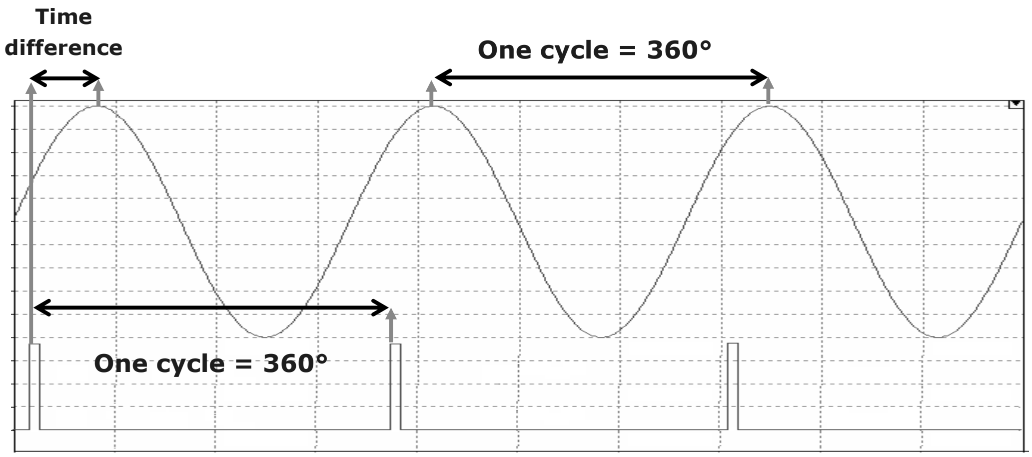


Figure 1-68 Measuring phase with a tachometer

Relative phase: Two channel

Another option for measuring phase is to use a two channel data collector and measure the phase difference between two vibration sensors. This is termed "relative phase." We can see a typical configuration in Figure 1-69 below.

Take a moment to look at this figure and remember the earlier example of the fan with the unbalance weight. What do you think the phase relationship will be between the two sensors? How much later will the weight pass one of them than the other one? Consider how the sensors are oriented in relation to each other. When you think you have an answer, take a look at Figure 1-70 and compare the waveforms to each other. How many degrees apart are they?

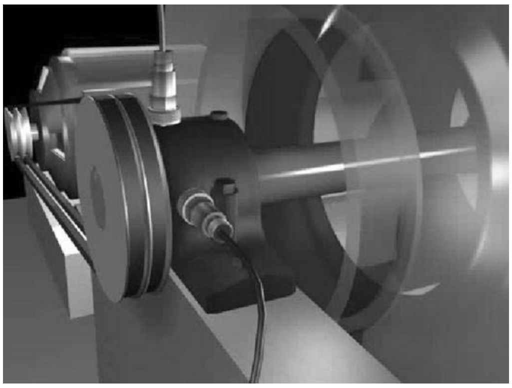


Figure 1-69 Measuring relative phase with two accelerometers

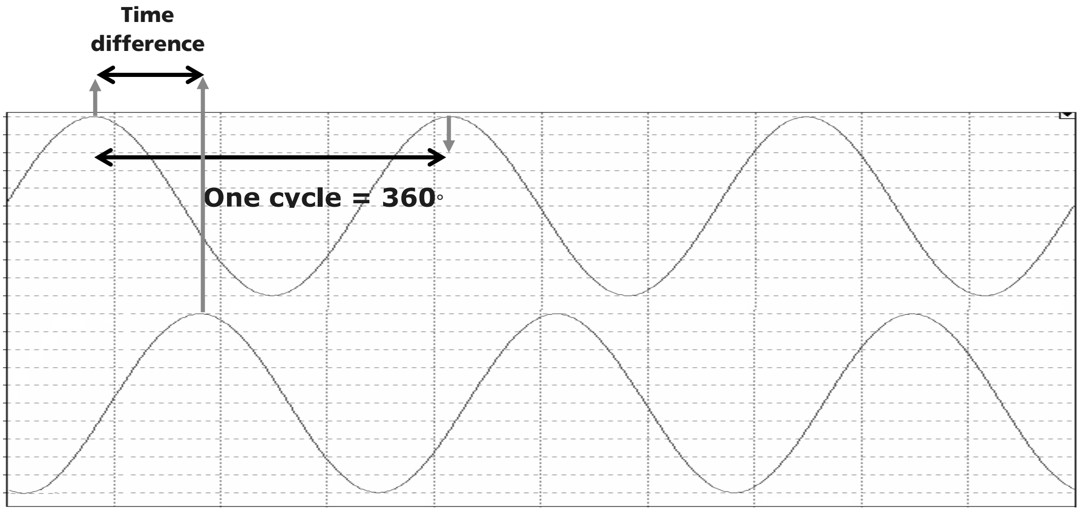


Figure 1-70 Relative phase between two accelerometers

Representing phase data

Phase is useful for diagnosing some mechanical faults in machines. We can just use the analyzer and make observations, or we can record the readings and analyze the patterns to see what they are telling us about the machine. If we will be collecting and storing data to show to or share with our colleagues or clients, we need to have some conventions for presenting the data.

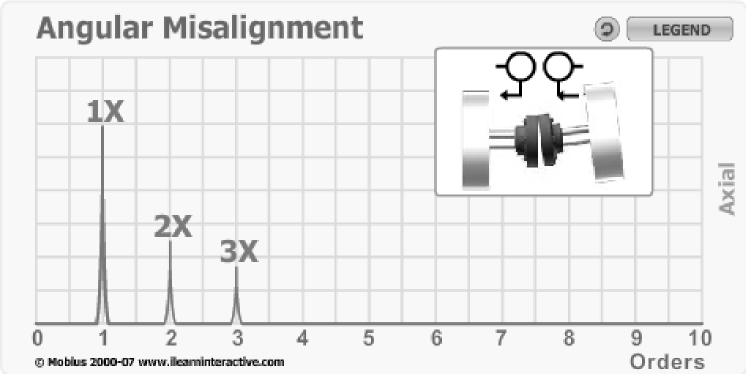


Figure 1-71 Angular misalignment with phase information

In the figure above we see a vibration graph and at the top right there are two circles with lines coming out of them that describe the phase relationship between axial readings on both sides of the couplings. These circles are referred to as a "bubble diagram."

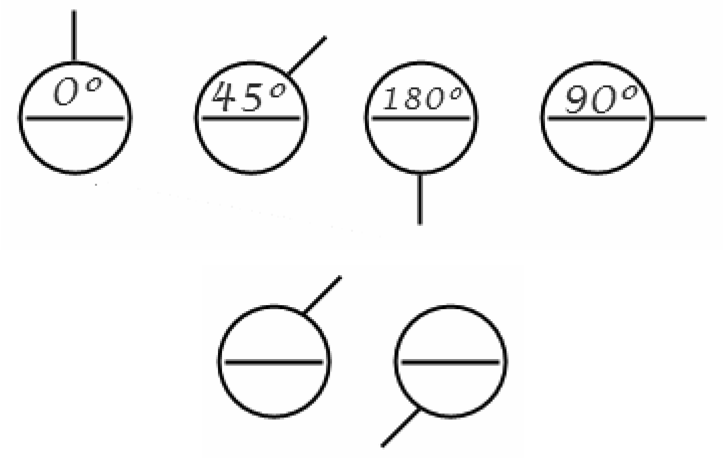


Figure 1-72 Bubble diagram for documenting phase readings

Note that in most cases we are interested in the relative phase readings between one or more points on the machine, such as in where the readings were taken on the axial faces of the machine components on either side of the coupling. In this case it is the fact that the readings are 180 degrees apart that confirms we have this fault. We are not interested in knowing that the reading on the right was 90 degrees or 135 degrees, we simply want to know how it relates to the reading on the left.

Therefore we have some options on how to use the bubble diagrams. We can write in the phase values as we see done in the top row (noting that the bottom half of the circle would contain the vibration amplitude) or we can just use the tails sticking out of the circles (bottom left) or inside the circles (bottom right) to display the phase. As described in the top row – zero degrees is shown with the tail pointing upwards, 90 degrees is 3:00, 180 degrees is 6:00 and 270 degrees is 9:00.

Summary of phase

Here is a brief summary of phase:

Phase describes the relationship between two signals, so we need a reference or two separate signals to calculate phase. Typically, a vibration sensor and a tachometer are used to collect **absolute phase** or two vibration sensors are used with a 2 channel data collector to collect **relative phase**.

Phase readings are used in many applications including:

- Balancing
- Diagnosing common machine faults
- Verifying resonance
- Understanding how structures move

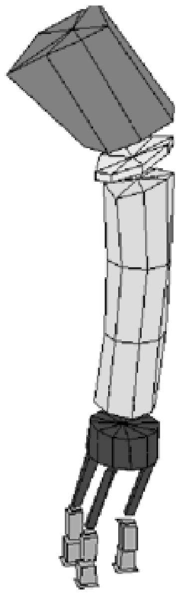


Figure 1-73 Animating the movement of a machine with phase data

Introducing orbits

If we look at a time waveform from a proximity probe (a sensor that measures displacement or the distance between itself and the shaft) we will see a familiar pattern (Figure 1-74)

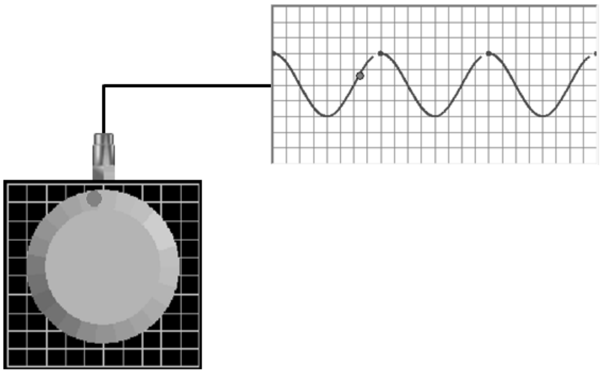


Figure 1-74 A sine wave produced by a proximity probe

And if we place the sensor in the horizontal direction we will see the same pattern shifted over 90 degrees (Figure 1-75).

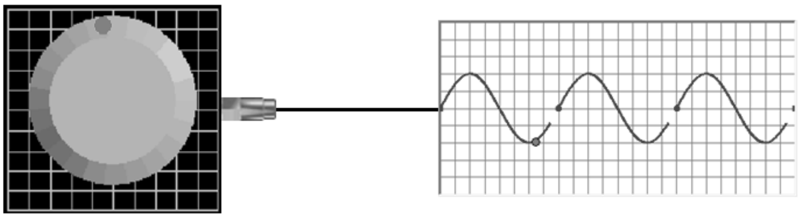


Figure 1-75 A sine wave with a 90 degree phase difference from the vertical reading

Now, if we plot the two waveforms against each other we will get a circle. This plot, the combination of the two waveforms, is called "Lissojous figure" or an "orbit" (Figure 1-76)

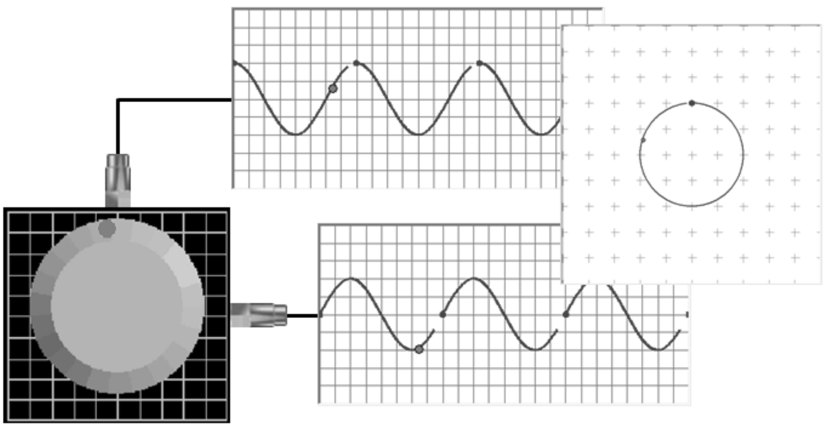


Figure 1-76 Combining two waveforms creates an "orbit"

The orbit plot is typically used when monitoring large journal or sleeve bearings using proximity probes. The orbit describes how the shaft is moving within the bearing. If the shaft moves more in the vertical than the horizontal direction, the orbit will change shape (Figure 1-77).

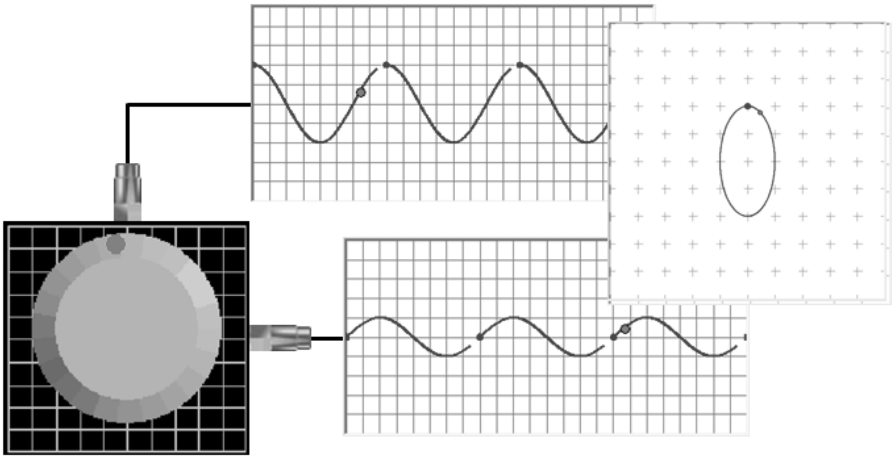


Figure 1-77 Vertical motion is greater than horizontal motion

Other interesting shapes are also possible (Figure1-78). In a general sense one can say that the shape of the orbit directly relates to how the shaft is moving and hence can give a good indication of the common problems associated with large journal or sleeve bearings. Typically orbits are filtered to only display vibration at 1x or the shaft rate. There are however applications where 2x or other frequencies may be collected and analyzed.

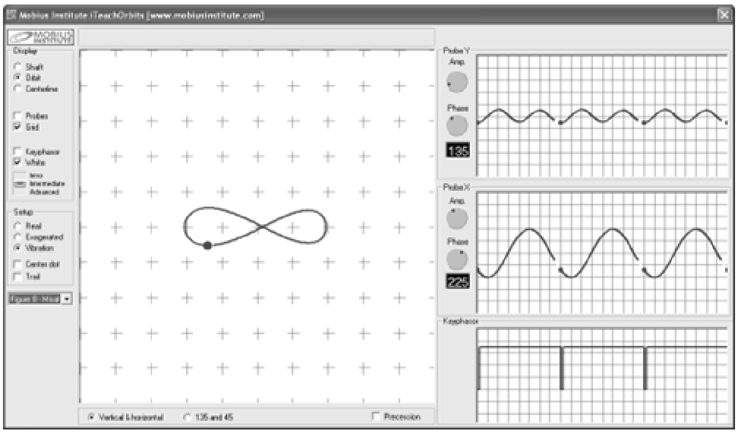


Figure1-78 A figure 8 indicates misalignment

Chapter 2
Understanding Signals

Objectives:

- Describe amplitude and frequency effects of mixing signals
- Recognize transients, clipped signals, and distortion and describe how they affect the spectral data
- Describe Amplitude Modulation and how to recognize it in waveform and spectra
- Describe Frequency Modulation and how it differs from Amplitude Modulation
- Describe ‘Beating’ and how it looks in waveform and spectra
- Compare and contrast ‘Beating’ and Amplitude Modulation
- Describe and calculate sum and difference frequencies

Understanding signals

The previous section focused on signals generated by specific forcing frequencies in the machine. They were fairly simple or mixed signals of only one or two sources which yielded a simple FFT or spectrum. However, not all the signals are that simple. Numerous signals can mix together and generate many peaks in the spectrum. This section looks at those results plus some other phenomena such as:

- Square waves
- Truncation
- Transients and impulses
- Amplitude and Frequency Modulation
- Beating

Although some of this may seem theoretical, if the concepts are understood, then when these patterns show up in data it will be clearly understood what is going on in the machine. This can lead to a better diagnosis. Many of the screen shots in this chapter that show the signals are taken from the iLearn software.

Classic Signals

THE PURE SINE WAVE

The pure sine wave is a single frequency producing a single peak in the spectrum

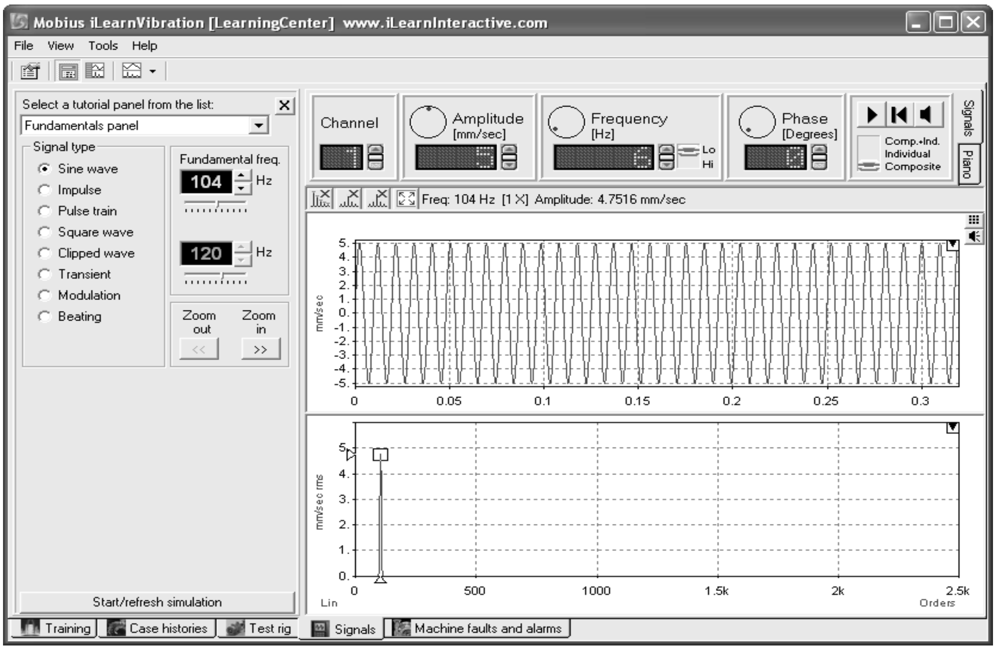


Figure 2-1 - Classic pure sine wave produces one peak in the spectrum.

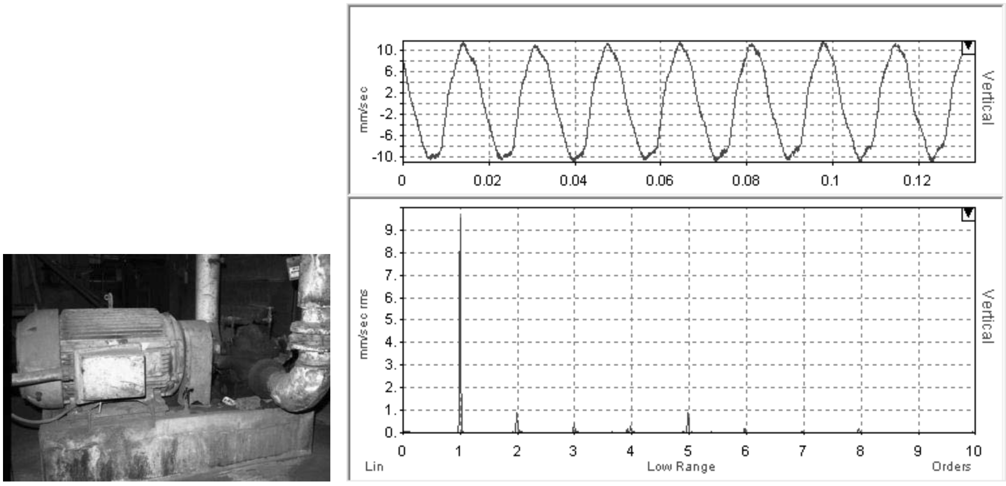


Figure 2-2 - A real signal with a dominant sine wave indicates a single dominant force - such as unbalance

When additional sources of vibration are added, it changes the waveform dramatically. The top graph has two signals, the first at 160 Hz and a second at 140 Hz. The signals have equal amplitude. The spectrum reflects this with two peaks of equal amplitude.

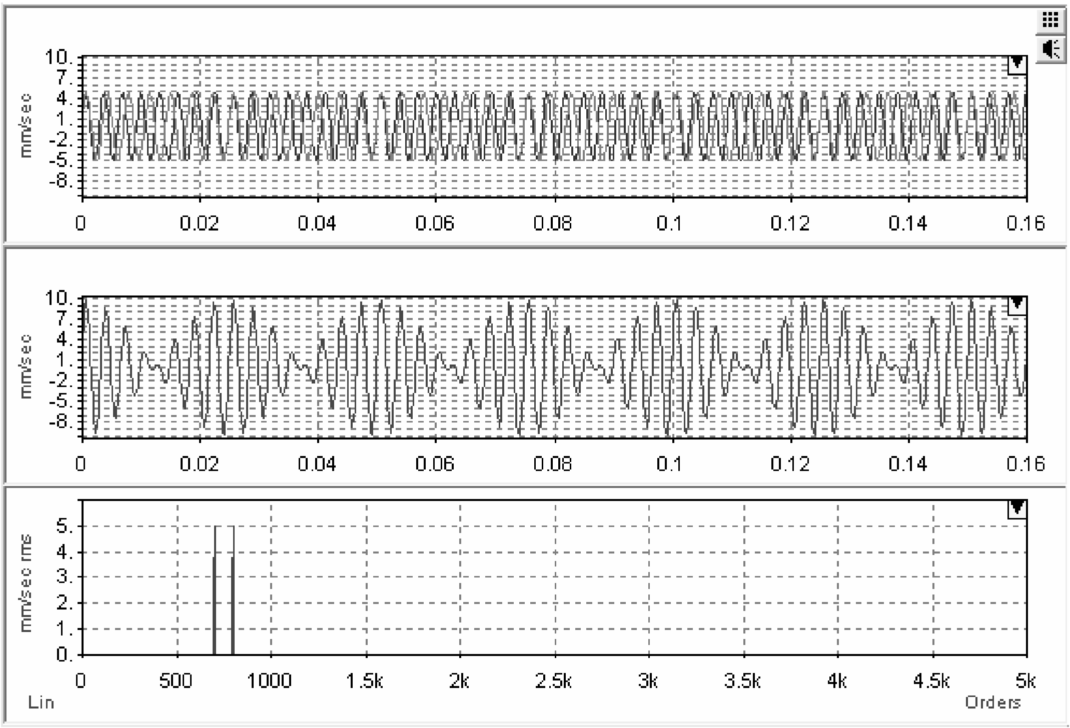


Figure 2-3 - The 160 and 140 Hz signals move in and out of phase, adding and subtracting

In Figure 2-3 the two simple signals add and subtract from each other as they move in and out of phase causing an unusual shape in the combined waveform. Notice how there are places in the combined waveform that the amplitude goes to zero. This is because the two signals are the same amplitude and cancel each other out. The maximum amplitude is 10… the combined values of both signals.

DIFFERENT AMPLITUDES

In Figure 2-4 the amplitude of one signal has increased so that one signal still is 5 mm/sec and the other is 10 mm/sec. Notice the changes in the resultant combined time waveform. The amplitude now sums to a maximum of 15 mm/sec and a minimum of 5 mm/sec. So there is no place where the amplitude goes to zero.

In the spectrum there are two peaks, each with amplitudes corresponding to their individual waveform amplitudes.

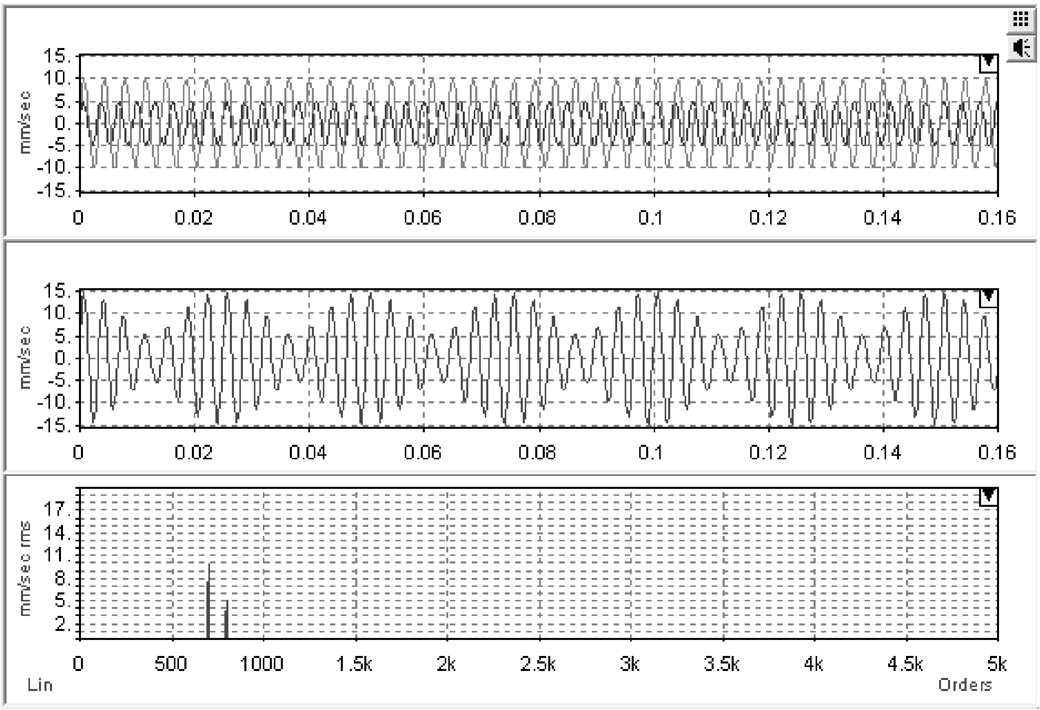


Figure 2-4 - The two signals are no longer equal so there is no place the combined waveform goes to zero.

SAME FREQUENCY, SAME AMPLITUDE, IN-PHASE

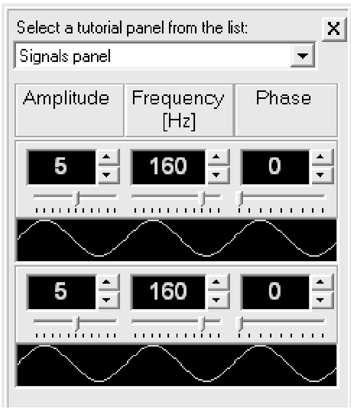


Figure 2-5

In Figure 2-6 the Frequency of both signals has been adjusted to 160 Hz. The amplitudes are the same, 5 mm/sec, and they are in-phase. Notice in the top graph how the two individual signals look as if there is only one.

The middle graph shows the combined waveform where the two signals are added together. Notice that they add to be only one frequency, but the amplitude is the combination of them, 10 mm/sec.

This situation can occur in machines when two signals of the same frequency are in-phase. Notice there is only one peak in the spectrum.

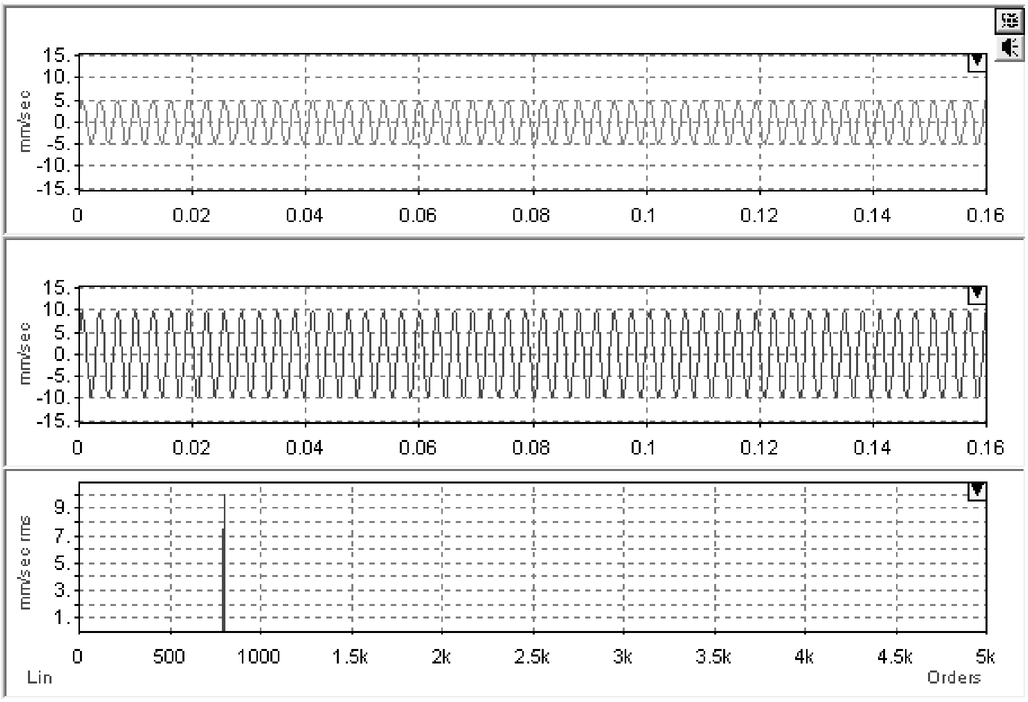


Figure 2-6 - The two signals are now the same frequency, 160 Hz, and the same amplitude, 5 mm/sec. They add together because they are in-phase.

Resolution - In reality, the source could be from two different frequencies that are close together, but if the resolution is not good enough they will combine and appear as a single frequency.

EFFECTS OF PHASE

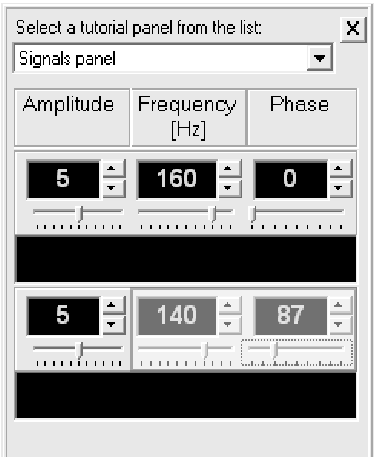


Figure 2-7

The two signals are now at frequencies of 140 Hz and 160 Hz with equal amplitudes. When the phase is varied, the individual sine waves in the top graph seem to "slide over each other." The pattern in the combined waveform adjusts or shifts as the phase changes.

The spectral data shows the two peaks of equal amplitude.

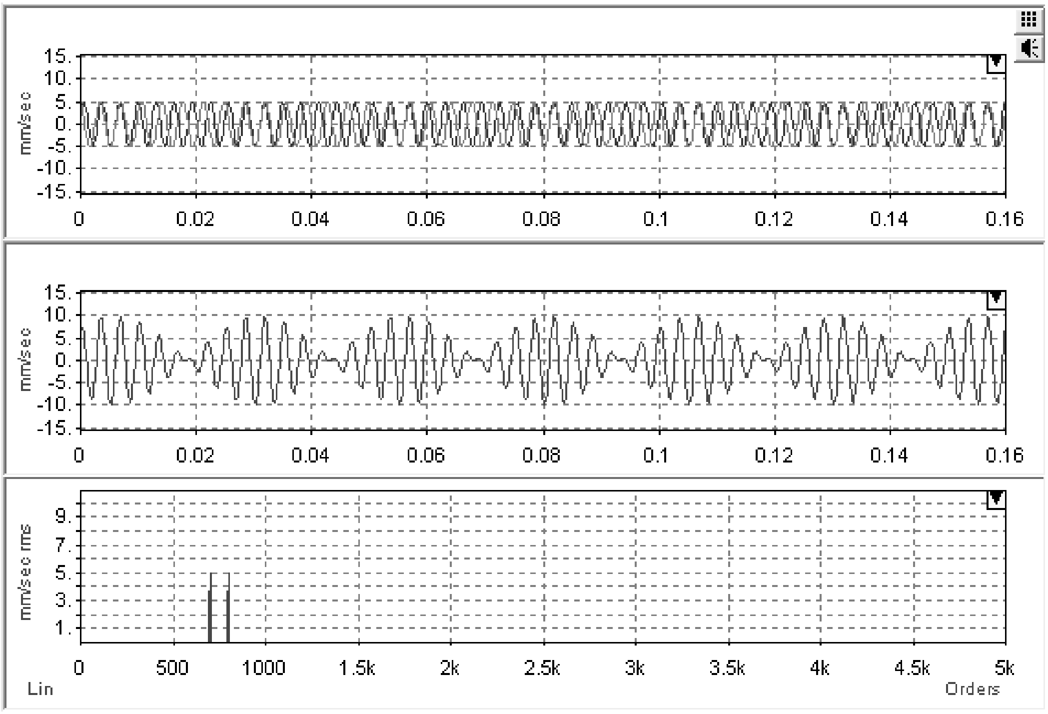


Figure 2-8 - A 160 Hz and a 140 Hz signal 87 degrees out of phase

PHASE CANCELLATION

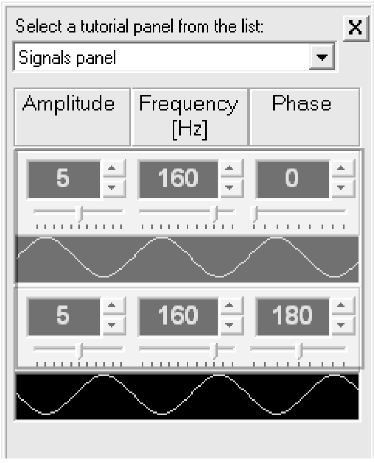


Figure 2-9

Two signals with the same amplitude and frequency, but 180 degrees out of phase with each other cancel each other out.

The top graph shows the two signals. The middle graph shows the combined resultant… as straight line. The spectrum shows the result of the combined waveforms… no peaks.

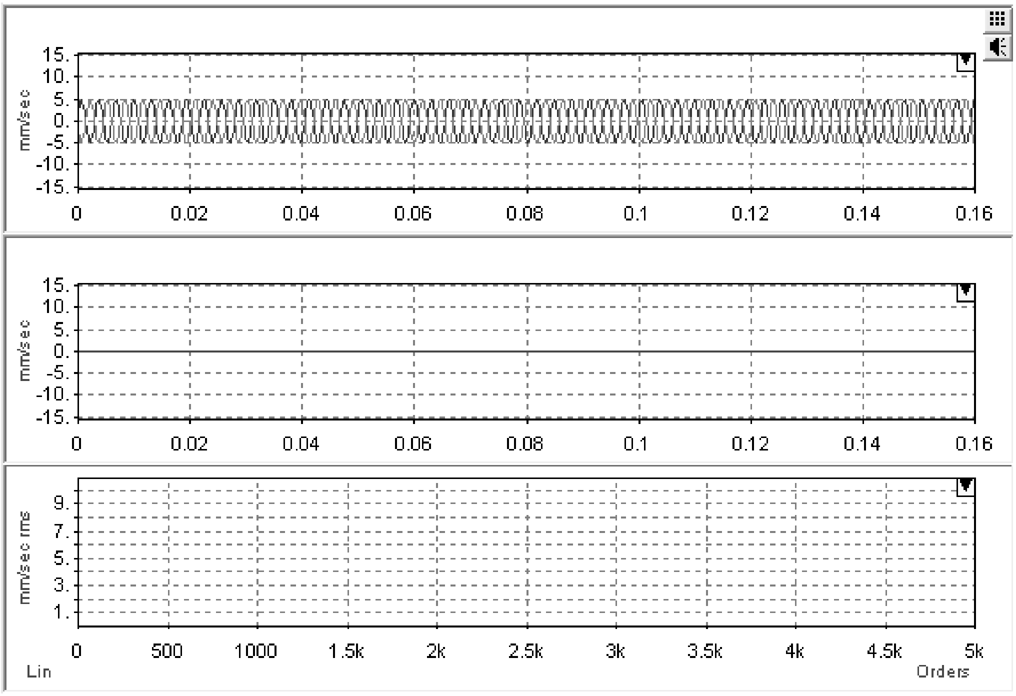


Figure 2-10 - The result of two signals of the same frequency and equal in amplitude...they cancel each other out.

Signal Rules

SINGLE IMPULSE

If the input signal is a single impulse, the waveform will have just one spike in it. It is not sinusoidal because it is not repetitive or "periodic." It is only a single impulse.

The result in the spectrum is a flat line. An impulse injects energy into all frequencies in the spectrum. Essentially it raises the noise floor of the spectrum to some amplitude.

An application for this single impulse is the "Bump Test" which is discussed in the "Natural Frequencies and Resonances" chapter. The machine is struck to inject energy at all frequencies.

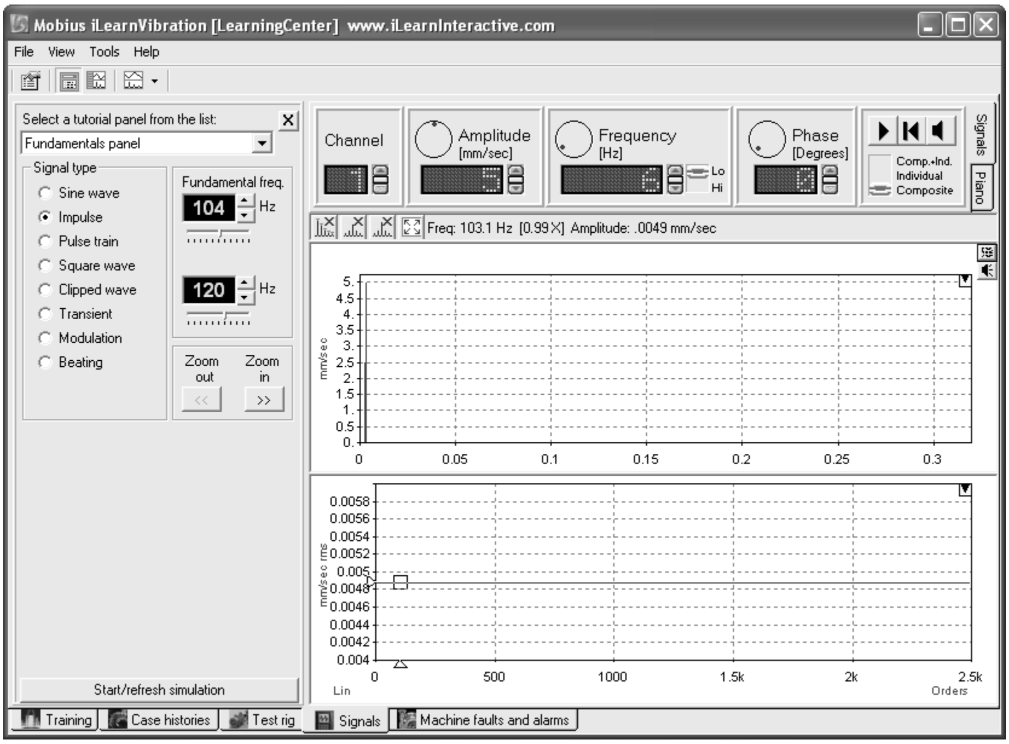


Figure 2-11 - An impulse injects energy into all the frequencies, raising the floor of the spectrum.

PULSE TRAIN

The result is very different when there are multiple pulses in the time window. Multiple peaks appear in the spectrum. This pattern is called "harmonics."

Harmonics are a series of evenly spaced peaks. Their amplitudes could vary, but the peaks will be spaced at integer multiples of the first in the series.

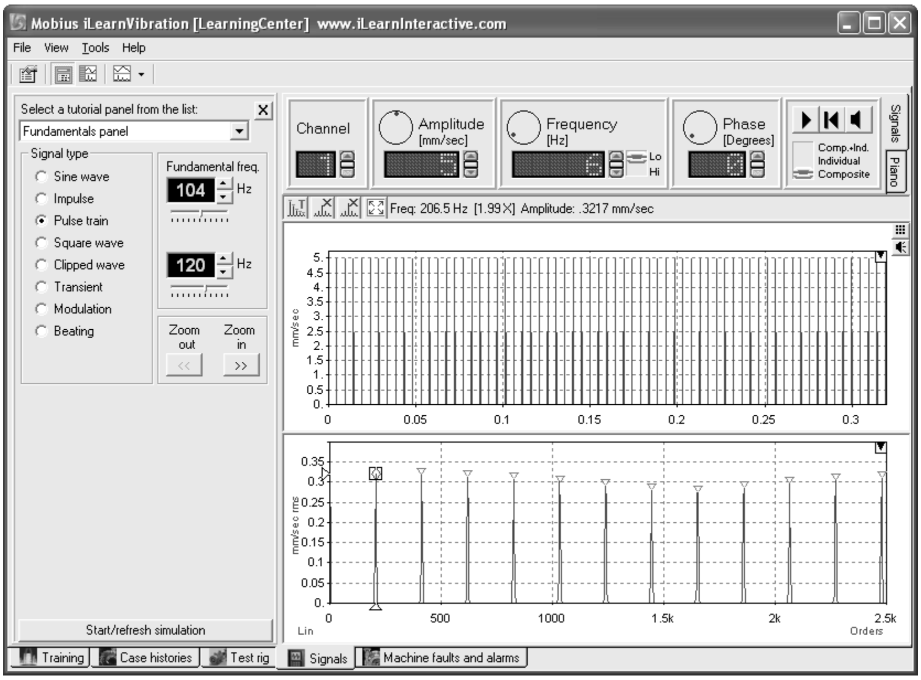


Figure 2-12 - Multiple pulses in the waveform produce harmonic peaks in the spectrum

SQUARE WAVE

A pure square wave generates odd harmonics…1x, 3x, 5x… Pure square waves are not common in machinery analysis.

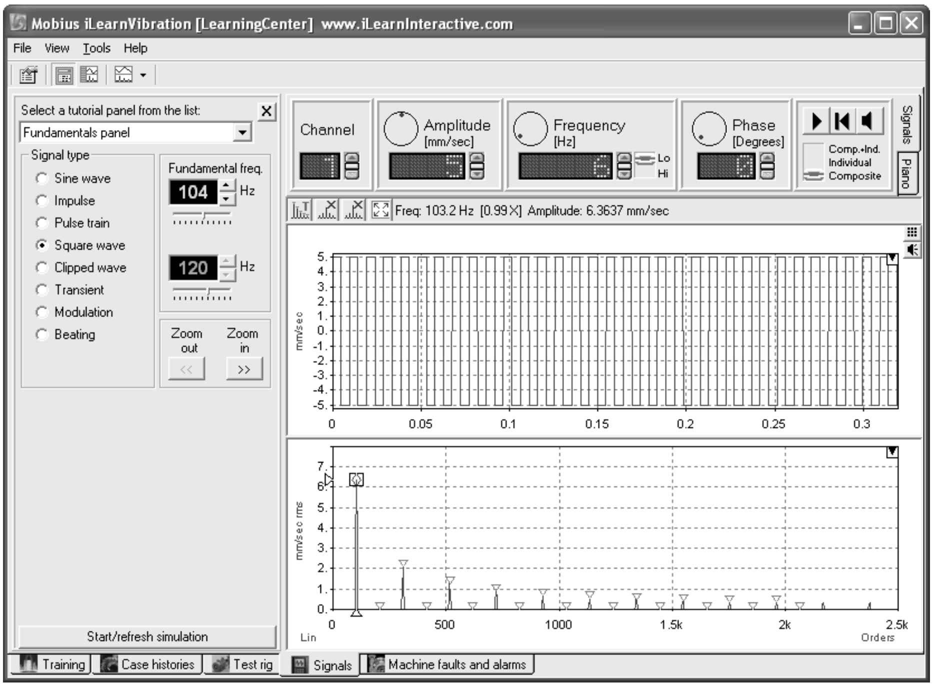


Figure 2-13 - A pure square wave produces odd harmonics in the spectrum.

CLIPPED WAVE - DISTORTION

Clipped waves are a little different than the square waves. They are a normal wave that appears as if it has been chopped off on one side. The signal is distorted. The result is harmonics in the spectrum.

Clipped waves are fairly common in machinery analysis. It occurs when the motion is restricted in one direction. This generates all harmonics, not just odd harmonics as the square wave.

When the waveform is clipped more, the harmonics are much stronger in amplitude. When the wave is clipped, the impact that occurs as the motion is restricted may be strong enough to produce a transient.

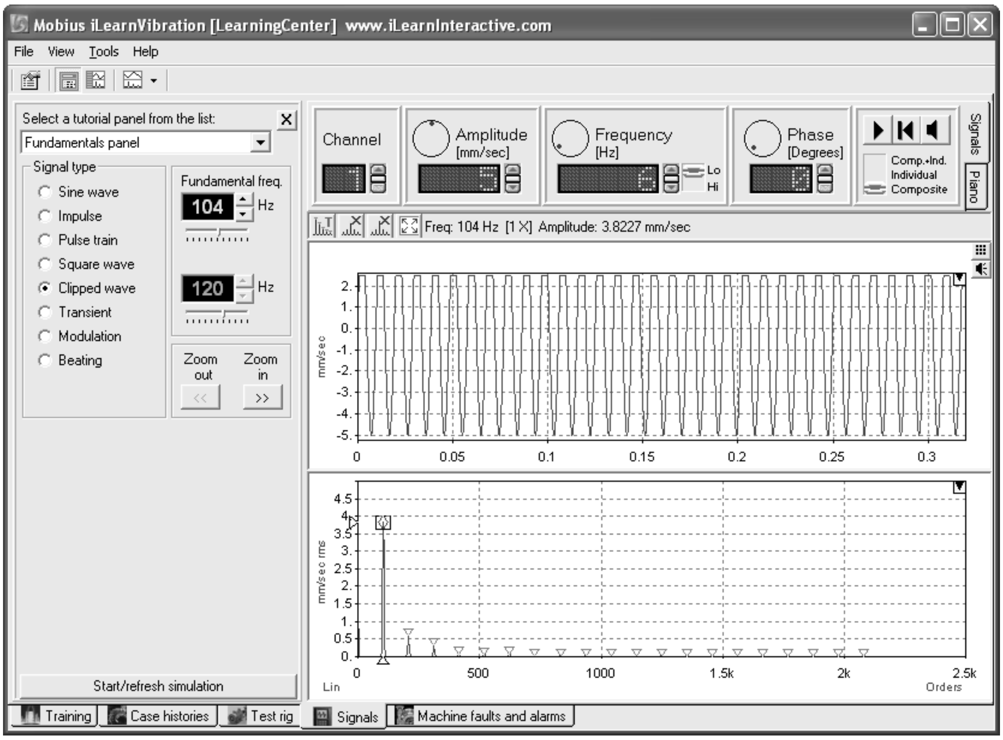


Figure 2-14 - A clipped wave produces harmonics at all integer multiples, not just odd harmonics.

IMPACTS

An impact is a very steep increase in amplitude with an equally steep decrease in level, possibly with some ringing afterwards. This occurs when there is extreme looseness in a machine. The impacts of rattling or banging produce strong harmonics. In extreme cases the harmonics will be half-order harmonics or even 1/3 order harmonics. Peaks will be at 0.5x, 1x, 1.5x, and so on.

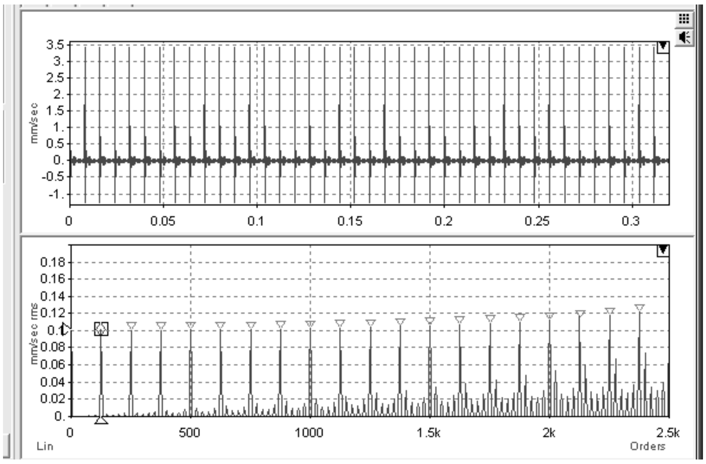


Figure 2-15 - The impact produces harmonics and in severe cases sub-harmonics...multiples of 0.5 orders

Example from the rotor kit.

A bearing pedestal was loosened on the rotor kit so that it rattled as it was running.

Waveform Data from the horizontal direction has strong impacts that are clipped. These transients produce harmonics. In this case the 1x harmonics are strong and there are small 1/3 order harmonics.

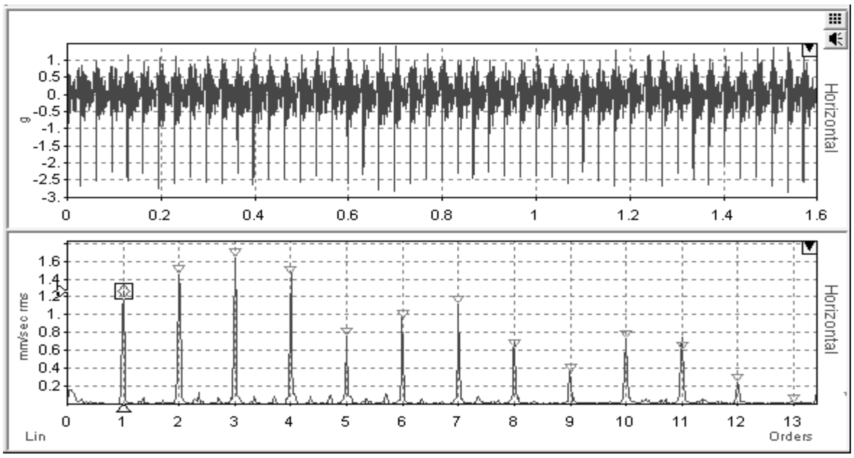


Figure 2-16 - Transients from a rotor kit produce strong 1x harmonics and small 1/3 order harmonics…1/3, 2/3, 1, 1 1/3, etc.

WAVEFORM SYMMETRY

In a pure sinusoid there is as much positive vibration as negative vibration

However, in some cases there is far more movement in one direction than another. See Figure 2-17 which is an expanded section of the waveform in Figure 2-16.

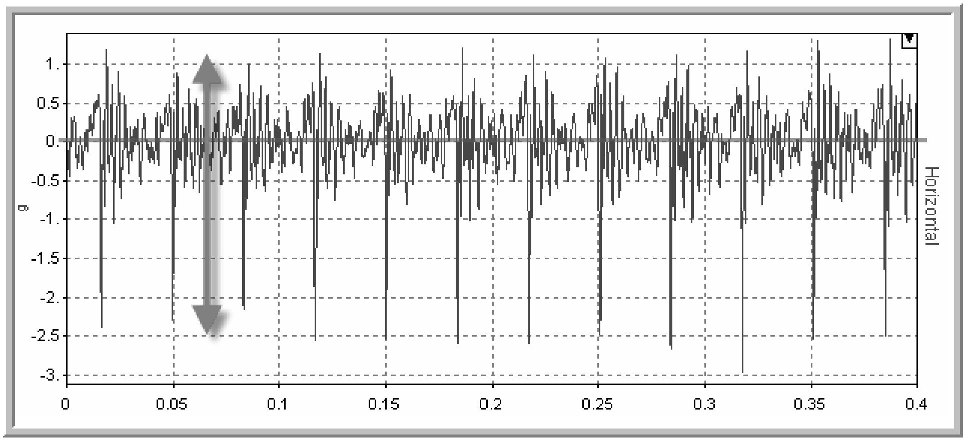


Figure 2-17 - The waveform indicates there is more freedom of movement in one direction.

DISTORTION VS. ACTUAL SIGNALS

If there are peaks at 1x, 2x, 3x, 4x, etc it does not necessarily mean there are actual signals in the machine at those frequencies. Signals as multiples of 1x can come from two sources:

- Actual sources of vibration – misalignment, blade pass, couplings, etc.
- Distortion – the FFT process creates harmonics in the spectrum

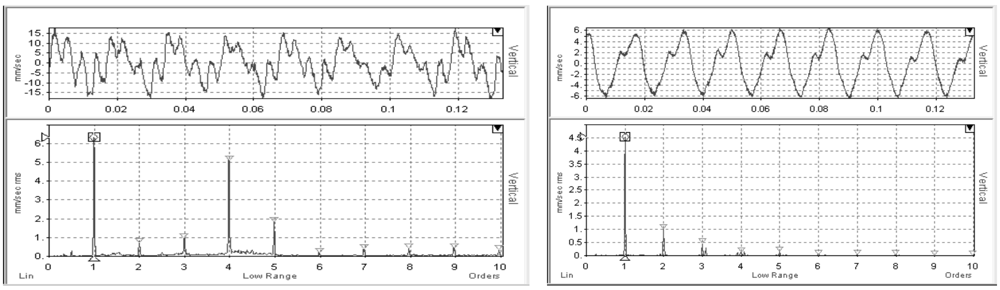


Figure 2-18 - Multiples of 1x can come from actual signals in the machine at those frequencies, but can be effects of Distortion in the FFT process

Amplitude Modulation

Two types of modulation are common in machinery analysis. They are the same ones as are on the radio dial, AM and FM. AM is Amplitude Modulation. This is discussed here. Frequency Modulation is discussed in the next section of this chapter.

Amplitude modulation occurs when two frequencies are "tied together" or locked together and one of them is varying in amplitude. The frequencies are not close together, just tied together in some way. This is explained later.

Of the two related frequencies, the higher frequency is varying in amplitude periodically. It is the Carrier Frequency. The lower frequency is the modulated frequency. Figure 2-19 - Amplitude Modulation produces peaks around a central Carrier Frequency. Figure 2-19 shows two signals, a 20 Hz and a 200 Hz signal. A Forcing Frequency in the machine is actually varying in amplitude. It is not merely two signals coming in and out of phase to generate the waveform shape as shown in Figure 2-10. Instead, the amplitude of a forcing frequency is actually fluctuating.

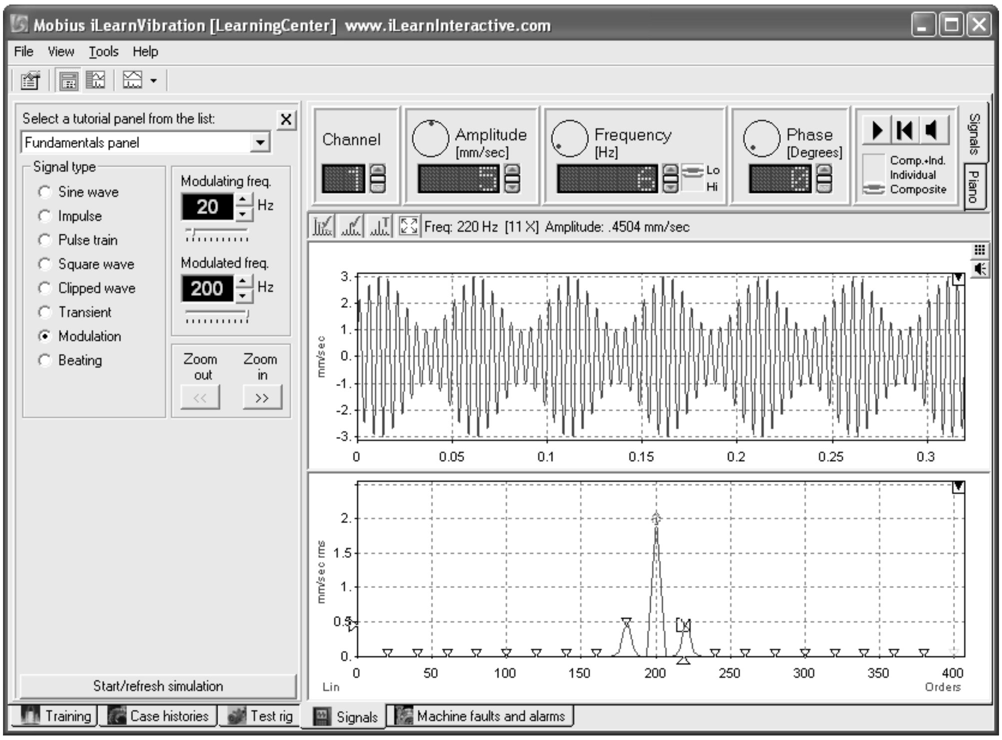


Figure 2-19 - Amplitude Modulation produces peaks around a central Carrier Frequency.

In the spectrum, the center peak is the Carrier frequency and it is surrounded by equally spaced peaks that are the modulated frequency and are called sidebands. The sideband spacing is equal to the modulated frequency, which in this case is 20 Hz.

It is typical to find multiple peaks on either side of the carrier frequency.

AM RADIO

In order to transmit the radio signal over long distances, and to distinguish one station from another, it is necessary to transmit using a high frequency (~500 - 1700 kHz). This is the "carrier".

The music and DJ babble is superimposed - or modulated - on the carrier.

When the signal gets to the radio, the carrier is removed (it is demodulated), leaving the lower frequency music and talk.

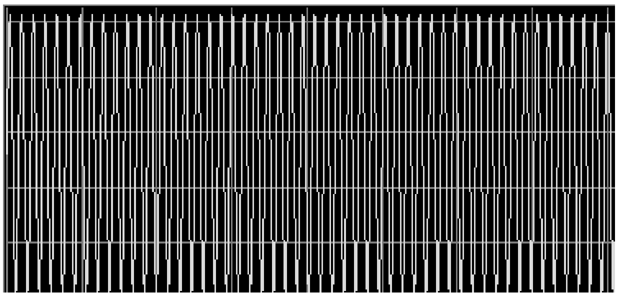


Figure 2-20 - The Carrier Frequency

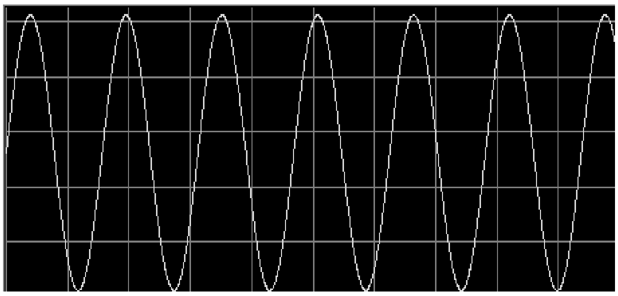


Figure 2-21 - The Modulated Signal

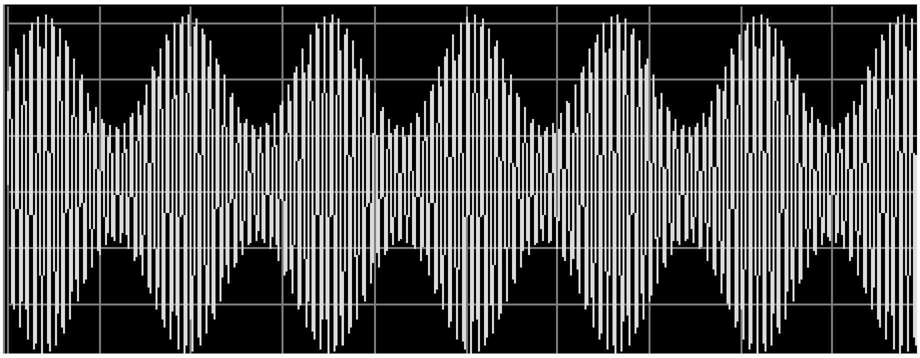


Figure 2-22 - The Amplitude Modulated signal.

Looking closer, the amplitude of the high frequency is varying with time periodically. The amplitude of the higher frequency signal is increasing, then decreasing, increasing, and so on. Could that happen inside a rotating machine? It sure can.

CLASSIC MODULATION – INNER RACE FAULT

Consider a ball bearing as it turns and there is a spall on the Inner Race. Since the inner race is locked onto the shaft, it rotates with the shaft. The balls impact the spall on the inner race every time one rolls across it. (The balls impacting the spall create a transient so there will be harmonics) The inner race spall is constantly moving in and out of the load zone as the shaft turns. Therefore the impacts of the balls on the spall vary in amplitude.

So, the balls striking the spall on the inner race are doing so periodically at the Ball Pass Inner Race Frequency (BPI). This is the Carrier Frequency which varies in amplitude as the shaft turns.

The inner race is "tied to" or locked onto the shaft. It is this frequency, then that is the modulated frequency, generating the sideband spacing. Sidebands indicate that modulation exists.

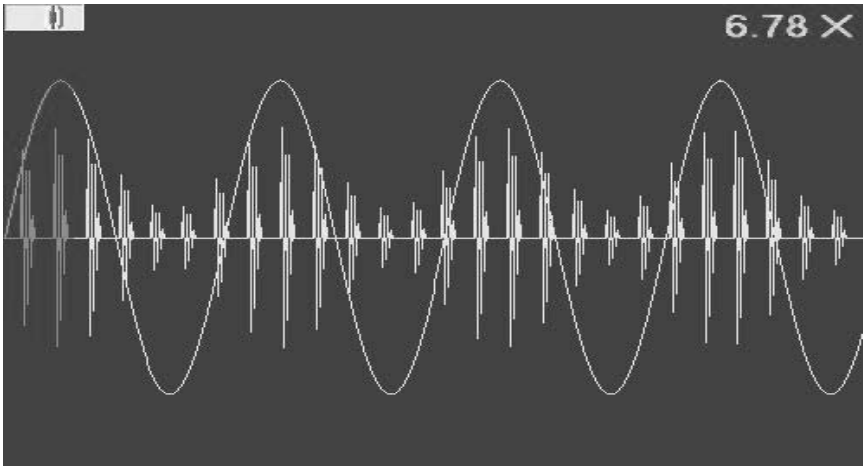


Figure 2-23 - The low frequency is 1x (the modulated frequency) the fluctuating, higher frequency is the inner race defect frequency.

It is not the purpose now to learn to diagnose bearing faults, rather to understand that Amplitude Modulation is very common when diagnosing machines. Frequency modulation is less common but does occur.

AMPLITUDE MODULATION - GEARS

Gears often generate Amplitude Modulation. If a gear is eccentric the forces will vary as the gears mesh. If a gear is on a bent shaft, the same situation occurs. Misaligned gears, (especially helical gears) tend to vary in load each shaft revolution. The carrier frequency is the gearmesh frequency (the shaft speed x the number of teeth on that gear).

If there is modulation there will be sidebands. But what will the sideband spacing be? Recall that amplitude modulation occurs between frequencies that are "tied together" or locked together. A gear is locked onto its shaft which is a different frequency than the gearmesh frequency. So the sideband spacing (modulation frequency) will be the shaft speed of the offending gear.

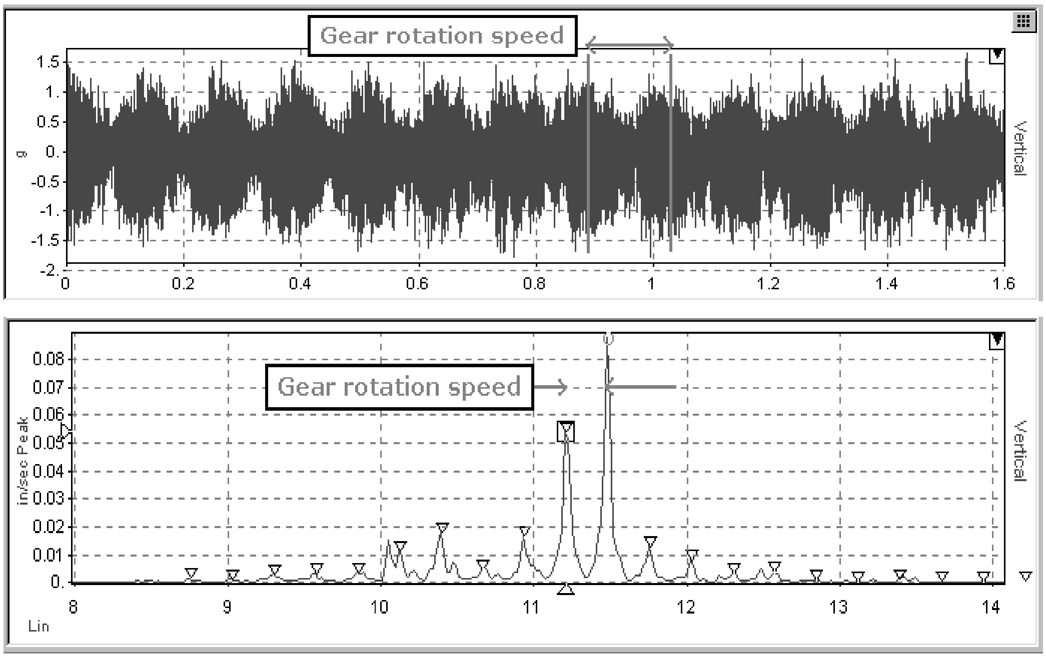


Figure 2-24 - Gearmesh Frequency is the carrier frequency. Shaft speed is the modulation frequency, generating the sidebands.

DISTORTED MODULATION

When an amplitude modulated signal is distorted there will be sidebands due to the modulation. But there will also be harmonics due to the distortion.

For example, an inner race bearing defect will cause impacting. If those impacts are distorted, then there will be harmonics of that BPI (Ball Pass frequency Inner race).

Harmonics of a modulated frequency reproduces the “family” of peaks centered at a multiple of the BPI.

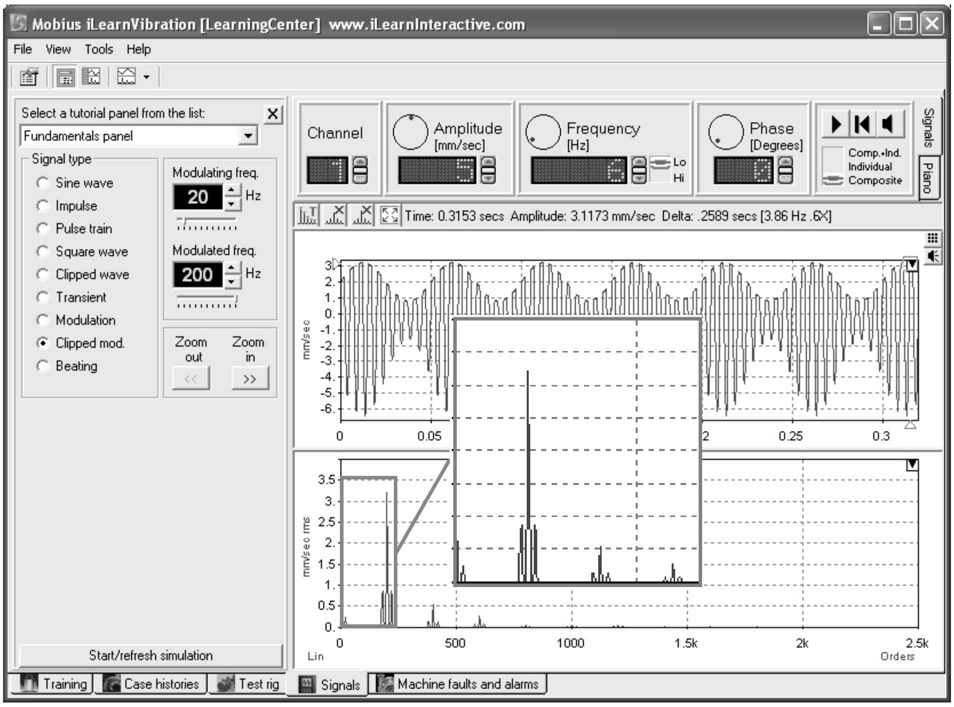


Figure 2-25 - Distorted Modulation generates the harmonics of the "families" of peaks.

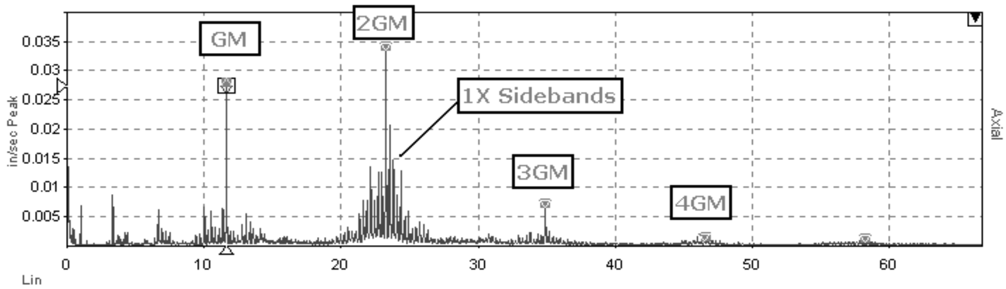


Figure 2-26 - Typical results of Distorted Modulation. Harmonics of gearmesh frequency and multiple sidebands. A "skirt" of sidebands is common.

Frequency Modulation

Frequency Modulation is common in machinery analysis. It differs from Amplitude Modulation in that the frequency varies in Frequency Modulation. This occurs when a machine changes speed periodically. It could be due to process loading or other causes. It occurs in gear boxes when they are set incorrectly allowing a little too much play. The gears rock back and forth as they turn, changing the frequency periodically.

The result of the frequency modulation in the spectrum is similar to amplitude modulation except that generally there will be a lot more sidebands.

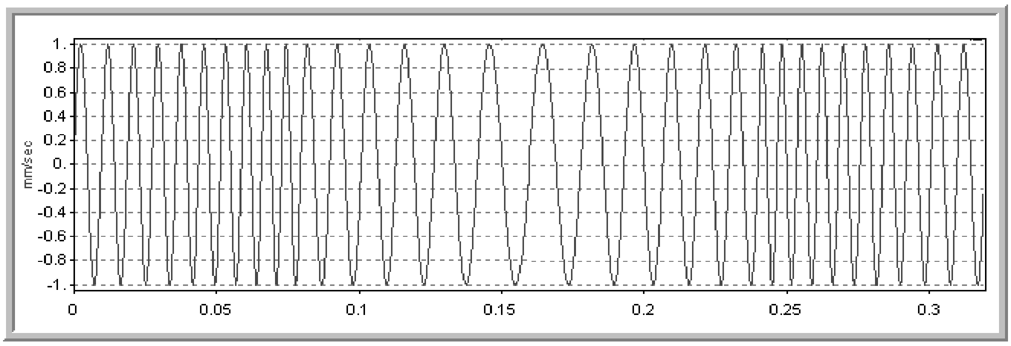


Figure 2-27 - Frequency modulation occurs when a frequency changes periodically.

Noise

Noise can be viewed a number of ways in vibration analysis:

- A source of random, non-periodic or external vibration that negatively affects our vibration readings
    - Signal problems, external machines, process sounds, etc.
- An indicator of a fault condition
    - Bearings, lubrication, cavitation, turbulence, etc.

We have to deal with noise during the measurement process:

- More averages reduces the impact of noise
    - Non-synchronous noise is removed with time synchronous averaging
    - Noise is averaged with linear or RMS averaging to improve repeatability
- Better dynamic range, transducer sensitivity, and measurement electronics improve signal-to-noise ratio (S/N)

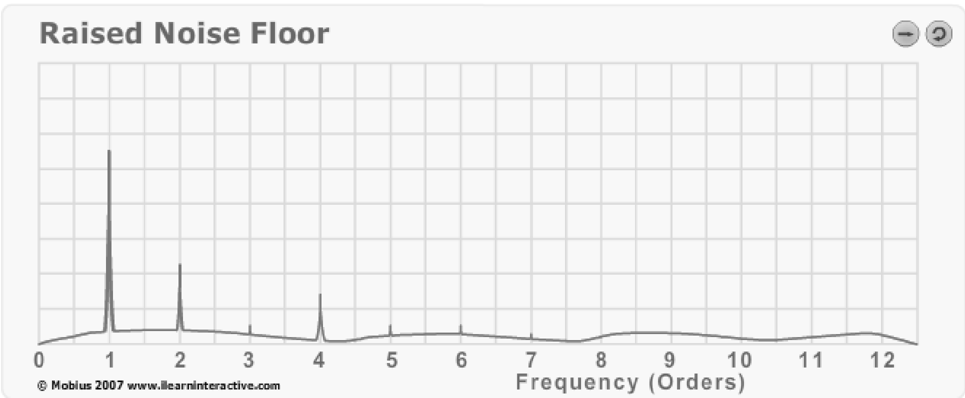


Figure 2-28

Beating

There are many instances in a plant when there is a droning sound that gets louder and then softer. The period between the loudest points could be from 10 seconds or (longer) or as little as a half second or less. The closer spaced occurrences may sound more like a throbbing sound than a droning sound.

While collecting data or monitoring a machine a peak in the spectrum may rise and fall periodically.

What is the source of these common occurrences?

It could be due to a phenomenon known as "Beating," two signals with similar frequencies. It can come from two identical machines near each other that run at exactly the same speed. The vibration from one machine transmits to another.

Let's start with the basic concepts. If one of two identical machines is running at 10 Hz and all the vibration is at running speed, then the waveform might be like the one at the right.

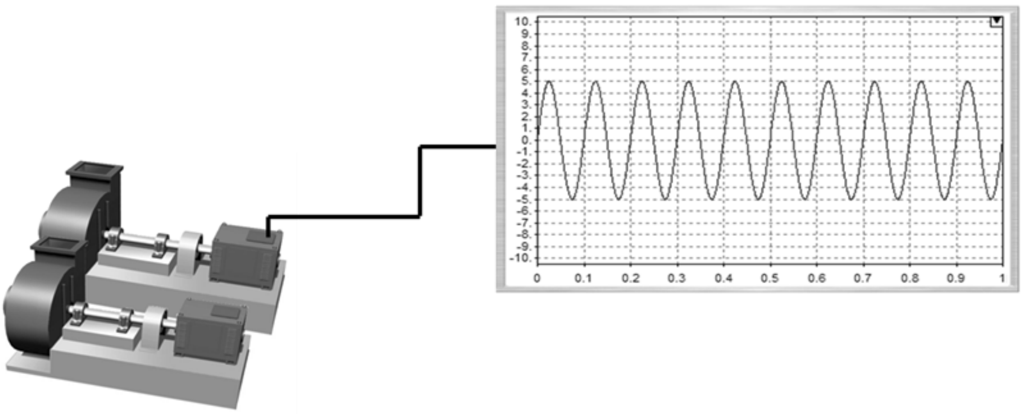


Figure 2-29 - Waveform of 10 Hz running speed.

In a one second time waveform there would be 10 cycles. We could hear a 10 Hz "tone" and the amplitude would be constant.

If the second machine is started, too, and they were exactly the same speed and both perfectly in phase, the waveform measured on the machine would be the sum of the two. The amplitude would be twice as high.

The sound coming from the machines would still be the 10 Hz frequency, but would be twice as loud.

So the frequency would not change, but the amplitude would double.

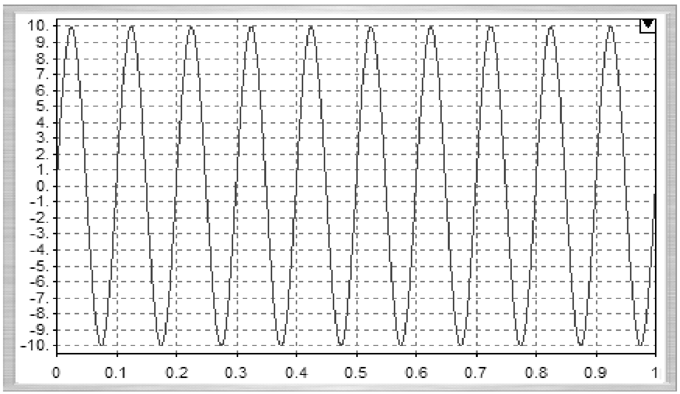


Figure 2-30 - The waveform from the two machines in phase with each other. The amplitude has doubled but the frequency is the same.

Now suppose the two identical machines were 180 degrees out of phase. What would the result be? It would be very quiet because the signals would cancel each other out. The waveform would be a flat line. But this is unrealistic.

The chances of the machines being 180 degrees out of phase are remote, and there would be losses in vibration from one machine to the other means that there would really be some amplitude to measure and hear.

To cancel each other out, the two machine speeds would have to be identical.

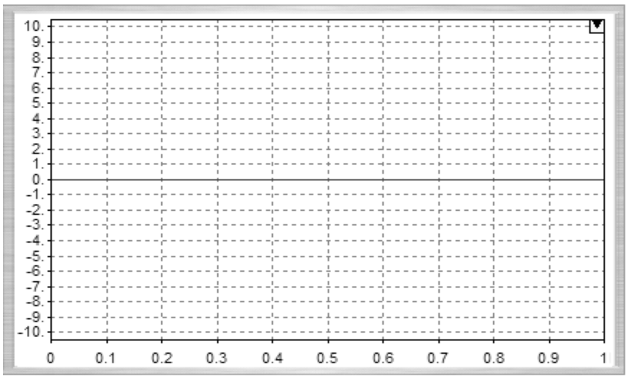


Figure 2-31 - The waveform from 2 identical machines 180 degrees out of phase results in a flat line.

What would happen if the two machine speeds were different?

Example: If two machines were running at different speeds, 600 RPM and 660 RPM, (10 Hz & 11 Hz), then what would happen? What would be measured and what would be heard?

Figure 2-32 shows the two individual signals in the top graph. The composite signal is in the bottom graph.

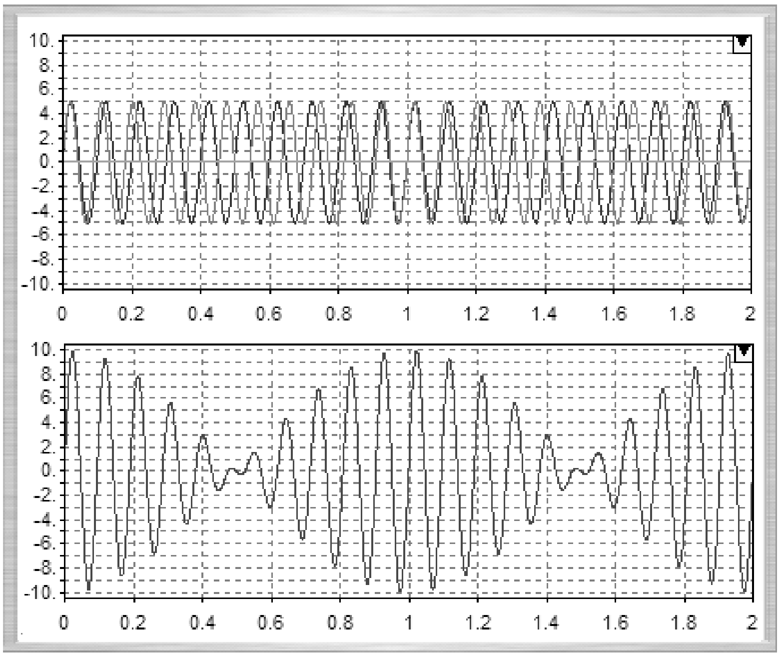


Figure 2-32 - A 10 Hz signal and an 11 Hz signal are in the upper graph. The composite signal is in the lower graph.

The composite graph is what would be measured on the machine. Note that the time waveform is rescaled to two seconds long.

What would be heard is "beating" - a droning sound with a duration of one second. The amplitude would rise from zero to full amplitude and back to zero in one second.

Figure 2-33 shows this more clearly. The top graphs shows that the two signals are in phase at one point and 0.5 seconds later they are 180 degrees out of phase. In between those times they are somewhere between 0 and 360 degrees. When they are in phase, the composite signal adds the signals giving twice the amplitude of each individual signal. When the signals are 180 degrees out of phase, the composite signal cancels them out. (There is zero amplitude in the graph at that point.)

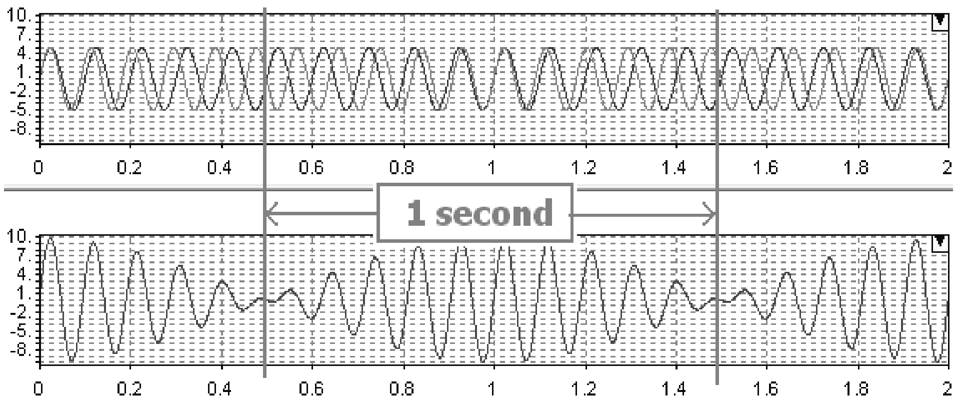


Figure 2-33 - The two signals add and cancel out depending on whether they are in-phase or out of phase.

Because the signal is periodically varying from zero amplitude to a maximum amplitude, we hear a pulsation – the pulses have a 1 hertz cycle – that is, there is one second between pulses. Note that the two signals, 10 Hz and 11 Hz, are separated by 1 Hz.

What if the speeds were closer together?

If one signal was at 10.5 Hz and the other at 11 Hz, what would it look like and how would it sound?

Figure 2-34 is the two signals just 0.5 Hz apart. Notice how the time for one cycle is now two seconds.

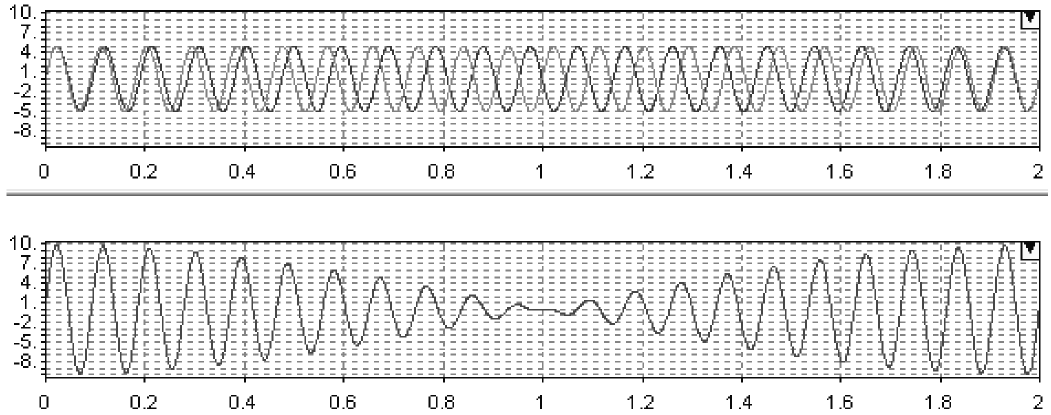


Figure 2-34 - the cycle time for the 0.5 Hz beat is 2 seconds.

The beat frequency is the difference in frequency between the two signals. So this is a half Hz beat. How is the time figured?

Recall from chapter two that the period for one cycle is the inverse of the frequency. The frequency is 0.5 Hz. Divided into one it yields 2 seconds.

Figure 2-35 shows 20 seconds in the time waveform. 10 cycles are clearly seen.

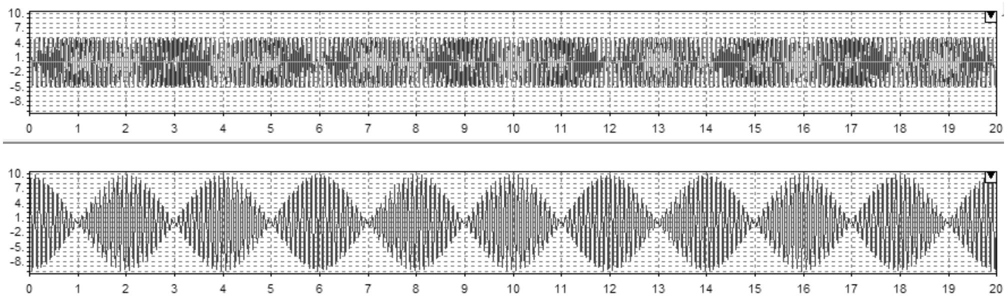


Figure 2-35 - 10 cycles in the 20 second time waveform

Two frequencies closer together

Figure 2-36 shows the result of two closer frequencies… 10.9 Hz and 11 Hz. The beat frequency is 0.1 Hz so the time for one cycle is 10 seconds (1/0.1).

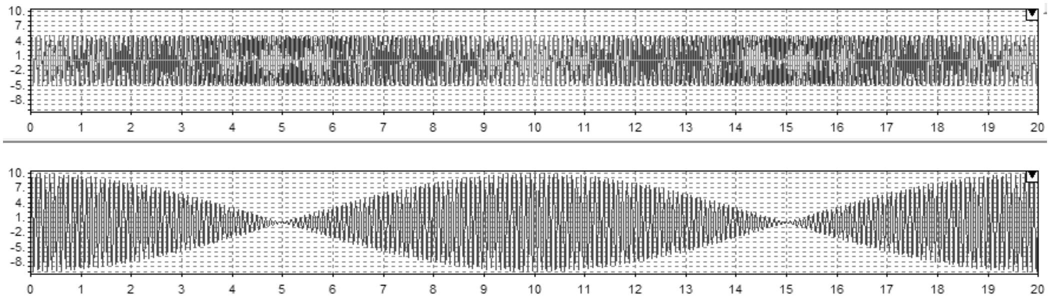


Figure 2-36 - A 10.9 Hz signal and an 11 Hz signal produce a waveform cycle 10 seconds long.

The pulse that would be heard is 10 seconds long.

Different Amplitudes

If the same two frequencies have different amplitudes, how would it differ? Figure 2-37 shows the result of the two signals if one signal is less than previous.

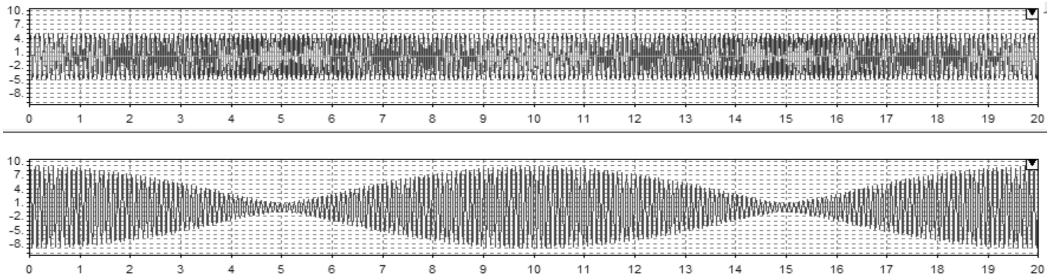


Figure 2-37 - the two signals are not equal amplitude. They don't cancel to zero nor go as high in amplitude.

When the two signals are 180 degrees out of phase they do not sum to zero. When they are in phase their sum is not as high in amplitude.

Beating vs. Amplitude Modulation

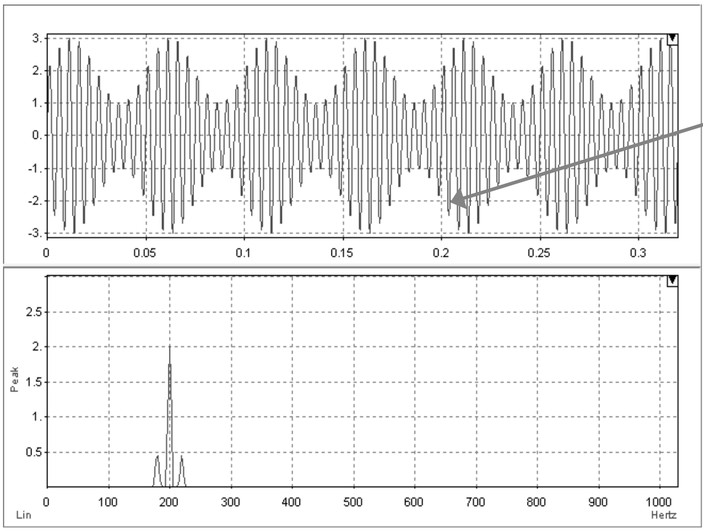
The waveforms of Beating and Amplitude Modulation may look similar but they are vastly different.

Amplitude modulation occurs between two frequencies that are far apart.

Beating occurs between frequencies that are close together. (Usually less than 4 Hz apart.)

In Amplitude modulation, one signal (the carrier) is being varied periodically by a second signal – this produces sidebands. In Figure 2-38 the 200 Hz carrier signal varies in amplitude 20 times per second.

Beating sums two similar frequencies as they go in and out of phase.



200 Hz "carrier" modulated by20 Hz – the 200 Hz varies in amplitude 20 times per second

Figure 2-38 - Amplitude modulation - a 200 Hz carrier frequency varying in amplitude 20 times per second.

Spectrum Analysis of Beating

It is difficult to tell from a stored spectrum if beating is occurring. It could be that a particular frequency changes drastically from one reading to another which could be the result of capturing the data when the signals were in phase during one measurement and out of phase the next measurement.

It is much easier to spot the beating while collecting measurements at the machine. If resolution is not sufficient to separate the two signals, a peak will be rising and falling. It could be due to the beating. If the two frequencies are 1 Hz apart, the peak will rise and fall once per second.

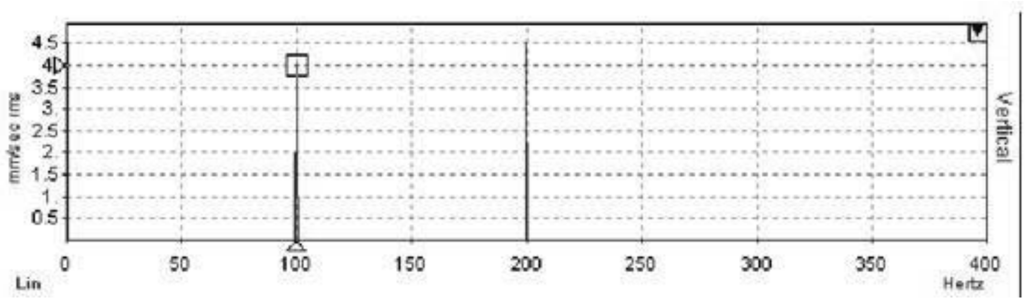
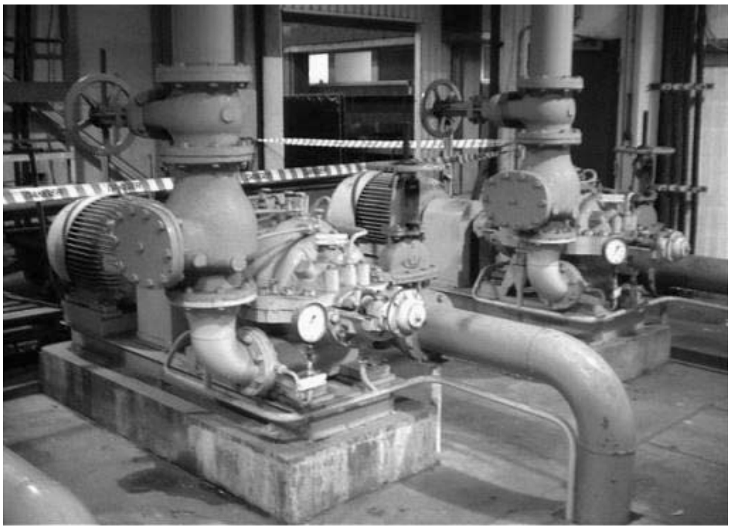


Figure 2-39 - If a peak is rising and falling during data collection, it could be due to beating.

How does this affect us in reality?

When two machines are running at very similar speeds, and there is a good mechanical transmission path between them, then beating could occur. The vibration level on both machines could cycle between quite low levels to quite high and therefore destructive levels.



In one recent example, the vibration level was cycling from 3 mm/sec (0.17 in/sec) to 20 mm/sec (1.1 in/sec) over a 10 minute period (the vibration level was normally about 10 mm/sec (0.56 in/sec). This is very destructive on the machine.

Correcting Beating

To correct the situation, the speeds must be controlled, or find out why the speed of one of the machines has changed. In the example above, one of the machines had been recently balanced, and when one of the balance weights was removed, the beating stopped.

The second way to correct the situation is to examine how the vibration is being transmitted from one machine to the other. In most cases, there will have been some recent change to one or both machines which resulted in this beating phenomenon. Isolating the bases and foundations helps to remove the transmission path between the two machines.

Intermodulation

When two closely spaced signals mix, they can beat, or they can modulate. Peaks may show up in the spectrum at Sum and Difference frequencies.

Assume there are two signals at frequencies of 99 Hz and 100 Hz. They may “Sum” together producing a peak at 199 Hz.

And on the other hand, they may also subtract from each other and produce the "Difference" frequency at 1 Hz. See Figure 2-40.

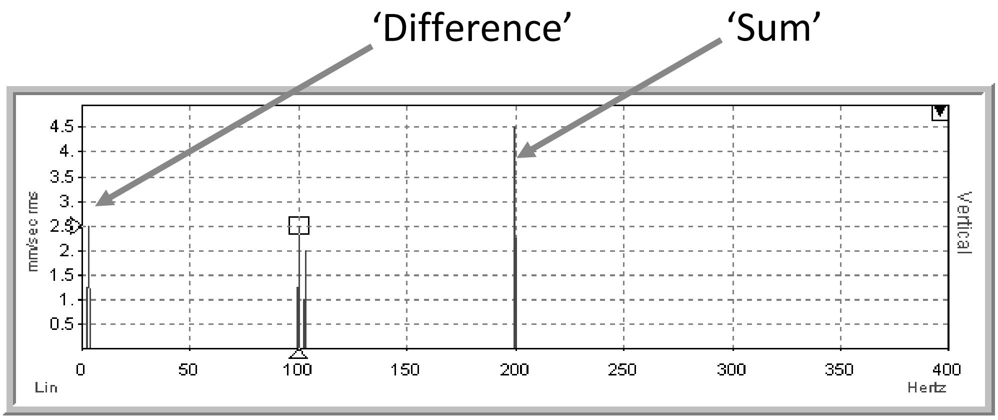
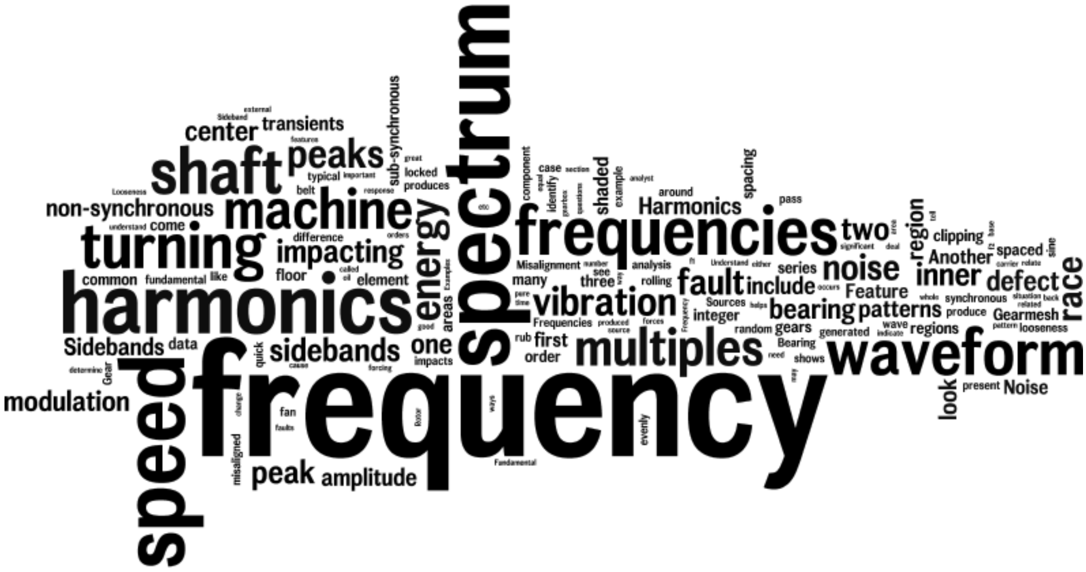


Figure 2-40 - 2 close frequencies may produce Sum and Difference frequencies.

It is common to see the beating and the peaks at Sum and Difference frequencies in the spectrum.



Chapter 3

Understanding Spectra

Objective:

- Understand the features of vibration
    - Pure frequency
    - Harmonics
    - Sidebands
    - Noise
    - Sum/difference frequencies
- Define spectral regions

There are three ways to approach fault diagnosis:

1. Understand the forces on the machine and determine how the vibration will change as a result.
2. Understand the common features of the vibration spectrum and "reverse engineer" the fault based on the patterns observed.
3. Remember all of the patterns for each fault condition (or keep a wall chart close by) and try to recognize the patterns

In order to be a successful vibration analyst it helps to understand why the vibration patterns change the way they do. What generates peaks in the spectrum? Where do harmonics come from? What is noise? Where do sidebands come from?

Now we will look at it from the opposite point of view – recognizing features in the spectrum and deciding what to do next.

We will review vibration from distinct sources, harmonics, sidebands, the noise floor, and sum/difference frequencies. If we can identify these patterns and relate them back to the root cause, then we can determine what is happening inside the machine

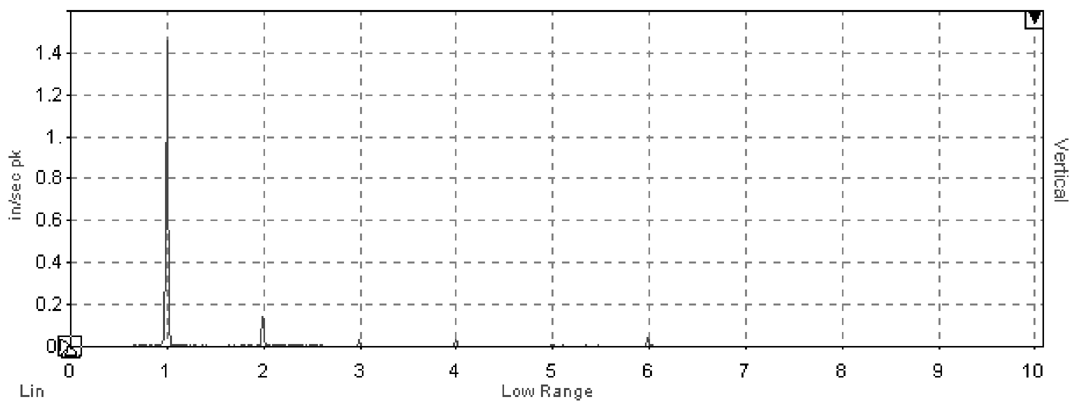


Figure 3-1 The vibration spectrum

Feature one: Pure frequency

As we learned in "Understanding signals" certain peaks in the spectrum are generated by the machine directly: unbalance, blade-pass-frequency, etc. We need to identify the frequency and relate it back to the machine and we need to know how it is related to the turning speed of the machine. We can then see if there are any harmonics or sidebands or sum-and-difference frequencies.

Feature two: Harmonics

One of the most common patterns seen in spectral data is harmonics. A waveform that is repetitive (periodic) but not a pure sine wave creates harmonics in the spectrum.

Harmonics are a series of evenly spaced peaks that are multiples of the first peak in the series. The first peak is called the fundamental frequency.

In Figure 3-2 the first peak (with the square on top) is the fundamental frequency of 1785 CPM which is shaft turning speed. The remaining peaks marked with a triangle shaped cap are integer multiples of that frequency. These are the harmonics. The Fundamental Frequency in the series is often referred to as the first harmonic.

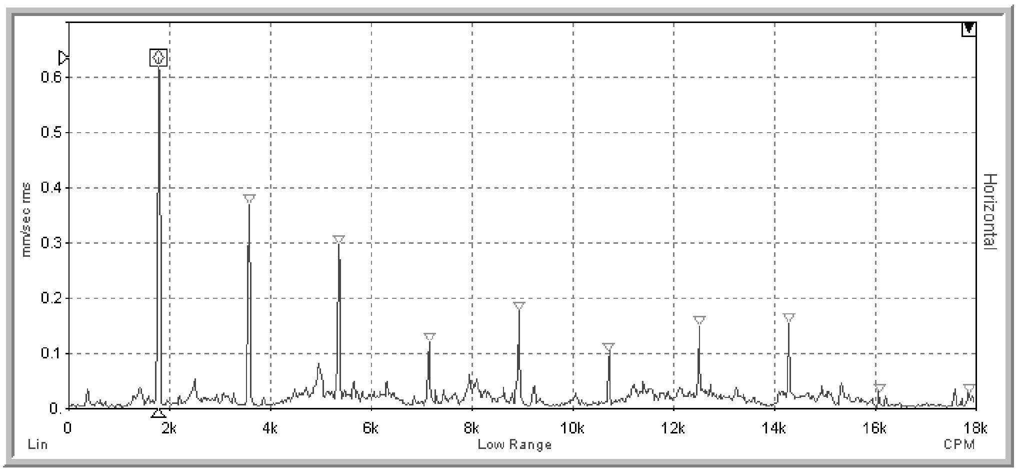


Figure 3-2 Harmonics are multiples of any frequency. The first in the series is the fundamental frequency.

The most common harmonics are multiples of the shaft turning speed. However, harmonics can be multiples of any frequency. Figure 3-3 shows harmonics of a non-synchronous peak which could be a bearing frequency. Harmonics can be multiples of belt rate frequency, or other frequencies.

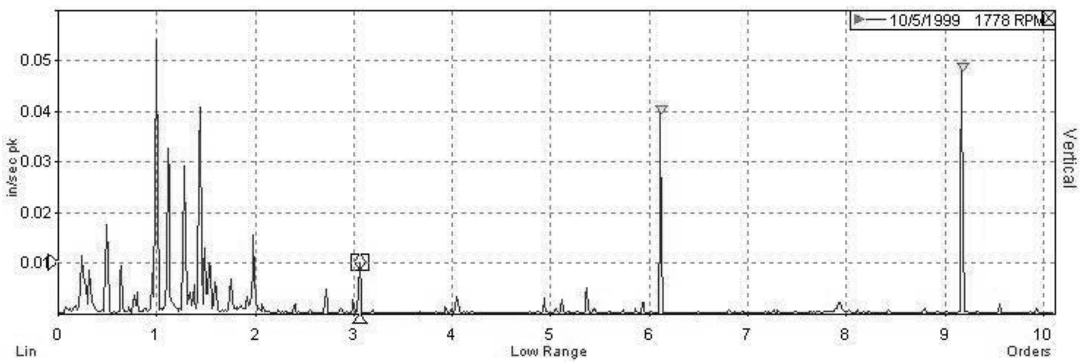


Figure 3-3 Harmonics can be multiples of non-synchronous peaks such as this. Bearing fault?

Recall from the "Understanding Signals" chapter, that harmonics in the spectrum is only half the story. Harmonics are generated when there is clipping, transients, or random impacting in the waveform. Whenever harmonics are present, it is a good idea to look at the waveform.

The waveform associated with the spectrum in Figure 3-3 does have significant transient impacting. We would expect to see harmonics in the spectrum.

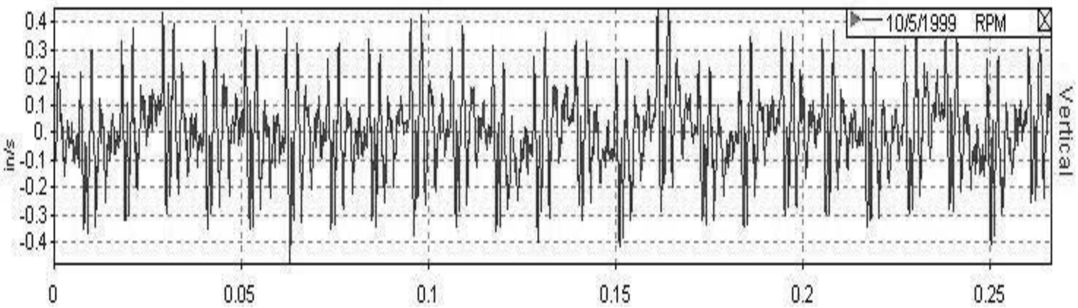


Figure 3-4 Waveform for spectrum in Figure 3-3 shows the transients.

The waveform in Figure 3-5 looks almost like a sine wave, but not quite. This is why there are some harmonics on the spectrum below it.

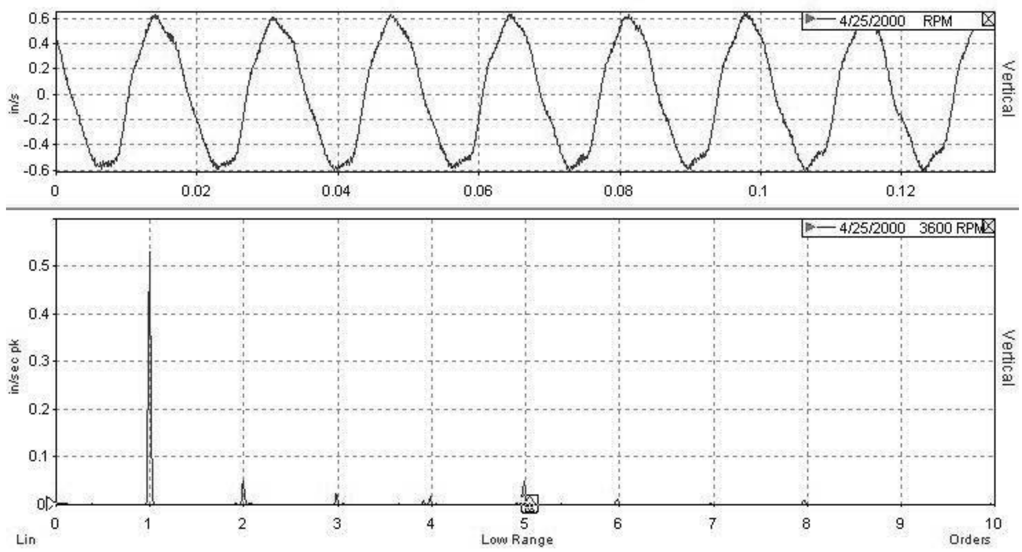


Figure 3-5 Because it is not a perfect sine wave, there are harmonics.

When the vibration is not pure, for example, it is distorted, has random impacting, has clipping or transients, there will be harmonics. Figure 3-6 has significant random impacting and produces a spectrum with many harmonics.

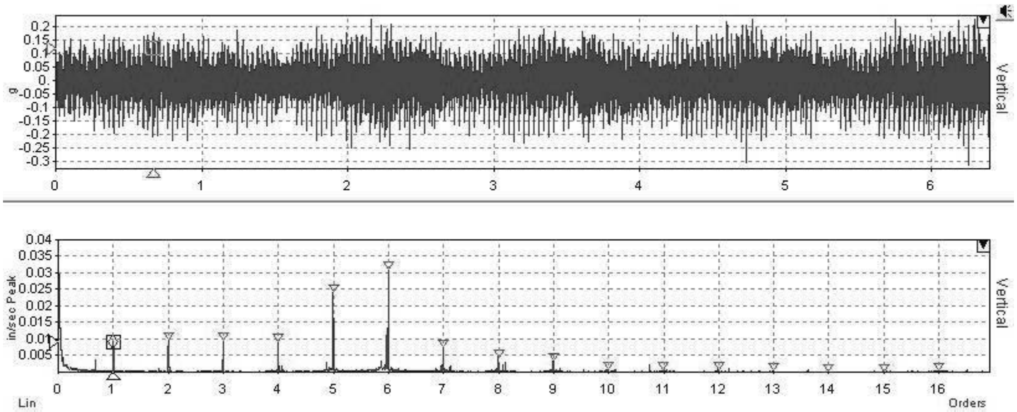


Figure 3-6 Impacting produces harmonics

Figure 3-7 is typical of structural looseness. The waveform has impacting with clear transients. The waveform autoscales from a positive 0.4 g's to a minus one g. This is also a typical clipping situation. Clipping occurs when there is more freedom of movement in one direction than the other. In this case there is considerably more impacting downward (away from the sensor) as the bearing pedestal bangs against the base.

Notice how there are harmonics of turning speed to 25 orders.

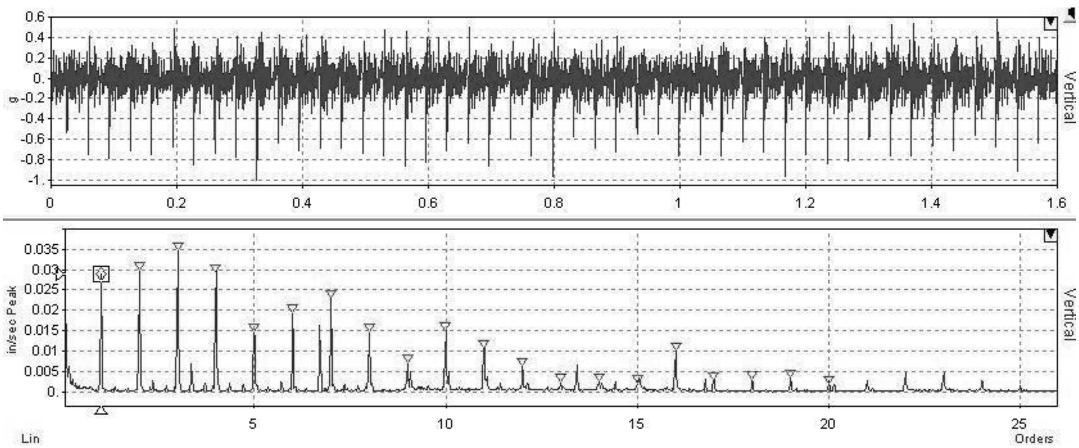


Figure 3-7 The impacting, transients, and clipping in the waveform produce harmonics of turning speed to 25 orders.

Conditions that can produce harmonics include:

- Looseness
- Misalignment
- Bearing Wear
- Gear Faults

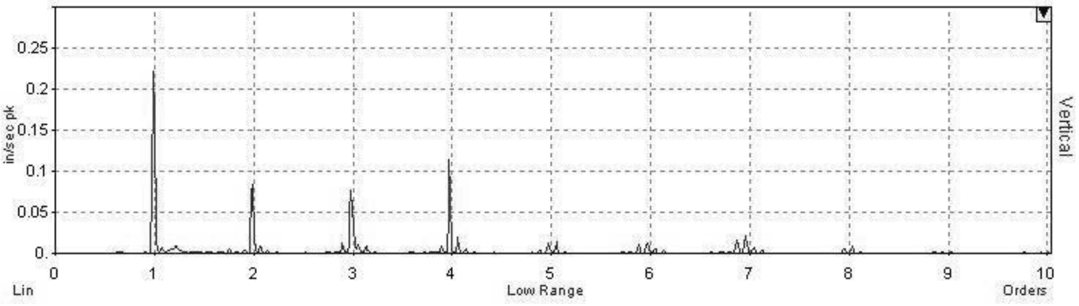


Figure 3-8

Feature three: Sidebands

Sidebands are a very important phenomenon to look for (and understand) in machinery analysis. Sidebands are a result of **amplitude modulation** between two signals. Sidebands are common when studying rolling element bearing, gearbox, electrical, and certain other fault conditions.

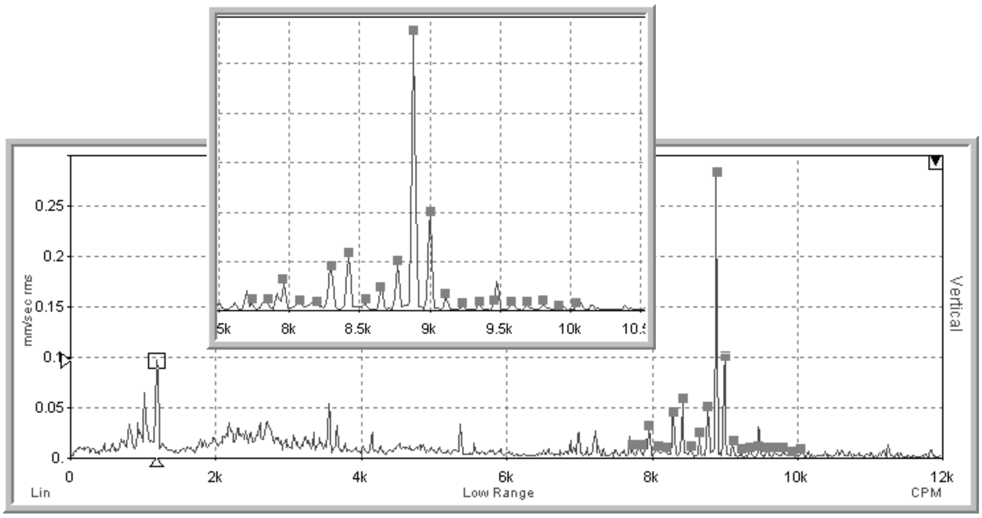


Figure 3-9 Sidebands are evenly spaced peaks around a center frequency or peak.

In the spectrum, sidebands look like evenly spaced peaks, centered on another peak called the "**center frequency**". Depending upon the situation, we may either be interested in the frequency of the "center frequency", or the frequency of the spacing between the sidebands, or both.

The center frequency is the "**carrier**" frequency and the sidebands are the "**modulation**" frequency.

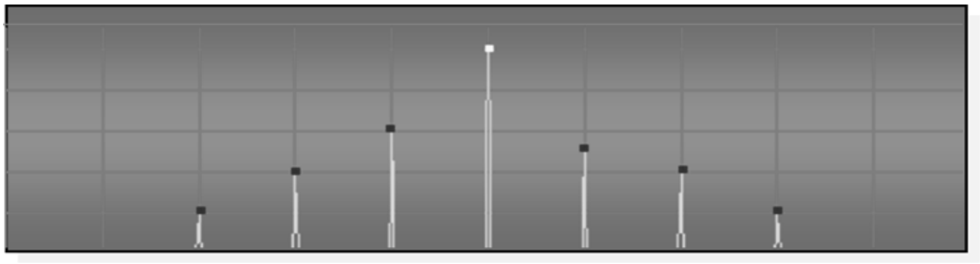


Figure 3-10 Sidebands around a center peak

Examples of Carrier Frequencies include:

- Gearmesh Frequency
- Bearing inner race frequency
- Rotor bar pass frequency

Examples of the Sideband Frequencies include:

- Running speed
- Fundamental train frequency (cage frequency)
- Pole pass frequency

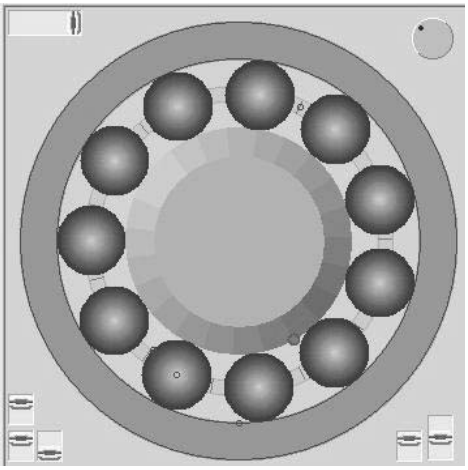


Figure 3-11

An example of modulation is an inner race defect on a rolling element bearing. In Figure 3-12 the waveform pattern is produced from the bearing with the inner race defect. The waveform is separated into its two component frequencies.

The sinusoidal waveform is the shaft turning speed and remains constant in amplitude. The higher frequency impacts are from the inner race defect. These impacts vary in amplitude because the inner race defect moves in and out of the load zone since the inner race is tied to the shaft.

Both frequencies are produced because these two frequencies are locked together. The inner race is locked onto the shaft, so it has the shaft speed associated with it. The frequency of the balls impacting that defect is the calculated inner race defect frequency.

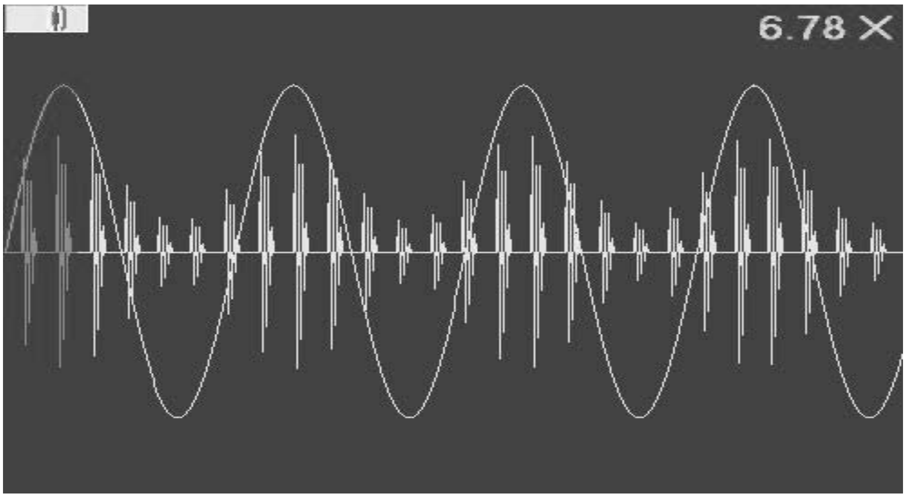


Figure 3-12 Inner race defect produces amplitude modulation

Another example is from Gear Misalignment. The modulation occurs in a gearbox due to two frequencies locked together. The two frequencies are the shaft turning speed and Gearmesh frequency. (Gearmesh frequency is the number of teeth on a gear times the speed of its shaft.)

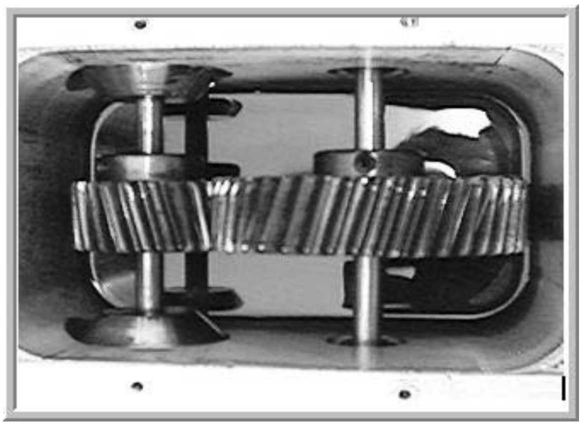


Figure 3-13 Gear box with misaligned gears

Because the shafts are misaligned, the forces against each other vary as the shaft turns producing varying amplitude. This varying amplitude is evident in the waveform pattern. See Figure 3-14.

The frequency of the modulation is the shaft speed of one of the gears.

The spectrum shows the Gearmesh frequency as the center or carrier frequency. The sidebands are spaced from the center frequency at spacing equal to the frequency of one of the shaft turning speeds. In this case, the sideband spacing is the frequency of the pinion.

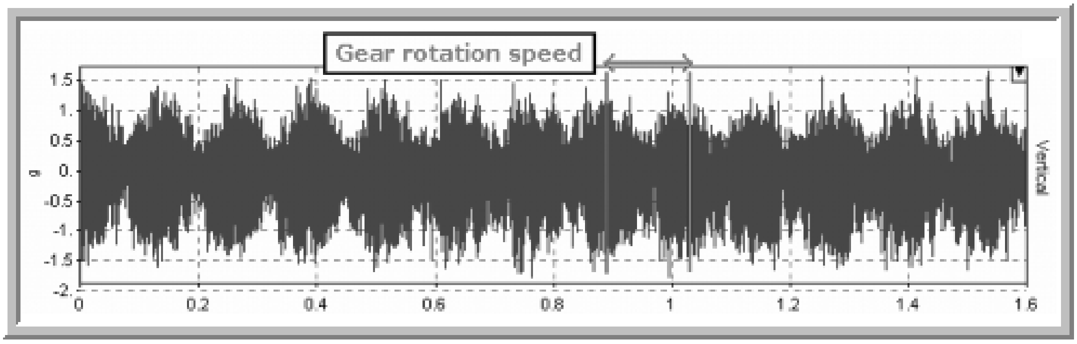


Figure 3-14 Amplitude modulation from misaligned gears.

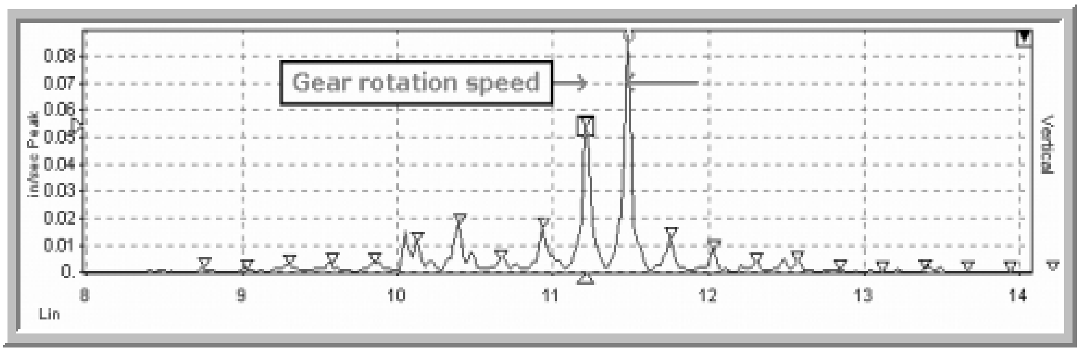


Figure 3-15 Sideband spacing around the gearmesh frequency is the shaft speed of one of the gears.

Feature four: Noise

Noise is generated by either:

- A single event in the waveform (like an impact) or
- Random vibration – a waveform that does not repeat itself

The "noise floor" of the spectrum can tell us a great deal about the machine. Noise can indicate that a fault exists (cavitation, bearing fault, rotating looseness, turbulence, poor lubrication, impacts and rubs). Noise can come from the internal process. Noise can come from an external machine.

If there was no noise the base of the graph would be flat.

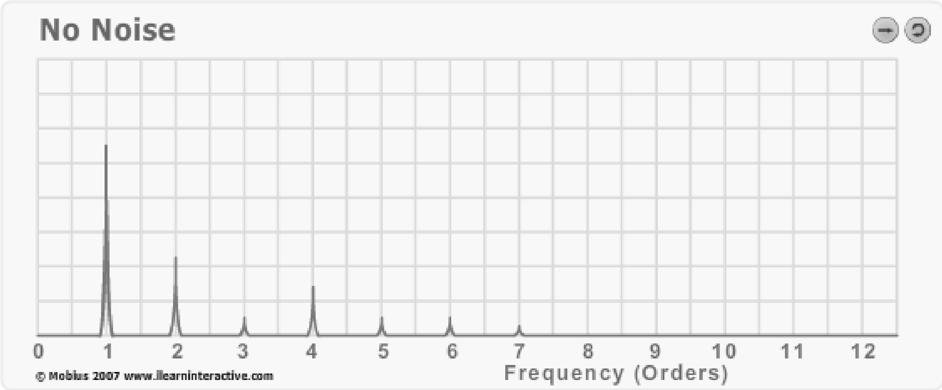


Figure 3-16

Sources of noise will cause the noise floor to rise up in different ways. The noise floor of the spectrum can tell us a great deal about the machine.

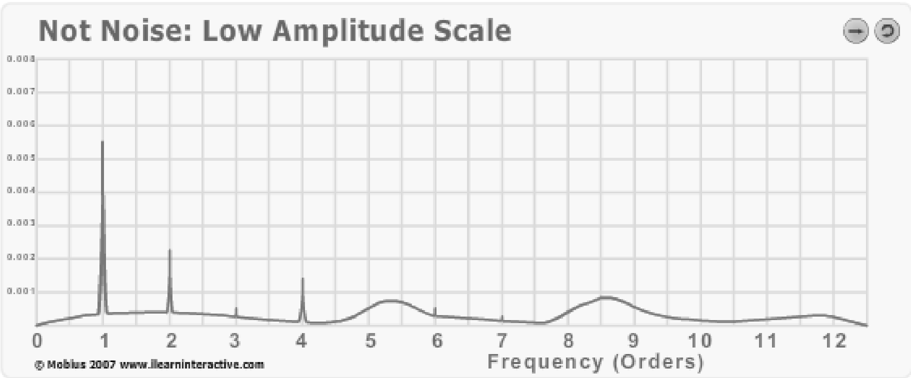


Figure 3-17

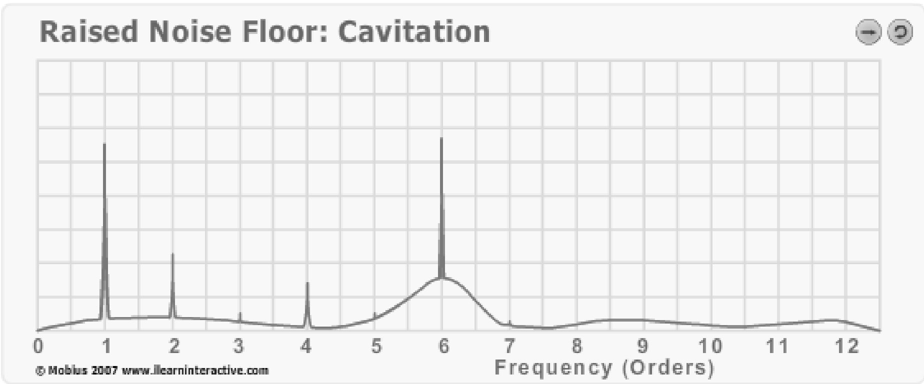


Figure 3-18

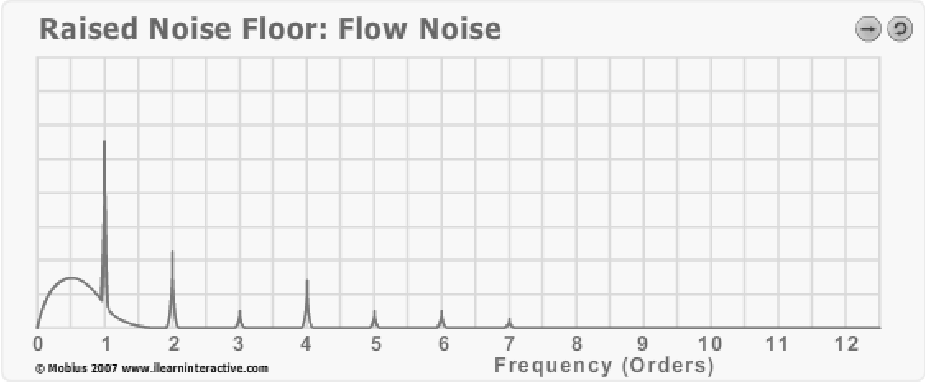


Figure 3-19

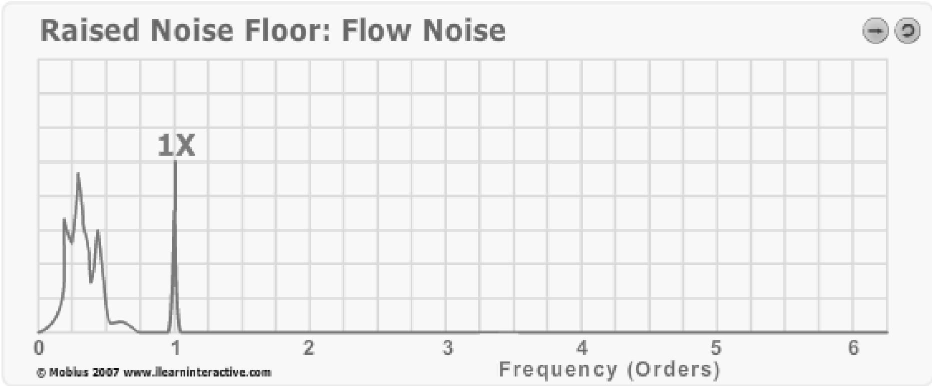


Figure 3-20

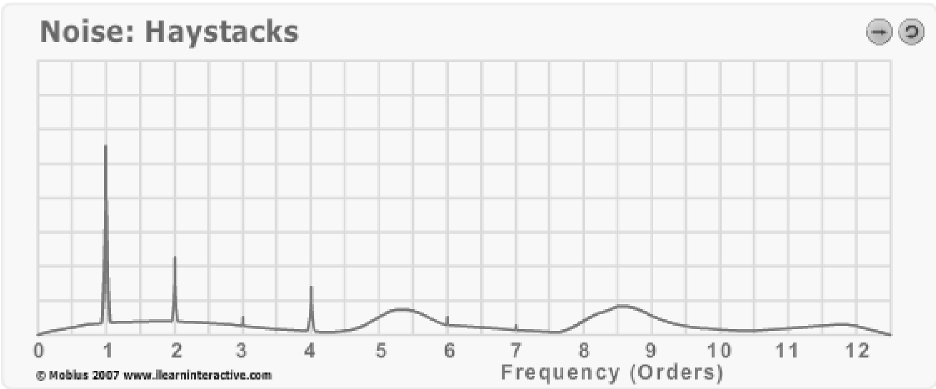


Figure 3-21

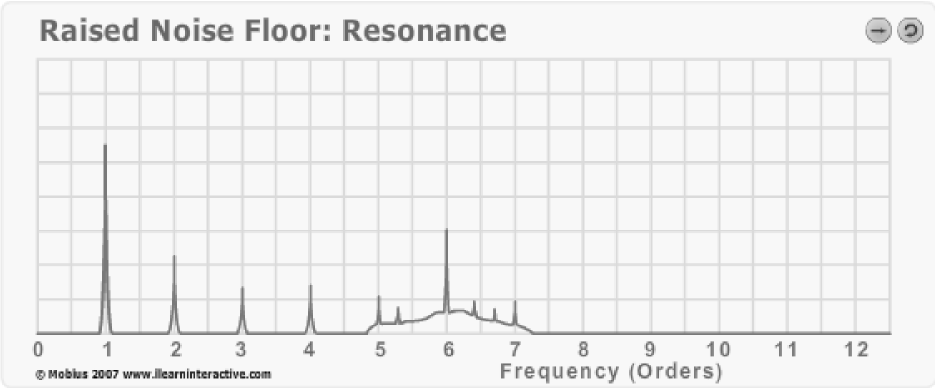


Figure 3-22

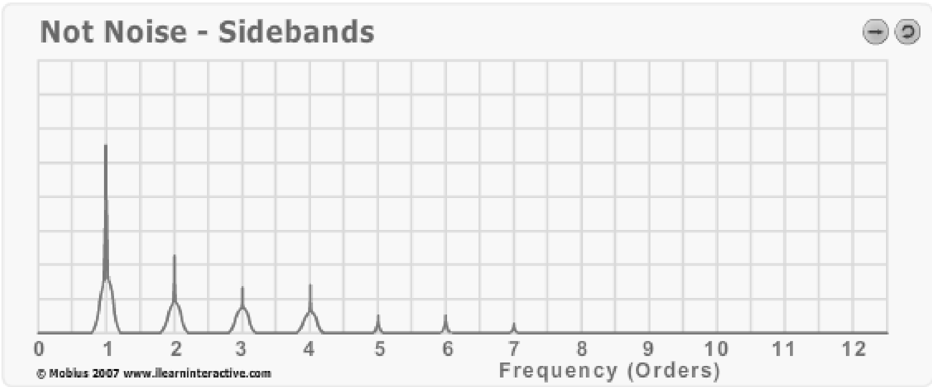


Figure 3-23

If the time record used to create the FFT has one large impact then the spectrum can look the same as if the vibration contained a lot of noise. In this case time waveform analysis is important.

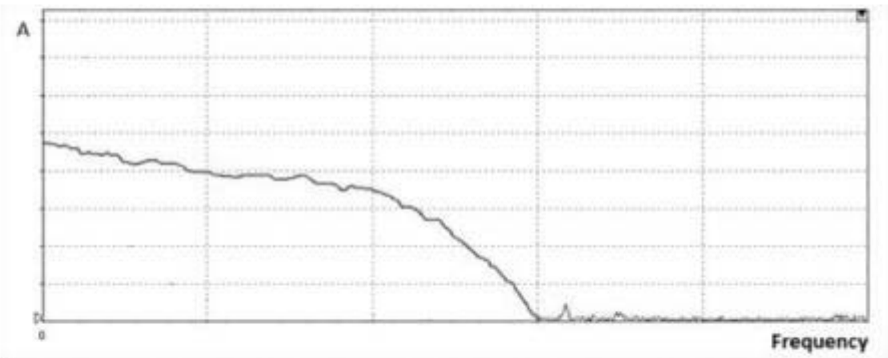


Figure 3-24

Feature five: Sum and difference

When you see peaks in a spectrum you may be at a loss to identify their source. Possibilities are unidentified forcing frequencies, external vibration, noise or harmonics or sidebands. We should consider another possibility: sum and difference frequencies due to intermodulation.

If two frequencies, f_1 and f_2, are present, the non-linear response would produce intermodulation products equal to the sum and difference of the two frequencies, that is $m f_1 \pm n f_2$, for any whole numbers m and n.*

Due to the ear's non-linear response we can hear tones that are not actually present. The same is true in a machine.

This does not indicate a fault condition; it is simply a way to explain the presence of unidentified peaks in a spectrum

Spectral Regions

A quick scan of three regions of order normalized spectra helps to categorize the fault types.

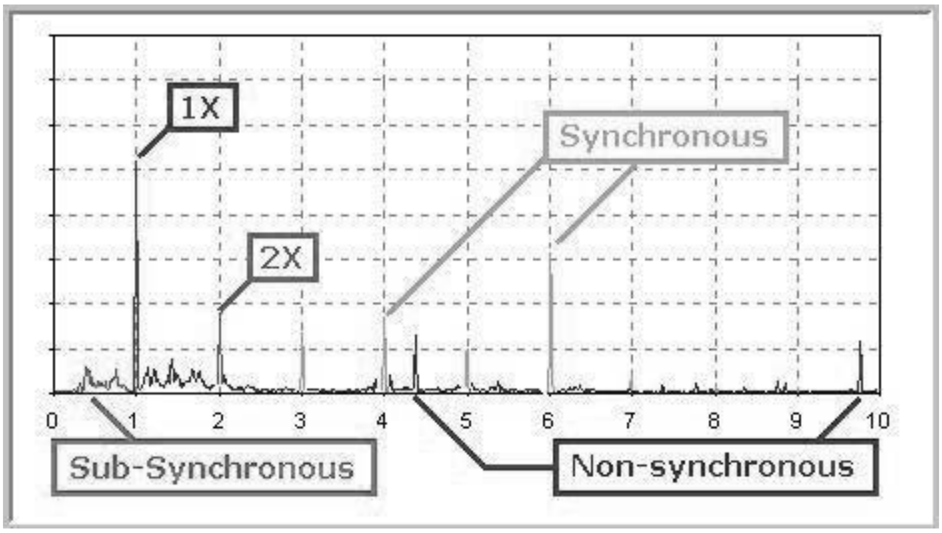


Figure 3-25 - The spectrum can be separated into 3 regions

The section on Forcing Frequencies showed how many faults are related to turning speed and its integer multiples (synchronous energy). Other faults are fractional multiples (non-synchronous) and still others show up at frequencies below the shaft turning speed (sub-synchronous). By taking a quick look at an Order Normalized spectrum, a good indication is given of the source of the peaks. Figure 3-25 is a typical spectrum with the three regions shown.

Sub-synchronous energy refers to any energy below the frequency of the shaft turning speed. The shaded area in Figure 3-26 is the sub-synchronous region of the spectrum.

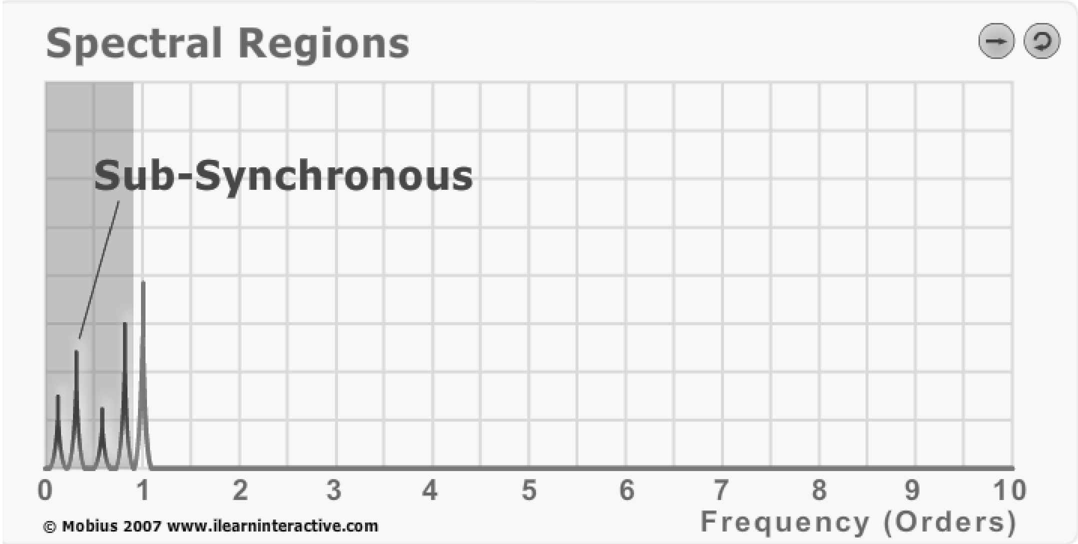


Figure 3-26 - Shaded area is the sub-synchronous region

Sources of energy in the sub-synchronous region include:

- Another component in the same machine (the fan speed of a belt driven fan unit)
- Another machine
- Belt frequencies
- Hydraulic instability in journal or sleeve bearings (oil whip and oil whirl)
- Rotor rub, shaft rub, compressor wheel rub
- Cage frequencies in rolling element bearings
- Severe looseness

The **synchronous** region is the whole number multiples of turning speed (integer multiples). The synchronous regions are marked in Figure 3-27 as shaded areas at each order of turning speed.

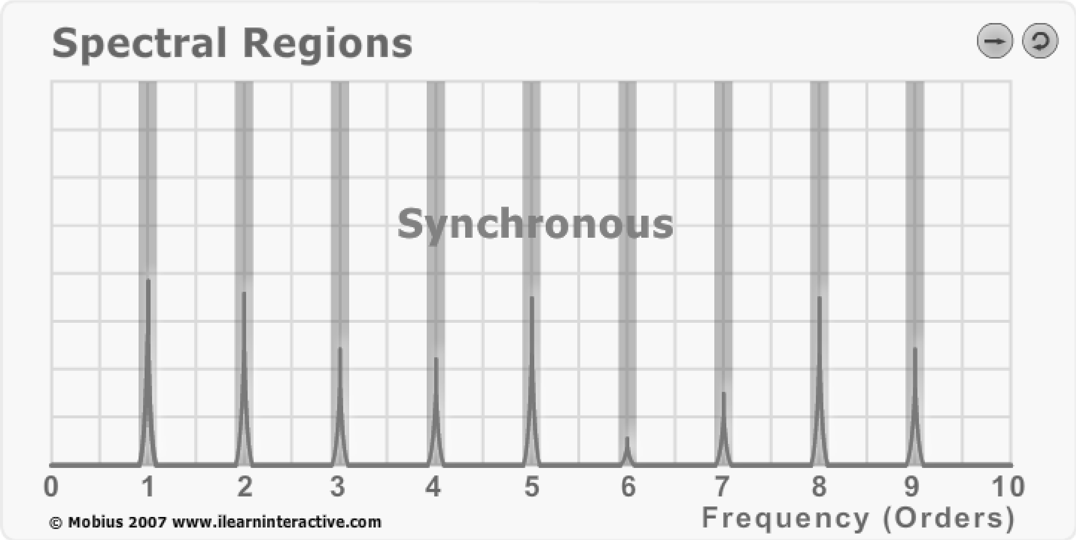


Figure 3-27 - Synchronous energy is the shaded areas at the order intervals

Many forcing frequencies generate energy at turning speed or its multiples. Some of them include:

- Unbalance
- Misalignment
- Bent shaft
- Looseness
- Blade and Vane pass frequencies
- Reciprocating motion
- Gears
- Slot frequency in motors
- Eccentricity

Non-synchronous energy is all energy above turning speed frequency that is not an exact integer multiple of turning speed. The shaded region in Figure 3-28 is the non-synchronous region.

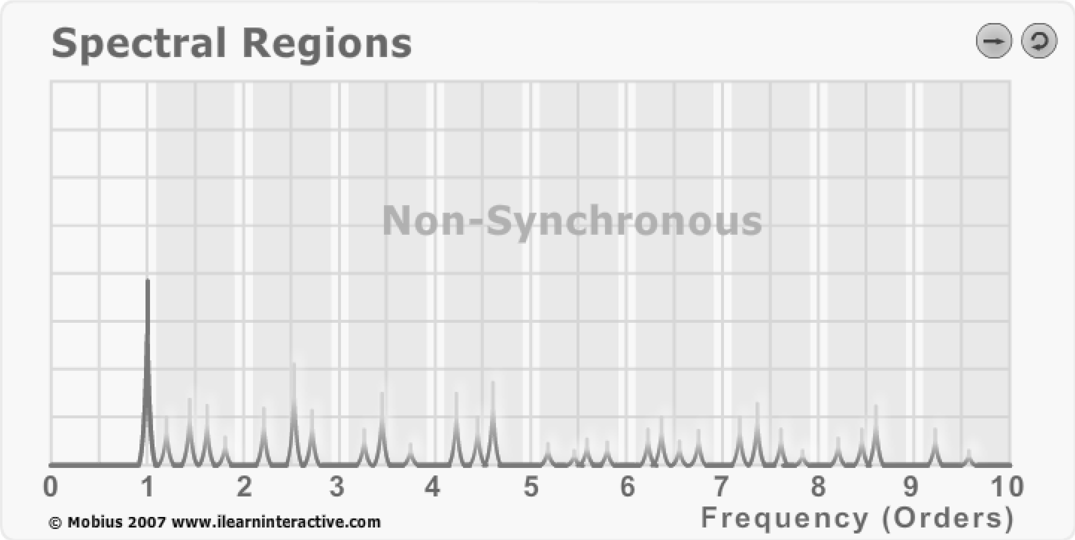


Figure 3-28 - The shaded areas are the non-synchronous areas

Non-synchronous energy is Non-integer multiples of running speed such as 3.1x, 5.65x, etc. Sources of non-synchronous energy include:

- Another component in the machine
- Multiples of belt frequency
- Rolling element bearing defects
- System resonances
- Cavitation
- Electrical
- Lube pumps
- Sliding surfaces
- Another machine

Using the information in this section, a quick analysis and generalizations can be made of the spectrum used in the illustrations. Whether the data is from a motor, pump, or fan, there is a process of evaluation that can follow a line of reasoning that asks questions about the potential sources of each peak and each mound of energy.

When the analyst knows the machine and measurement point the data came from, many possibilities will be eliminated to narrow the options even more. The waveform data can be viewed to confirm many of the questions.

Chapter 4

Signal Processing

Objective:

- Describe the purpose of filters and what they do
- Describe how sampling affects spectral data
- Determine Lines of Resolution needed to resolve frequencies
- Explain the purpose for Windowing
- Describe three Averaging methods and how they work

This chapter discusses what really happens to the signal from the transducer; the processes related to making the measurement from the perspective of capturing and manipulating the signal to produce the data we need. It involves digitizing the signal and producing the spectrum. Issues related to that are covered such as filters, sample rates, resolution and windowing.

A Quick Overview

Inside the Data Collector/ Analyzer the "analog" signal from the sensor is converted to provide usable data: waveforms, spectra, rms readings and more

There are many settings and we need to know what is happening inside the box in order to better understand the information it is giving.

Step One Power the sensor - The ICP accelerometer requires power sent to it. The signal from the sensor is an electrical signal – an analog signal.

Step Two Digitize the signal – The Analyzer is a digital instrument so the data must be converted into a digital signal so the analyzer will have numbers to work with.

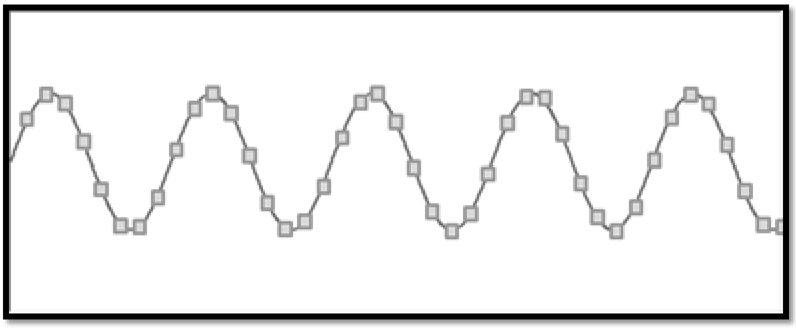


Figure 4-1 - The analog signal must be digitized so the analyzer has numbers to work with.

There are a number of considerations:

- Does the voltage from the sensor match the input range of the analyzer?
- How quickly should the signal be sampled?
- How many samples are kept?

Step Three – Process the signal - generate other data that can be worked with: spectra, rms overall readings, envelop spectra

Considerations in the processing:

- Filtering
- Integration
- Windowing
- Averaging

Filters

When a transducer is mounted on a machine, the electrical output is a continuous analog signal representing the vibration at that location. It is important to understand what filters do and how they are used in the field of vibration.

There are four types of filters:

- Low pass filters: let low frequencies pass through
- Band pass filters: let frequencies within a band to pass through
- Band stop filters: blocks frequencies within a band from passing through
- High pass filters: lets high frequencies pass through.

Figure 4-2 is spectral data from a compressor. This data is unfiltered data from 0 to 10kHz.

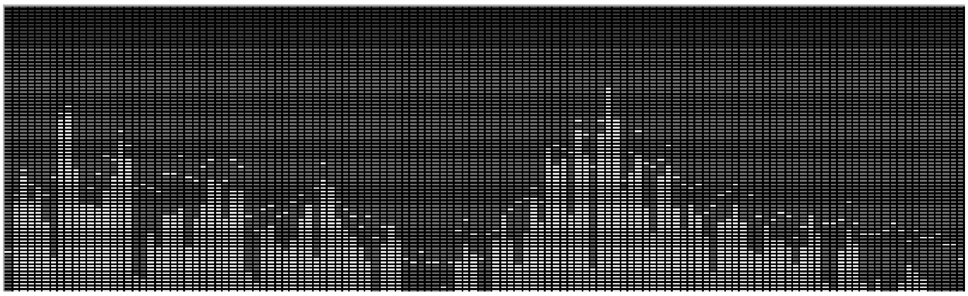


Figure 4-2 - Data from a machine, no filter applied.

Figure 4-3 is the same data but with a Low Pass filter applied. It is letting the low frequencies through while blocking the high frequencies.

Low-pass filters are by far the most common filter type, earning wide popularity in removing alias signals and for other aspects of data acquisition and signal conversion. For a low-pass filter, the pass-band extends from DC (0 Hz) to the frequency specified, and the stop-band lies above the specified frequency.

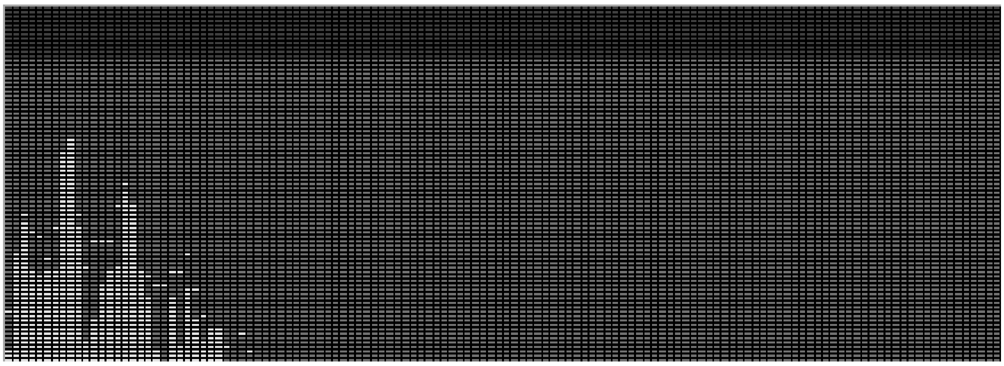


Figure 4-3 - Same data but with a Low Pass filter applied.

Band-pass filters transmit only those signal components within a band around a center frequency. An ideal band-pass filter would feature brick-wall transitions at fL and fH, rejecting all signal frequencies outside that range. Band-pass filter applications include situations that require extracting a specific tone, such as a test tone, from adjacent tones or broadband noise. See Figure 4-4

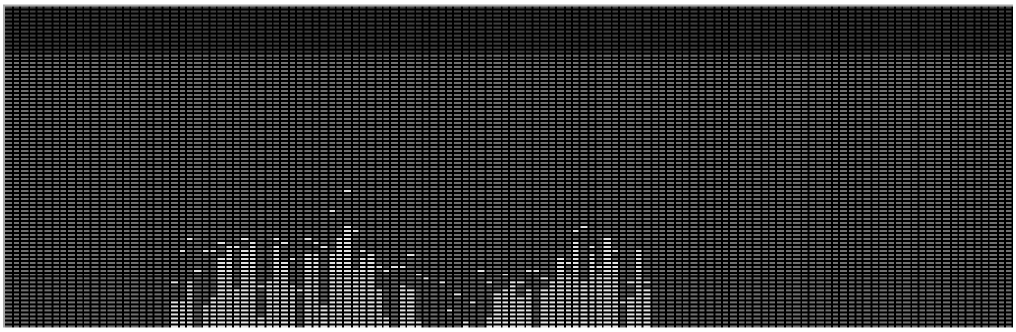


Figure 4-4 - Band Pass filter applied. It effectively blocks the high and low frequencies.

Band-Stop (sometimes called band-reject or notch) filters transmit all signals except those between specified frequency ranges. These filters can remove a specific tone - such as a 50 or 60 Hz line frequency pickup - from other signals.

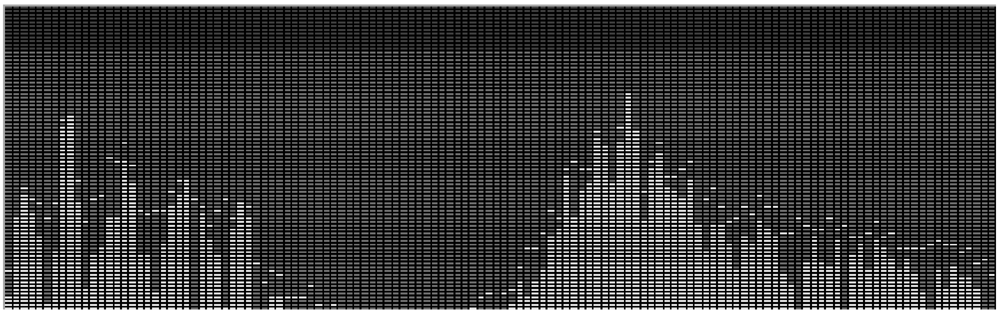


Figure 4-5 - Band Stop filter blocks the frequencies in a particular frequency band.

In a **High-Pass filter**, the pass-band lies above a specified frequency, while the stop-band resides below that point.

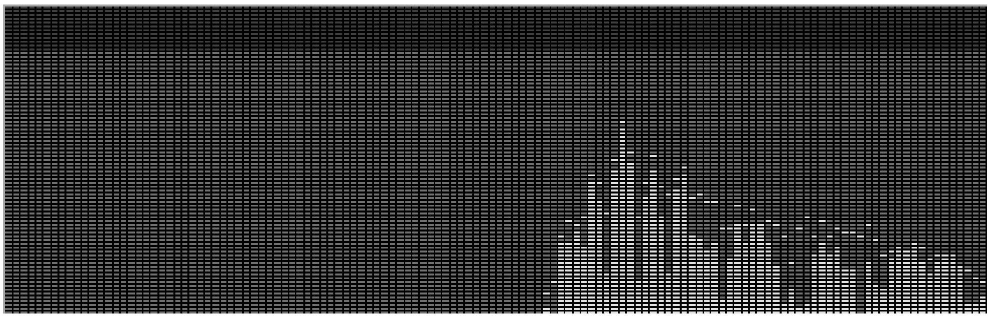


Figure 4-6 - High Pass filter blocks all frequencies below the specified frequency.

High pass filters are normally used in early bearing wear detection. Manufacturers have their names for this feature: Spike energy detection, HFD, and others. It blocks the high amplitude, lower frequencies to enable the scaling to adjust to the low amplitude levels of early bearing wear in the higher frequencies.

FILTER CHARACTERISTICS

Two types of filters are common used in the field of vibration, Digital and Analog.

Digital filters are achieved with special "digital signal processing" (DSP) chips or software.

Analog filters are created with electronic components such as capacitors and resistors. Many analyzers still have some analog filter.

There are two issues to be concerned about:

- The filter characteristics (the cut-off)
- Settling Times

Ideally, the filters would block the unwanted frequencies like putting up a brick wall to stop them. See Figure 4-7. This would provide a clean break and keep out all unwanted signals. However, this is not the case.

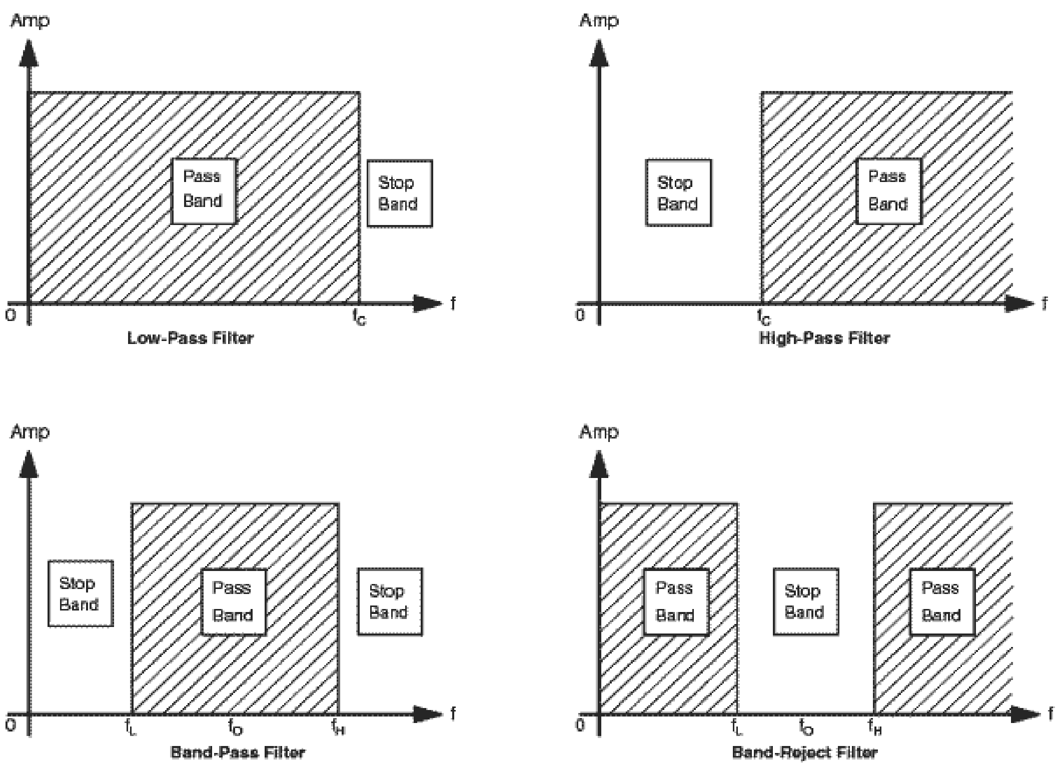


Figure 4-7 - Ideal Filter Design would block out all unwanted frequencies.

In reality, especially with analog filters, there is a region where some frequencies will be attenuated, but not blocked. There are other ways the data is massaged to get the results needed.

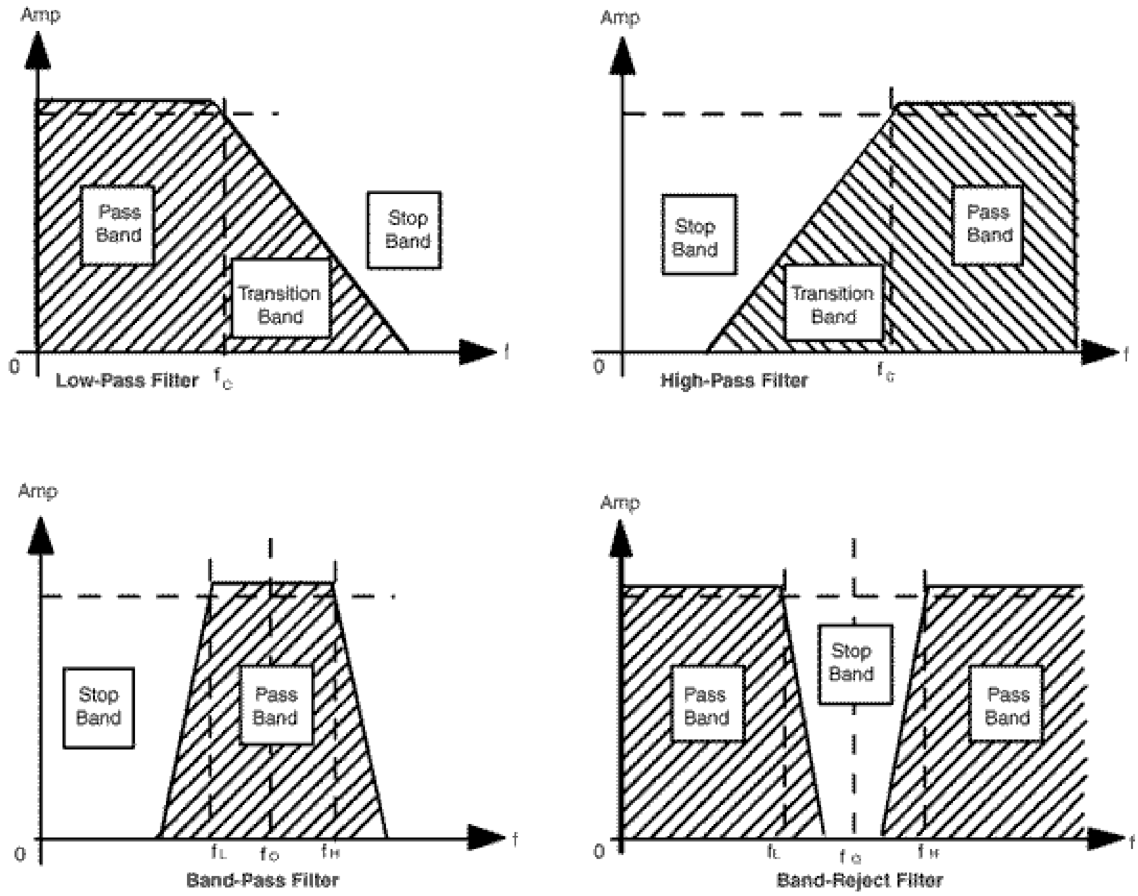


Figure 4-8 - Actual Filter design has a transition band that lets some of the frequencies through.

Settling time for filters - there is one concept that is important to understand regarding filter electronics. When data is applied to the filter circuitry, it causes the output of the circuit to "ring". This requires the settling time to be set for a duration long enough to allow the circuit to settle to normal levels before capturing the data.

Filters are used in a number of applications inside the data collector. Four of the most obvious application are anti-aliasing filters, integration (converting the acceleration signal to velocity), tracking filters, and filters used in high frequency bearing analysis (demodulation, PeakVue, acceleration enveloping).

Sampling and Aliasing

What really happens to the signal from the transducer? Sophisticated software and hardware capture and manipulate the signal to yield data that can be used for accurately determining machinery condition.

In the past, discussions regarding signal processing and sampling have been somewhat confusing. This is probably due to the many similar terms that have very different meanings and applications. The confusing terms include the word "Sample" or "Sampling." There are three of these and they will be discussed in a manner which will hopefully add clarity. The three terms are:

1. Sample Rate or Sampling Rate

2. Sample time

3. Number of Samples

There is a number which is applied in two ways in working with sampling and resolution. The number is 2.56. An attempt is made to give a clear picture of how it is used and why it is so important.

Sampling the Signal, Sample Rate

When the transducer is placed on the machine, its electrical output is a continuous analog signal representing the vibration at that location.

It contains all of the information in either velocity, displacement, or acceleration

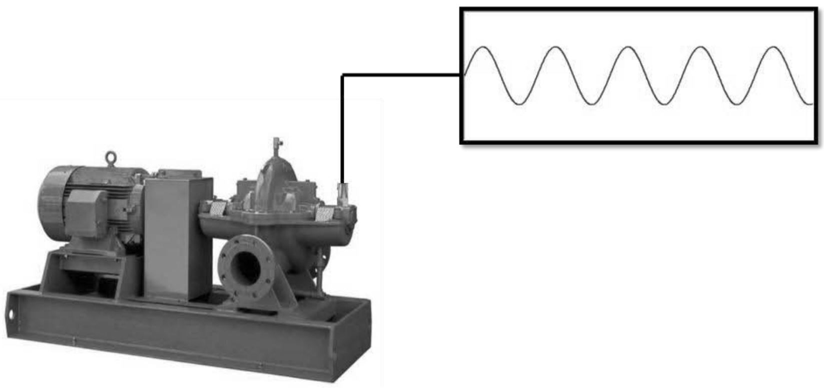


Figure 4-9 - Continuous Analog signal from the sensor

The signal must be digitized to enable the box to record and store the vibration. To do this the continuous analog signal is converted into a series of discrete numbers called a **time record**.

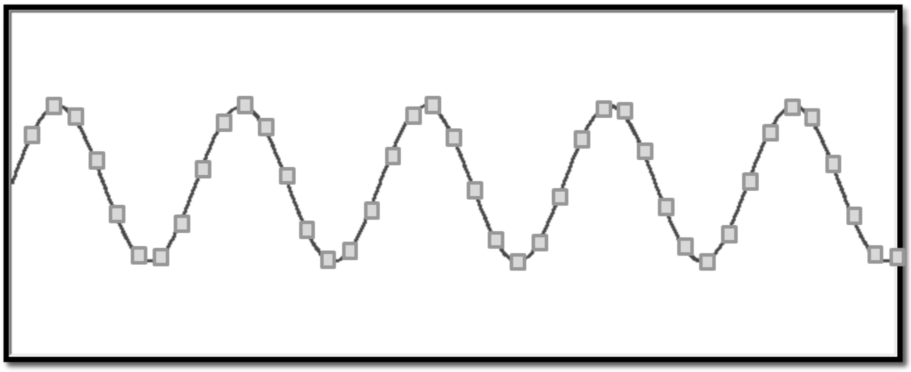


Figure 4-10 - The time waveform is sampled at discreet intervals. Each point is an individual sample. The rate of the sampling is called the Sampling Rate.

Figure 4-10 shows the process of digitizing a standard analog waveform. It has been marked where individual samples have been made at discreet time intervals. The samples are converted to numbers and stored so the waveform can be rebuilt later and build the FFT.

The Rate at which the waveform is sampled is called the **Sampling Rate or sample rate.** It is the number of samples made in one second.

To represent the time waveform, the data collector only has the samples to work with. Anything that happened between the samples is lost. When the data collector or software draws the time waveform it simply "connects the dots." This is called the **Time Domain.**

What is actually stored in the data collector is therefore not as detailed as the original continuous analog signal.

Fast Fourier Transform

The FFT process takes the time waveform and creates the spectrum which is referred to as the Frequency Domain.

Fast Fourier Transform or FFT

The vibration spectrum is the result of a Fourier Transform, named after the mathematician who developed the equation which turns a complex wave into its harmonic components.

Two other mathematicians, Cooley and Tukey, in 1965 developed an algorithm to effectively speed up the processing of the Fourier Transform and thus the name Fast Fourier Transform or FFT.

$$f(t) = a_0 + \sum_{i=1}^{\infty} a_i \sin(2\pi f_i + \phi_i)t$$

Fourier, Jean Baptiste Joseph - French baron, physicist, mathematician 1768 - 1830

The FFT works best on waveforms that include only periodic signals. It will build sine waves out of those signals. Each peak in the spectrum represents a sine wave with an amplitude a_i, frequency f_i and phase Θ_i.

For the FFT to work correctly we have to deal with three problems:

- Aliasing - we need at least two samples per cycle if we wish to resolve that frequency.
- Leakage - a loss of resolution if the cycles are not perfectly periodic in the time record.
- Resolution (picket fence effect)

We only have data at specific frequencies – we don't know what happened between those frequencies

Aliasing

The Sampling Rate affects the range of waveform that can be reconstructed.

In Figure 4-11 a 50Hz signal is sampled at a sampling rate of 50 times per second (50 Hz). This yields one sample during each cycle. Connecting the sample dots yields a straight line. The waveform cannot be reconstructed. There are not enough samples.

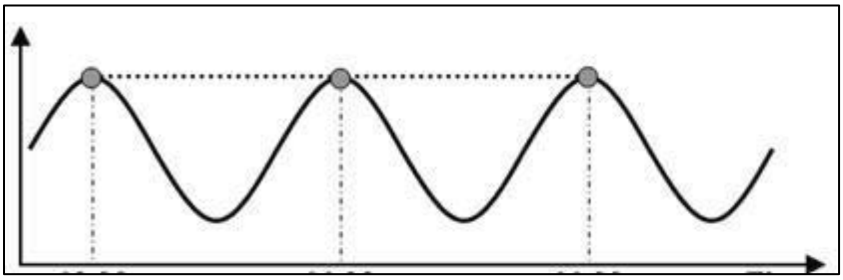


Figure 4-11 - A 50 Hz signal sampled at a rate of 50 times per second yields one sample per cycle. Connecting the dots yields a straight line.

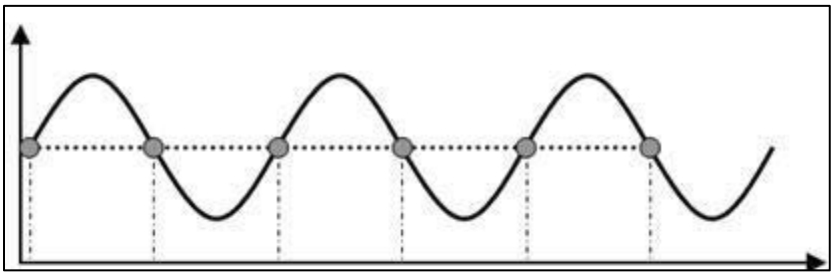


Figure 4-12 - The 50Hz signal sampled at a rate of 100 Hz. Connecting the dots still yields a straight line.

In Figure 4-12 the sampling rate has been doubled to 100 Hz. But connecting the dots again yields a flat line. There are still not enough sample points to reconstruct the waveform. The waveform must be sampled at a rate greater than twice the frequency of the waveform.

The **Nyquist Theorem** or **Criterion** states that the waveform must be sampled at a rate greater than twice the input signal. This frequency is often referred to as the Nyquist Frequency.

Most digital analyzers sample at **2.56 times the maximum frequency of interest.**

A digital vibration measurement consists of a large number of samples, enough to be representative for the time record. The speed at which these samples are taken depends on the maximum frequency to be covered when the spectrum is produced. See Figure 4-13.

The sampling frequency must be larger than twice the maximum frequency in the spectrum. To get a spectrum line at 400 Hz it must sample at faster than f_s = 800 Hz, that is collect more than 800 readings per second. This means that there is less than 0.00125 seconds for each reading.

Another example, if the Maximum Frequency of interest (often called the Fmax) is 400Hz, the **sampling rate** would be 400*2.56 = 1024 times each second.

Sampling time: ts

Sampling Frequency: fs (fs = 1/ts)

Nyquist theorem: fs > 2 x Fmax

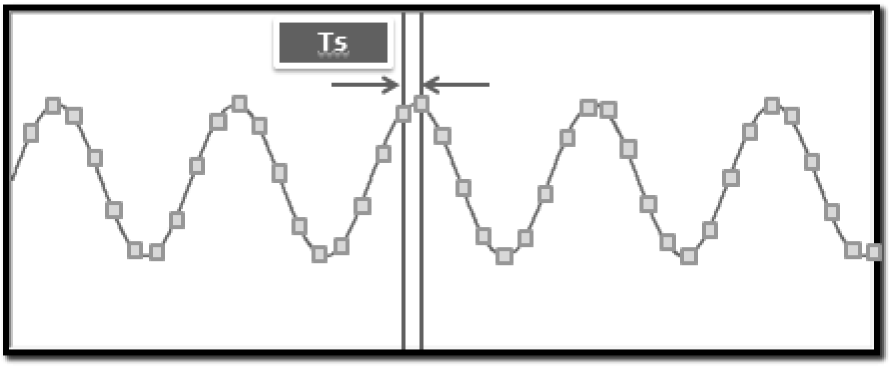


Figure 4-13 - the sampling frequency depends on the maximum frequency to be displayed in the spectrum

Sampling means that we measure a continuous real time event in a number of points and then try to reconstruct the event by connecting these points. If we do not sample fast enough, the reconstruction will not match the event. In Figure 4-14, the reconstructed curve has the right amplitude but the wrong period. This effect is called “aliasing” and the effects must be corrected.

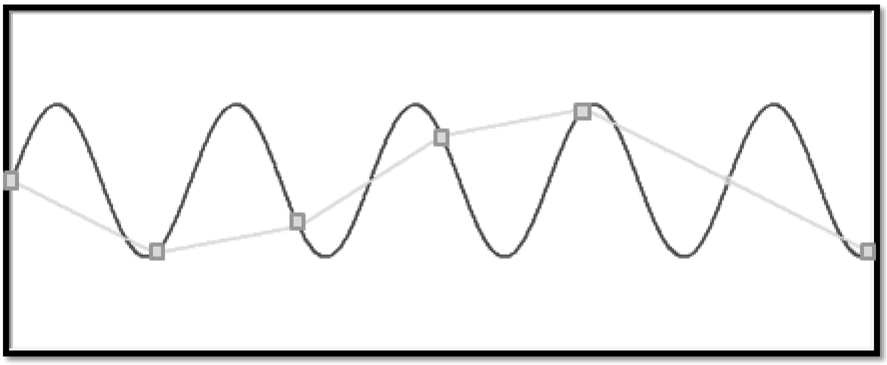


Figure 4-14 - The effects of aliasing is that a false signal is constructed. It has the correct amplitude but the wrong period.

Two signals are said to alias if the difference of their frequencies falls in the frequency range of interest. This difference frequency is always generated in the process of sampling

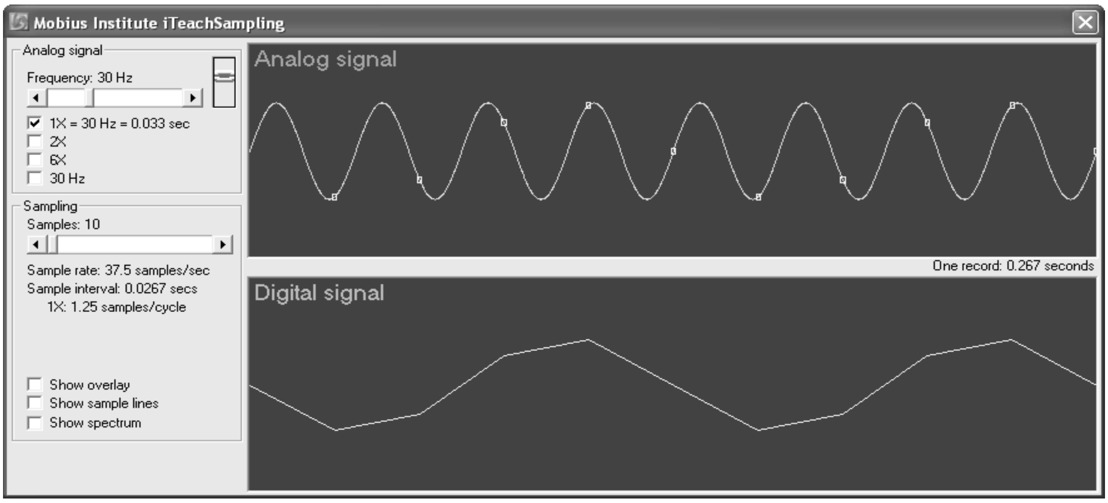


Figure 4-15 - The analog signal showing points of sampling and the digital reconstruction.

In Figure 4-15 the analog signal is not sampled at a rate fast enough to reconstruct it. Instead, it generates a waveform at a much lower frequency. Note that the analog signal is a real signal, but there are not enough samples to reconstruct it accurately. The reconstructed waveform is at a much lower frequency, it is an aliased signal; a signal that appears real in the spectrum but does not really belong there.

Figure 4-16 shows an analog signal with a 1x signal of 48 Hz and a 6x signal at 288 Hz. The Sampling Rate is 960 samples per second. So the 1x peak is sampled 20 times per cycle and the 6x pk is sampled 3.333 times per cycle.

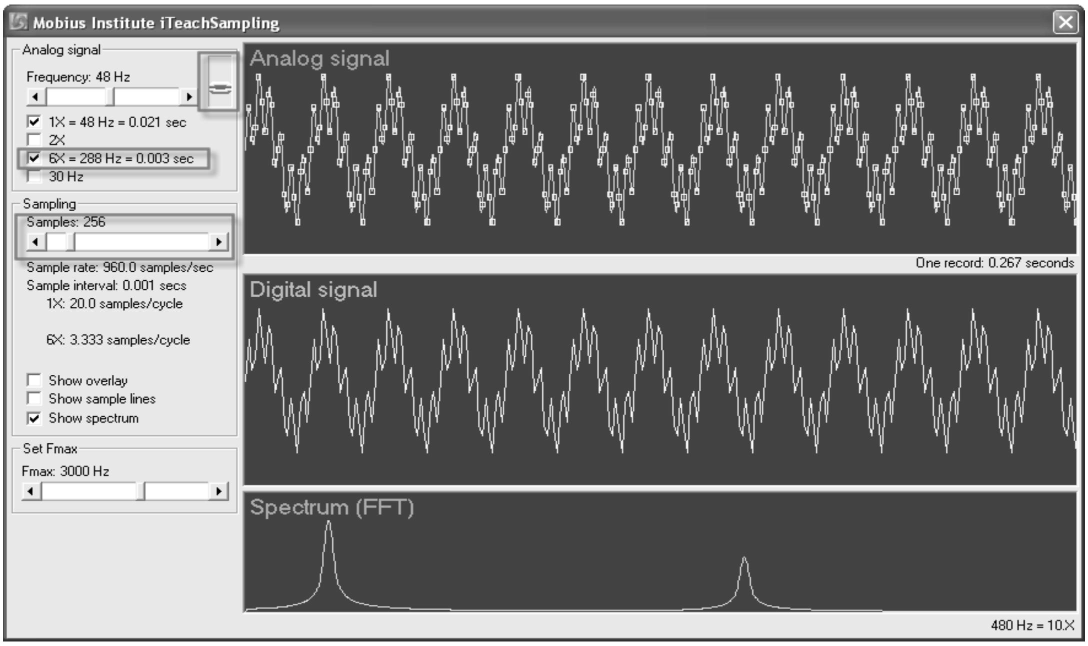


Figure 4-16 - The 1x and 6x signals are sampled more than twice per cycle so they are reconstructed accurately in the digital waveform.

Since both signals are sampled more than 2x their frequencies, they are both reconstructed accurately in the digital waveform. Both signals are also accurately reflected in the spectral data.

Another example, Figure 4-17, has a 65 Hz signal. The spectrum has an Fmax of 100 Hz so the sampling rate is 256 samples a second. The peak in the spectrum is at 65 Hz.

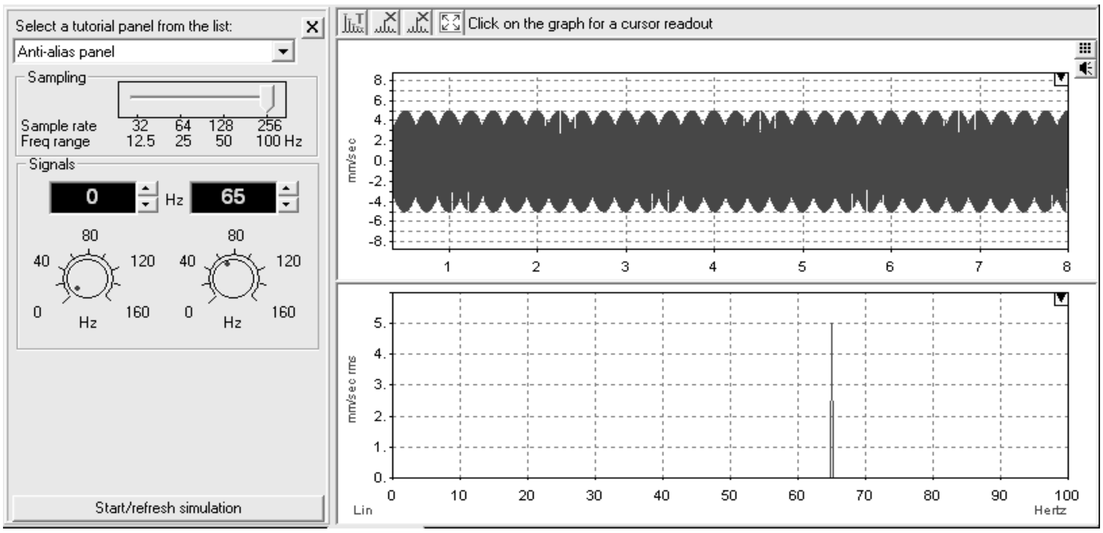


Figure 4-17 - A signal is sampled at a rate of 256 times per second includes a 65 Hz signal. It is at 65 Hz in the spectrum.

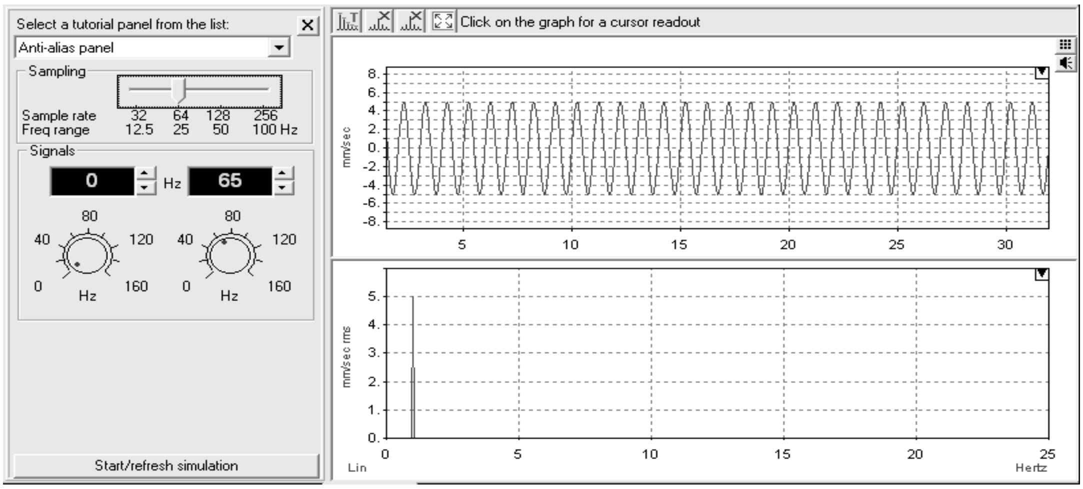


Figure 4-18 - A 65 Hz signal is sampled at a rate of 64 times per second. The aliased peak shows up at 1 Hz.

Figure 4-18 is the very same signal but sampled at a rate of 64 samples per second. The frequency range is 25 Hz. The 65 Hz signal is in the analog signal but cannot accurately be reconstructed because the sampling rate is too low. It placed the aliased signal at 1 Hz.

COPING WITH ALIASING

In the previous example there was a signal of 65 Hz but in the spectrum there was a peak of 1 Hz. The peak looks real and can easily be mistaken for a real peak. However, it not real. It is the aliased peak.

But there is no way to know what frequencies are contained in the signal. It must be assumed that there will always be some signals that will fold-over into the spectrum.

To eliminate the effect of aliasing, all signals that are too high for our sampling rate must be filtered out. A low pass filter is put in place that filters out any signal above the sampling rate. So the low pass filter is set at the Fmax to correct this phenomenon. This filtering is called an **Anti-aliasing Filter**.

All spectrum analyzers use some form of anti-aliasing filter.

The cause of Aliasing- a frequency is present in the signal that is not being sampled fast enough (more than 2 times the frequency).

The Effect of Aliasing - If the difference frequency (the difference between the Fmax and the actual frequency) falls below the Fmax, it will show up in the spectrum.

Correcting Aliasing –Filter out all frequencies above the Fmax.

Another Example

Using the example fan from the "Principles of Vibration" chapter, and adjusting the speed to 1 Hz, the blades produce a signal at 8 Hz. So the combined signal is a 1 Hz signal plus the 8 Hz signal. It's easy to see that if that signal is sampled at a rate of 1 Hz, the result would be a flat line.

According to the Nyquist Criterion, if it is sampled at three times per second, there would be enough data to resolve the 1 Hz signal, but still not enough for the higher frequency 8 Hz signal.

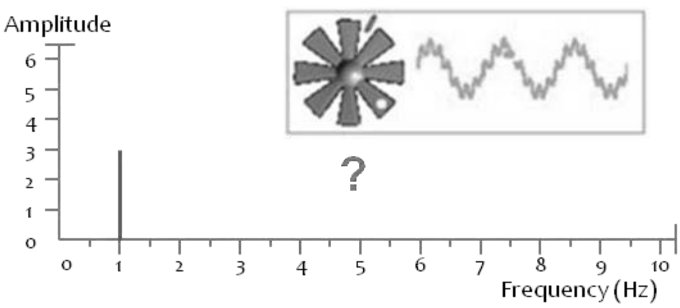


Figure 4-19 - Sampling at 3 times/sec ignores the 8 Hz signal, but it will show up somewhere in the spectrum.

If the 8 Hz signal is expected, and we wanted to measure it, the signal would have to be sampled ABOVE 16 times per second (8 * 2 = 16). But if we did not know it was there, and only sampled at 3 times per second, there would be a peak in the spectrum at a "ghost" frequency from the 8 Hz component.

The only solution to eliminate the ghost frequency is to filter out everything above the frequency that is known to be valid in the data. Sampling at 3 times per second, filters out everything above 6 Hz. (3 * 2 = 6)

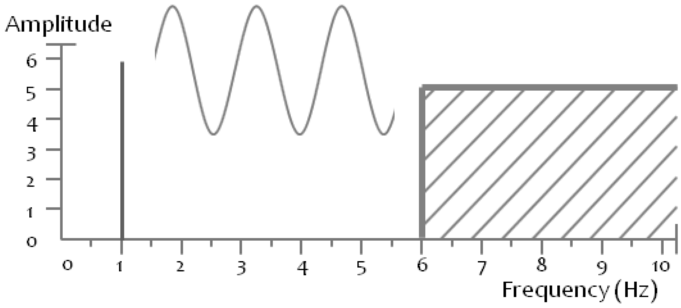


Figure 4-20

Unfortunately filters are just not that good. They are unable to pass everything below a frequency, and reject everything above a frequency. If the filter were perfect, every frequency above the sampling rate would be completely removed. But in reality the filters are not that good. Instead some signal level at those frequencies makes it through, so all of the spectrum cannot be used.

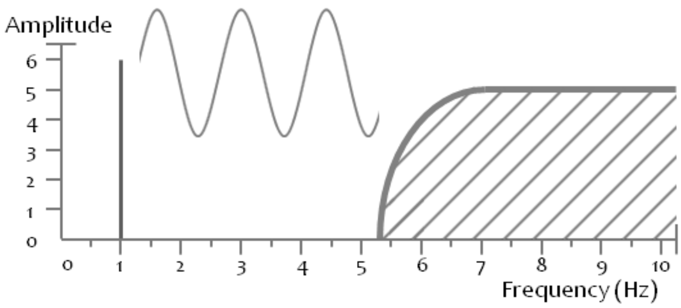


Figure 4-21

THE BOTTOM LINE ON ALIASING

Two steps are taken to eliminate the effects of aliasing. First, a low pass filter is applied to the Fmax before sampling so that the sampling is still limited to those frequencies for the spectrum.

Secondly, recall that for N samples in the time record the FFT produces a spectrum with N/2 lines. So a 2048 sample time record produces a 1024 line spectrum. However, to correct for anti-aliasing, only 800 lines are kept. The accepted rule is that the number of lines in the spectrum is equal to the number of samples in the time record divided by 2.56.

- $N = 2^8$ = 256 ⟶ 100 lines
- $N = 2^9$ = 512 ⟶ 200 lines
- $N = 2^{10}$ = 1024 ⟶ 400 lines
- $N = 2^{11}$ = 2048 ⟶ 800 lines
- $N = 2^{12}$ = 4096 ⟶ 1600 lines
- $N = 2^{13}$ = 8192 ⟶ 3200 lines

DELTA-SIGMA

There are actually two ways to solve the Aliasing dilemma. One is to use an analog anti-aliasing and the other is to use "digital" filtering, often called the "sigma-delta" method.

Most modern data collectors now employ the sigma-delta method, as they have better performance, and are less expensive to manufacture. In effect, it produces a much better filter which would allow many more lines to be kept in the spectrum. For convention, the industry has kept the same number of lines as we used in the past.

Sampling and Resolution

We can control 2 parameters: The Sample Rate and the Total Number of Samples collected.

The Sample Rate is controlled by selecting the Fmax.

The Number of Samples is defined by the Lines of Resolution.

Taking a closer look at the sampling process leads to some conclusions. The Sampling Rate must be 2.56 times the highest frequency of interest. To recreate a signal of 1000Hz requires a Sampling Rate of 2560Hz. This produces a spectrum with an Fmax (Maximum Frequency) to 1000Hz. This yields Rule #1.

Rule #1

Sample Rate is proportional to frequency range

(The Sample Rate is 2.56 x Fmax)

Another relationship to remember involves the waveform and spectrum. The number of samples in the time waveform determines the bandwidth or resolution in the spectrum. In most analyzers the **bandwidth or resolution** is set by the **"number of lines"** chosen by the operator. This number is referred to as **Lines of Resolution** or **LOR**.

One line is one data point in the spectrum. It is sometimes referred to as a cell or bin. Two frequencies cannot be resolved if they are closer together than the lines. The bandwidth or resolution of each line, bin, or cell is calculated by dividing the Fmax by the number of lines.

Resolution = Fmax / LOR

For example, a spectrum with a maximum frequency of 1000Hz and 400 lines results in a maximum resolution of 1,000/400=2.5Hz. This means that two discreet frequencies will appear as one if they are closer than 2.5Hz apart.

If the Lines of Resolution is increased to 800 lines, the resolution is 1000/800=1.25Hz.

The Number of Lines is proportional to resolution

But this is not the whole story. There is a link between the number of lines and what is called the **Number of Samples**. The Number of Samples required is **2.56 times the number of lines (LOR).** (There's the 2.56 number again) For 400 lines in the spectrum, the Number of Samples is 400 x 2.56 = 1024. Rule #2 in its broadest scope then is:

Rule #2

The Number of Samples is proportional to resolution

(Number of Samples = 2.56 x LOR)

The net result is that good resolution can be achieved. It requires a greater number of samples and increases the time to collect the data.

To resolve frequencies that are close together, more samples are needed.

Number of samples 'N' is proportional to resolution

- N = 256 ⟶ 100 lines
- N = 512 ⟶ 200 lines
- N = 1024 ⟶ 400 lines
- N = 2048 ⟶ 800 lines
- N = 4096 ⟶ 1600 lines
- N = 8192 ⟶ 3200 lines

Sample Time

There is a trade off in results vs. time. The faster the data is sampled, the more data that is collected – which has to be stored, transferred, graphed, and used in calculations. The more data collected to get greater resolution, the longer it takes to collect it, which means more time at the machine.

The **time required to collect the samples** is equal to the number of samples required, divided by the sampling rate. (**Number of samples/sampling rate**)

For example: an Fmax of 400 Hz sets the Sampling Rate at 1,024Hz. A resolution of 400 lines sets the Number of Samples at 1,024. So the Sample Time is 1,024/1,024 = 1 second.

Another way to compute the Time for one average is to use the numbers that we can control, i.e. the Fmax and LOR. So **T = LOR / Fmax.** Using this simplified formula for the previous example yields... T = 400LOR/400Hz Fmax = 1 second.

Another example using the simplified formula for an Fmax of 100 Hz and 12,800 lines of resolution.... T = 12,800LOR/100Hz Fmax = 128 seconds to collect one average.

Figure 4-22

Example: Sampling Rate

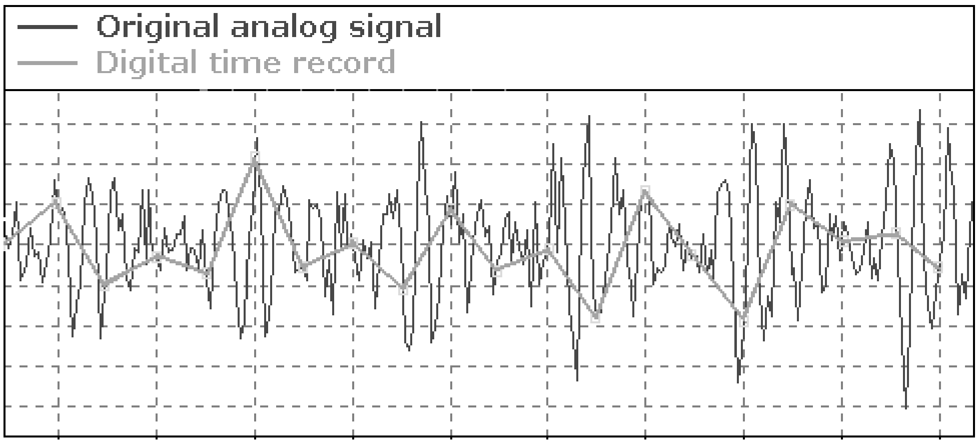


Figure 4-23 - The sampling rate is set too low capture the high frequency occurrences in the waveform.

In Figure 4-23 the sampling rate is too low to capture any of the high frequency signal. Information contained in the waveform signal will be lost.

Example: Resolution

Figure 4-24 is a waveform which contains 3 signals, a 6 Hz, an 8 Hz, and 380 Hz signal. However, there are only two peaks in the spectrum.

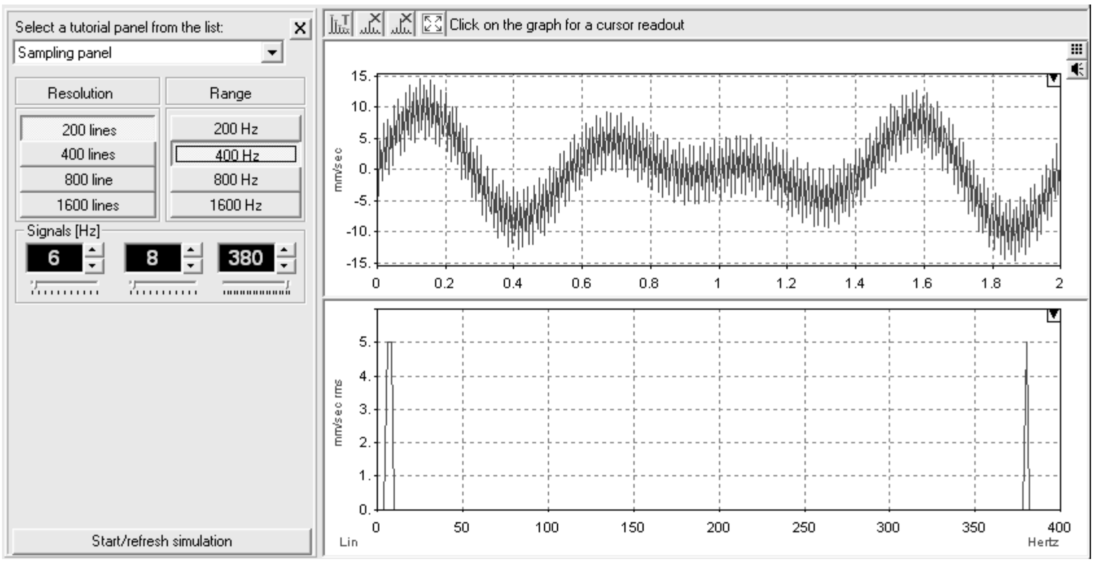


Figure 4-24 - There are three signals but only two peaks

The frequency range is set to 400 Hz which is a sampling rate of 1024 samples per second. This captures the 380 Hz signal.

However the number of samples is 512 which produces a 200 line spectrum. The 6 Hz and 8 Hz signals combine and produce a peak around seven Hz. The resolution is not good enough to separate the peaks. The resolution is 2 Hz (400 Hz Fmax /200 LOR). This only separates signals greater than 2 Hz apart.

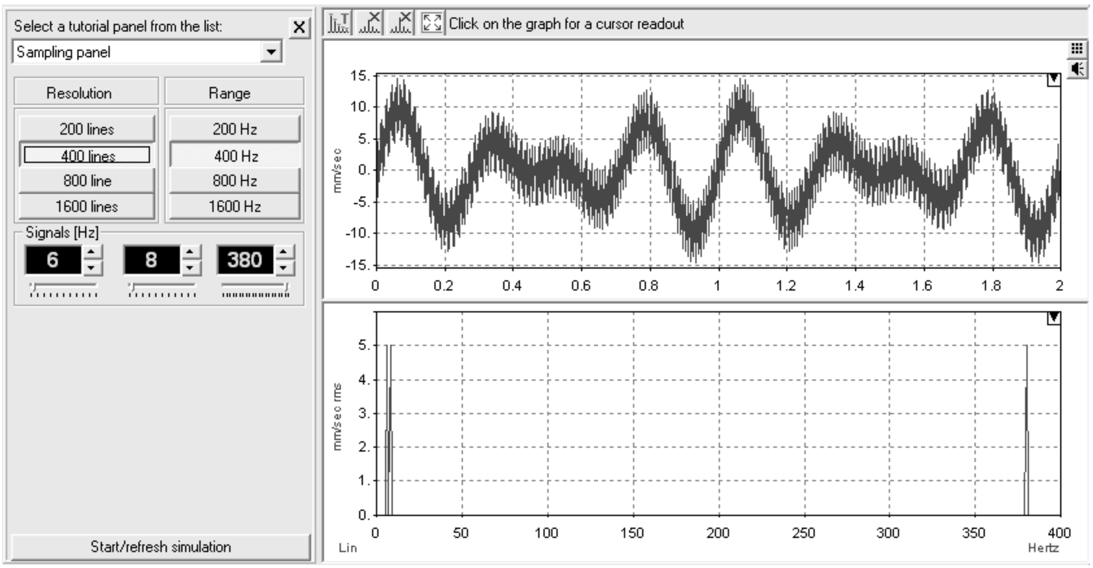


Figure 4-25 - The Resolution is increased to 400 lines and the two signals are now separate peaks

Figure 4-25 is the same signal but with 400 lines of resolution, 1024 samples in the waveform. The resolution is now 1 Hz and the two signals have their own discreet peaks.

It takes a combination of Sampling Rate and No. of Samples (Fmax and LOR) to capture and see the data that may be needed.

Figure 4-26 has two signals, a 15 Hz and a 17 Hz signal. The Fmax is set to 800 Hz and the number of lines is 800 lines. The resolution is one Hz. The spectrum is zoomed in to show only the first 25 Hz. The dotted lines indicate the lines of resolution.

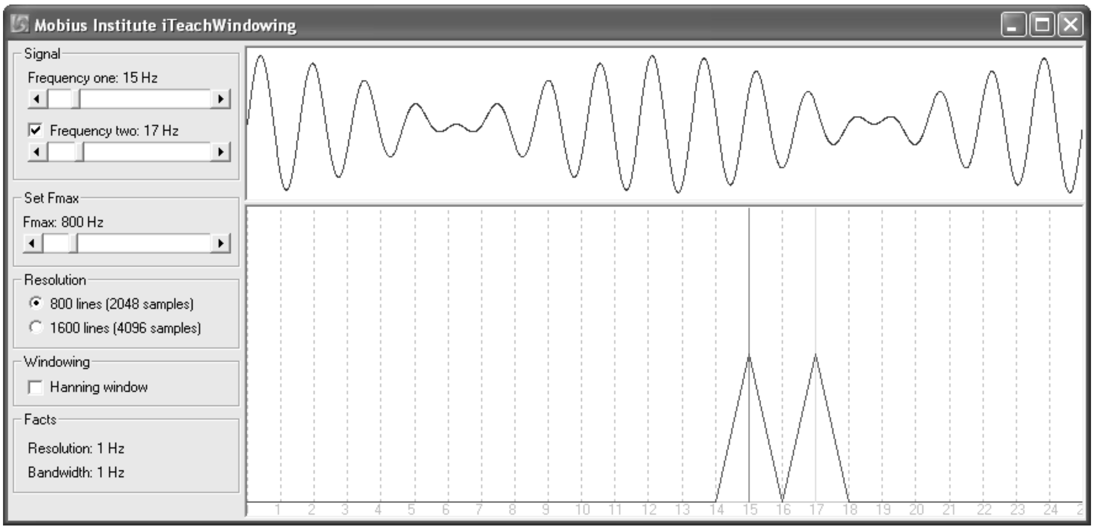


Figure 4-26 - Two peaks are separated but smeared over two lines of resolution.

The peaks are "smeared over two lines of resolution because there is not enough resolution to make them more precise. So although the resolution is 1 Hz, these peaks are 2 Hz wide.

*NOTE: The Resolution is further limited by the Window factor which will be discussed in a later section. See **Window Type Affects Frequency Resolution** in the Windowing section.*

TIME FOR ONE MEASUREMENT

The time required for collecting one spectrum is calculated by dividing the number of lines in the spectrum by the Maximum Frequency of the spectrum.

$$T = ts \times N = \frac{N}{fs} = \frac{N}{2.56 \times F_{max}} = \frac{lines}{F_{max}}$$

Example: A spectrum with 800 Lines has an Fmax of 1000 Hz. The time to collect this spectrum is:

T = 800 / 1000 = 0.8 seconds

Resolution: In a spectrum, the measured vibration is shown as evenly spaced lines. The distance between these lines, in Hz, is the resolution of the spectrum. It is obtained by dividing the range through the number of lines in the spectrum.

If the resolution is 1 Hz, all vibration frequencies between e. g. 99.5 Hz and 100.5 Hz (approximately) will be shown at 100 Hz, because in such a spectrum there is nothing in between 99 Hz, 100 Hz, and 101 Hz.

Example: for a spectrum with an Fmax = 200 Hz…		
Lines – number of lines in the spectrum	**Resolution – width of each bin**	**Time required to compute one spectrum**
200	1	1 sec
400	0.5	2 sec
800	0.25	4 sec

Table 4-1

The illustration below shows part of a spectrum with a range to 200Hz. It has 400 lines of resolution. The frequency resolution is the Fmax / number of lines or **Fmax / LOR**. 200/400=.5Hz

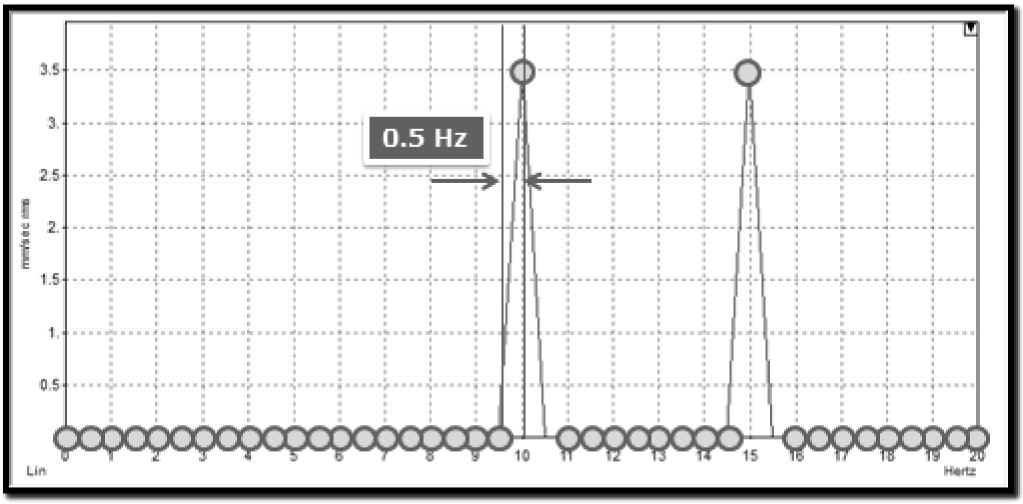


Figure 4-27

TEST YOUR KNOWLEDGE:

What is the Number of Samples required to get this spectrum?

What is the Sampling Rate used to yield this frequency range?

What is the Sample Time for each average to compute this spectrum?

0-200			0-500			0-1000		
lines	*res. (Hz)*	*time (s)*	*lines*	*res. (Hz)*	*time (s)*	*lines*	*res. (Hz)*	*time (s)*
200	1	1	200	2.5	0.4	200	5	0.2
400	0.5	2	400	1.25	0.8	400	2.5	0.4
800	0.25	4	800	0.625	1.6	800	1.25	0.8
1600	0.125	8	1600	0.3125	3.2	1600	0.625	1.6
3200	0.0625	16	3200	0.15625	6.4	3200	0.3125	3.2

0-2000			0-5000		
lines	*res. (Hz)*	*time (s)*	*lines*	*res. (Hz)*	*time (s)*
200	10	0.1	200	25	0.04
400	5	0.2	400	12.5	0.08
800	2.5	0.4	800	6.25	0.16
1600	1.25	0.8	1600	3.125	0.32
3200	0.625	1.6	3200	1.5625	0.64

Table 4-2 lists the increasing number of lines for a constant Fmax showing the resolution or bandwidth, the sample rate, the number of samples and the sample time.

Number of lines	Fmax (Hz)	Resolution (Hz)	Sample rate	Number of samples	Sample time (sec)
400	1000	2.50	2560	1024	0.4
800	1000	1.25	2560	2048	0.8
1600	1000	0.63	2560	4096	1.6
3200	1000	0.31	2560	8192	3.2
6400	1000	0.16	2560	16384	6.4

Table 4-2 - Effect of resolution on the Sample Time. Fmax and therefore the sample rate are constant.

Dynamic Range

Up to this point the focus has been on the accuracy of the frequency. But there are concerns regarding the resolution of the amplitude. Vibration waveforms from machinery contain a great deal of information, and some signals are very low in amplitude compared with the more dominant frequencies.

Remember, when analyzing the data do not just look at the high peaks, but also look at the lower peaks. Harmonics of bearing tones, for example, can be quite low in amplitude, but they are still critical to the diagnostic process.

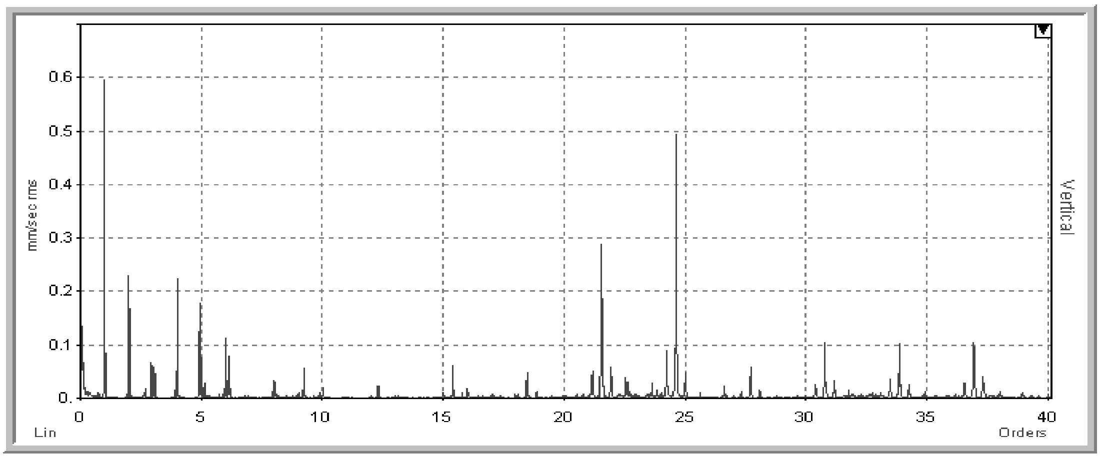


Figure 4-28 - low level bearing peaks in the presence of other moderate peaks.

The signal is digitized by recording the voltage level at discrete time intervals. The analyzer can record a maximum voltage and a minimum voltage. For example, +or – minus 1 volt. But what happens if the signal is only 0.1 volts?

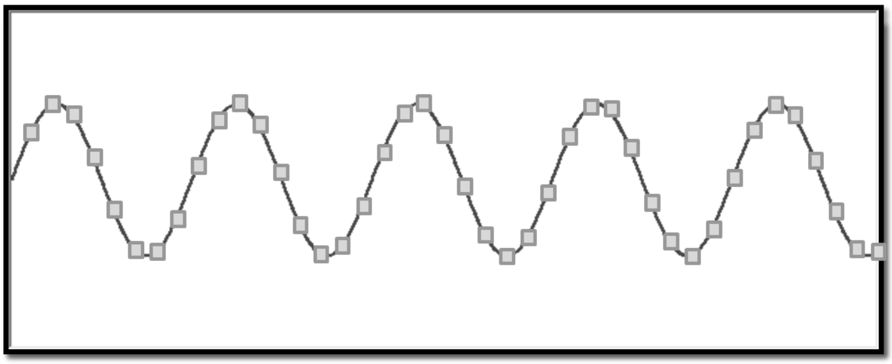


Figure 4-29

When the data collector digitizes the incoming signal it can only assign a limited number of amplitude values to each sample it takes. Let's take the extreme. If a digitizer (the A/D) was only 5 bit, that means it can assign one of 32 numbers (2^5 = 32) to the sample.

If the maximum voltage was 1 volt, which is called the input range, our amplitude resolution would be (1/32 * 1000) 31.25 mV. The smallest amplitude measured would be 31.25mV and the largest would be one volt. The reading would all be increments of 31.25 mV.

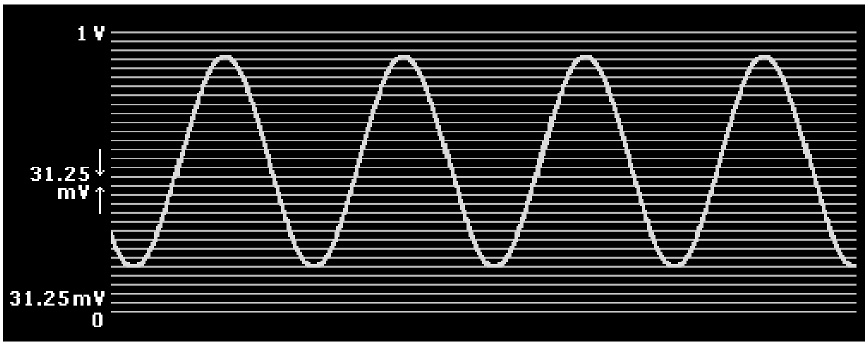


Figure 4-30 - the incremental values of the 5 bit digitizer is increments of 31.25 mV.

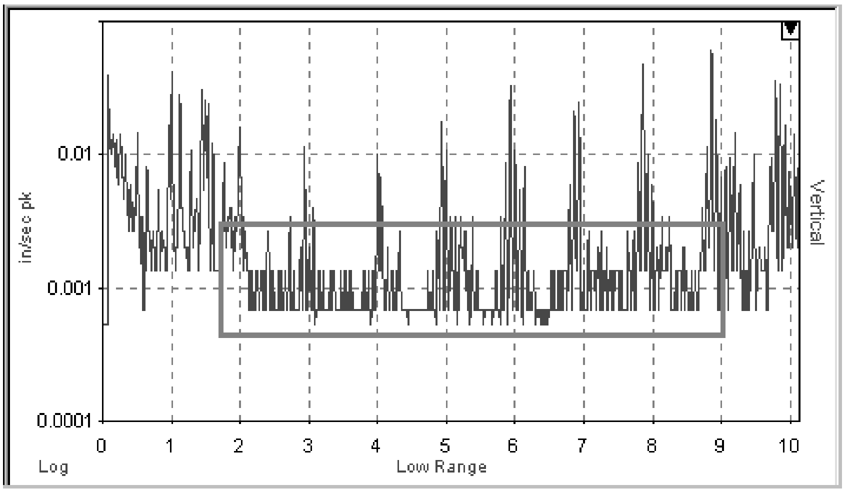


Figure 4-31 - Chunky spectrum due to poor dynamic range.

This is not nearly enough dynamic range. The waveform is chunky and the spectrum will be similar.

Dynamic range is a measure of the ability to 'see' small signals in the presence of large ones. It is a ratio of the smallest signal to the biggest, and is typically represented in decibel (dB).

Dynamic range = 20 x log (smallest / Biggest signal)

A wide dynamic range means very small signals can be resolved in the presence of very large signals. In theory the dynamic range is solely dependent on the resolution of the A/D converter. However, in reality, data collector electronics (including the signal conditioning, amplification and filtering components) add noise to the system thus reducing the dynamic range (because the noise swamps the lower amplitude signals).

Many of the older data collectors on the market have 12 bit A/D converters, which gives them a theoretical dynamic range of 72 dB.

12 bit gives 4096 possible values ($2^{12} = 4096$)

72 dB is the dynamic range of 20 x log(1/4096)

The more recently developed data collectors have 16 bit A/D converters which provides a dynamic range of 96 dB. This really means that if the input signal was 1 volt, we could still detect changes as small as 0.015 mV - which is excellent.

Data collectors do not have only one input range. There is usually an amplifier (a gain stage) before the A/D to increase the amplitude to best suit the input range of the A/D converter.

If we go back to our 5 bit A/D example (Figure 4-29,) the ideal situation exists when the signal is actually 1 volt in amplitude. All of the available input is being used.

The measurement quality of a data collector is actually a measure of the resolution of the A/D (for example 16 bit), and also its ability to amplify the signal so that the collector utilizes the majority of its input range. We would like to be able to change the range settings in small steps to cater for the wide range of possible signals.

While most data collectors will allow you to manually set the input range settings, in most cases you will allow the collector to auto-range - it will determine the best range automatically. It actually does this by trying each range setting to find which is best.

SETTING THE INPUT RANGE

If we go back to our 5 bit A/D example, the ideal situation exists when the signal is actually 1 volt in amplitude. All of the available input is being used.

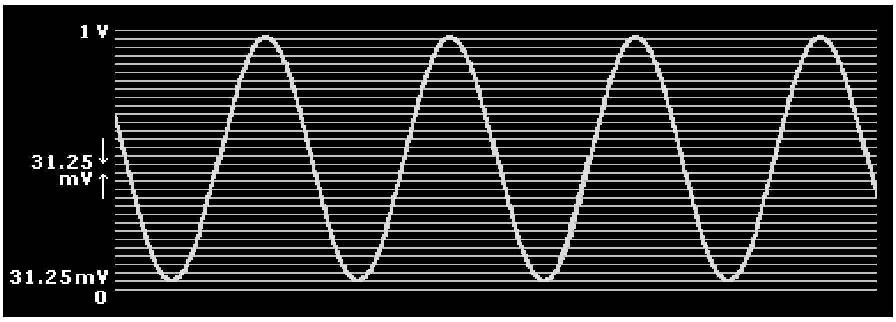


Figure 4-32 - In this 5 bit example, all of the available input is being used. The signal actually is 1 volt.

But what would happen if the input is only 0.05 volts? Our smallest signal measurable is still only 31.25 mV, so the actual dynamic range is greatly reduced to 20 * log (31.25/50) = 4 dB. What a terrible measurement!

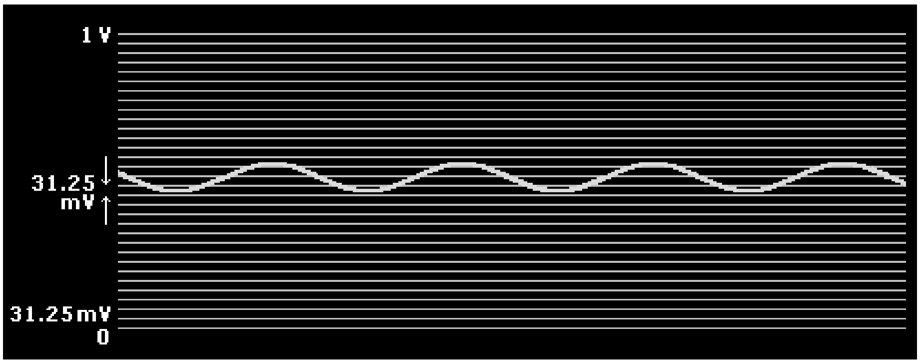


Figure 4-33 - A fixed range with only a 0.05 amplitude signal. It is very poor.

Fortunately, analyzers have the ability to auto –range. That is, they sample the signal before actually collecting the data, and set the gain for the best fit. They do this by actually trying each range setting to see which has the best fit.

Additionally, some analyzers have the ability to manually set the gain, or pre-set the input range, which can be very useful in some special tests.

The measurement quality of a data collector is actually a measure of the resolution of the A/D (for example 16 bit), and also its ability to amplify the signal so that the collector utilizes the majority of its input range. Modern data collectors have excellent amplitude resolution and they have multiple gain settings – therefore they have excellent dynamic range.

AUTO-RANGE VS. AUTO-SCALE

Many people confuse auto-ranging with auto-scaling. They do not mean the same thing at all.

As just described, auto-ranging is the process of selecting the optimum gain setting. Auto-scaling is a simple graphical routine which takes the spectrum and displays it with the optimum display range.

The two spectra below are the same data and illustrate the effects of auto-scaling. One takes advantage of the full graph scale so that low amplitude frequencies can be seen more clearly.

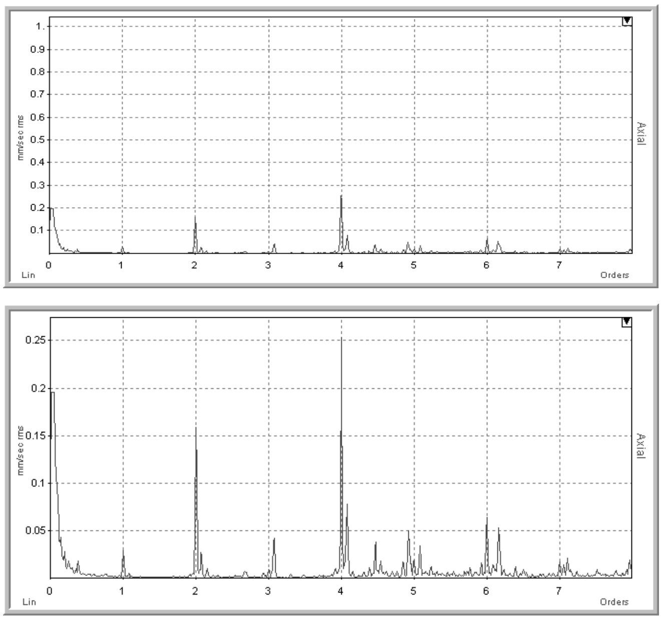


Figure 4-34 - Auto scaling merely adjusts the display to utilize the full display area.

Windowing

There is another property of the FFT which affects its use in vibration analysis. You may remember that the FFT is performed on a block of samples called the time record. One assumption made in the FFT calculation is that the time record is continuous. That is, the signal just before the captured time record, and the block immediately after our time record are identical.

In this example, although we are performing the FFT on the block of data with the black background, the FFT calculation "assumes" that the data continues endlessly before and after this block of data - as shown with the data with a gray background.

In this example it is true that the single frequency sine-wave begins and ends at zero amplitude. Four complete cycles live within the time record

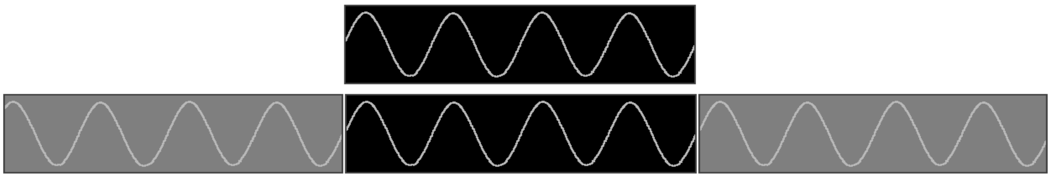


Figure 4-35 - The FFT process assumes the block of data goes to zero at the ends. In this case it does.

If we are analyzing a pure sine wave, i.e. just one frequency, and there is an integer number of cycles in the time record, then this assumption is correct.

However it is seldom true that the time record starts and ends at zero. More commonly they are similar to Figure 4-36.

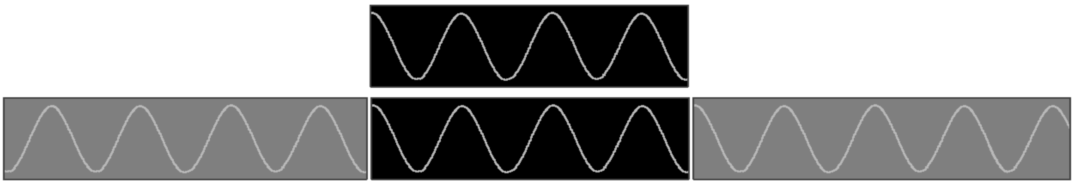


Figure 4-36 - The time window does not start nor end at zero.

When the FFT calculation is performed the signal is discontinuous. It seems to have a step increase in level and looks similar to an impact to the FFT calculation. It generates a peak that is spread over a wide frequency band similar to an impact.

Recall from the chapter on Introduction to Vibration that an impact generates energy in a wide frequency range. That is not what we want to see as a result of this.

The real data in Figure 4-37 shows that the ends of each sample do not have the starting and ending amplitude at zero.

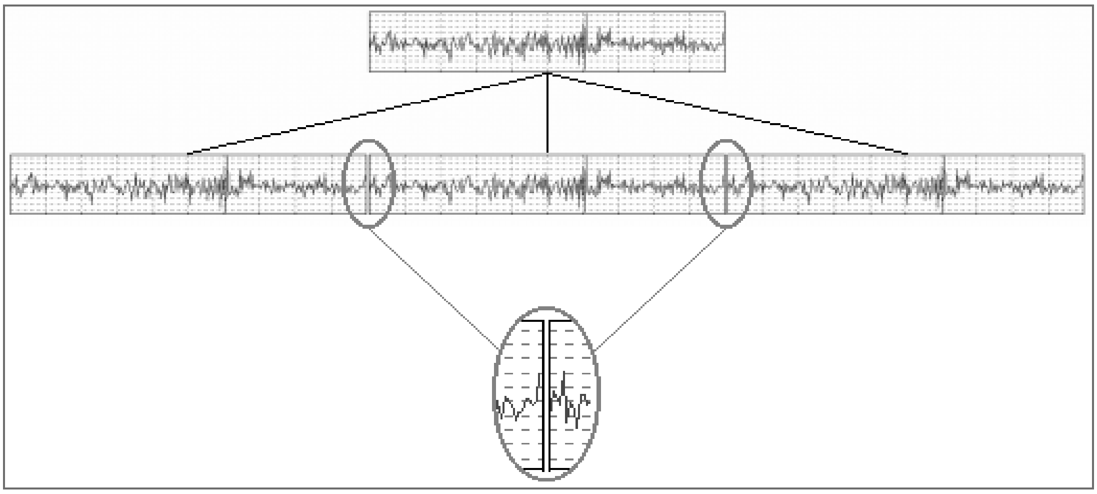


Figure 4-37 - Real data example where the ends of the sample blocks do not end at zero amplitude.

This phenomenon is called **Leakage.** The result is a broadening of peaks within the spectrum.

Figure 4-38 shows a signal of 10.9 Hz where it does not end at zero. No window is applied. The result is a widening of the bottom of the peak (the skirt). Notice it is in Linear scale. The inset is in Log scale shows how the frequency is smeared over several lines of resolution.

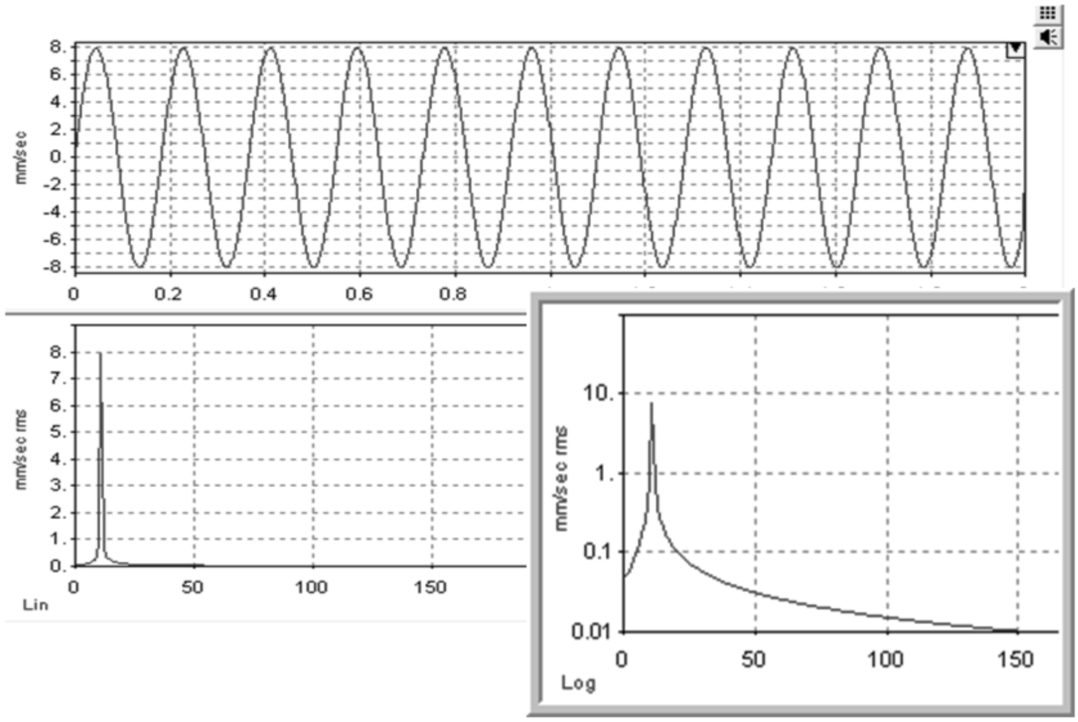


Figure 4-38 - A signal that is non-periodic within the time window and the resulting spectrum in Linear and Log scaling.

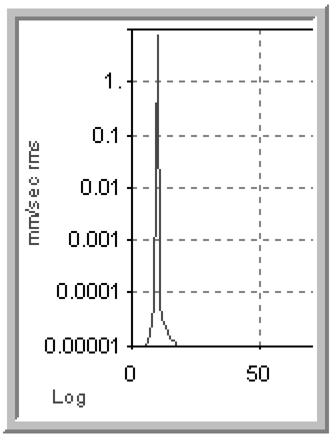


Figure 4-39 - After applying the Hanning Window

Figure 4-39 is the data after applying the Hanning Window. Notice how the skirt is now much narrower so that it is not smeared nearly as much.

To get around this problem the shape of the time record is actually changed so that there is no data at the beginning or end of the record. The ends are pushed to zero amplitude. This is known as **windowing** the data. The window has a minimal effect on the frequency content, but it does affect the shape of the spectral peaks and the amplitude levels.

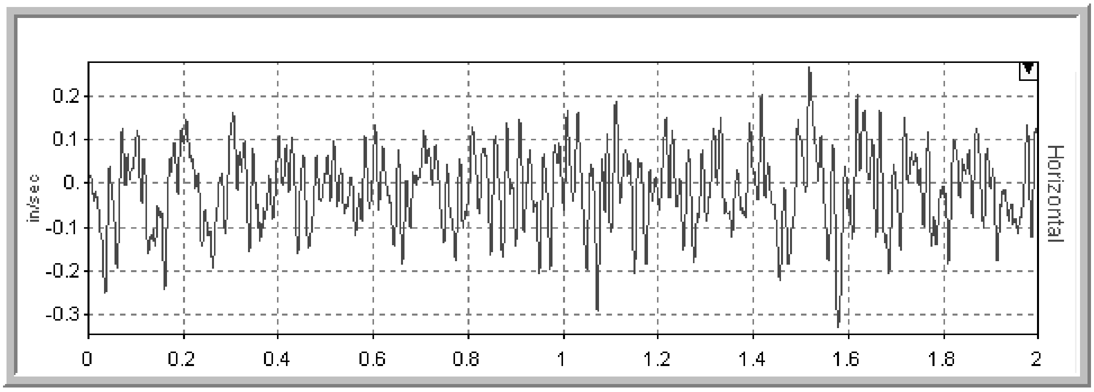


Figure 4-40 - This is the raw data before any window is applied.

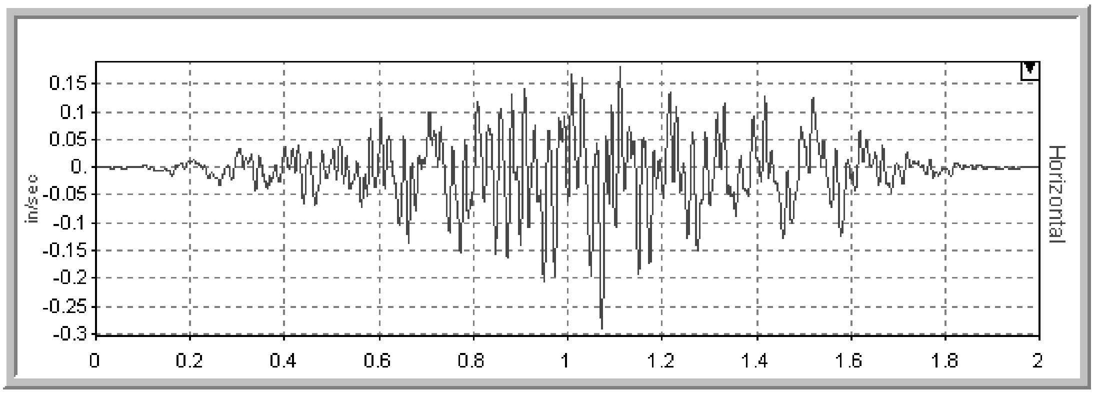


Figure 4-41 - This is the same data with the Hanning Window applied. Both ends have zero amplitude

In this illustration, (Figure 4-41) it is clear that when the windowed time records are placed end to end there is no longer any sudden change in amplitude at the start or end of the record, and thus there is no leakage.

Resolution and Accuracy

Using a window has its effect on the spectrum. Depending on the type of window and the spectrum resolution, the spectrum lines shown, both their number and their amplitude, are more or less true pictures of the measured vibration, i. e. of the time record.

The Hanning window will result in a correct amplitude when the frequency corresponds exactly with a spectrum line. However, there will also be "leakage": a spectrum line with a reduced amplitude to either side of the "true" line.

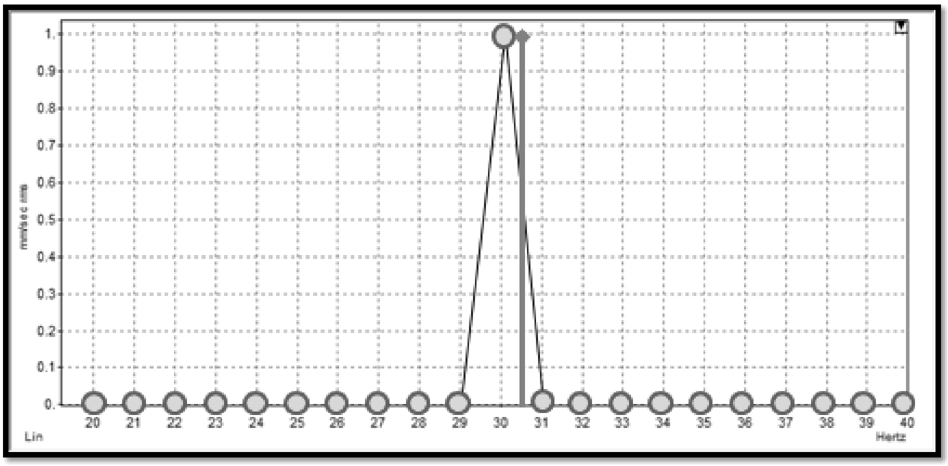


Figure 4-42 - A 30.5 Hz signal with an amplitude of 1, but bins at 30 Hz and 31 Hz

In this example the vibration occurs at 30.5 Hz but can only be shown at 30 Hz or 31 Hz, the Hanning window will produce two lines, both with an amplitude of 85%, plus small leakage. The accuracy of the peak is ± ½ the resolution.

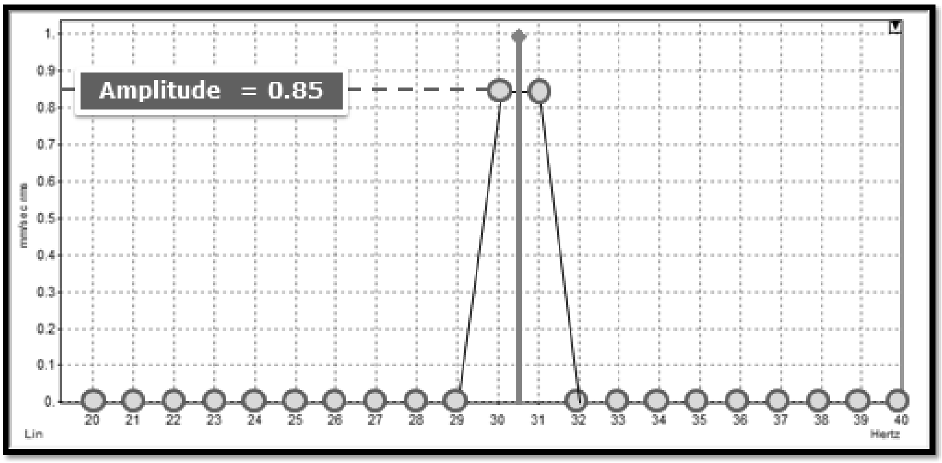


Figure 4-43 - The peak will change shape - the amplitude will be 15% less

Window Type and Bandwidth

Although the Hanning Window has the best frequency accuracy, it still affects it. To correct for this phenomenon, a Window Factor is used. The Window Factor is a multiplier to apply when determining resolution.

The Window Factor for the Hanning Window is 1.5. This means that after determining resolution in the normal manner, the result is multiplied by 1.5. This effectively decreases the resolution. The term Bandwidth is used to describe this. For example:

If the frequency range is1600 Hz and the Lines of Resolution is 800 lines, then the formula for determining resolution is 1600/800=2. Now multiply that by 1.5 and the Bandwidth using the Hanning Window is 3Hz. This means that two discreet frequencies that are closer than 3 Hz will appear as one peak.

Another way to express the resolution formula using a Hanning Window is:

Resolution = 1.5 x Fmax/Lines

There are a number of window functions to choose from, each with a different shape, and each having a different effect on the resultant spectrum. The "**Hanning**" window is most commonly used in vibration analysis of rotating machinery. It yields the best frequency accuracy although it does affect the amplitude.

Other Window types include:

- Flat Top Window – has better amplitude accuracy but the frequency accuracy is poor.
- Hamming Window – similar to Hanning window except the ends do not go to zero amplitude.
- Rectangular, Uniform, or No Window – the Rectangular and Uniform are the same as having no window at all. These can be used during special testing situations and are discussed in Category II.

Most analysts use only the Hanning Window for routine measurements.

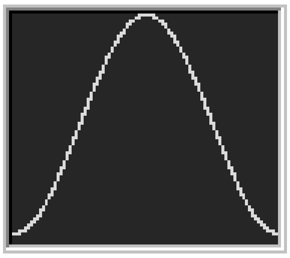


Figure 4-44 - The shape of the Hanning Window

Example: For a spectrum with an Fmax of 400 Hz and 400 lines of resolution **RES= Fmax / LOR = 400 / 400 x Window Factor = 400 / 400 x 1.5 = 1.5 Hz**

A drawback of the Hanning window is the Amplitude Accuracy is poor. If the frequency in question is periodic in the time record, its peak will fall exactly on a spectral line, and its amplitude in the spectrum will be accurate. Otherwise, the level may be attenuated by up to 16% (1.5 dB).

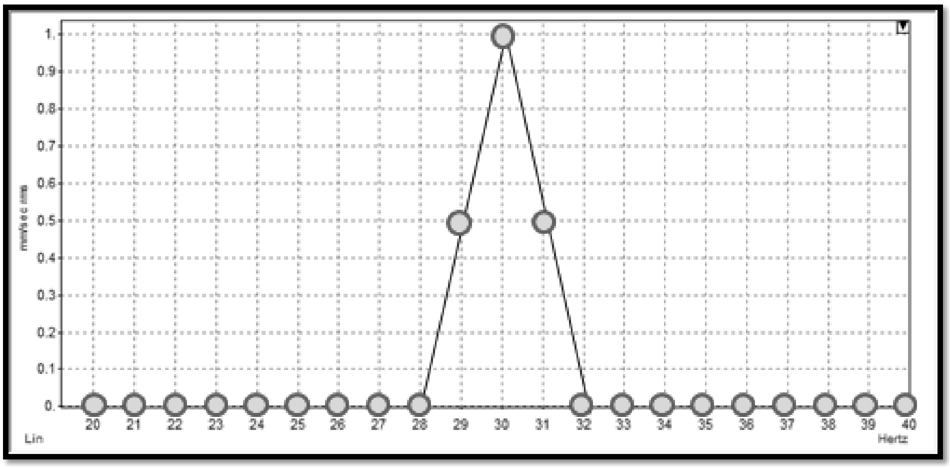


Figure 4-45 - When the frequency falls on a line, the amplitude is accurate.

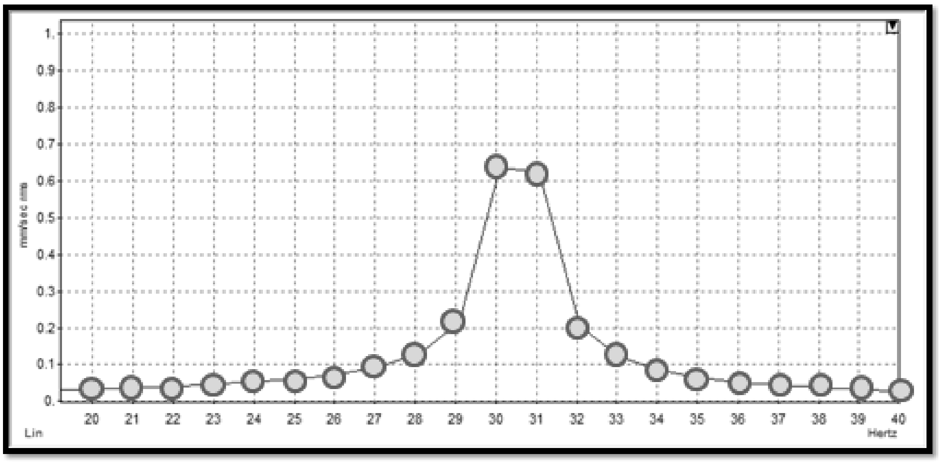


Figure 4-46 - When the frequency does not fall on a line, the amplitude is incorrect by as much as 16%.

If the speed of the machine is exactly the same from test to test, then the error will be repeatable and you can compare levels safely. But machines are rarely that stable.

The good news is that the nature of the inaccuracy is well understood, so most analysis programs have an option to compute the exact frequency and amplitude of a peak. Most automated routines that compare spectral levels to alarm limits also perform this correction.

FLAT TOP WINDOW

There is another option called the Flat Top window. It has greater amplitude accuracy with only 1% (0.1 dB) error, however the frequency resolution is poor.

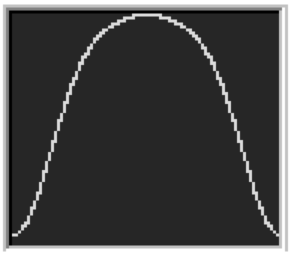


Figure 4-47 - Shape of the Flat Top window

Because vibration analysis of rotating machinery involves the detection of key frequencies, and it is quite common to have closely spaced peaks, most analysts only use the Hanning window.

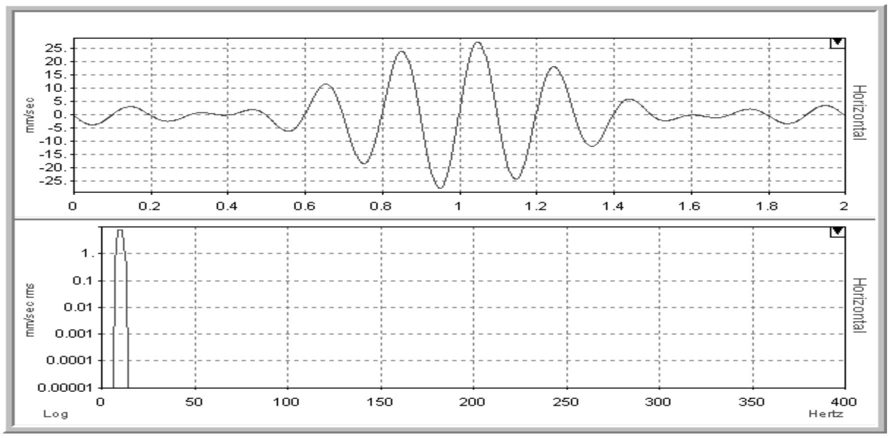


Figure 4-48 - Flat Top window has greater amplitude accuracy, but poor frequency accuracy.

RECTANGULAR OR UNIFORM WINDOW

Some analyzers provide the option for a Rectangular or Uniform Window which is actually the same as having no window at all. It is used when performing transient analysis tests such as an impact test or a bump test. Because the impact response is located at the beginning of the time record, other window types such as Hanning, Hamming, and Flat Top destroy the most important part of the waveform.

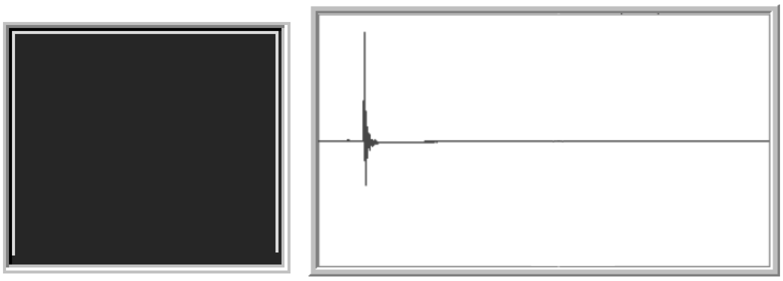


Figure 4-49 - The Rectangular or Uniform window is used for impact tests.

Transient data is typically zero at the beginning and end of the record anyway (especially if the test has been set up correctly), so leakage is not a problem.

EXPONENTIAL WINDOW

Another window type used primarily for impact testing is the Exponential Window. It is used primarily in modal analysis to window the response of a structure after impact.

Many analyzers do not offer this option.

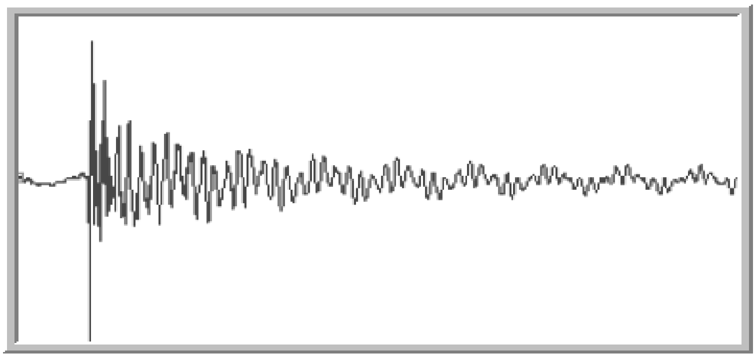


Figure 4-50 - Exponential window is used primarily for impact testing.

RESOLUTION, WINDOW, AND BANDWIDTH

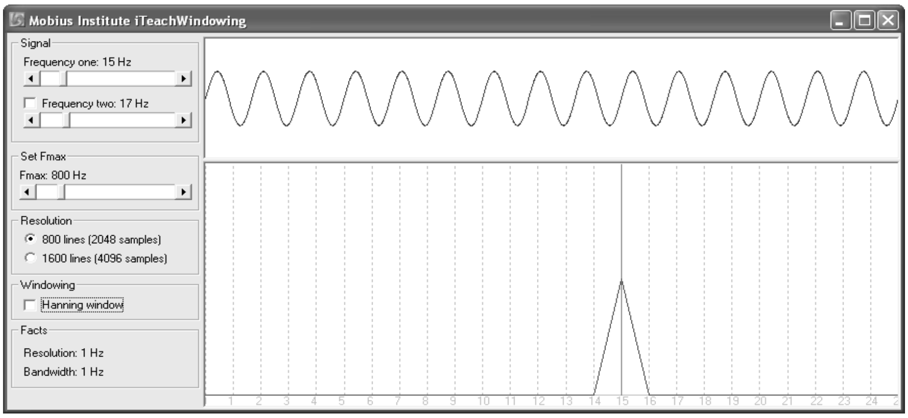


Figure 4-51

When deciding upon the data collection parameters for each machine it must be looked at individually to determine the rotation rates of the important components, and decide what the lowest and highest frequencies of interest will be. Remember to analyze harmonics of key frequencies, which will dictate the highest frequency of interest - a gearmesh frequency or rotor bar rate for example.

- If Fmax is not high enough you can miss faults
- If resolution is insufficient you cannot distinguish between peaks
- If resolution is too high, you will spend too much time collecting data
- Setting the Fmax is discussed in the analysis section

These are all very important decisions as they do influence the usefulness of the data.

Averaging

In an ideal world, the data collector would collect a single time record free of noise from a never changing vibration signal, then produce the FFT and store it. But the vibration is constantly changing slightly and there is noise in the signal. Changes occur as rotating elements go through cycles and there is random noise from inside and outside the machine.

There is a way to minimize the effects of the noise and keep more of the changes due to cycles inside the machine. The process used to correct this is called **Linear Averaging**. It simply computes the Average Value for each line in the spectrum. This process helps in maintaining repeatability so that two measurements taken 5 minutes apart should be the same.

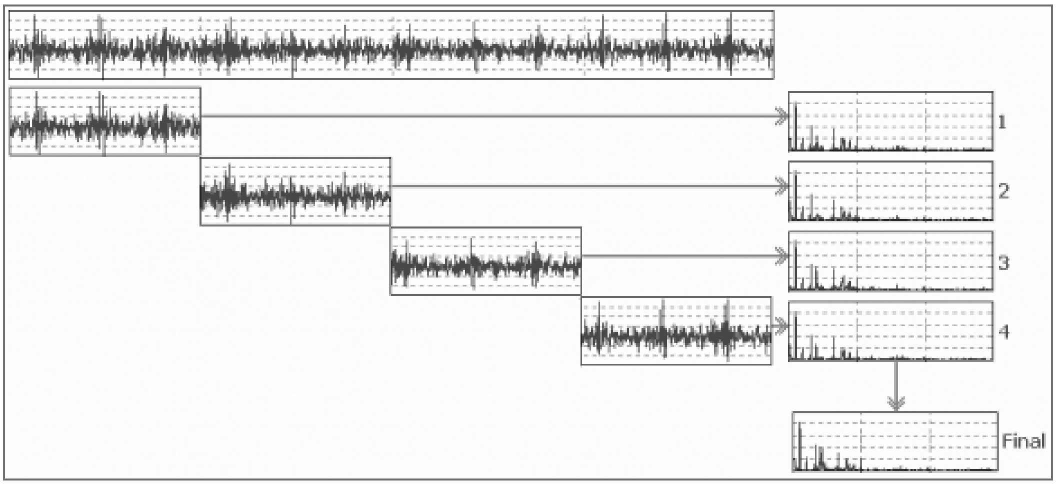


Figure 4-52 - Linear Averaging with 4 samples

Linear Averaging collects a time record, produces the FFT and holds the FFT or spectrum. The second time block is collected immediately after the first and produces the FFT and holds it. This is repeated for each average (usually 4). The spectra are then averaged to yield an Averaged Spectrum. It is stored and the rest are dropped from memory.

When the measurement calls for storing the waveform, most analyzers store either the first or last time record without any window applied. This way the analyst can see the actual time block that produced the spectrum. Some manufacturers collect and store a separate waveform after the spectral data has been averaged and stored.

Reducing Noise

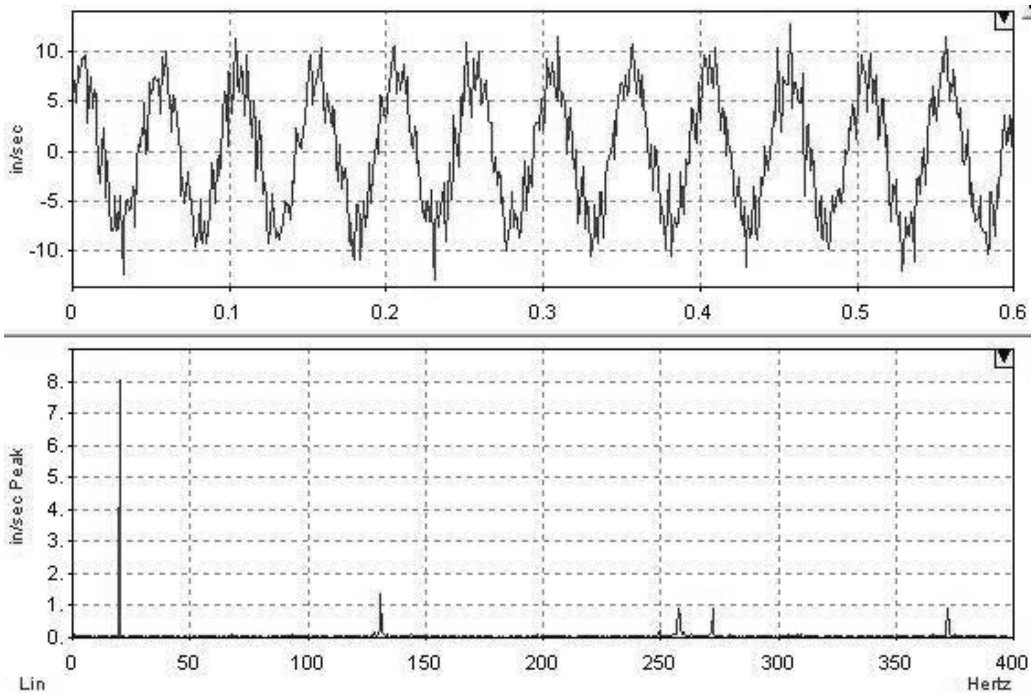


Figure 4-53 - One average. Are all the peaks real?

Figure 4-53 shows a waveform and the resulting spectrum. It is one average, or essentially no averaging. Notice the dominant peak. It is likely that it is the frequency of the background sinusoidal waveform.

But what about the other peaks? Are they real? Are they really a result of conditions in the machine such as bearing frequencies?

Additional averages help in resolving that. If it is due to a mechanical or electrical vibration it should be in every block of data. If it is random noise it will be averaged down.

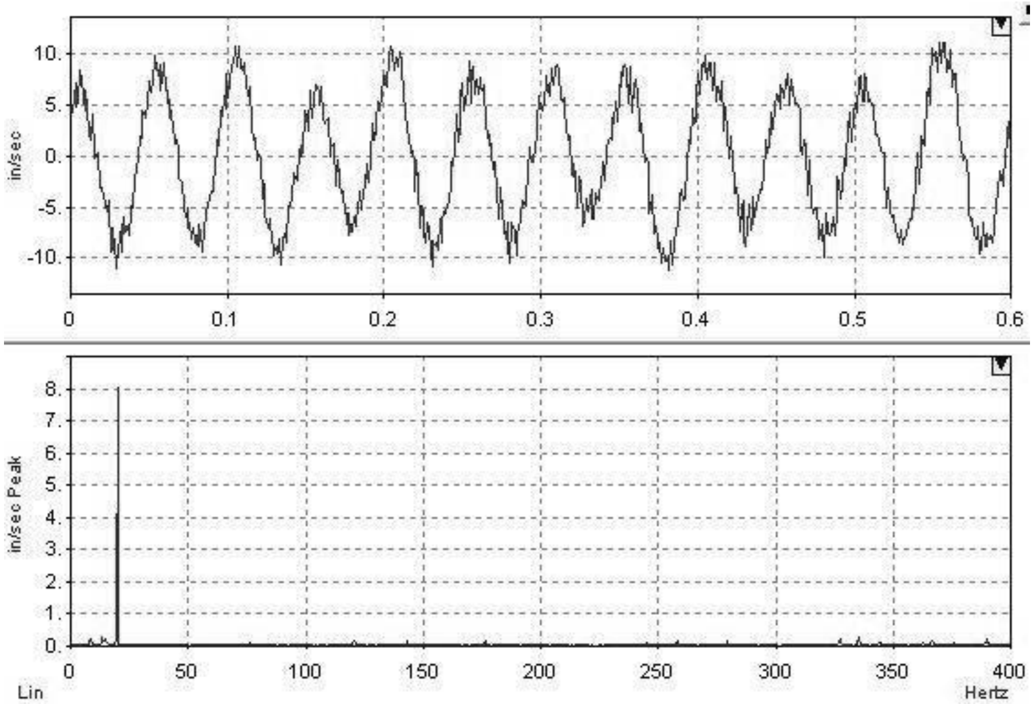


Figure 4-54 - Four averages. Most of the peaks are drastically reduced. They are random noise.

The spectrum in Figure 4-54 is the result of 4 averages. Notice how the spectrum has changed. The primary peak is still present but all the other peaks have been reduced significantly. They were random noise and were averaged down to $1/4^{th}$ the amplitude because of the averaging (4 averages). Notice that the noise is NOT removed, it is only reduced.

How many Averages should be used?

The number of averages is dependent on the machine being measured, but generally should be long enough to see the shaft rotate 6 to 10 times. If there are transient shock loads, use more averages. The goal is to average all the sources of vibration and noise.

Figure 4-55

One way to find out what is needed for a specific machine is to set the analyzer to free run and watch the data.

Another way is to set the number of averages to 30 and see how long it takes for the spectrum to settle out.

Typical machinery such as motors, pumps, fans, etc., do well with 4-10 averages with 4-6 being the most common. A key point to keep in mind is that the noise floor will have more energy in it with 4 averages. 6 averages reduces the noise 30% over 4 averages. Reducing the noise floor helps in reducing the potential for investing valuable time trying to analyze noise related peaks.

Increasing the number of averages reduces the noise further. Figure 4-56 shows the effect after 10 averages. Note that the spectrum is in Log scaling which amplifies these frequencies.

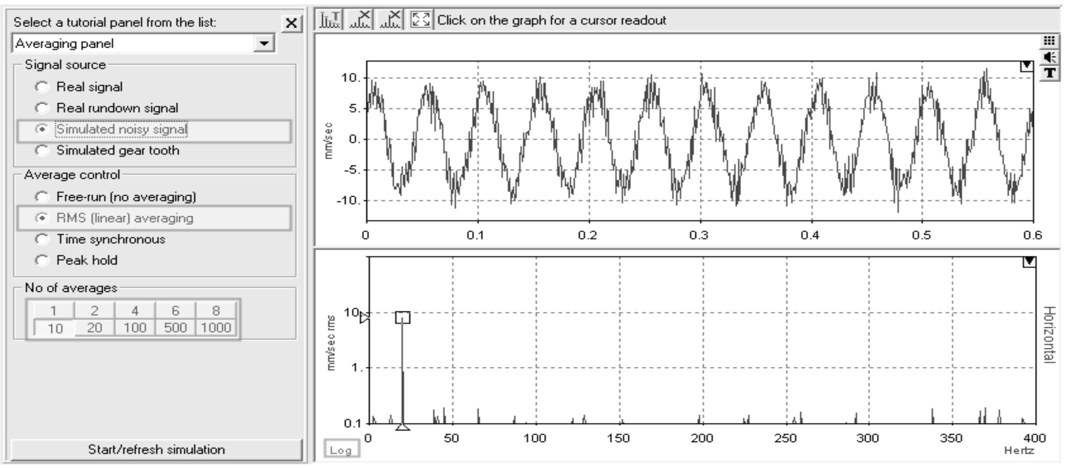


Figure 4-56 - 10 averages have been used in this spectrum to reduce noise. Note that this is in Log Scaling.

HOW AVERAGING WORKS

Linear averaging works by collecting a time record, windowing the data, converting it to an FFT, stores it in temporary memory, then collecting another time record, producing the FFT, and adding it to the average. This is repeated for each average. The averaged spectrum is then displayed. Note that the spectra are averaged by averaging each bin or line of resolution in the spectra.

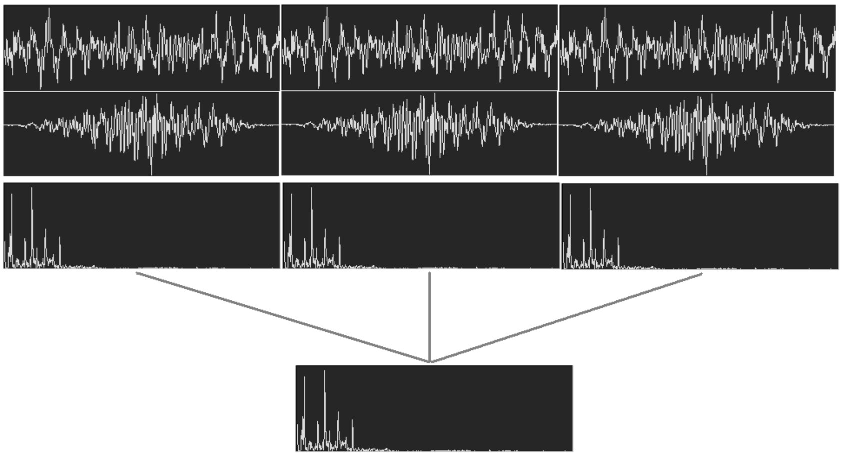


Figure 4-57 - Linear averaging captures a time record, windows it, computes the FFT, stores it temporarily, averaging the specified averages.

Notice that the person collecting the data must wait the total duration of time for all samples or averages to be completed.

But recall from the discussion on windowing, the start and end of the time record is windowed and pushed to zero. The vibration information at the start and end of the record is being "wasted." So what if the waveforms are overlapped? Instead of using an entirely new time record for each average, part of the previous record is used.

Figure 4-58 uses a 50% overlap. That is, 50% of the previous time record is used for each average. The data collector collects 1024 samples, windows them, creates the FFT and starts the average, then it collects just 512 samples, appends them to the last 512 samples of the previous time record, performs the window, produces the FFT and adds it to the averages.

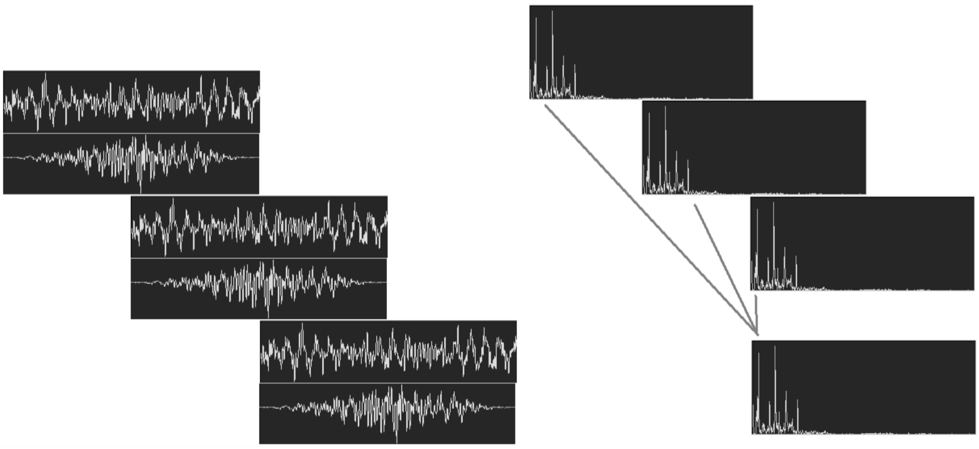


Figure 4-58 - Time Records with 50% overlap saves data collection time

The net effect of this overlap is twofold:

1. It requires less time at each measurement point to collect the data

2. The data that was "wasted" at the beginning and end of each time record is now fully represented in the spectrum.

Below is another example illustrating the time saved for each measurement point. This may seem miniscule but when multiplied by 500 points to collect in one day, that is significant.

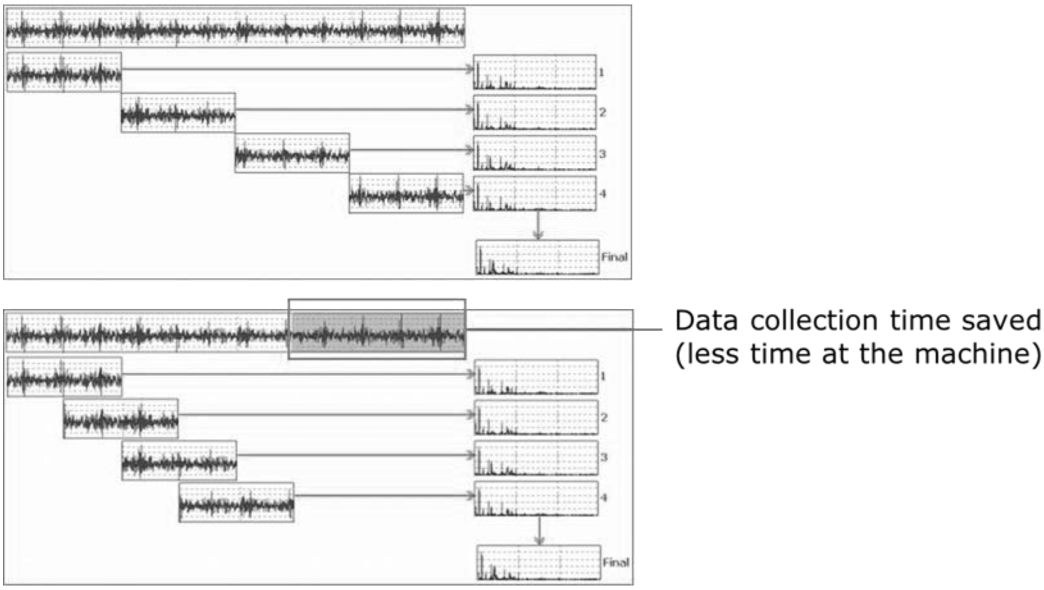


Figure 4-59

Peak-Hold averaging

Another method of averaging data is to use the Peak-Hold averaging. It is not really averaging at all but is a comparison of values in each bin of the spectrum. It **holds** the highest **peak** for each line of the spectrum.

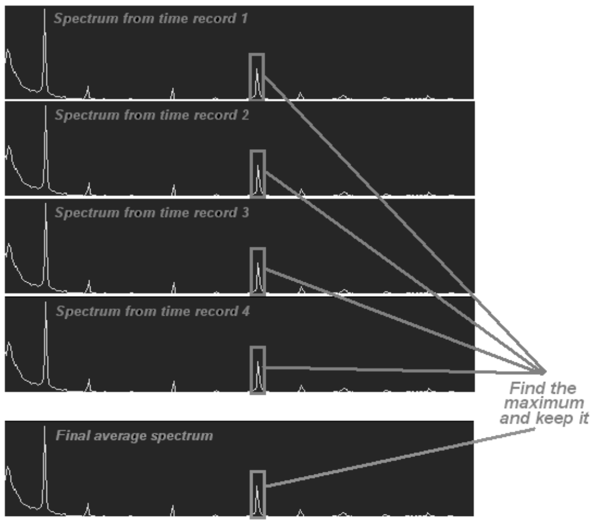


Figure 4-60 - Peak-Hold Averaging

For a spectrum with 4 averages using the Peak-Hold method, it works as shown in Figure 4-60.

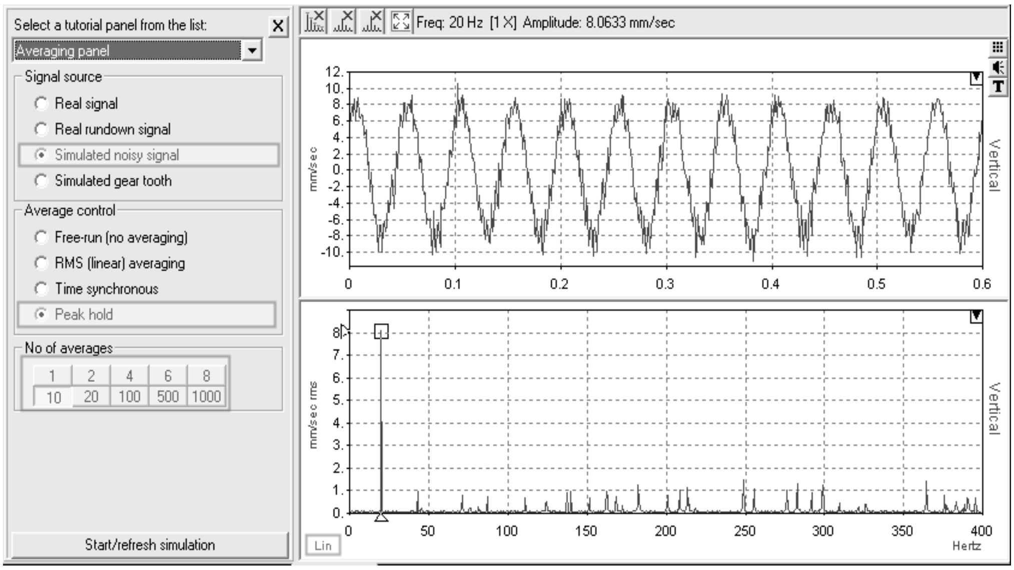


Figure 4-61 - Peak Hold averaging with 10 averages - holds the highest value in each line. The noise floor is not reduced.

Peak hold averaging is normally not used in routine data collection. Instead it is used for special tests such as Run-up, Coast Down, and Bump Tests.

Time Synchronous Averaging

To this point all the averaging methods are averaging in the spectrum or frequency domain. Time Synchronous Averaging averages in the Time Domain.

Recall that linear averaging does not actually *remove* noise, it just *reduces* noise, improving the statistical accuracy of a noisy spectrum. Averaging the Time data can actually reduce the level of noise by removing noise from the data. It therefore can be used to uncover low level signals that may have been obscured by noise. This noise is not just the random noise from surrounding machines or from the processes, but it really means sources of vibration that are not wanted in the measurement.

Figure 4-62

Each time record is captured and averaged with previous time records. The final averaged time record is often analyzed and/or used to create a final spectrum.

Time Synchronous averaging uses a trigger to synchronize the beginning of the record. The trigger must be synchronous with the signal of interest. This is typically done with a tachometer once-per-revolution transducer. When analyzing a gearbox, for example, it is necessary to trigger off the shaft of the gear of interest.

Synchronous signals, the signals that are fixed in the time record, will remain in the waveform (because they occur at the same time relative to the trigger reference). Non synchronous signals will eventually average to zero.

This method is commonly used to extract signals from gearboxes.

Recall from Understanding Signals, that waveforms add and subtract from each other depending on the relative phase. Averaging waveform in the normal way can cancel out to zero. The illustration below shows this well.

Figure 4-63 is waveform averaging with non-triggered data capture. The top two waveforms are 180 degrees out of phase therefore cancelling each out. The bottom two are also 180 degrees out of phase and cancel each other out.

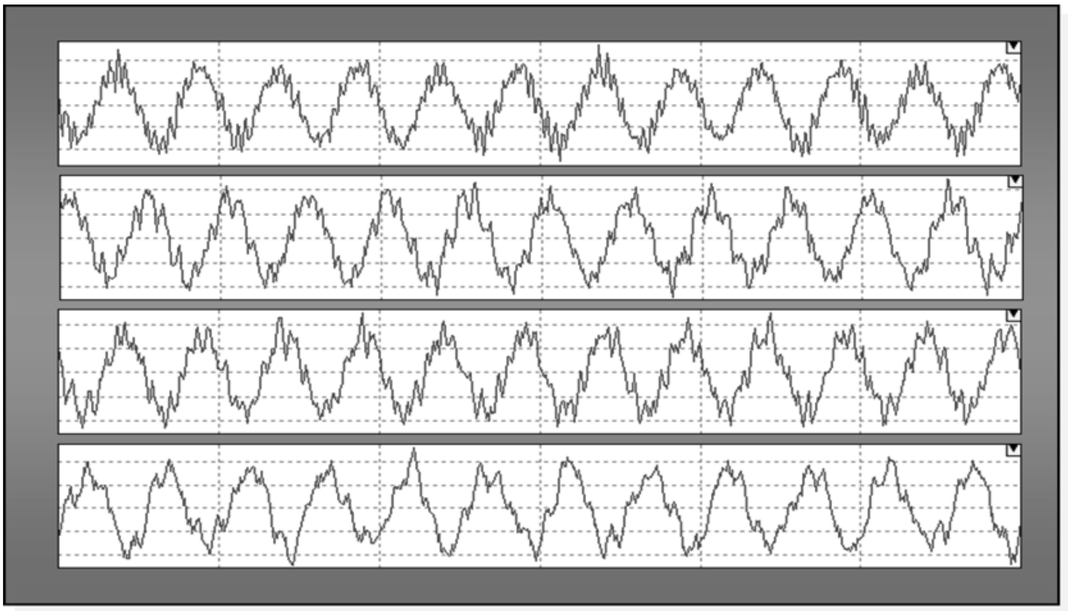


Figure 4-63 - Non-triggered waveform averaging: the top two waveform are 180 degrees out of phase and sum to zero. The bottom two do the same, resulting in a zero value.

When the data collection is triggered so that each waveform starts and ends at the same relative phase, the data is very useful.

Figure 4-64 uses the same data as in the previous example, but synchronizes the data capture with a trigger. This data can be averaged.

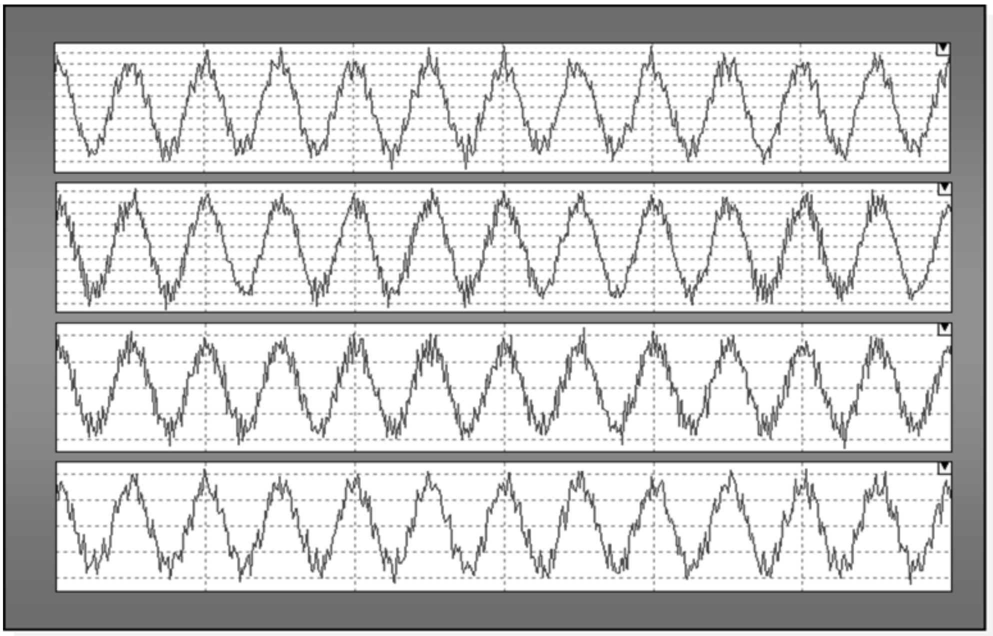


Figure 4-64 - Time Synchronous Averaging triggers the data collection so that all time records are in phase and can therefore be averaged.

Time synchronous uses many averages to remove the noise and yield a useful spectrum. Typically 100 averages or more are used.

The simulator demonstrates how this works. A noisy signal is averaged in the waveform using 100 averages. The result is a time waveform without much noise. The spectrum is shown in Log rather than linear scaling, showing the noise is still there. In linear scaling, these would appear to be gone. Frequencies that are not harmonics of shaft speed are removed. See Figure 4-65.

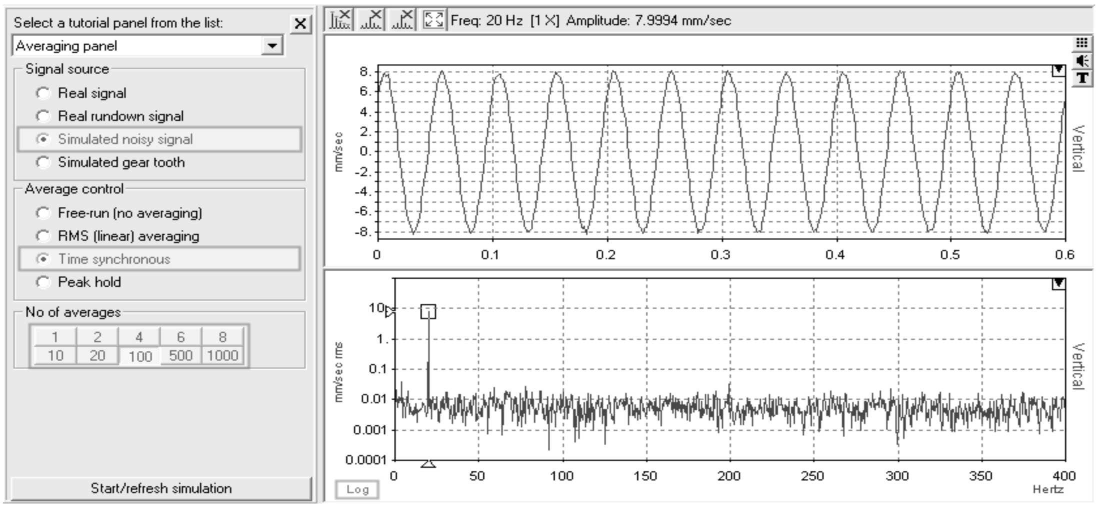


Figure 4-65

Time Synchronous averaging can be very useful in determining if a signal is coming from the machine being measured or a nearby machine. If the vibration from the nearby machine is not exactly the same frequency, or if it is out of phase with the triggered signal, it will be averaged out.

Another good application is in gearboxes.

Figure 4-66 shows an example of Time Synchronous on a shaft of a gearbox. Every 10th revolution has a higher amplitude value which is clearly seen in the waveform. In fact, because the data is captured using a trigger, it can be seen that the 6th tooth from the tach reference is the one with the problem.

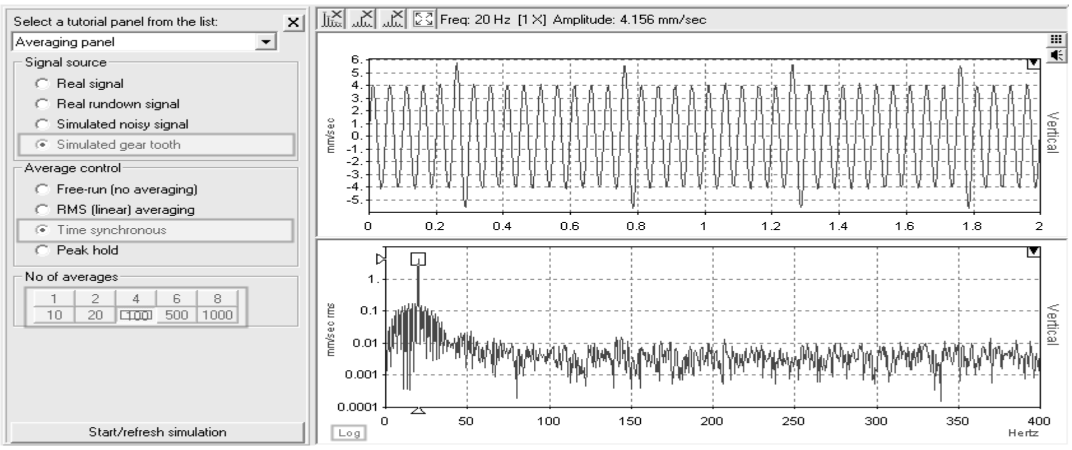


Figure 4-66 - Time Synchronous Averaging of gear with a bad tooth. Waveform is very useful.

Notice how little information is present in the spectrum. Although this is only simulated data, similar results can be expected from real data. If a problem is suspected from the gearbox, look in the time waveform.

The need for the trigger/synchronizing signal (the tachometer signal) means that this method is not often used. In addition, a large number of averages are typically required, so the measurement can take a long time. However, in certain situations (such as gearbox analysis), the additional effort will pay dividends.

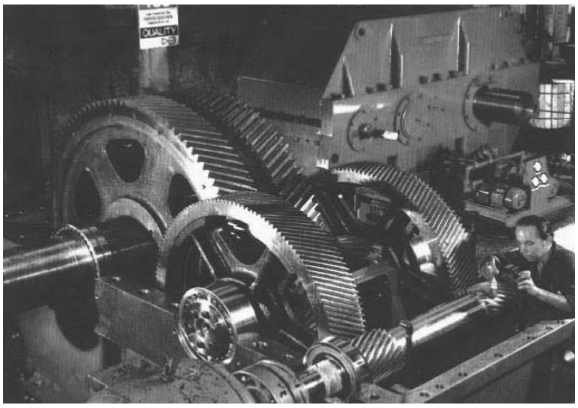
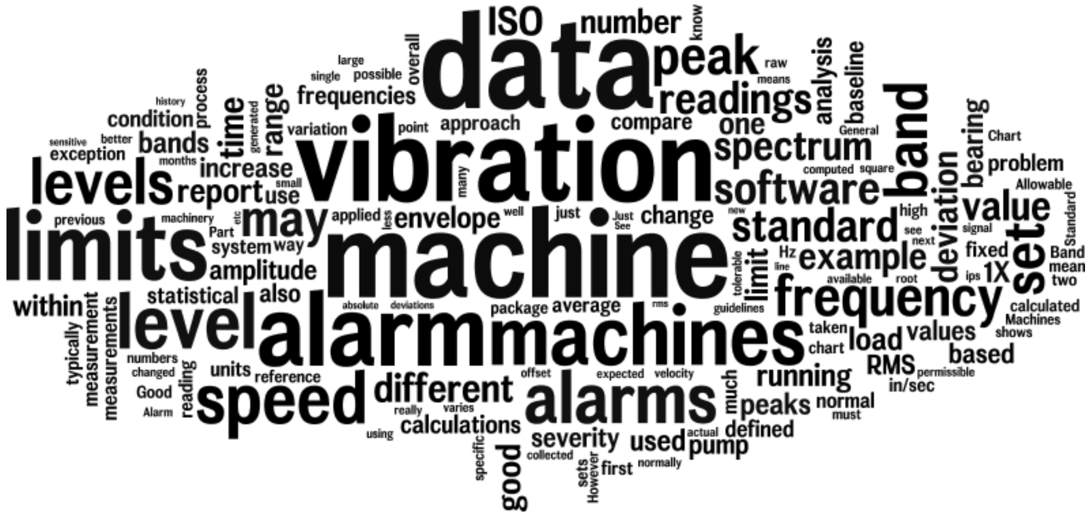


Figure 4-67



Chapter 5

Time Waveform Analysis

Objectives:

- Calculate Resolution, sample rate, and record length
- Select the best units for the application
- Describe 3 distinct waveform patterns
- Describe Beating, Modulation, and Transients
- Describe how to analyze waveforms with spectra
- Describe or Recognize Looseness, belt damage, and cavitation in waveform
- Describe how waveform is used to analyze gearboxes

Time waveform analysis is a complex topic, but one that the analyst should understand in order to correctly diagnose faults. There are classic waveform patterns and steps to analyze the waveform. Faults can be used to diagnose machinery faults.

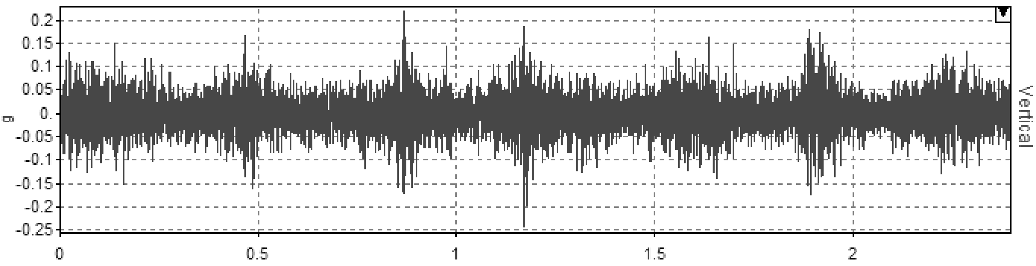


Figure 5-1

Waveform analysis is essential in the analysis of gearboxes and bearings. The waveform can confirm findings in spectral data.

A simple Time Waveform

This time waveform has a strong sinusoidal pattern and therefore one dominant peak in the spectrum corresponding to the running speed of the machine. This pattern is from a machine that is out of balance. The time waveform and spectrum are in velocity units.

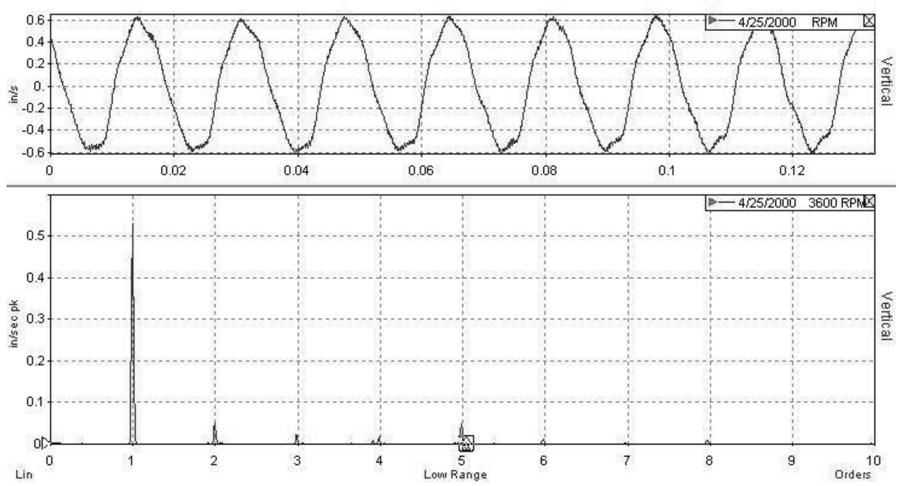


Figure 5-2 - The strong sinusoidal waveform produces a dominant peak in the spectrum

A More Complex Waveform

This second example is from the drive end of the motor on a belt-driven fan. The belt is damaged so that every rotation of the belt hits the pulley.

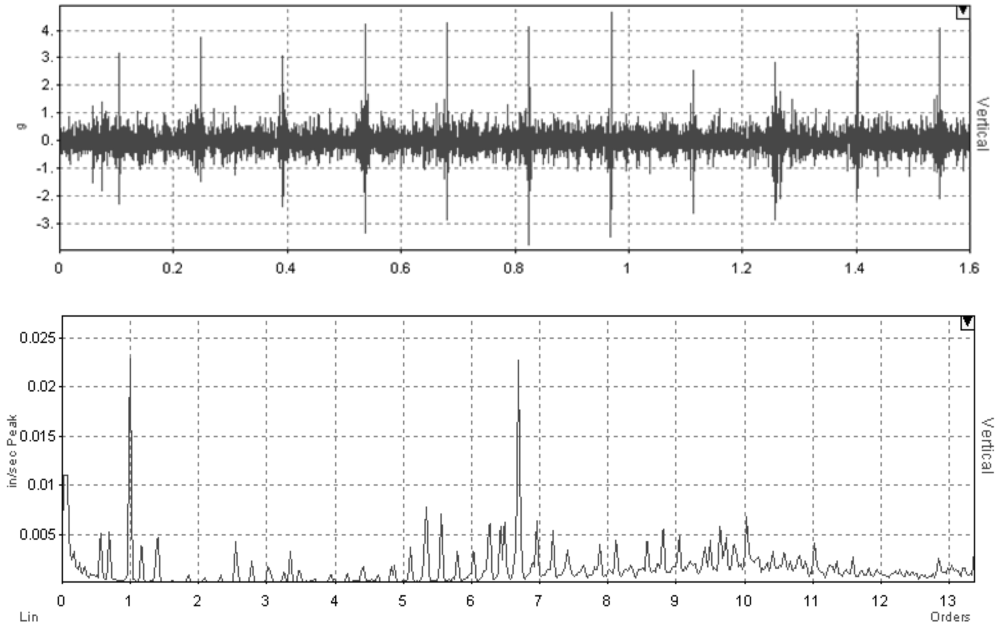


Figure 5-3 - A damaged belt produced this waveform. It is much clearer than the spectrum.

The time waveforms in both examples tell us a great deal. In the second example, the waveform provides more information than the spectrum. The waveform often reveals the key to accurate diagnosis.

Where does the waveform come from?

If the output of the sensor was displayed on an oscilloscope, it would be a time waveform similar to the pattern that comes from a microphone. Sometimes the signal can appear quite simple, like a "sine wave". At other times, however, the pattern can appear quite complex.

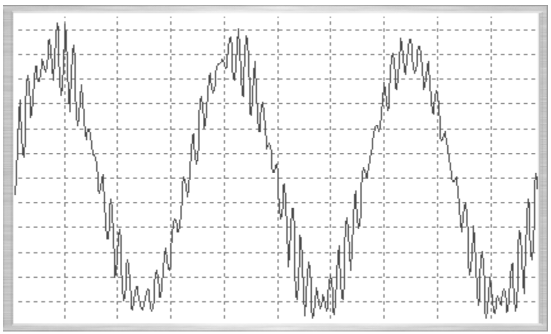


Figure 5-4

The waveform displayed on the data collector or computer screen is not the original, pure analog waveform that came from the sensor. It is digitized so the computer can store and manipulate it.

There are three parameters that can be controlled:

- The sample rate
- The length of the time record
- Whether to integrate the signal

If the signal was sampled a zillion times a second for one minute, it would provide a very clear picture of how the waveform changed as the shaft turned and the balls rolled around the raceway, but the data collector would run out of memory. On the other hand, the signal could be sampled at 20 times per second for a tenth of a second, but the waveform would reveal no information at all. Obviously, the ideal setting is somewhere in between.

The two sources of vibration in Figure 5-5 were sampled at different rates and look very different.

The first time waveform is 16384 samples over 4 seconds. The second time waveform is also 16384 samples, but over a 12 second time span.

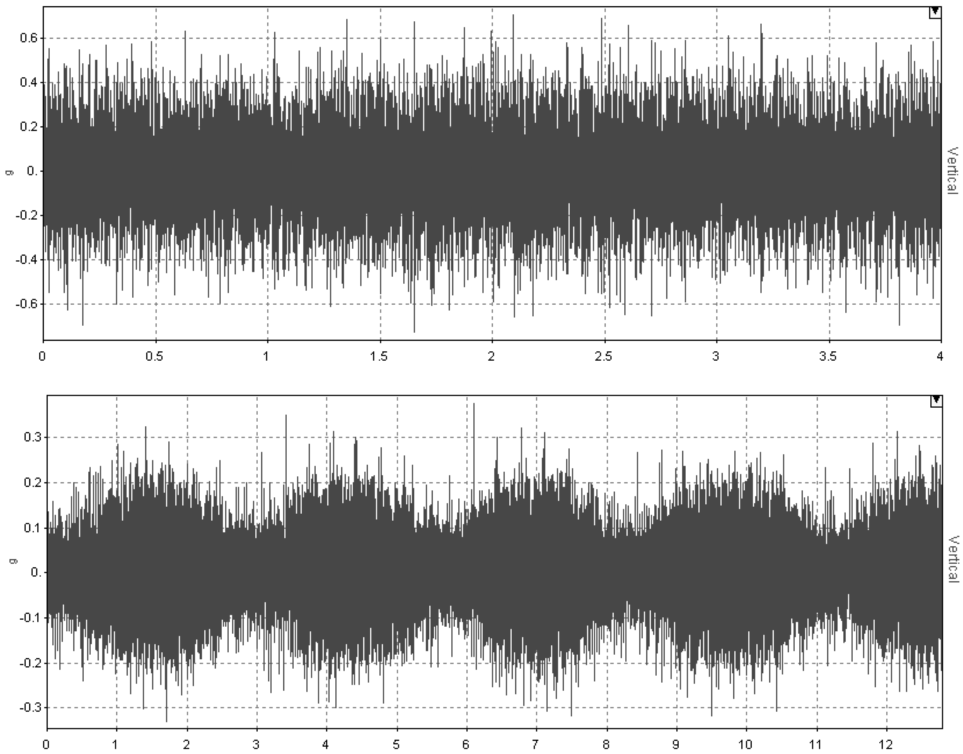


Figure 5-5 - Two waveform with same # samples for differing time spans.

For routine data collection, the recommend settings for the time waveform are to collect 6 shaft revolutions with 2048 or 4096 samples. (The 2048 or 4096 samples is equivalent to selecting 800 lines or 1600 lines of resolution in the spectrum.)

<u>**Time Window and Number of Samples**</u>

Example:

Shaft speed = 1770 CPM or 29.5 Hz

Time for one revolution = 1/29.5 = 0.034 seconds

Six shaft revolutions = 6 x 0.034 seconds = 0.205 seconds = desired time waveform length.

For 2048 samples and a time window of 0.205 seconds, the sample rate =2048/0.205=9990 samples per second.

Therefore the **Fmax** =9990/2.56 ~3900 Hz.

(For 4096 samples, the Fmax would be ~7800 Hz)

If the analyzer does not have an option for 3900 Hz, use the next HIGHER Fmax.

Many systems collect the time waveform as part of the spectrum measurement. Either the first or last set of 2048 samples (for an 800 line spectrum) are saved in the data collector and then saved to the database. So in that case the spectrum definition defines the length of time in the waveform.

Time = Number of Samples / Sampling Rate

Since both the Number of Samples and the Sampling Rate have a multiplier of 2.56, it can be dropped out to simplify the formula to:

Time = Lines of Resolution /Fmax.

For routine data it is easy to set the time window too long. Too many cycles of rotation hampers analysis unless a large number of samples is also collected. (Number of samples = LOR x 2.56)

Don't set the time window too long.

Example:

Shaft turning speed = 1770 CPM or 29.5 Hz

Fmax = 20 orders = 20 x 29.5 = 590 Hz

Lines of Resolution (LOR) = 800

Time = LOR / Fmax = 800 / 590 = 1.36 seconds

1.36 seconds ~40 shaft revolutions

Note: Fmax = # samples / (Time window * 2.56)

Use Acceleration or Velocity or Displacement?

The other issue to consider is whether to look at the time waveform in units of acceleration, velocity or displacement. A lot could be said on this topic, but in most cases it is best to use acceleration (e.g. g's). The acceleration waveform will highlight (retain) the important high frequency impacting information. If the signal is "integrated" to velocity, the higher frequency information is reduced or lost, and double integrating to displacement will remove the remaining high frequency information. (The rules are different for journal bearing machines – displacement is the unit of choice.)

The two signals in Figure 5-6 in come from the same position and axis of the same machine. You can easily see that the high frequency information is far more obvious in the acceleration waveform (and spectrum).

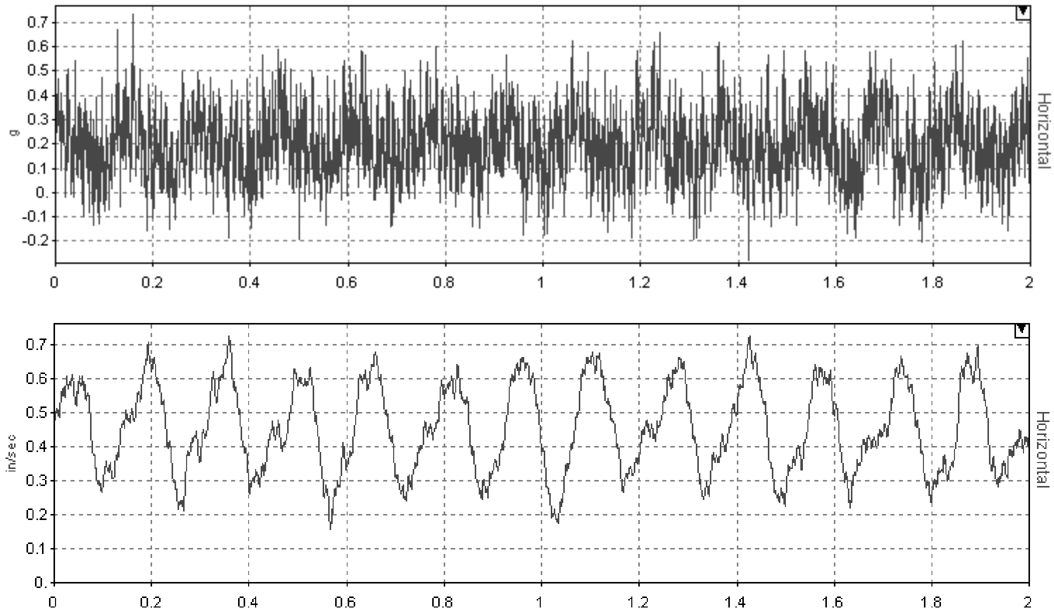


Figure 5-6 - Top waveform is in Acceleration - g's and the bottom is in Velocity. Same measurement point on the machine.

It is usually best to collect and store the waveform in Acceleration. Recall that Acceleration is proportional to the Forces in a machine. Force of the vibration impacts can be very destructive and degrade the condition rapidly. Those forces can be from a degraded bearing, cavitation, misalignment or looseness problem. The fact that there is considerable force warns the analyst that the machine is being damaged and that failure is imminent.

The Vibration Signal

When analyzing time waveforms there are a few terms that must be familiar: the peak, rms, peak-to-peak, measurements and how they are calculated. Also the period and frequency. These are all shown below, and are discussed in more detail in the "Vibration Analysis" chapter.

A key issue is that if the time between two "events" in a time waveform is measured, and then the inverse is computed, it yields the frequency. With that information the spectrum can be searched for the corresponding peak, and more importantly, it can be compared with what is known about the machine: speed, bearing forcing frequencies, belt rate, etc.

Hertz = Hz = Cycles per second

RPM = Revolutions per minute

CPM = Cycles per minute

CPM = RPM = Hz x 60

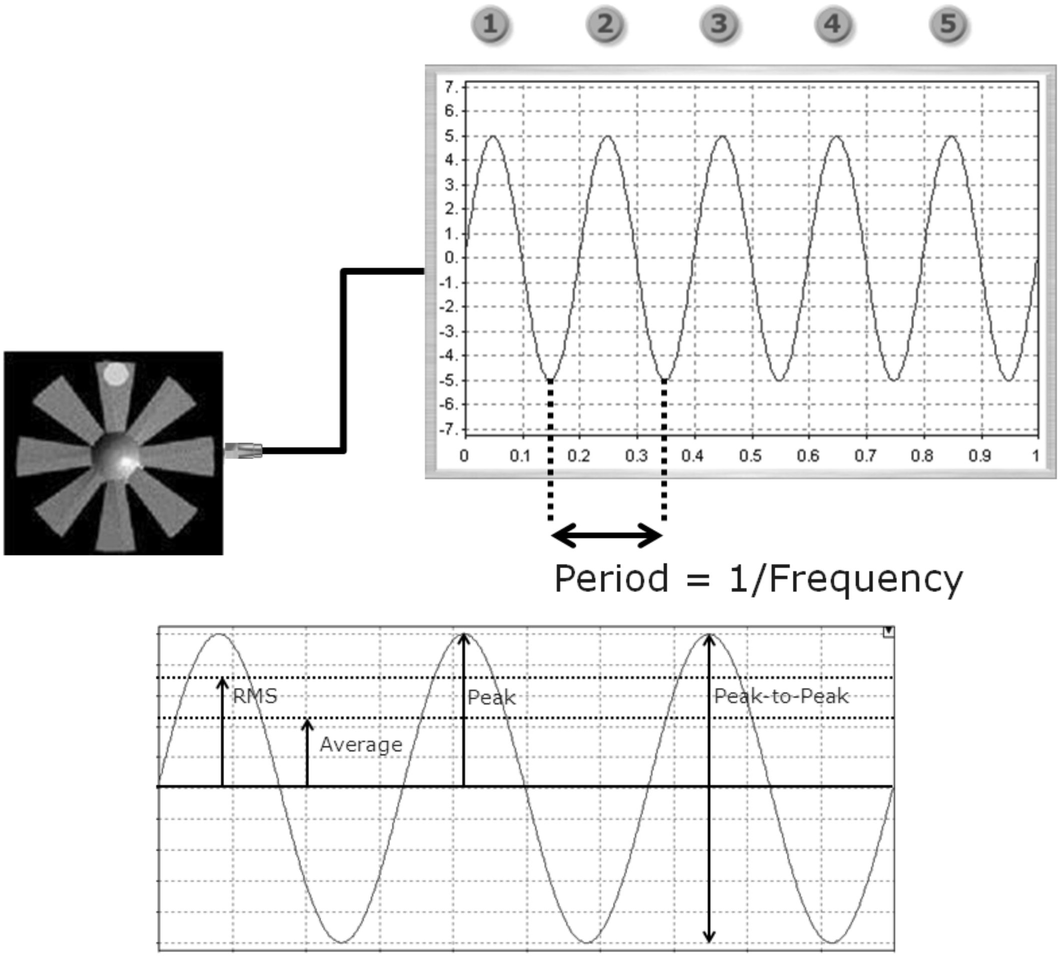


Figure 5-7

Waveform Patterns

In order to perform time waveform analysis it is VERY helpful to understand the relationship among mechanical phenomenon, signals, and the spectrum.

The analyst should recognize sinusoidal patterns, transients and pulses in the waveform (and harmonics, sidebands, or a raised floor in the spectrum) and know why and how they got there and relate that to what is happening inside the machine.

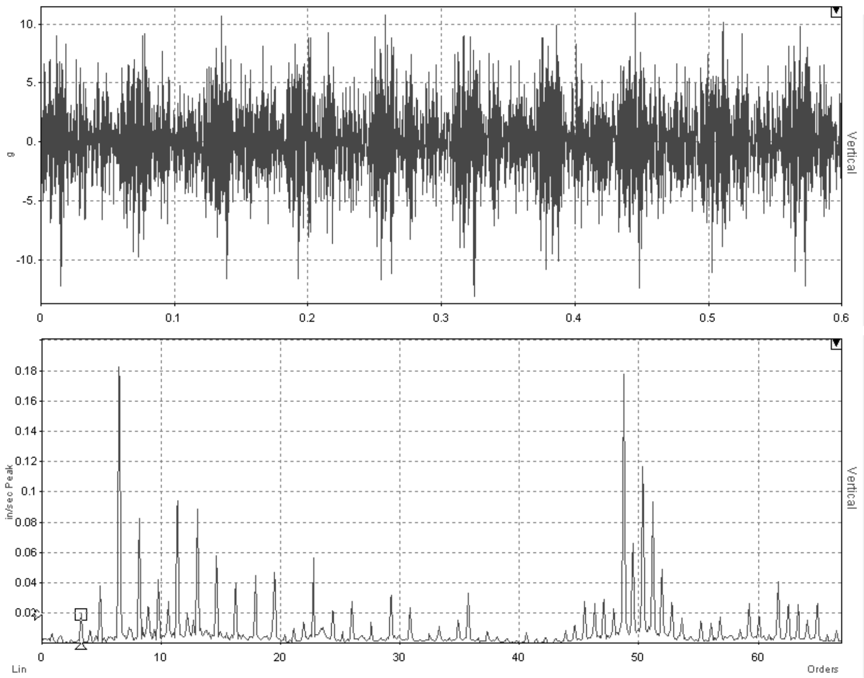


Figure 5-8

Waveforms and Spectra

A single frequency waveform produces a single peak in the spectrum.

As new signals are added, with different amplitudes, the waveform becomes more complex, and the spectrum has additional peaks.

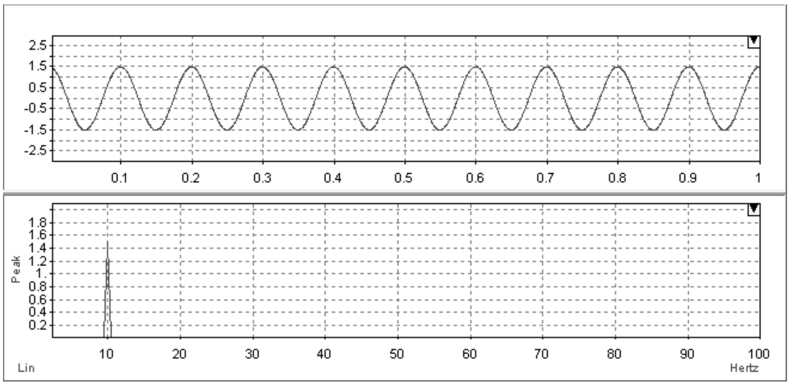


Figure 5-9 - Simple 10 Hz signal

Figure 5-10 has two signals, a 10 Hz and a 15 Hz signal with zero degree phase difference.

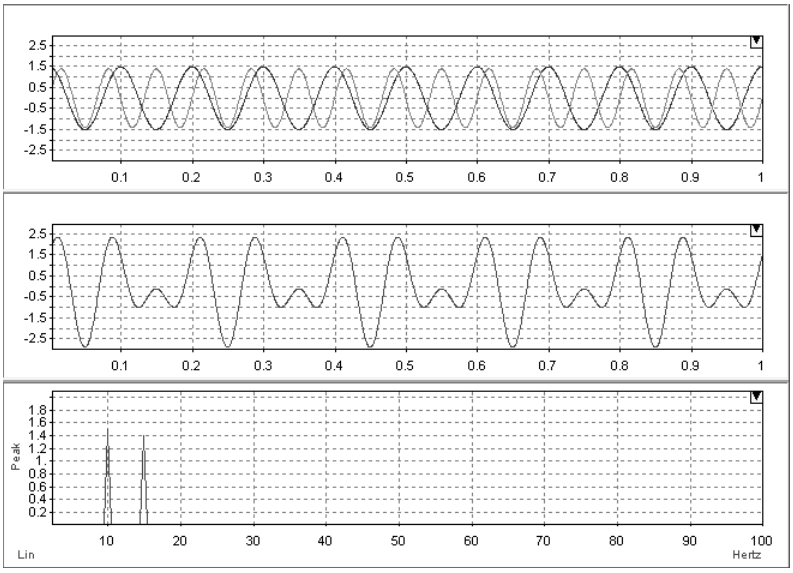


Figure 5-10- A 10 Hz plus a 15 Hz signal with 0 degrees phase difference

The resulting waveform appears very different from the two individual waveforms.

The spectrum has a peak for each of the two signals.

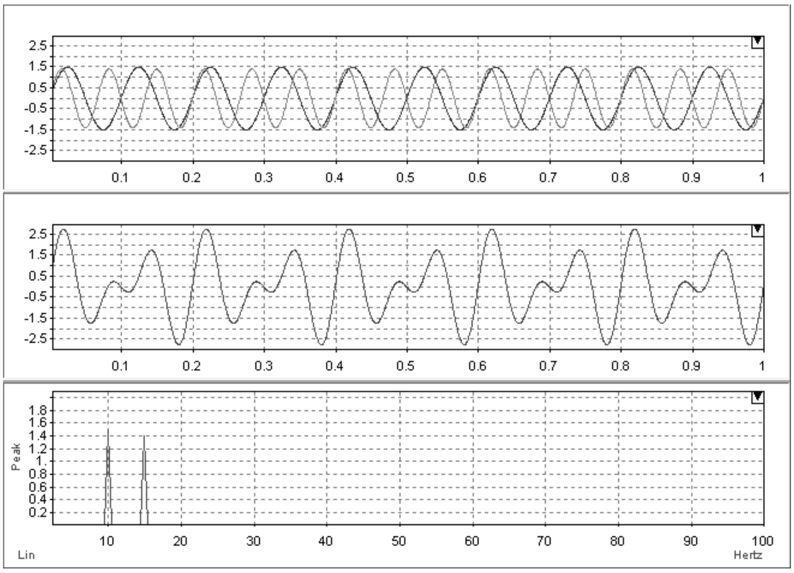


Figure 5-11 - The same 10 Hz and 15 Hz but with a 90 degree phase difference.

Figure 5-10 and Figure 5-11 are made up of the same signals and produce the same spectrum.

However, their composite waveforms are very different.

Phase of the signals do make a difference in how the composite waveform appears.

"Beating"

The term "beating" describes the phenomena that occurs when two signals are less than 4 Hz apart

In this Example the two signals are a 200 Hz signal and a 202 Hz signal. They will go in and out of phase, adding and subtracting from each other. The sound that is heard is a droning sound that rises and falls in amplitude. The frequency of the droning depends on the spacing of the peaks. Closer peaks cause a longer droning frequency.

There is 0.5 seconds between the peaks in the waveform which is a frequency of 2 Hz. The beating sound will rise and fall 2 times per second.

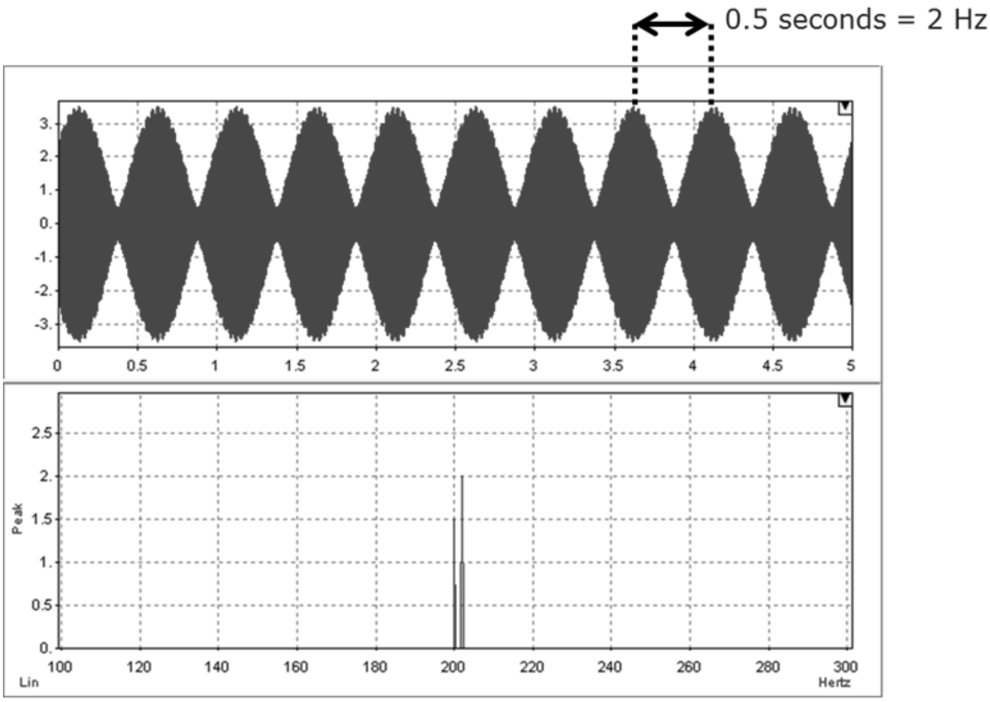


Figure 5-12 - Beating occurs when two signals are closely spaced. They go in and out of phase.

Note that "Sum and Difference" frequencies may be in the spectrum. The Sum frequency is equal to the sum of both frequencies: 200 Hz + 202 Hz or 402 Hz.

The Difference Frequency is the difference between the two: 202 Hz – 200 Hz = 2 Hz. So peaks at 2 Hz and 402 Hz may be present.

Amplitude Modulation

Amplitude Modulation may at first glance look similar in the waveform to a Beat Frequency. But it is vastly different and the spectrum is different. It occurs when two signals, far apart from each other in frequency, are tied together and one of them varies in amplitude.

The signal that fluctuates in amplitude periodically is called the carrier frequency. The other frequency is called the modulating frequency. In the example, the two frequencies are a 20 Hz signal and a 200 Hz signal. The high frequency signal is the "carrier" and is modulated by the 20 Hz signal. In other words the 200 Hz signal fluctuates 20 times a second.

So the waveform looks similar to the one associated with "Beating." However, the underlying mechanism is quite different.

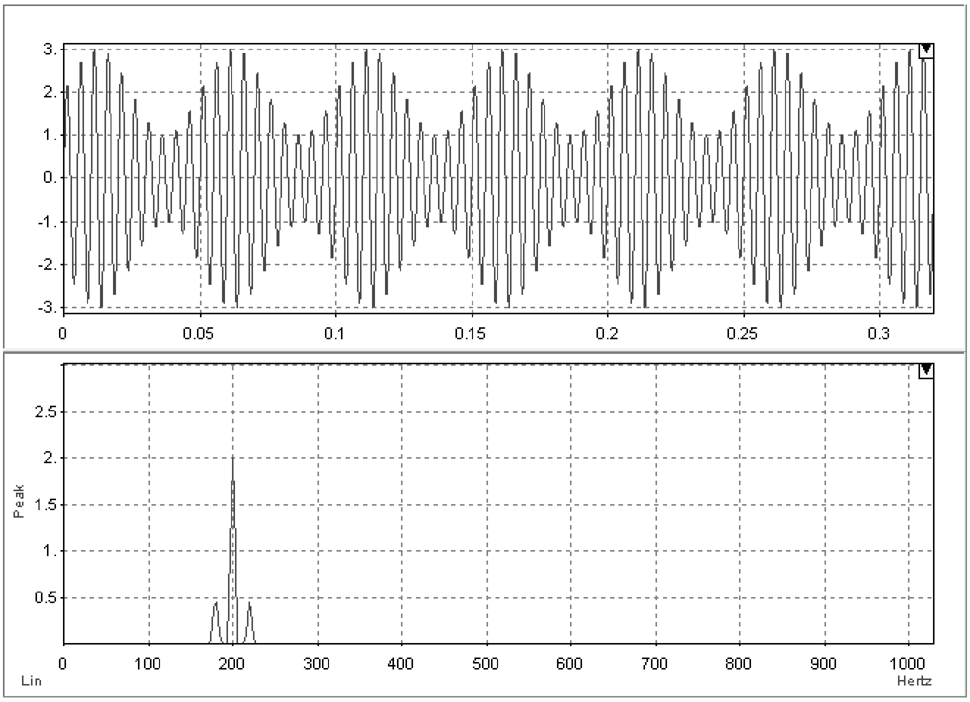


Figure 5-13 - Amplitude Modulation of a 200 Hz "carrier" and a 20 Hz signal.

The spectrum will have a peak at the frequency of the carrier with sidebands spaced at the frequency of the modulated signal, which in this case is 20 Hz.

So there will be peaks at 180 Hz, 200 Hz, and 220 Hz. Additional sidebands may be present spaced at 20 Hz apart.

What produces this phenomenon in the real world of our plant machinery?

It is especially common in rolling element bearings and gearboxes.

Here is a typical scenario that occurs in machines. A ball is rolling around inside a bearing – a ball with some damage. As it rolls around, the damaged area impacts the inner and outer race, which generates vibration. The frequency of the vibration can be calculated, as it depends upon the size of the ball and the diameter of the inner and outer race. But the amplitude of the impacts is not constant – and therefore the vibration is not constant. This occurs because, as the ball rolls into the load zone of the bearing, the forces are greater, reaching its greatest level when it is in the center of the load zone. But as it rolls out of the load zone, the force of the impacts is reduced, reaching its minimum level when the ball is on top of the inner race (out of the load zone).

The result is that the amplitude of the vibration rises and falls periodically. The "carrier" frequency is the ball spin frequency, and it is modulated by the cage frequency (the rate at which the ball moves around the bearing). The spectrum has a peak at the "ball spin" frequency, and another peak at the "fundamental cage" or "FT" sidebands.

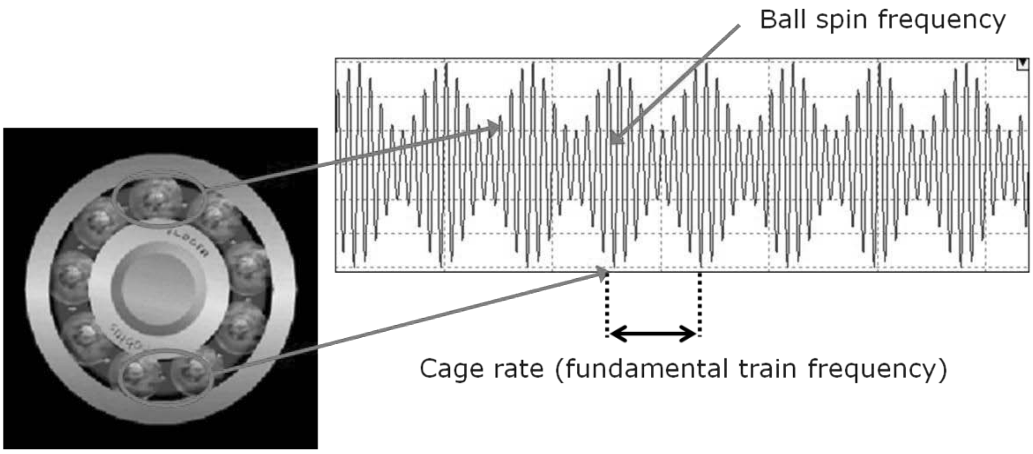


Figure 5-14 - The Ball-spin frequency is the "carrier" frequency modulated by the cage frequency.

The same scenario occurs with the inner race defects. Since the inner race is tied to the shaft, it goes in and out of the load zone, changing the amplitude at the rate of the shaft speed. The carrier frequency is the inner race defect frequency and it is modulated by shaft turning speed.

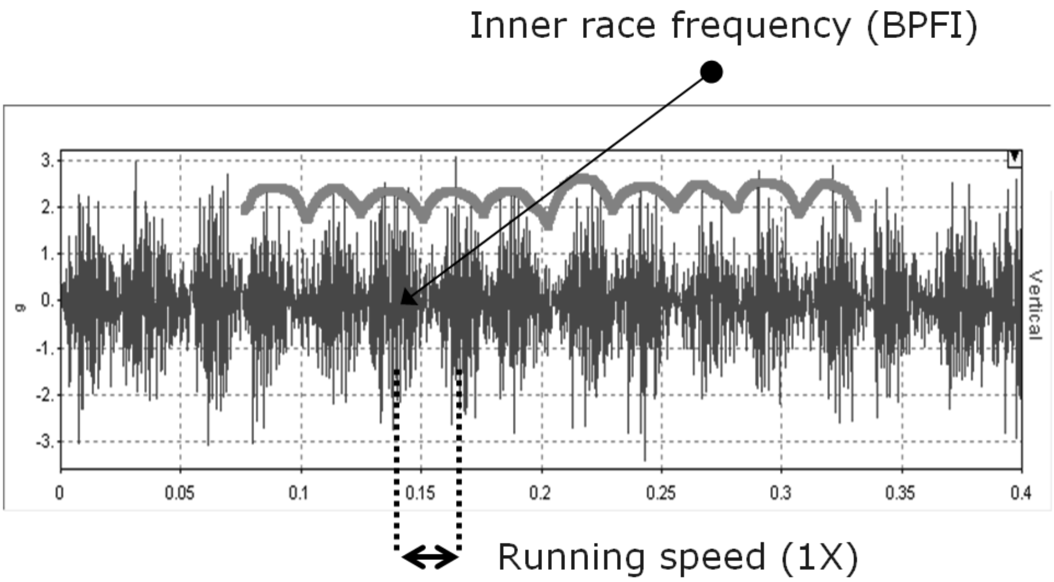


Figure 5-15 - An inner race defect being modulated by shaft turning speed.

Amplitude Modulation and Gears

Another classic case of Amplitude Modulation is misaligned gears. In this particular case there is modulation of both gears. The larger, slower gear has the highest amplitudes and dominates the waveform, but the pinion is there, too.

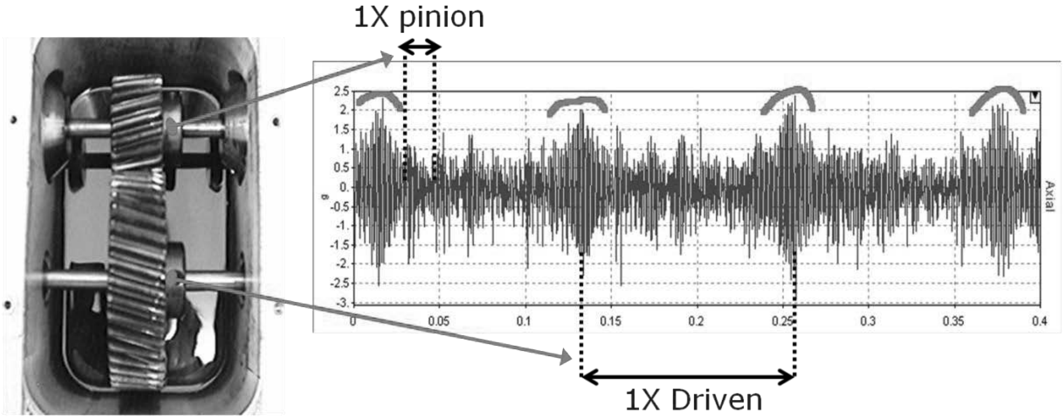


Figure 5-16 - Amplitude modulation is visible for both gears.

The spectrum has multiple 1x sidebands of the slower shaft around the carrier frequency which is the gearmesh frequency.

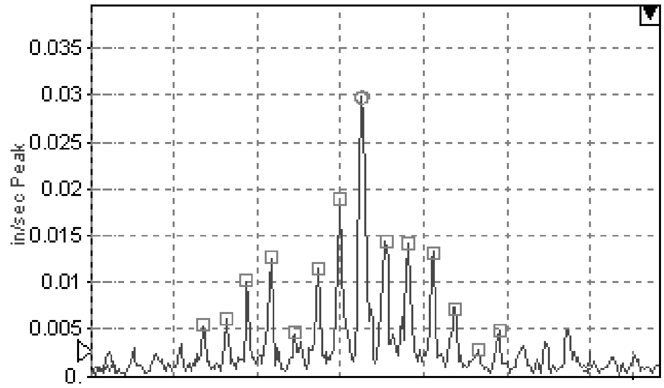


Figure 5-17

"Non-linear" clipped vibration

Another very common phenomenon is "non-linear" vibration, or truncated waveforms. If movement in one direction is restricted, then instead of a classic sinusoidal signal, the waveform is clipped.

In this example there are two 100 Hz signals, one clipped, and one normal. The normal sine wave has just a single peak in the spectrum, but the clipped waveform has harmonics – peaks at 100 Hz, 200 Hz, 300 Hz, and so on.

In more severe cases, the harmonics will appear at 1/2 the frequency, or 1/3 the frequency.

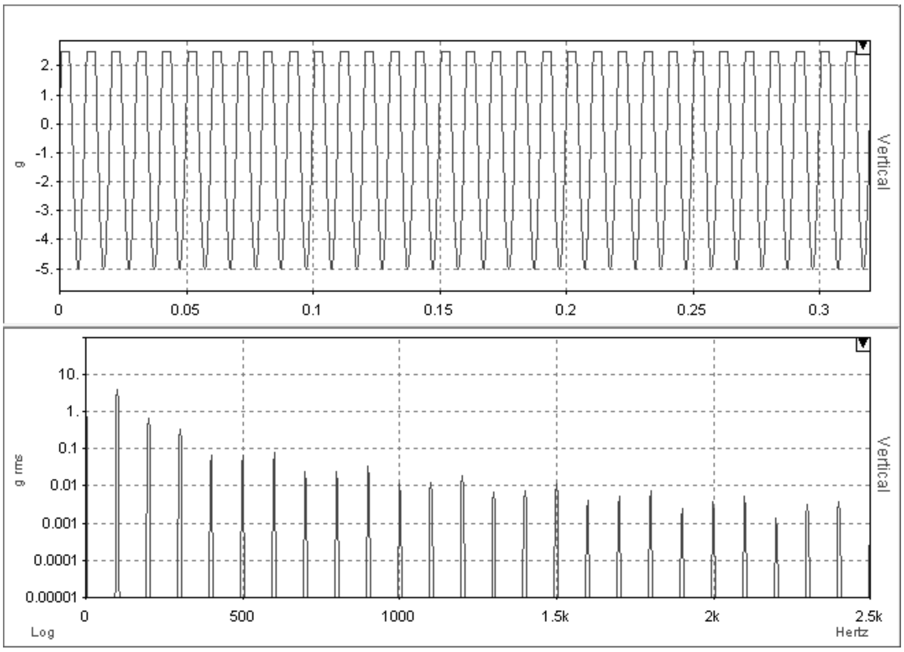


Figure 5-18 - A clipped signal produces harmonics in the spectrum.

Impacting

Impacts can easily be seen in the waveform. Impacts can occur as defects on rollers strike the inner and outer race of a bearing; impacts occur as loose parts rattle; and there are other causes. If we were to strike a machine and look at the waveform it would have one pulse or transient, and the spectrum would have energy at all frequencies. The spectrum does not have any peaks, because there are no repeating patterns in the waveform, but the noise floor has been raised. So, if impacting occurs we would expect the noise floor of the spectrum to be raised.

In this example, there was actually an electrical spike in the waveform. The raised noise floor is somewhat apparent, and the classic "ski-slope" at the beginning of the spectrum is there. If a ski slope is present, it can be assumed that there was a severe impact or some other kind of transient. The measurement should be repeated and reason for the ski-slope should be found (there are a few possibilities).

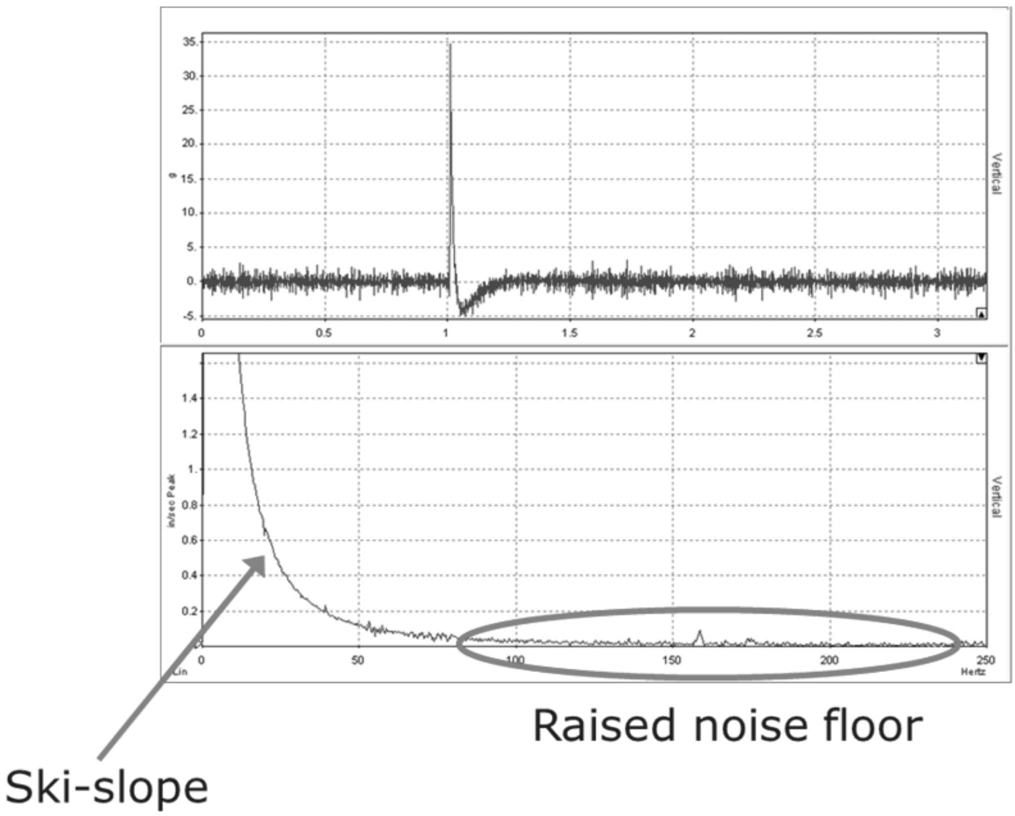


Figure 5-19 - Impacting produces no peak in the spectrum except the ski slope, but raises the noise floor.

Rotating Looseness

Some conditions produce repetitive, periodic impacts. One of these conditions is rotating looseness. Three characteristics will be present in the spectrum.

First, because there is impacting, the noise floor will be raised. This is due to the fact that impacts impart all frequencies into the spectrum, meaning there will be some amplitude at all frequencies. That raises the noise floor.

The second characteristic is that because the impacting is repeating periodically, there is a peak in the spectrum at the periodic rate, which in this example is shaft turning speed.

The third characteristic is that because the impacts are non-linear (clipped or not even + and -), there are harmonics of that frequency.

In severe looseness examples sub-harmonics can occur. They can be ¼ and ½ harmonics or 1/3 and 2/3 harmonics.

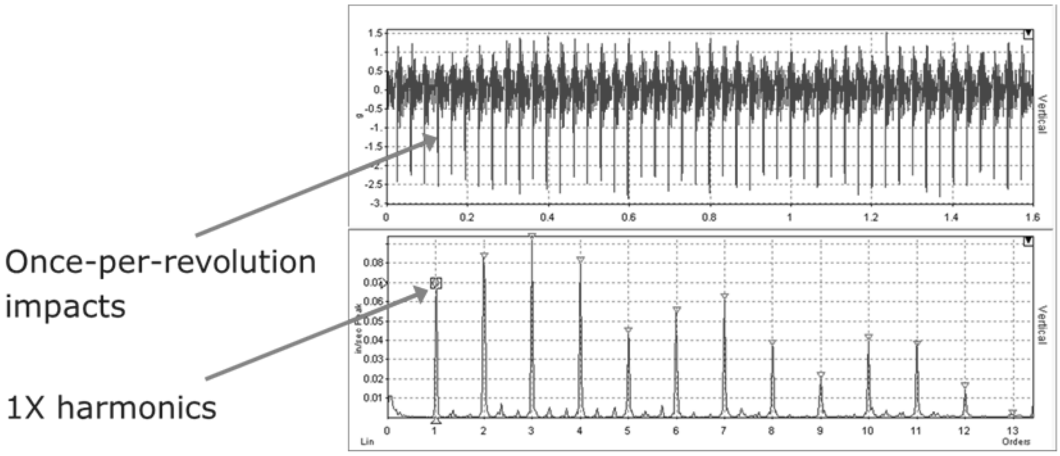


Figure 5-20 - Repeating Periodic impacts produce harmonics and a raised noise floor.

Relating the waveform to the spectrum

Sometimes it seems it is difficult to relate occurrences in the waveform to peaks in the spectrum. But taking the time to zoom-in can help us find what is going on in the spectrum.

Looking at the difference in time between two impacts gives the period. The inverse is the frequency which then can be compared to the spectrum to see what may match up.

In the example here, the period between impacts related to a non-synchronous peak that corresponds with the outer race frequency of a bearing. Since the impacts of this outer race are repetitive, it has a peak at that frequency, and since the impacts are non-linear, there are harmonics of that frequency.

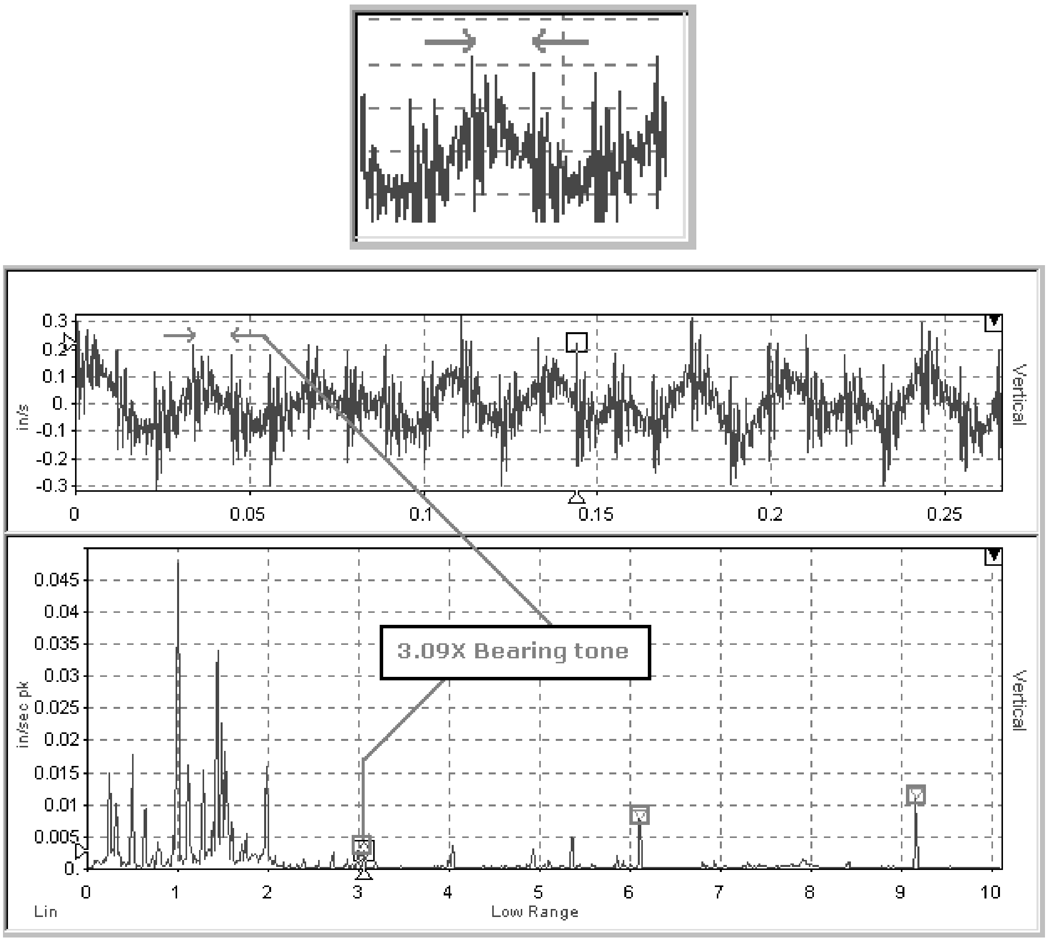


Figure 5-21 - Expand the waveform to see the peaks. Convert the time span to frequency and compare to known machine frequencies.

Belt Damage

When a belt is damaged and a chunk of rubber is missing, that defective area makes an impact as it passes over the pulleys. It is common for it to show up at 2x the belt frequency (or belt rate) because the defect passes over two pulleys in one rotation of the belt.

In this example the time between the pulses is 0.1441 seconds which relates to 6.94 Hz or 416 CPM. The running speed of the machine is 1792 CPM, so this frequency is 0.23 orders.

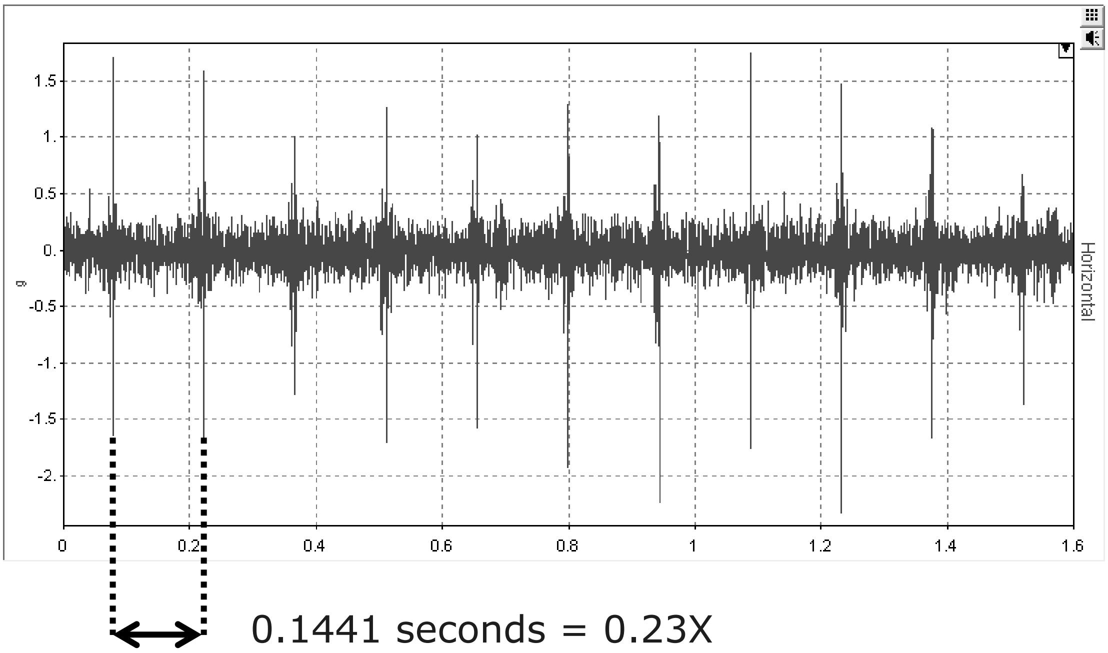


Figure 5-22 - The time between impacts coincides with the belt rate which is 0.23X.

The Belt Rate calculation is shown below.

$$Belt\ rate = \frac{\pi \times Sd}{\sqrt{c^2 - (R2 - R1)^2}}$$

Where

Sd = sheave diameter

c = the center to center distance between the two pulleys

R1 = radius of one pulley

R2 = radius of the other pulley

In the simplest explanation, the belt rate is a comparison of its circumference to the circumference of the pulleys. More specifically, it is the pitch length of the belt compared to the pitch circumference of the pulleys.

Pump Cavitation

When this waveform data from the pump is played back through speakers it sounds as if it pumping gravel. There are many random impacts.

The waveform data shows many pulses, but unlike other examples used, these pulses are not evenly spaced. They are not periodic. They are random.

In the spectrum the noise floor is raised, especially at the higher frequencies.

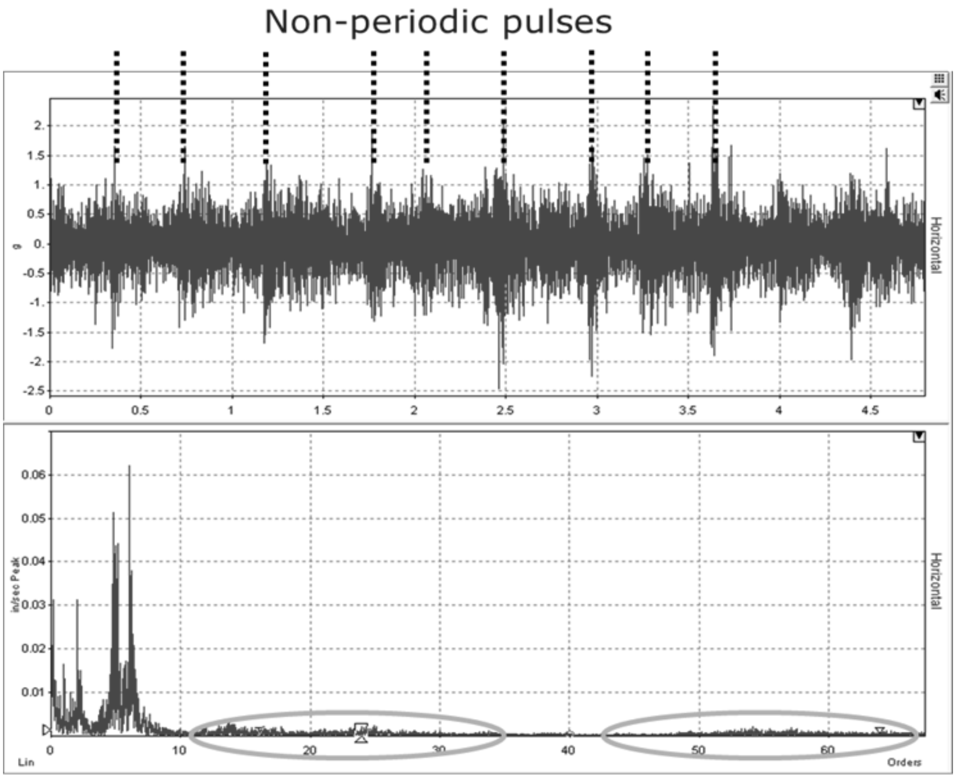


Figure 5-23 - Cavitation produces non-periodic pulses. The floor of the spectrum is raised. Fmax is 1500 Hz.

Classic Looseness

This is the same data used in the Rotating Looseness example, but is used here again to emphasize the nature of looseness.

On the machine, the pedestal bearing is loose, so as the shaft turns it is rattling up and down. If the hold down bolts were extremely loose, and there were no impacts, there would be no harmonics – just a high 1x peak as it rocks back and forth.

But there is severe impacting so there are many harmonics. In the waveform the impacts are clearly seen and the harmonics are in the spectrum. In fact there are 1/3 harmonics present, a sure sign of severe looseness.

Another interesting point about this data is that the negative going peaks are much greater than the positive going peaks. The positive side has amplitudes of 1 – 1.5 g's. The negative side has amplitudes of 2 -2.7 g's. This means the bearing has more freedom to move in one direction than the other, which is consistent with the looseness diagnosis.

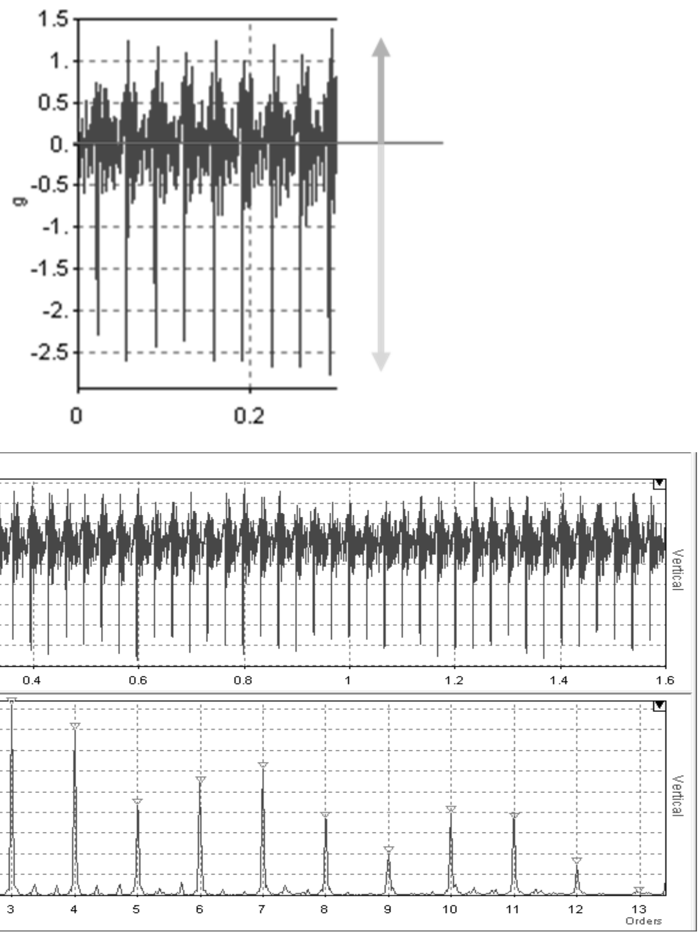


Figure 5-24 - Non-linear impacts produce harmonics. Severity is revealed by 1/3 harmonics.

Comparing Vibration Patterns and Levels

This example shows how important it is to collect a waveform for all the points on the machine and in all directions. If the waveform had been collected in the horizontal direction only, the severity of the condition would have been missed.

Some faults may only show up in certain axes and can often best be understood when comparing the level or pattern across axes (or between points on the same machine). Therefore it is best to perform comparisons with waveform data in the same manner that it typically done with spectral data.

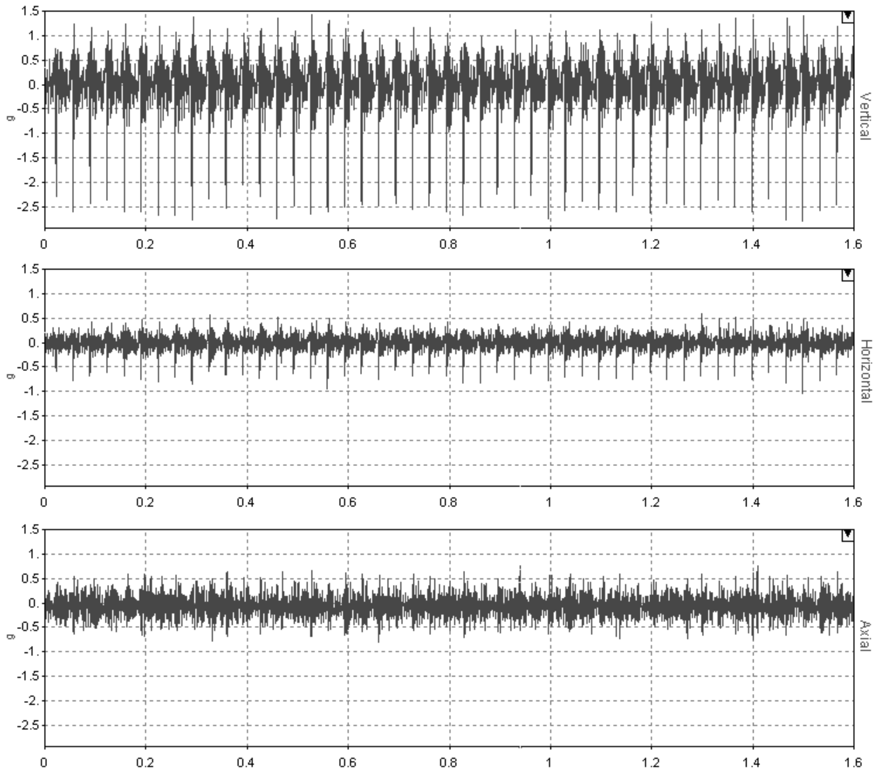


Figure 5-25 - Comparing data from different directions shows how important it is to store waveform data on all measurements.

Gearbox Analysis

The spectrum has a raised noise floor, but the source of the problem is not necessarily real clear. There were no alarms on this spectrum. However, the waveform shows 4 clear impacts that repeat themselves. The impacts are at the interval of the RPM of the gear with spalling.

Note that this is the first set of data on this gearbox, so no trends, nor historical data is available. Trending and narrow band alarming may have alerted to the spalling developing on the gears.

Note that the peaks in the waveform are nearly 3 g's on the positive side to the same on the negative side. That is nearly a 6 g peak to peak swing which indicates a lot of force is going on there.

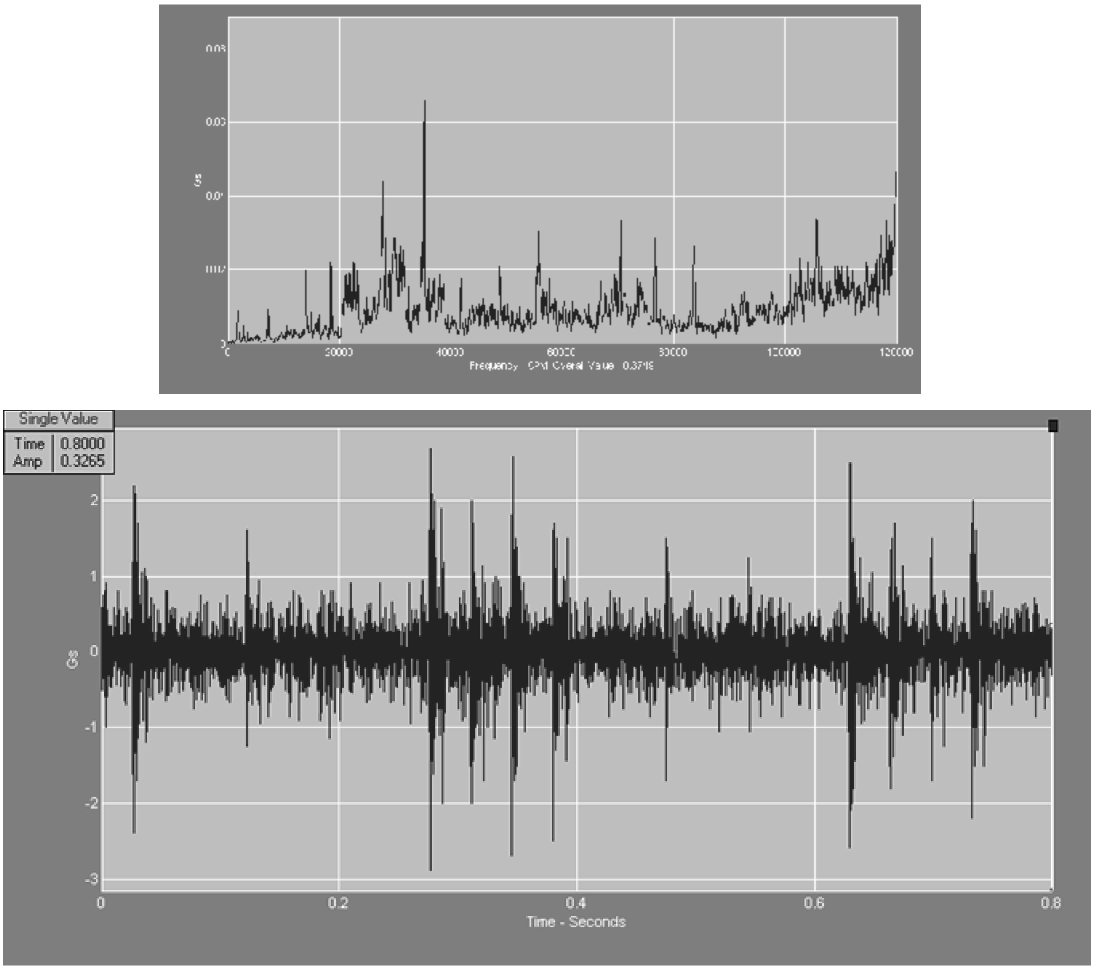


Figure 5-26 - The four peaks in the waveform are at the RPM of the gear with the spalling. Notice the high g levels.

Gearbox Analysis Example 2

Here is another example of gears with spalling on the teeth. The spectrum does not make it very clear what the problem could be, but the waveform clearly reveals the problems. There is not a peak over 0.006 in/sec.

The spacing of the impacts in the waveform are at the RPM of the gear with spalling. In this case notice the low g levels that would not trigger an alarm.

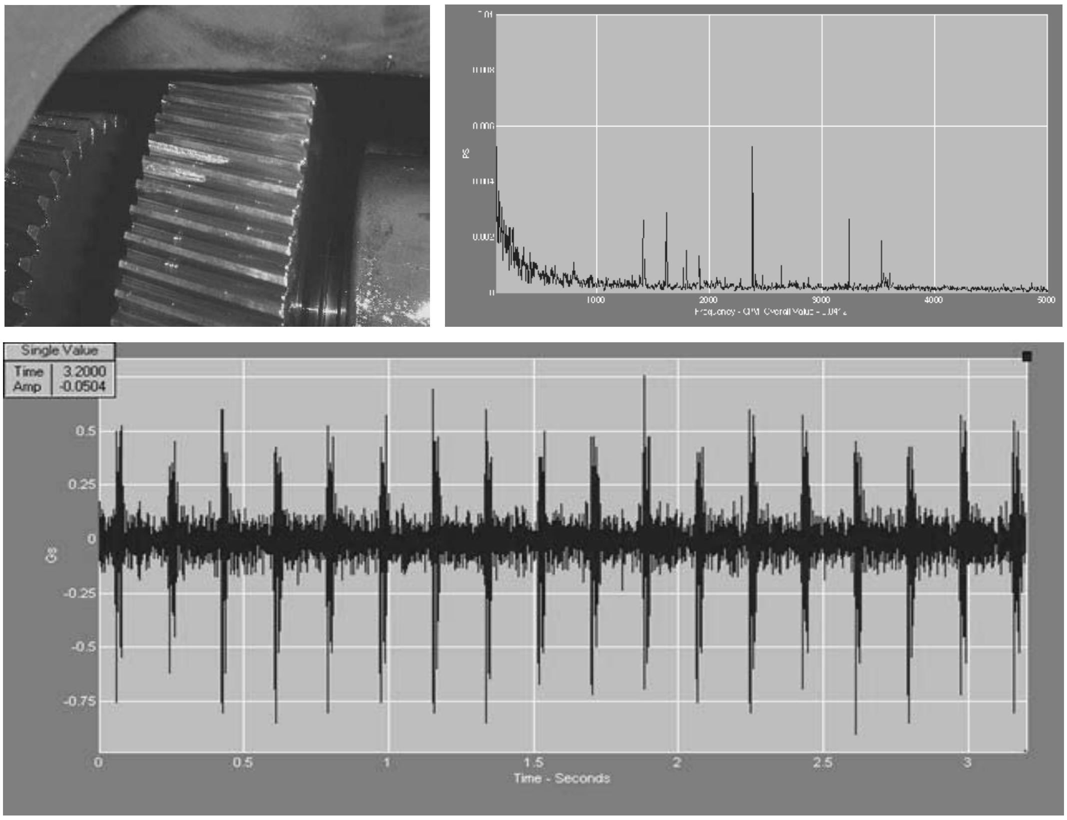


Figure 5-27 - The impacts are spaced at the RPM of gear with spalling. No alarms were triggered.

Healthy Gearbox

This waveform is from a healthy gearbox. There are no pulses or spikes but is continuous and fairly even.

Notice the levels are from +1 g to -1 g or a 2 g swing.

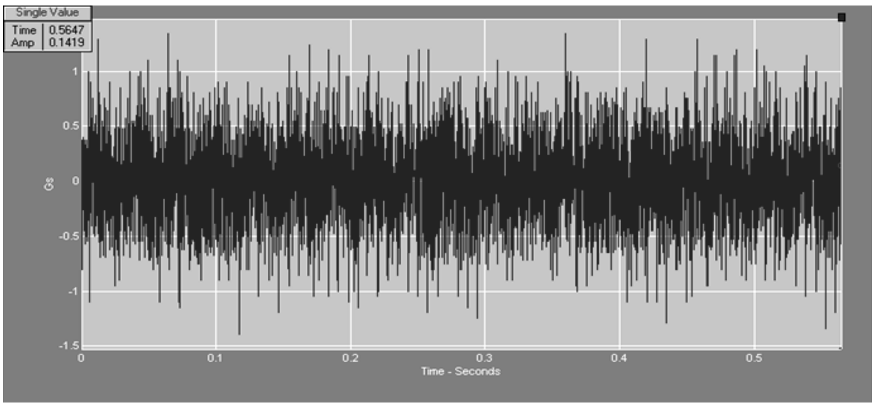
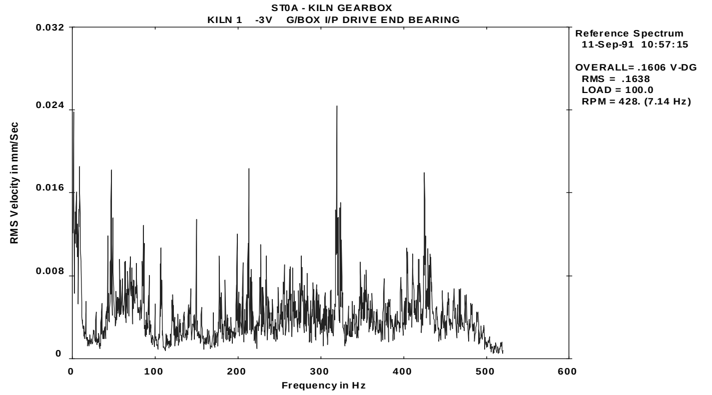


Figure 5-28 - Healthy gearbox.

Gearbox Example 3

The spectrum indicates there must be a lot of impacting due to the raised floor and there must be some non-linearity due to the hundreds of harmonics. But what is the source of the problem?

The waveform clearly identifies the problem with the periodic pulses spaced at the RPM of one of the gears.



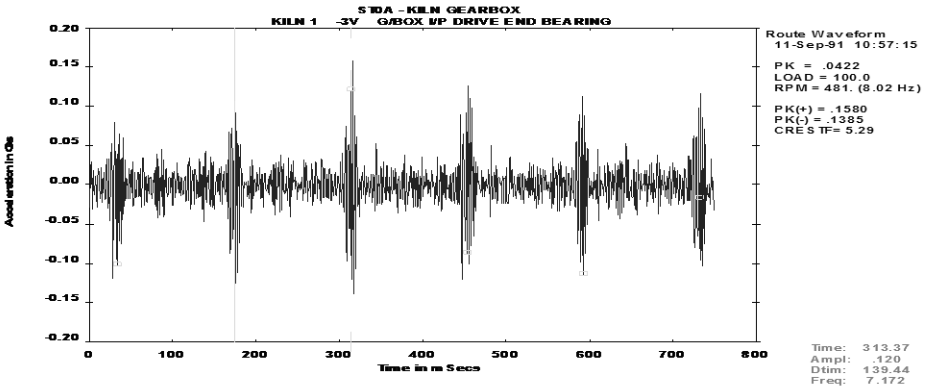


Figure 5-29 - Another case where the waveform makes it clear as to the cause of the raised floor and many harmonics in the spectrum.

Chapter 6

Data Acquisition

Objectives:

- List three factors that affect repeatability
- Select the proper sensor for the measurement
- Select the best measurement location
- Select the best mounting method for the measurement
- List three measurement point naming conventions
- Describe How to Recognize Bad Data
- Describe how to make phase measurements

Measuring vibration is the most important aspect of the vibration analysis program. Without good data, good results cannot be achieved. The analyst and automated software rely on good data to make accurate diagnosis. Clichés abound, but there is a good reason people say, "Garbage in, garbage out".

Vibration measurements can tell a great deal about the machine. They can indicate whether a fault condition exists and whether maintenance is required. With good measurements, and appropriate analysis, the exact nature of the fault, and the severity of the fault condition can be determined.

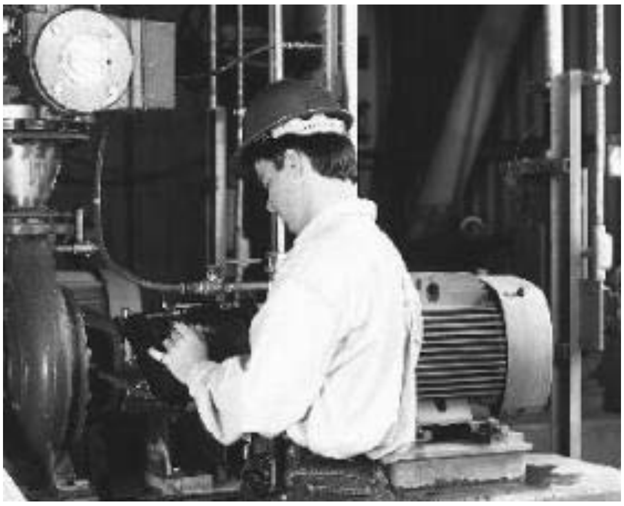


Figure 6-1

MAKING A GOOD MEASUREMENT

Making a good measurement involves:

- Selecting the correct transducer for the type of measurement. The measurement units are in units of Acceleration, Velocity or Displacement.
- Selecting the model that suits the environment, such as low speed or high temperature, etc.
- Mounting the sensor correctly. It must be in the correct location and the correct mounting type that produces the required results.

For most people and in most plants only two transducers will be required and the mountings used will be fairly uniform. However, high speed and low speed machines generally cannot be measured the same. Machines operating in hazardous environments, or with unsafe access require special treatment. Turbines may have sleeve or journal bearings requiring a different sensor type. One size does not fit all.

Figure 6-2

The data should be collected the same way every time it is measured. It should be under the same operating conditions and the same load, with the same mounting method.

One very valuable aspect of data collection is the **observations** about the machine that are noted during data collection. Most data collectors have a Notes option that stores comments or notes regarding the machine. This feature should be used often. These notes can lead to maintenance action and are extremely useful to the analyst.

The automated software and the analyst assume the data was collected correctly, i.e.

In many plants the person collecting the data is the only one to see some machines and some areas of the plant. It is imperative that this person keeps eyes and ears open and tuned for hints of abnormalities. Notes should be entered into the analyzer for reference later.

Figure 6-3

Most machines will be measured every 30 days. One key aspect the software and the analyst will be looking for is a change in the vibration pattern.

Measuring Vibration

A transducer (sensor) is required to capture the vibration and convert it to an electrical signal. This waveform can then be used by the software for signature analysis.

Figure 6-4

First, revisiting the basics of vibration may be beneficial. In the example used in the "Principles of Vibration" chapter a shaft had a vibration at shaft turning speed. The **Displacement** sine wave generated by this 1x vibration is very sinusoidal.

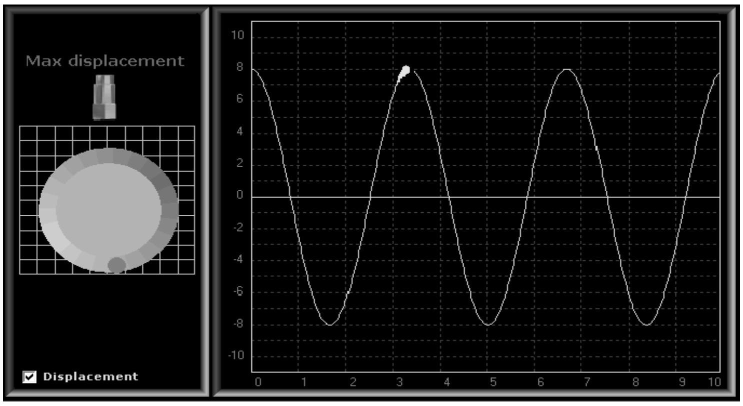


Figure 6-5 - the Displacement measurement is a measure of the distance the shaft moves relative to the sensor.

Looking closely at the movement of the shaft reveals that the measurement is actually measuring the distance from the shaft to the sensor. As the shaft moves, the distance between the shaft and the sensor is growing smaller, then larger, over and over. Measuring that changing displacement and graphing it versus time, produces a displacement time waveform.

If the shaft was turning relatively slowly, and the shaft could move inside the bearing, such as in a journal (or sleeve) bearing, then this would be an excellent measurement.

But what if the shaft was turning very quickly inside rolling element bearings? It is not likely that there would be very much displacement to measure inside the bearings. Sure, there would be some movement (displacement), but another measurement is better.

The Velocity measurement measures the speed of the shaft's up and down movement. Note that this measurement is also very periodic and produces a time waveform that is sinusoidal.

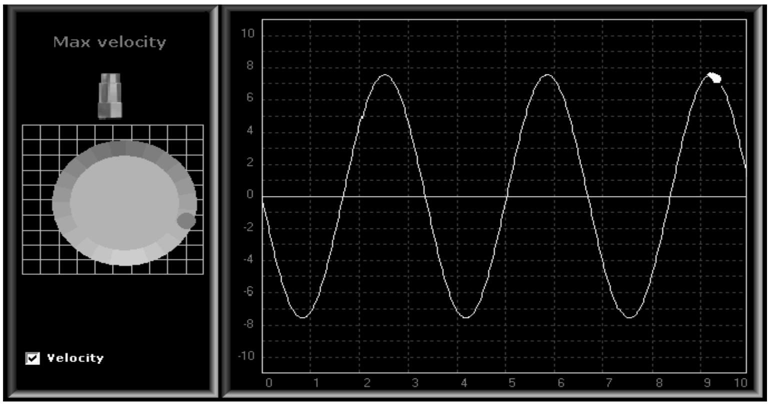


Figure 6-6 - Velocity measures the speed of the shaft movement.

Comparing Velocity and Displacement, the shaft is moving fastest while the displacement is zero. When the displacement is greatest the velocity is zero. They are out of phase. Phase is discussed later in this chapter.

The Acceleration Measurement reflects the force of the change of direction as the shaft accelerates toward the bottom, decelerates, comes to a stop and then accelerates back toward the top.

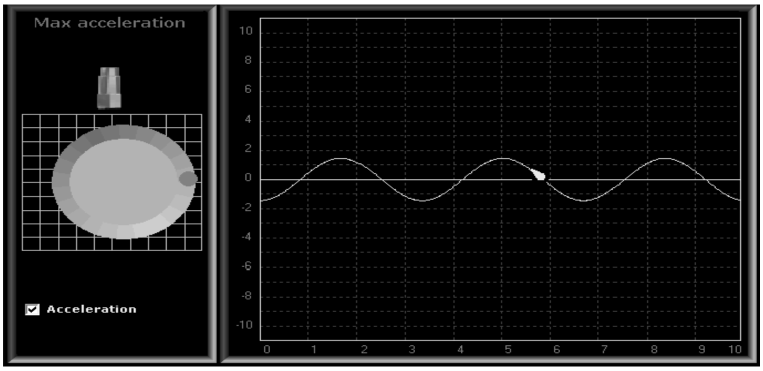


Figure 6-7 - Acceleration measures the force as the shaft changes direction

At low speed there is very little acceleration. Speeding up the shaft increases the acceleration

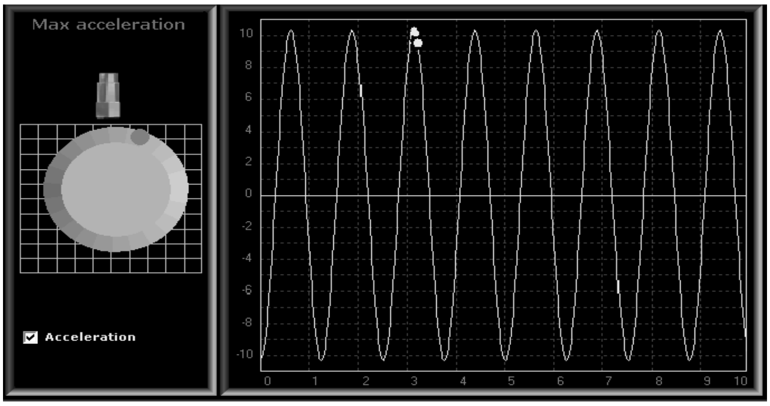


Figure 6-8 - Acceleration increases with an increase in speed.

Unless the forces are huge, the shaft movement will be very small. The velocity will be high, but can you see that the velocity of the shaft is changing very rapidly? It is moving quickly in one direction, then it moves quickly in the other direction. As the shaft increases in speed, this acceleration and deceleration becomes greater, and is the best way to measure the vibration.

Showing all three units, with velocity remaining constant, the relationship can be seen between acceleration and displacement. As speed increases (while keeping velocity constant) the acceleration increases but the displacement decreases.

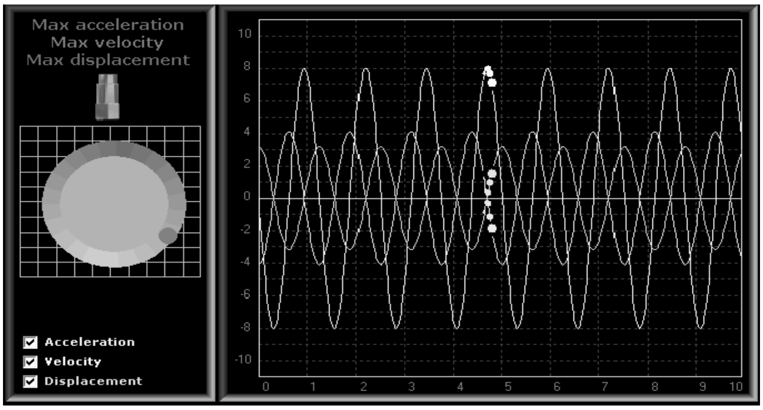


Figure 6-9 - The relationship of all three.

Acceleration (or deceleration) is highest when the velocity is zero - when it has to change direction. So the acceleration is also out of phase with the velocity.

Further, the acceleration is out of phase with the displacement, because when the acceleration is at its greatest, the displacement is at its minimum, and vice versa. So they are completely out of phase - or 180 degrees out of phase.

Considering all three measures; displacement, velocity and acceleration, displacement and acceleration are 180 degrees out of phase with each other, and velocity is 90 degrees out of phase with both of them.

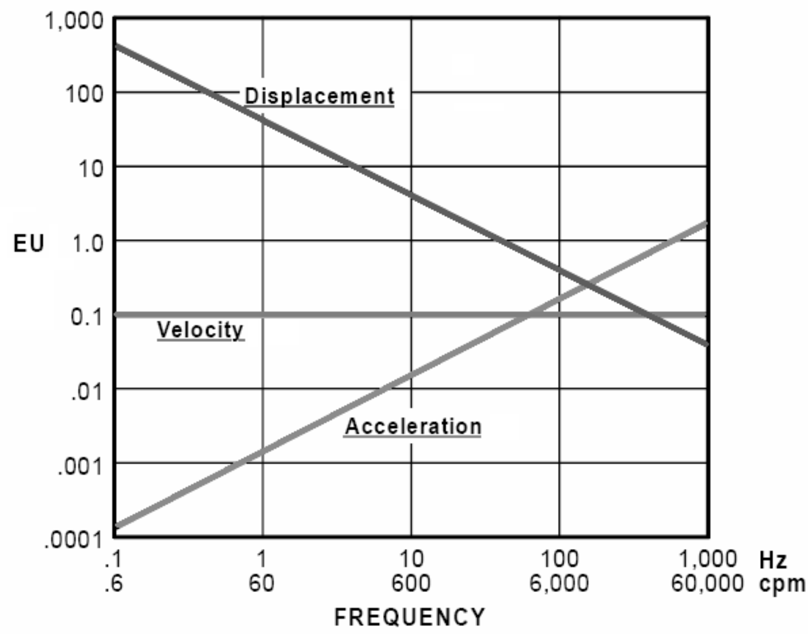


Figure 6-10

This graph shows the relationship among displacement, velocity and acceleration for a fixed vibration level of 0.1 IPS. Notice that the x and y-axis are logarithmic scaling. The y-axis indicates vibration level, while the x-axis indicates frequency. It indicates that at low frequency very little acceleration vibration is produced, whereas much larger displacement amplitudes are produced. The opposite is true at higher frequencies.

The graph indicates that if there is a signal at 10 Hz, the acceleration value is just over 0.01 while the value in velocity is 0.1. In displacement the amplitude of the peak would be equivalent to over 5.0 on this scale. Displacement is good for low speeds, while acceleration is good for very high speeds, and velocity is a good general purpose measurement for most standard ranges of industrial vibration.

The difference between Displacement, Velocity, and Acceleration is clearly seen in these spectra of the same data.

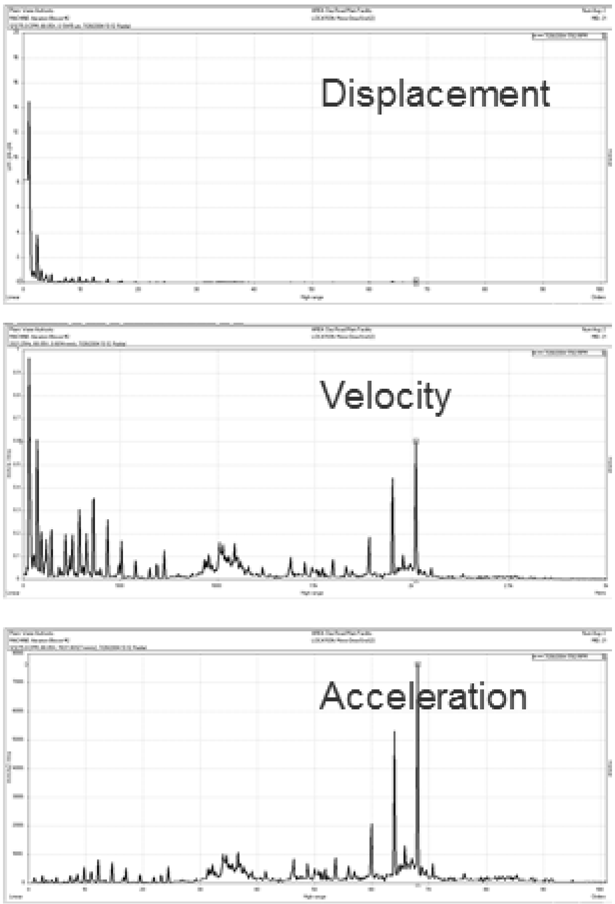


Figure 6-11 - The same data displayed in the 3 vibration units.

In Chapter three the value of the vibration units was illustrated using the spectra in Figure 6-14. The sensor type clearly has an impact on the signal recorded. The sensor type chosen will depend on the machine and what data is needed.

The first step is selecting the right type of transducer. Select among displacement, velocity, and acceleration. Issues such as machine speed and bearing type come into play.

Next, select the brand/model that suits the measurement environment. Consider the operating temperature, space restrictions, presence of water or other liquids, and other factors.

Then consider mounting techniques. The type of machine, the nature of the machine's surface and other environmental factors, the frequency range required, and accessibility issues are important. Study the machine and make sure to identify the best location for the transducer.

There are three types of sensors commonly used: Displacement “proximity” probes; Velocity probes; and accelerometers. The sensor is selected based on machine type and machine speed.

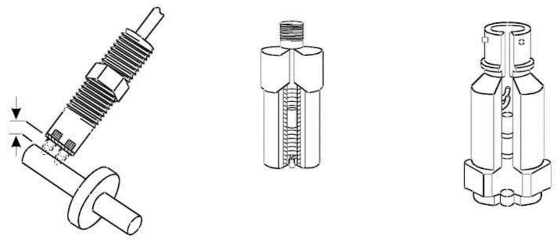


Figure 6-12

Converting Units - Sometimes one transducer is used but the data is displayed in other units. To convert from acceleration to velocity, or from velocity to displacement the math process of integration is used.

Differentiation is the math process used to go from displacement to velocity or from velocity to acceleration.

The most common integration is from acceleration to velocity. The process involves a phase shift that suppresses the higher frequency information and amplifies the lower frequency information.

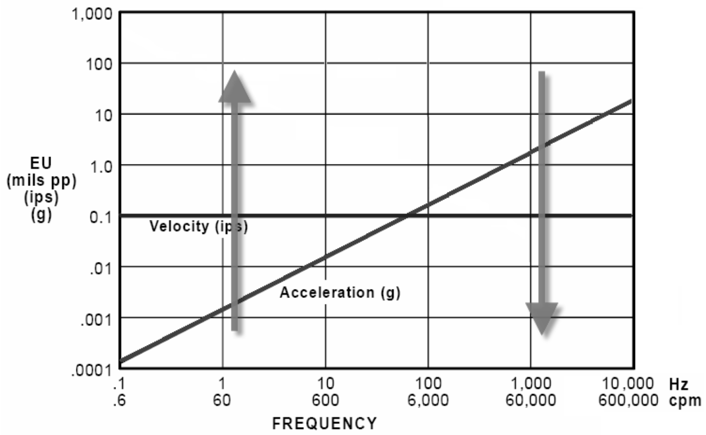


Figure 6-13

The very low frequency information is amplified so much that it has to be filtered out either electronically or in the software with the use of a high-pass filter.

The issue of acceleration, velocity and displacement is very important when it comes to transducer selection. Measuring displacement is best for low speed machines, and acceleration is best for high speed machines, and machines with high speed rotating components. Velocity fits the middle ground and most common plant equipment.

Comparing the units – Acceleration is highest when the velocity is zero, that is, when the shaft has to stop its travel in order to change direction. So acceleration is out of phase with velocity.

Acceleration is also out of phase with displacement because when acceleration is at its maximum, displacement is at its minimum.

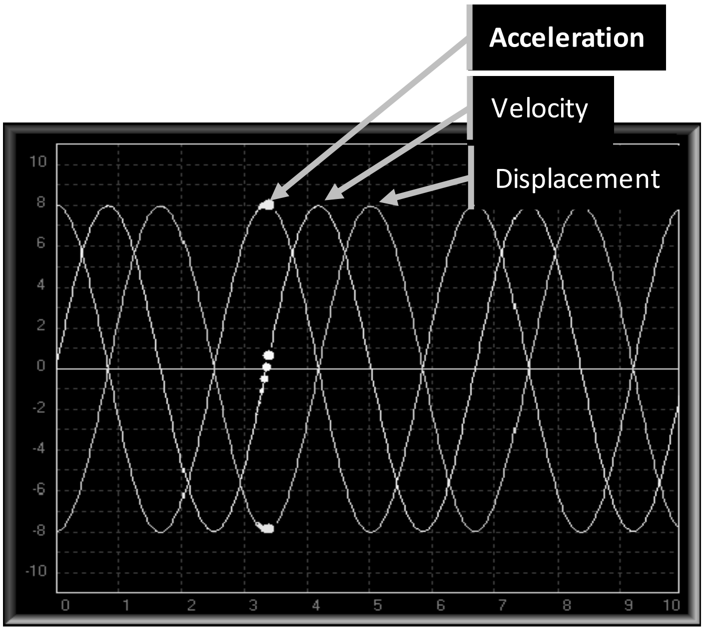


Figure 6-14 - Acceleration is 90 degrees ahead of velocity and 180 degrees ahead of displacement.

This phase relationship per se does not really affect us in our day-to-day condition monitoring activities. It is one of those things that should be understood. However it is important when performing more advanced time waveform analysis and balancing.

Knowing what integration means, and how it affects the vibration is important, since this term is used when dealing with the data collector and in some vibration analysis software.

CONVERTING UNITS

Most vibration software programs convert the units without problems. A set of formulas is shown below. Note that frequency does matter, it is in the formulas. PI is 3.1416.

Velocity = 2 PI f D = A/2PI f

Acceleration = 2 PI f V = (2 PI f) ^{2} D

- Where D = peak displacement (inches or mm)
- f = frequency – (Hz or Cycles/sec)
- V = velocity (in/sec or mm/sec)
- A = Acceleration (in/sec^2 or mm/sec^2)

Note: All values are peak values. Multiply by .707 for rms values or by 2 for Pk-Pk values.

Displacement Transducers

Displacement transducers measure the relative movement between the shaft and the tip of the sensor. So rather than being mounted on the outside of a machine, they are drilled into the journal or sleeve bearing. Therefore, they are permanently mounted.

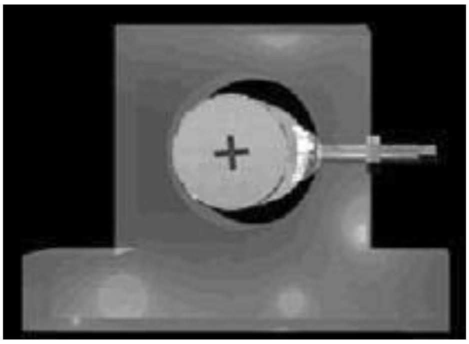


Figure 6-15

Displacement transducers are typically used in permanent monitoring (protection) of plain bearing machines such as turbines, large pumps and large fans. However it is usually possible to connect a portable data collector to the dynamic output to perform "normal" spectrum and trend analysis.

Displacement transducers are called **Eddy Current Probes**. They are also known as non-contact probes and commercially as Proximity probes. They all work on the same principle.

There are three components: a driver, a probe, and a cable between them. A voltage is supplied to the driver that produces an RF signal. That signal is transmitted through the cable to the probe. A coil inside the probe tip acts as an antenna and radiates the high frequency energy into the gap - its sets up a magnetic field. Any conductive material within the magnetic field absorbs the energy - eddy currents are set up in the material (the shaft), hence the name.

Figure 6-16 - Eddy current probes have 3 components: the probe, a driver, and the cable between them.

The absorption of the field causes the output of the probe to decrease in proportion to the gap distance. As the distance to the shaft changes dynamically, so does the output signal.

The driver then acts as a "demodulator" and "detector" and has two outputs. The dynamic output produces the time waveform, from which we derive the spectrum and overall level.

There is also a DC voltage proportional to the average gap distance. The DC gap signal is also used in monitoring systems to determine where the shaft is located relative to the bearing.

Typically a gap reading is taken when the shaft is at rest, and then as the machine (a turbine is a typical example) runs up to speed, the gap voltage is monitored. From this information we can detect shaft rubs, and produce a "centerline diagram".

On most turbines, two eddy current probes are installed on the machine 90 degrees apart (usually at 45 degrees and 135 degrees). This enables the analyst to see exactly how the shaft is moving within the bearing.

Figure 6-17

Orbit diagrams are commonly used to display this movement. The trained eye can detect unbalance, misalignment and a host of other fault conditions.

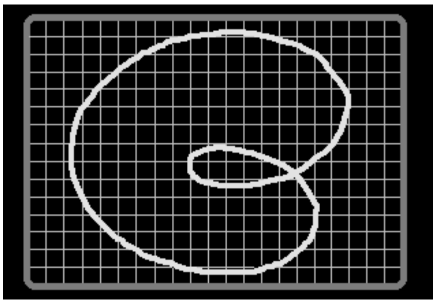


Figure 6-18 - Orbit Diagram

Cracks in the shaft can be detected this way, as the dynamics of the shaft change. Advanced software packages exist that look at whole shaft dynamic motion by examining the signals from all the eddy current probes, plus casing accelerometers if available. Orbit diagrams are covered in Category III vibration training.

The transducer will have an output sensitivity: 200 mV/mil or 7.87 V/mm. If the shaft moves 5 mil pk-pk per rotation, the voltage will vary 1 V pk-pk.

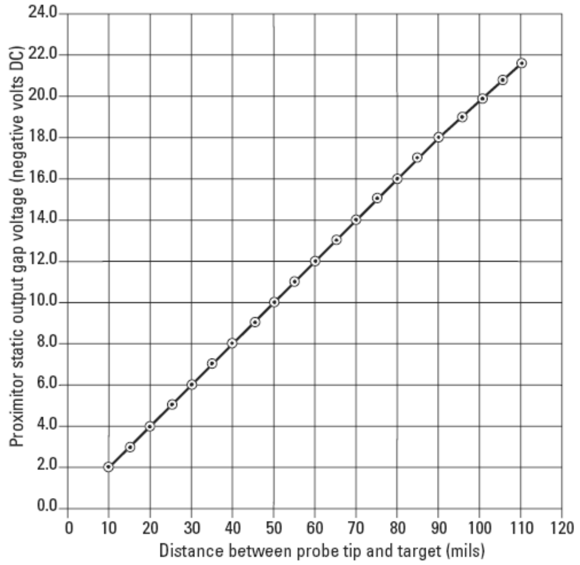


Figure 6-19 - As the gap between the surface and the probes is decreased, the voltage will increase

The transducer will have a linear range. The "set point" should be set in the middle of the linear range. Dynamic displacement from that point will remain in the linear range of the transducer. The sensitivity of both transducers must be the same.

Advantages of Displacement Probes:

- Wide frequency response (0 Hz to 10,000 Hz)
- It measures the actual relative displacement of the shaft within the bearing (1mm – 10mm or 40mil to 400mil).
- They are reliable (when installed correctly).
- Broad temperature range: -50 C – 200 C (-60 F to 390 F) or -50 C – 125 C (-60 F to 260 F) if using integrated oscillator.

Disadvantages:

- They are difficult and expensive to install
- Their calibration (determining the ratio between output voltage and actual displacement) is dependent upon the shaft material (different materials absorb the energy at different rates).
- Shaft runout and surface glitches produce false signal

Applications:

- Used in large journal bearings / protection systems.
- They are useful as a key phasor (a reference signal proportional to running speed) for dynamic balancing and analysis.
- Used for dynamic balancing due to filtering at 1X RPM

Units:

- The units are mils pk-pk or microns pk-pk.
- (1 mil = .001")

Velocity Transducers

While we often analyze vibration data in units of velocity, we typically do not actually use "velocity" transducers per se. Velocity transducers are still used in industry to some extent. Today there are accelerometers with built-in integrators to yield a Velocity output. Here are the pros and cons.

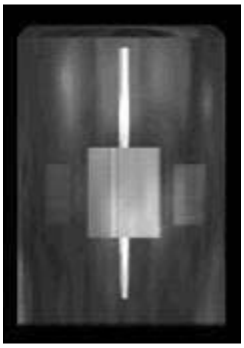


Figure 6-20

The "electrodynamic velocity sensor" is basically a suspended magnet, mounted between a spring and a damper. A coil surrounds the magnet. As the case of the transducer vibrates, the magnet remains stationary due to inertia. Therefore you have movement of the magnet within a coil, which generates electricity proportional to the velocity of the mass.

The construction can also involve a coil attached to a mass, surrounded by a stationary magnet. The result is the same; electricity is generated when there is relative movement between the magnet and coil.

Advantages:

- No external power required - it generates electricity
- The signal output is powerful
- It is easy to use - not as sensitive to mounting problems
- Ability to operate at higher temperatures (-50 C – 200 C (-60 F to 390 F))

Disadvantages:

- Not suitable for very low or very high frequency measurements: 1Hz – 2000Hz.
- Calibration changes with temperature.
- Given that moving parts are involved, internal wear can shorten its life.
- Velocity transducers are quite large.

- Not as accurate due to comparatively lower signal to noise ratio.

Applications:

- While velocity transducers were very popular, they are no longer the transducer of choice. Accelerometers that are internally "integrated" to velocity are more common as "velocity sensors".

Units:

- Typical units are in/sec, mm/sec rms or VdB.

Accelerometers

Accelerometers are by far the most common types of transducers used in machinery vibration analysis. All portable data collectors are supplied with an accelerometer, even though most people actually "integrate" the signal to velocity and view the data in units of velocity. Permanent monitoring systems also use accelerometers, except where eddy current probes are specifically called for.

Figure 6-21 - Accelerometers come in various shapes and sizes.

There are actually a number of different types of accelerometers. Several years ago, most accelerometers were **charge mode piezoelectric** which required an external charge amplifier.

Today's accelerometers are ICP accelerometers or Integrated Circuit Piezoelectric accelerometers meaning the amplifier is inside the unit. They are mounted externally, typically on the bearing housing. The method used to mount the transducer is very important. They come in all shapes and sizes to satisfy different installation applications.

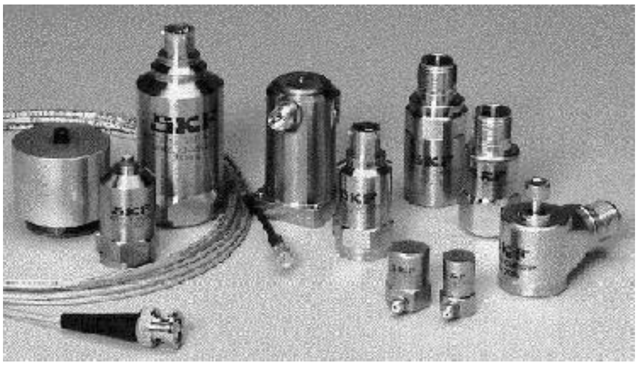


Figure 6-22

How do they work? Do you remember the old record players? The "needle" was a crystal, and as the record track moved beneath it, the crystal was compressed and decompressed as it "bounced" over the vibration pattern imprinted in the track.

Figure 6-23 - crystal and mass in accelerometer

This compression caused a charge to be produced, which was amplified, and played through your speakers as Deep Purple or Frank Sinatra - depending upon your taste!

That is similar to how modern piezoelectric accelerometers work. The piezoelectric material (crystal) is placed under a load using a mass. As the transducer vibrates the crystal is compressed and a charge is produced.

The charge output is proportional to the force, and therefore acceleration (Newton's second law; force is proportional to the acceleration of the mass). An amplifier is then required to convert that charge output to a voltage output.

Figure 6-24 - Charge Mode amplifier for older style accelerometers.

While an external charge amplifier was common several years ago, in modern accelerometers the amplifier is actually inside the transducer, and is powered by the data collector. These are known as "internally amplified piezoelectric accelerometers" (or Integrated Circuit Piezoelectric - ICP). This amplifier is powered by a DC polarization of the signal lead itself, so no extra wiring is needed. The data collector therefore needs to have this DC power available to this type of transducer.

Note: There is a setting called "ICP Power" or something similar that is used when specifying the set up - only accelerometers require this power. (Eddy current probes also require power, however that is supplied by an external device.)

Most data collectors monitor this DC "bias" voltage in order to determine if the transducer has a fault, or if there is a fault in the cable.

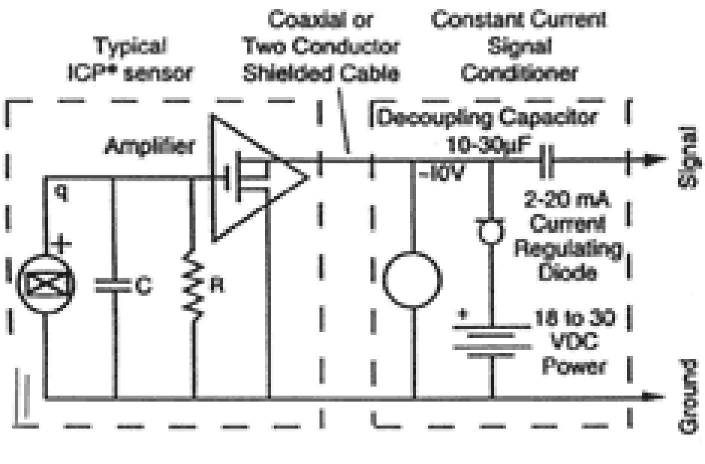


Figure 6-25 - courtesy IMI

The existence of the amplifier results in limited low frequency response (meaning that they normally cannot be used at very low frequencies). The "low frequency roll-off" of the amplifier is typically at 1 Hz for most generally available ICP units. There are some that are specially designed to go to 0.1 Hz if very low frequency data is required.

The amplifier is powered by the data collector using 18 to 30 VDC bias voltage. The data collector can check the sensor and cable to verify whether they are ok via the "bias voltage."

Accelerometer Configurations – Accelerometers come in different configurations for many applications and requirements such as Accessibility, cost, and environmental issues.

Three basic structural designs are used in manufacturing industrial accelerometers. They are the flexural, compression and shear designs. All three designs contain the basic components of the piezoelectric element, seismic mass, base, and housing.

In the flexural design the piezoelectric element is secured to the seismic mass in the form of a double cantilever beam. Figure 6-26 shows the sensing element/mass system being driven at the fulcrum or the base. Flexural designs have lower resonant frequency and are generally not well-suited for machinery monitoring applications. Because of their very high output (up to 100 V/g), flexural designs excel in low level, low frequency seismic applications. The flexural element is often epoxied which limits its application in high shock environments.

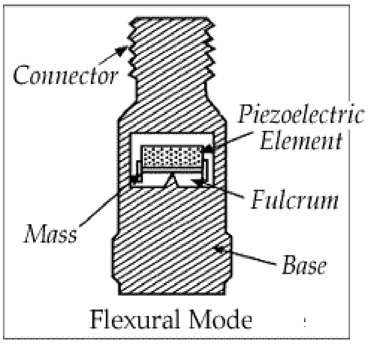


Figure 6-26

The compression design is generally the simplest and easiest to understand. The crystal, quartz or ceramic, is sandwiched between the seismic mass and the base with an elastic pre-load bolt. Motion (vibration) into the base squeezes the crystal creating an output. Compression designs are much more suited than flexural designs for industrial machinery monitoring applications because of their high resonance and more durable design. Compression designs generally have thick bases and should be used on thick walled structures because of base strain and thermal transient sensitivities.

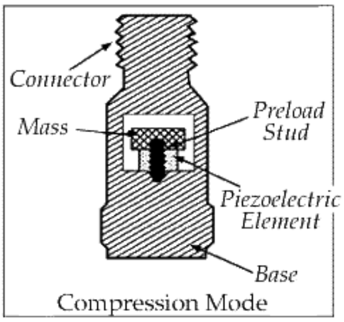


Figure 6-27 - Compression Mode accelerometer

The shear design subjects the sensing element to a shear stress. The piezoelectric via a retaining ring as shown in Figure 6-28. This preload produces a stiff structure with good frequency response and greater mechanical integrity. As the sensitive axis is not in-line with the mounting surface, adverse environmental conditions such as base strain and thermal transients do not produce false signals as in the other designs.

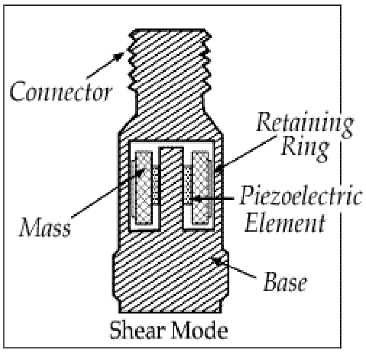


Figure 6-28 - Shear Mode accelerometer

FAQ: *Which is the best accelerometer operating mode, compression, shear or flex?*

Answer: In recent years, shear mode sensors have gained popularity with sensor designers while compression mode operation often considered as "old technology." Meanwhile, flexural mode sensors, once considered too fragile for industrial applications, are now being incorporated in an increasing number of new sensors using special design techniques.

Sensor design engineers weigh the advantages and disadvantages of each operating mode, minimize the disadvantages and design sensors to best perform in various applications. When purchasing sensors for a quality manufacturer, the construction method of a sensor is less important than its performance. Many times the difference between the three operating modes can be seen in such specifications as base strain, weight, thermal transient response and shock resistance.

Low Frequency characteristics - The existence of the amplifier results in limited low frequency response (meaning that they normally cannot be used at very low frequencies). The "low frequency roll-off" of the amplifier is typically at 1 Hz for most generally available ICP units. There are some that are specially designed to go to 0.1 Hz if very low frequency data is required.

FAQ: Why don't all piezoelectric vibration sensors have a frequency response down to 0 Hz (0 CPM)?

Answer: Piezoelectric materials generate an internal charge when stressed by motion (vibration). Inherent to all piezoelectric materials is internal resistance, which dissipates the charge over time. Zero frequency operation would require no dissipation and is therefore not possible. Additionally, very low frequency outputs from the crystal due to environmental effects such as mechanical parts thermal expansion must be eliminated.

To get rid of these unwanted signals, a high pass filter is incorporated in the electronics of all piezoelectric accelerometers. The filter has a resistor and capacitor in series and the value of these components, RC, determines the low-end cut-off. Also known as the discharge time constant (DTC), the larger the RC value, the lower the frequency response. The DTC also defines the sensor response to abrupt changes in sensor powering such as turn-on and signal overload. When the sensor is turned on or begins to recover from an overload, the time it takes to become usable is directly related to the DTC value. Therefore, the low-end cut-off is inversely proportional to the turn-on time (and shock recovery time). In other words, the lower in frequency the sensor measures, the longer it takes to turn-on or recover from an overload. For general-purpose sensors, some low-end frequency performance is sacrificed in favor of better turn on and shock recovery response.

FAQ: What are the differences between general purpose and low frequency accelerometers?

Answer: Low frequency accelerometers employ a larger seismic mass to increase the output from the sensing element assembly. This reduces the electronic noise from the amplifier and allows higher voltage outputs from the sensor. The higher voltage outputs of low frequency sensors help overcome data collector noise when measuring low amplitude signals.

An internal low pass filter is also used in low-frequency sensors to help eliminate undesirable high frequency signals that can create distortion from saturation.

Sensitivity

There are a number of factors to consider when selecting an accelerometer. The first is sensitivity. The sensitivity is a measure of the output voltage levels in response to a set vibration level. Accelerometers quote their sensitivity in mV/g. For example, if the sensor was the common 100 mV/g, and the vibration level was 1 g, it would produce 100 mV (or 0.1 Volts) of voltage.

For a precision machine, like a machine tool, or a very low speed machine where the vibration level is low, a more sensitive accelerometer is needed. It needs to produce enough voltage for the data collector to measure. A typical sensitivity in this case would be 1 V/g (1000 mV/g).

On the other hand, for very large noisy machines, the sensitivity needs to be much lower, perhaps 10 mV/g.

Most of the accelerometers supplied with condition monitoring systems, and used in permanent monitoring applications use 100 mV/g accelerometers. These sensors take the middle ground, providing sufficient dynamic range for the majority of test conditions.

Due to the design of sensor electronics, the frequency response of the sensor is not the same for highly sensitive accelerometers as it is for less sensitive units. (This is independent of the effects of different mounting methods.)

A common question arises: why not use a low frequency, high sensitivity accelerometer for all the measurements? First, due to the low frequency specifications of the internal high pass filter, the transducer takes far longer to settle after a mechanical, thermal or electronic (power-on) shock.

Also, they typically have a low pass filter, limiting their high frequency response, which makes them unacceptable for many "general purpose" applications. And finally, due to their high gain amplifier, they are susceptible to overload.

So, while these sensors play an ever more important part in condition monitoring, they should only be used when the application specifically calls for their unique capabilities.

Another common application for accelerometers is in very hot environments, such as the dryer section of a paper machine. Standard accelerometers can survive in high heat conditions, however due to their internal amplifier they have a limit. Over approximately 300 degrees F (150 C), it is normally recommended to use a charge mode accelerometer.

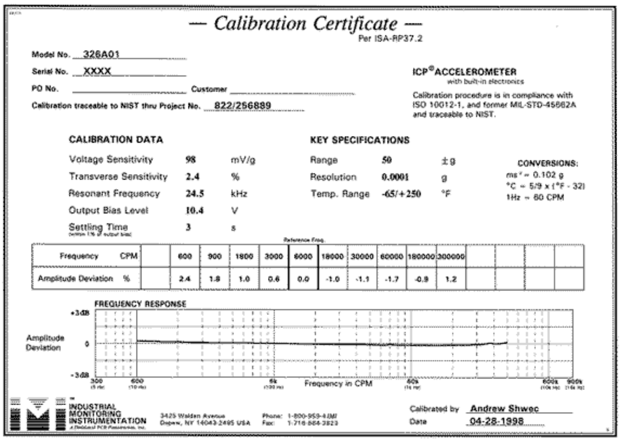
Charge mode accelerometers use an external "charge amplifier", located away from the sensor, in a more hospitable environment. A special heat resistant cable is then used to join the two.

Note: As with all charge mode accelerometers, care must be taken to secure the connecting cable (while the measurement is in progress), as any movement will induce noise on the cable.

In summary, internally amplified, 100 mV/g accelerometers are the most commonly used sensor in machine vibration monitoring. However, there are many situations when additional sensors designed specifically for the unique vibration level, operating conditions or frequency range should also be used.

Calibration

Excessive heat will damage an accelerometer, as will dropping them on a hard surface. If an accelerometer is dropped more than a few feet on to a hard surface, the crystal inside the accelerometer can crack. This affects the sensitivity and frequency response. Given the environment in which most accelerometers are used, it is a good idea to have them checked and calibrated once each year.



— Calibration Certificate —
Per ISA-RP37.2

Model No. 326A01
Serial No. XXXX
PO No. Customer
Calibration traceable to NIST thru Project No. 822/256889

ICP® ACCELEROMETER
with built-in electronics
Calibration procedure is in compliance with ISO 10012-1, and former MIL-STD-45662A and traceable to NIST.

CALIBRATION DATA

Voltage Sensitivity	98	mV/g
Transverse Sensitivity	2.4	%
Resonant Frequency	24.5	kHz
Output Bias Level	10.4	V
Settling Time	3	s

KEY SPECIFICATIONS

Range	50	±g
Resolution	0.0001	g
Temp. Range	-65/+250	°F

CONVERSIONS:
ms² = 0.102 g
°C = 5/9 x (°F - 32)
1Hz = 60 CPM

Frequency CPM		600	900	1800	3000	6000	18000	30000	60000	180000	300000				
Amplitude Deviation %		2.4	1.8	1.0	0.6	0.0	-1.0	-1.1	-1.7	-0.9	1.2				

INDUSTRIAL MONITORING INSTRUMENTATION
3425 Walden Avenue
Depew, NY 14043-2495 USA

Calibrated by Andrew Shwec
Date 04-28-1998

Figure 6-29

Accelerometer Settling Time

When an ICP accelerometer is connected to the power source, it takes a few seconds for the amplifier to stabilize, and during this time, any data the unit is collecting will be contaminated by a slowly varying voltage ramp. For this reason, there must be a time delay built into data collectors to assure the unit is stable.

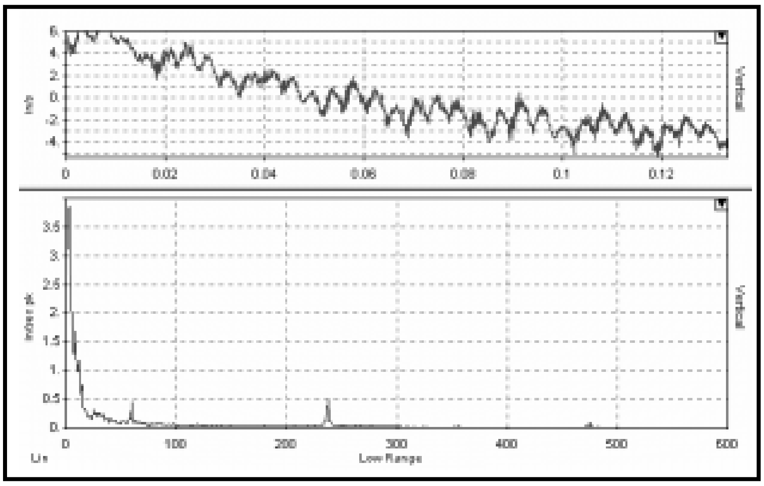


Figure 6-30 - The effect of powering on the transducer and the characteristic ski slope in the spectrum.

If the delay is too short, the time waveform will have an exponentially shaped voltage ramp superimposed on the data, and the spectrum will show a rising very low-frequency characteristic sometimes called a "ski slope". This should be avoided because the dynamic range of the measurement is compromised.

Many data collectors have a setting called "**settling time**". This is the phenomenon to which they are referring. However, when the transducer is placed on a very hot machine (after testing a cooler machine), the transducer will go into a thermal transient and will take a longer time to settle. In this case longer settling time must be selected.

Piezovelocity Transducers

Because standard velocity transducers are not suited to industrial applications, transducer companies make accelerometers with an output proportional to velocity.

They are called "piezovelocity" transducers

Figure 6-31

The sensor contains both the amplifier and the integrator circuit. They can be used down to 1.5 Hz (90 CPM)

COMPARING DISPLACEMENT MEASUREMENTS

Acceleration can be double integrated to displacement, but that does not mean the data will be the same as data taken with a displacement probe.

It can become confusing when talking about eddy current probes measuring displacement, and the fact that accelerometers are available that internally integrate twice to output displacement. If both sensors were mounted on a turbine journal bearing, would they measure the same thing? No.

The eddy current probe is measuring the relative movement between the bearing and the shaft. The accelerometer measures the vibration on top of the bearing, and then "converts" that to displacement. There are situations where the relative movement between the shaft and bearing is small, while the entire bearing is vibrating a lot - the eddy current probe will not measure that, the accelerometer will.

The two sensors are measuring two very different phenomena. For this reason, many professional analysts will monitor the eddy current probe and place an accelerometer on the bearing case. They can then see how the bearing is vibrating relative to ground, and how the shaft is vibrating relative to the inside of the bearing - they have the complete picture.

Triaxial Accelerometers

One interesting variation to the standard accelerometer package is the "triaxial" accelerometer. In this case there are three accelerometers mounted orthogonal to each other. When mounted to the machine the package is capable of measuring the vertical, horizontal and axial vibration from the single location.

Portable data collectors from some manufacturers are actually able to sample from all three accelerometers simultaneously, resulting in a test that takes the same time as a standard single-axis measurement, but gaining far more information.

Figure 6-32 - Triaxial accelerometers have 3 accelerometers in one unit

Other types of accelerometers include strain gage, piezoresistive, and variable capacitance. And even the modern internally amplified sensors can use different crystal materials and different arrangements of crystal and compression mass.

Advantages:

- Very wide frequency range: 0.1 Hz - 30 kHz. Higher frequencies supported by special models.
- Wide amplitude range.
- Broad temperature range: up to 125°C/260°F. Higher temperatures supported by special models.
- Typically very rugged, and designs exist for a very wide range of applications
- Velocity and displacement available as output (through internal integration)
- They remain stable and thus maintain calibration for a long time (for the same vibration, they output the same signal).

Disadvantages:

- Not responsive down to 0 Hz (low frequency accelerometers are available).
- Temperature limitations due to use of internal amplifier

Applications:

- Accelerometers enjoy the widest application in industry. From portable data collectors to permanent monitoring systems, including special modal tests, accelerometers are available for a wide variety of environments and machine speeds.

Units:

- G rms or AdB

Testing the machine

We need to collect enough data to detect changes in machine health OR collect enough data to identify specific machine faults.

- Too much data: more time to collect + analyze = $$ spent
- Too little data: not enough information to act on
- Collect repeatable data for trending
- Collect good quality data

Very important issues:

- Limitations of the accelerometer
- Limitations of the mounting method
- Mechanical transmission path
- Importance of the mounting location

Goals:

- Repeatability
- Capture vibration that will tell us about machine condition
- Time/cost issues
- Safety

Selecting a Transducer

Even if an accelerometer is to be used, it is still necessary to select the most appropriate accelerometer, and decide whether the signal should be integrated to velocity for analysis.

Frequency response

In an ideal world, if a transducer is placed on a machine which vibrated an equal amount at all frequencies from 0 Hz to 1 MHz, the transducer would produce an output equal to that level of vibration at all frequencies.

- 1 g at 10 Hz would generate 100 mV
- 1 g at 1,000 Hz would generate 100 mV
- 1 g at 10,000 Hz would produce 100 mV

But they do not. The transducer output is not “flat” at all frequencies. Instead, the physics of the transducer, the method used to mount the transducer, and the electronics involved all conspire against us. All transducers by themselves have practical limits. And that limit is often further compromised by the mounting technique (a hand held sensor is worse than a stud mounted sensor for example).

The frequency response is represented by a curve of input signal versus measured signal. For a known (and constant) input at all frequencies, the output of the sensor is examined. In the ideal world, the curve would be flat.

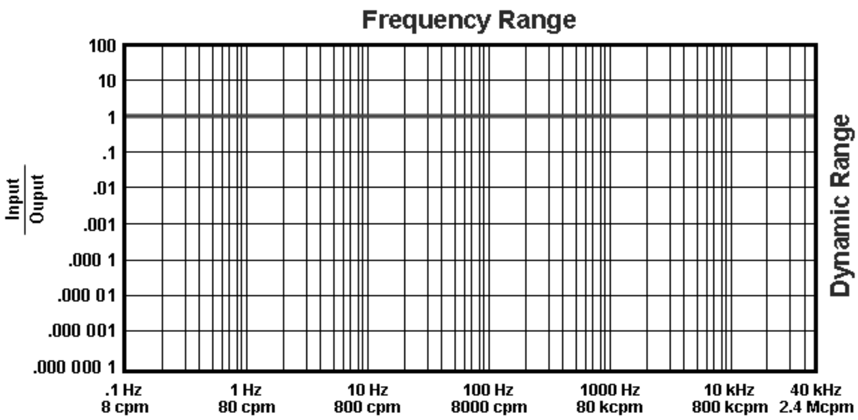


Figure 6-33 - The Ideal Response Curve

The reality is the response curve is not flat. There is a low frequency limit, a flat usable (or linear) region, and a sensor resonance at the high end after which the response tends to fall away.

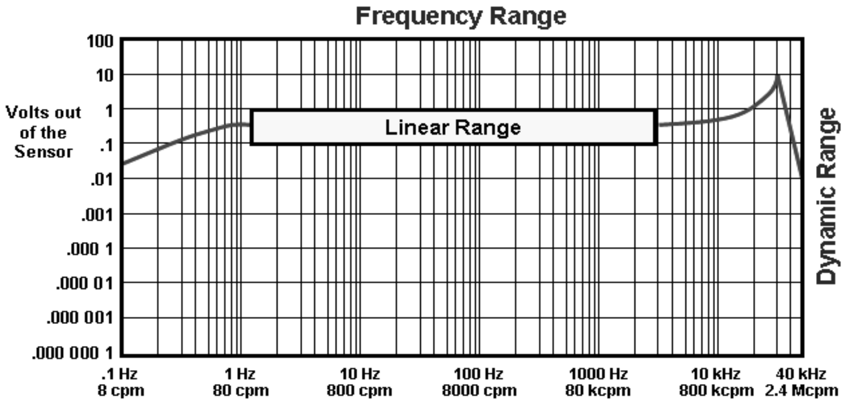


Figure 6-34 - Typical frequency response function

Some measurements actually utilize the resonance range of the sensor to amplify very low levels in the high frequency range. These include: shock pulse, spike energy, HFD, SPM, SEE and demodulated spectra.

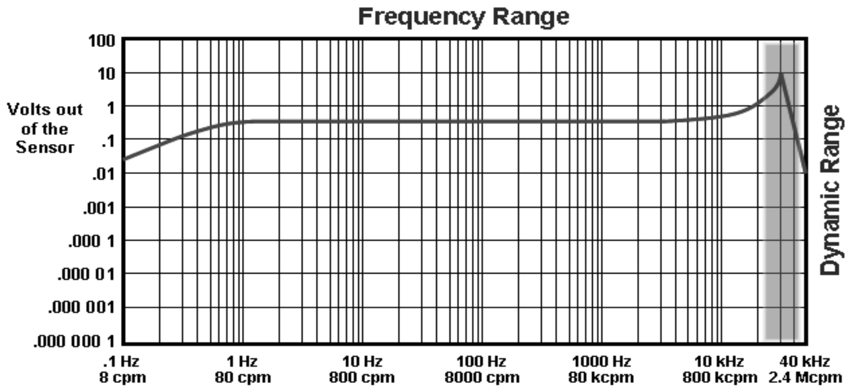


Figure 6-35 - The non-linear range of the sensor

Operating Range - Sensors have amplitude limits too. The upper limit is the point where the output begins to exceed the capabilities of the sensor. The output becomes “saturated” if vibration levels exceed output specifications. The lower limit is level below which the noise of the electronics becomes too great and swamps the data.

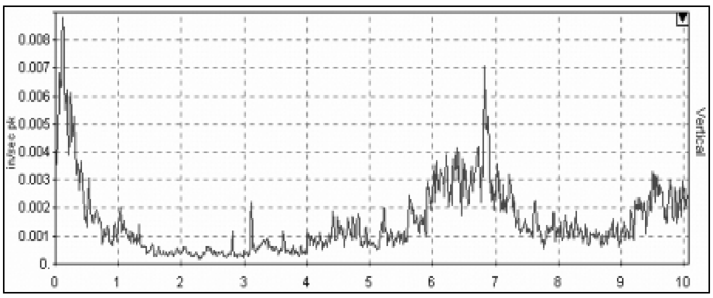


Figure 6-36 - When output is exceeded the spectrum noise floor may be lifted, and/or there may be a ski slope.

Seeing this in the spectrum, some may believe that the sensor is faulty. However, this phenomenon can be used as a diagnostic aid - it is often an indication of very high frequency, high amplitude vibration, as a result of cavitation or other conditions. So don't automatically assume there is something wrong with the sensor - the machine may be trying to tell you something.

In this case, remount the transducer with a rubber washer between it and the machine, which will dampen the higher frequency vibration. If the signal looks OK, the problem is saturation.

Operating Range: Low Amplitude - At the other end of the scale, is the need to be sensitive to the lower limit of the transducer's dynamic range. For low speed machines, where vibration levels are very low, a very sensitive sensor is needed that can amplify the signals sufficiently, and it must have very low electronic noise.

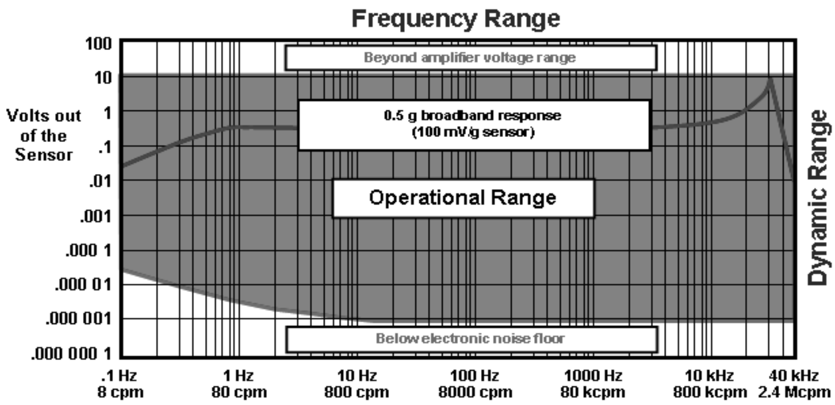


Figure 6-37 - Low amplitude vibration requires another sensor

Selecting the Sensor – The sensor must be selected based on:

- Frequency range required (both high and low)
- The sensitivity and noise
- Environmental conditions (temperature, moisture)
- Mounting Type

As noted earlier, eddy current probes are used on large machines with journal bearings, or in very low speed machines. Eddy current probes are very common on turbines. The frequency response of the displacement transducer extends from DC (0 Hz) to about 1000 Hz.

Figure 6-38

For most other applications an accelerometer is used. Now it becomes a question of the exact type and model number of accelerometer, and the units used to analyze the data.

There are a large number of sensors to choose from. One key issue is sensitivity. The standard accelerometer is 100 v=mV/g. Precision machines and low speed machines require low noise, very sensitive sensors with a much higher sensitivity such as 1 V/g.

Machines with high levels of vibration may require a much less sensitive sensor with a sensitivity of 10 mV/g.

Due to the design of sensor electronics, the frequency response of the sensor is not the same for highly sensitive accelerometers as it is for less sensitive units. (This is independent of the effects of different mounting methods.)

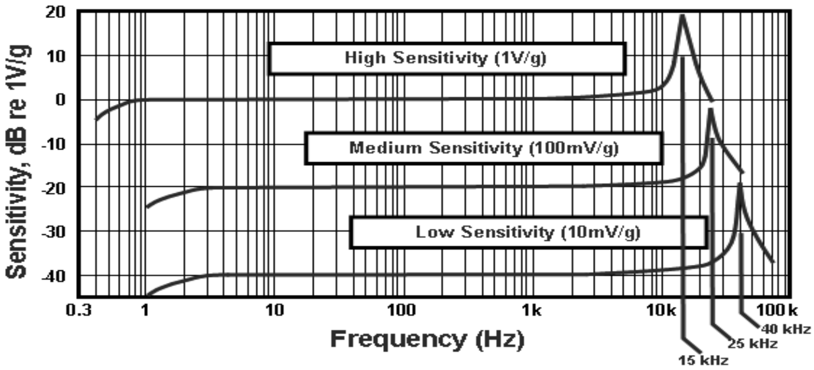


Figure 6-39 - The sensitivity affects the low and high frequency ranges.

There are a great many applications for highly sensitive, low frequency accelerometers. In days gone by, sensor and data collector electronics were not designed to take such measurements. Many applications require these sensors, including machine tool, petrochemical, and paper.

The requirements on the transducer are a highly sensitive amplifier (to give sufficient output to the data collector - typically 500 to 1,000 mV/g), very low internal electronic noise (otherwise the noise will be amplified by the more sensitive amplifier), and a low frequency cut-off (often down to 0.1 Hz or 6 CPM).

With their high sensitivity and consequently lower amplitude range, these transducers are susceptible to overload, especially in the presence of significant high frequency vibration. For this reason, some manufacturers use an internal low pass filter to attenuate the high frequency signals.

You may be wondering why you would not simply use a low frequency, high sensitivity transducer for all your tests. There are very good reasons. First, due to the low frequency specifications of the internal high pass filter, the transducer takes far longer to settle after a mechanical, thermal or electronic (power-on) shock.

Also, as mentioned earlier, they typically have a low pass filter, limiting their high frequency response, which makes them unacceptable for many "general purpose" applications. And finally, due to their high gain amplifier, they are susceptible to overload.

So, while these sensors play an ever more important part in condition monitoring, they should only be used when the application specifically calls for their unique capabilities.

LOW FREQUENCY ACCELEROMETERS (COURTESY OF WILCOXON)

Low Frequency measurement is critical for many industries. The petrochemical, machine tool, and paper industries use low frequency for both condition monitoring and process measurements. Other applications include slow speed agitators, cooling towers, semiconductor lithography, and structural testing.

Low frequency measurements and low levels of vibration are closely related. Acceleration levels decrease at low frequencies. In order to have adequate voltage signals at the acquisition equipment, low frequency sensors have greater output sensitivity (usually 500mV/g) than general-purpose sensors. Additionally, the low-end frequency cut-off is improved (down to 0.1 Hz @ –3dB) in order to read slow speed vibration signals.

Hot Environments - Another common application for accelerometers is in very hot environments, such as the dryer section of a paper machine. Standard accelerometers can survive in high heat conditions, however due to their internal amplifier they have a limit. Over approximately 300 degrees F (150 C), it is normally recommended to use a charge mode accelerometer.

Charge accelerometers use an external "charge amplifier", located away from the sensor, in a more hospitable environment. A special heat resistant cable is then used to join the two.

Figure 6-40

Note: As with all charge mode accelerometers, care must be taken to secure the connecting cable (while the measurement is in progress), as any movement will induce noise on the cable.

Wet or Humid Environments – Not all sensors are hermetically sealed and should not be used for underwater applications, nor on paper machines where steam can be a problem. Moisture penetrates the sensor, damaging the electronics and changing the performance and frequency characteristics.

Accelerometer Mounting

Another consideration regarding the sensors is the way they are mounted. Their mounting method greatly affects the accuracy of the measurement.

Several years ago most analyzer manufacturers supplied an accelerometer with a "stinger" or probe tip. The user then simply held the accelerometer to the bearing and started the measurement. It was fast, easy, and allowed measurements to be taken in otherwise inaccessible locations (because the tip could reach in to tight spaces). But the frequency response and repeatability were unacceptable.

Figure 6-41 - Stinger or Handheld probe used with the sensor

The typical choices include a handheld probe, magnets, and stud mounts. Each mounting method affects the response range of the sensor. If it is not mounted correctly the data will not be useful. Repeatability and Frequency Response will be an issue.

Instead, the measurement must accurately represent the state of the balance condition and the bearing condition, the degree of misalignment, and other faults. Steps must be taken to ensure the quality of the data or it will not provide the information needed for accurate and thorough analysis.

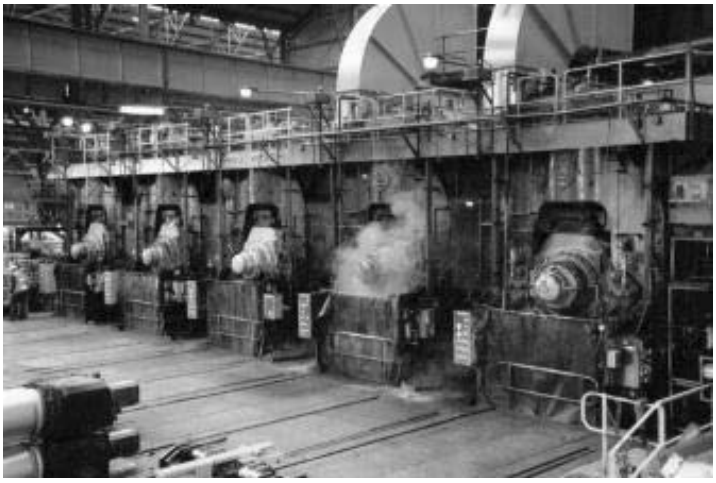


Figure 6-42

Another issue is **Repeatability**. The concern should not be centered on today's data alone, but the analyst needs to see how it has changed since the last reading, and today's reading must be compared to the next reading.

If the readings are taken correctly every time, and the machine is operating in the same way every time, then the only explanation for changes in vibration will be a change in condition.

However, if the way the sensor is mounted changes from measurement time to measurement time, the changes seen can merely be a reflection of the change in sensor mounting method. It is just not practical to go out and collect additional readings every time the readings change.

The sensor must directly contact the machine surface on a place that is smooth and flat. The stronger and stiffer the connection, the more likely it is to get a reading with acceptable frequency response. If there is excess paint, paint chips, rust or grit, the readings will be compromised.

Figure 6-43 - Machine surface should be clean and smooth.

There are a number of mounting configurations that are used to attach the sensors to the machine surface. The method chosen is dictated in part by the application, that is, whether a temporary mounting is used or a permanent mounting for online monitoring.

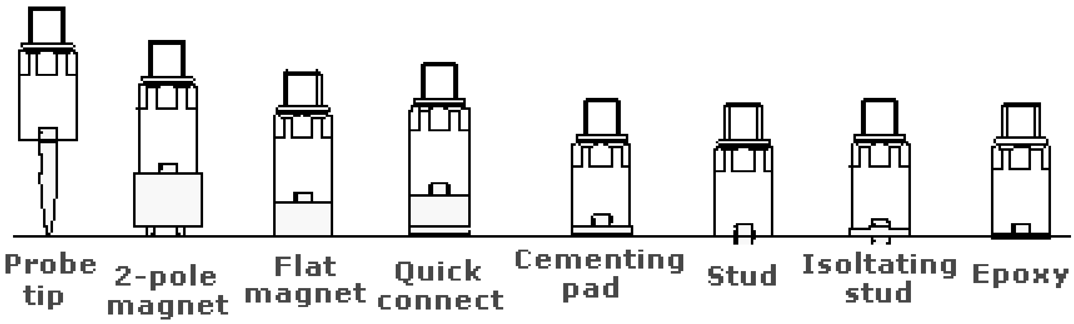


Figure 6-44

Stud mounting

The best option is Stud Mount. This gives the best frequency range. All other mounting methods reduce the upper frequency range of the sensor.

Removing the sensor from direct contact with the machine and inserting mounting pieces such as adhesive pads, magnets, or probe tips, a mounted resonance is introduced.

The level response range of the sensor is the useful range of the data. The mounting methods in Figure 6-45 are shown in progressive order of worst to best response range.

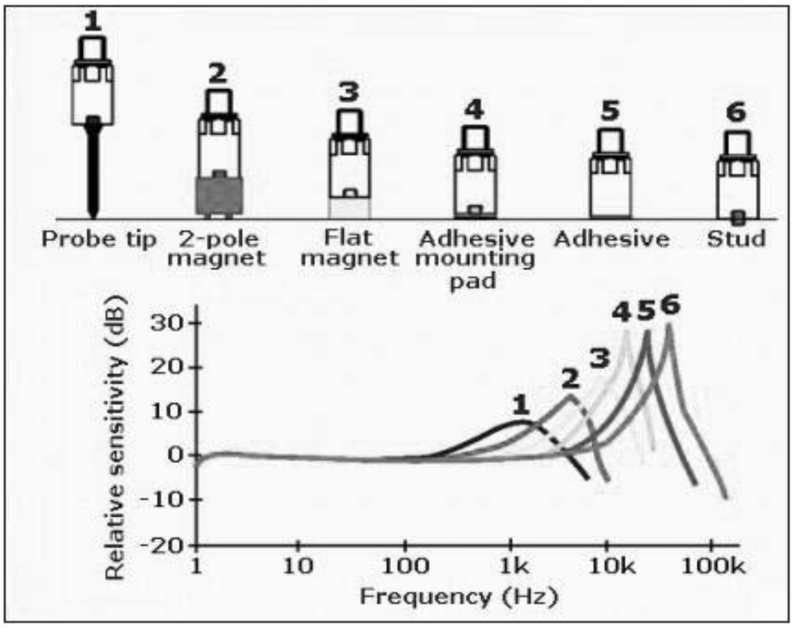


Figure 6-45

Notice that the Probe tip induces a mounting resonance at approximately 1000 Hz but the level range is only to about 500 to 800Hz. Frequencies above that are amplified and therefore not accurate.

All of these mounting methods are shown individually below.

Temporary Mounting

Although **Stud Mounting** is practical with permanently mounted online systems, it is impractical with portable, walk-around data collection. That leaves four options:

- Handheld or Stinger probes
- Magnets
- Quick Connect
- Screw-in

There are various styles in each of these mounting types. Each has its unique characteristics regarding the influence on the data collected.

HANDHELD PROBES

The length of the probes available are from about 2 inches to 8 inches. They were used in the early days of data collection due to their convenience. However, the data varied considerably depending on the amount of pressure used to hold the stinger, the angle it was held, and the position.

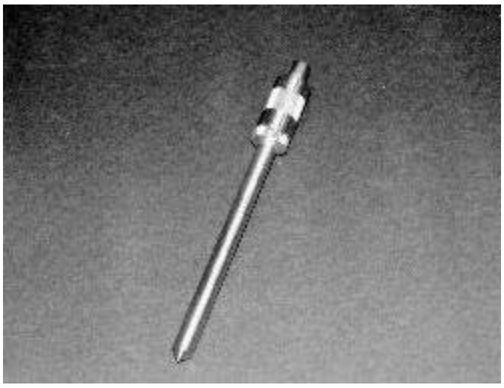


Figure 6-46 - Handheld probe

Therefore the **repeatability** is poor and the level response range is poor. Its convenience is not worth the data loss.

There are times when "some data is better than no data" and they can be used. One instance is reaching through an expanded metal guard to reach a pillow block bearing that is otherwise inaccessible. A better method is to permanently mount sensors on the pillow block bearings and wire them to a junction box for data collection.

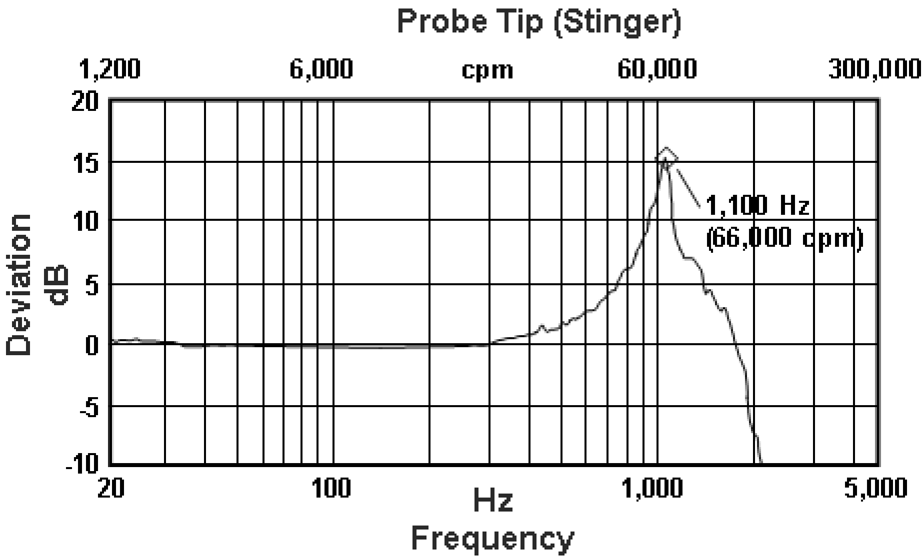


Figure 6-47 - The response curve of a Handheld Stinger

Magnet mounts provide much better results. Magnet technology has improved over the years ensuring a much stronger hold on the machine (which of course has to be magnetically attractive). Used properly, two pole magnets can be successfully applied to slightly curved surfaces, although we would still recommend that the target area be machined flat if possible.

If the surface is painted, it must be kept clean and well maintained. So must the magnet surface. All grit, metal particle and burrs must be removed.

Figure 6-48

Coupling fluids, such as heavy machine oil or beeswax, greatly improve the transmission of the higher frequencies, and ideally should be used with flat bottom magnets.

The magnets vary in size and the amount of pull they have to hold onto machinery. Large two pole magnets for curved surfaces can have as much as a 120 lb pull. Other two pole magnets typically range from 10 lb to 90 lb. pull.

The flat "Rare Earth magnets range in size from 10 lb. pull to 60 lb. pull. They should be used only on flat surfaces such as a mounting pad or a surfaced area of the machine. Surfacers mill a circle on the machine surface the same diameter as the magnet so that it provides a smooth, flat target area for repeatability.

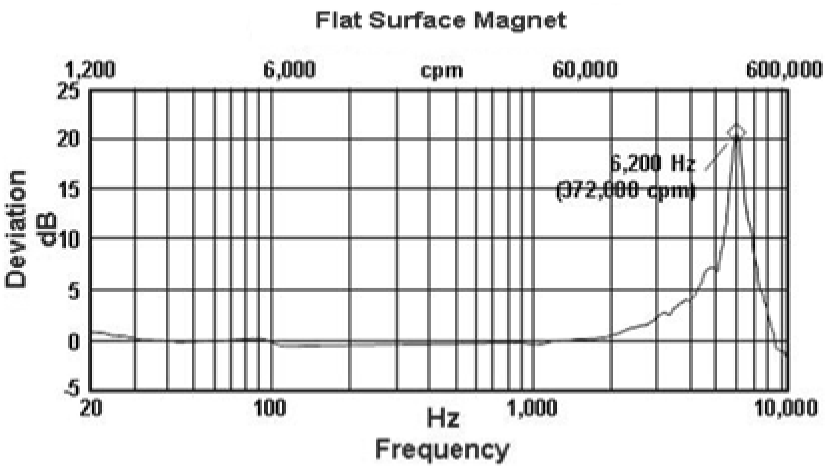


Figure 6-49 - The flat mount magnets provide a useable range to about 4000 Hz.

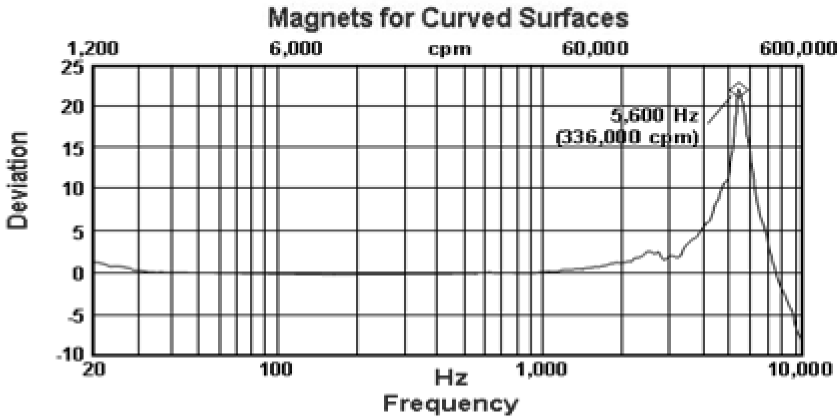


Figure 6-50 - two pole magnets for curved surfaces have a useful range to more than 3000Hz (depending on size)

Magnets have a very strong pull. Be careful when placing them on the machine so that they do not “thump” down. This can damage the sensor. Also avoid getting a finger caught under the edge as the stronger magnets can severely pinch it.

A good method of placing the sensor is set one edge down and then roll it over to make full contact.

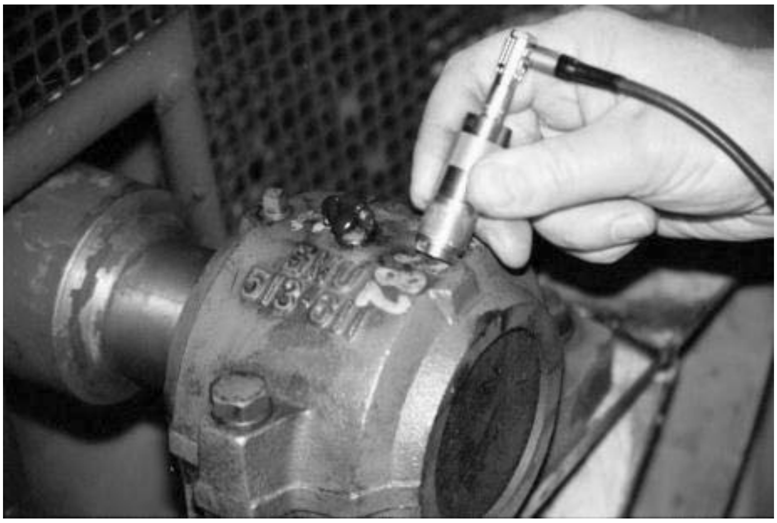


Figure 6-51 - Set one edge of the magnet down and then roll it into full contact with the machine.

MOUNTING PADS OR TARGET PADS

Alternatively, "target pads" can be permanently mounted on the machine. The pads themselves may be bonded to the machine via a stud or epoxy (discussed later). The pads provide a clean, flat area for the magnet to attach (especially useful on machines without magnetically attractive surfaces).

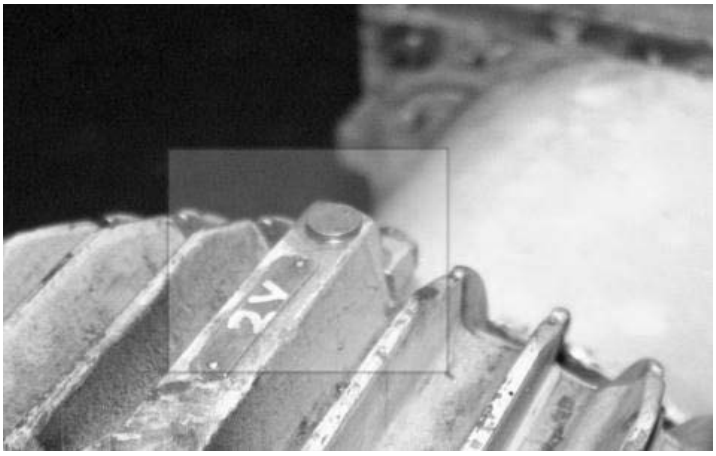


Figure 6-52

Pads improve measurements making them more repeatable with a higher frequency response.

	Model	Description
	**SF5**	Cementing pad, anodized aluminum, 10-32 integral stud, 0.5 inch hex
	**SF8**	Cementing pad, 1/4-28 integral stud, 1 inch diameter
	**SF8-2**	Cementing pad, 1/4-28 tapped hole, 1 inch diameter
	**SF11**	Magnetic mounting pad, 416 stainless steel – will accept <1 inch magnetic base
	SF20-3	Cementing pad 3/8 – 24 integral stud
	**QB-1**	QuickLINK® sensor base, adapter mates to 1/4-28 sensor, for walkaround data collection. Allow quick mount of sensors in less than one turn
	**QP-1**	QuickLINK® mounting pad, 1/4-28 tapped hole base, for use with QB-1
	**QP-2**	QuickLINK® cementing pad, flat base, for use with QB-1

Figure 6-53 - Pads come in all shapes and sizes

	Model	Description
	**FM101**	Fin mount, 0.5000 fin width, 1.25 length
	**FM102**	Fin mount, 0.500 fin width, 2.00 length
	**FM103**	Fin mount, 0.250 fin width, 1.75 length
	**FM104**	Fin mount, 0.250 fin width, 1.00 length

Figure 6-54 - Fin pads are designed to fit between the cooling fins of a motor.

Sensors are attached to the pads via a magnet, or it can be threaded for a stud mount. The sensor is then bolted to the pad which is itself bonded to the machine. This method is highly recommended when triaxial sensors are used. It is extremely useful for repeatability.

Figure 6-55 - Triaxial Accelerometer and mounting pad.

Figure 6-56

Quick-connect mounts are now offered by many of the data collector companies. These devices are permanently mounted to the machine and employ a locking method. The accelerometer is then attached to the pad with a half-turn.

These devices offer excellent repeatability and some are even coded so that the data collector can automatically identify the matching point in the route. They tend to provide good accuracy response if maintained properly.

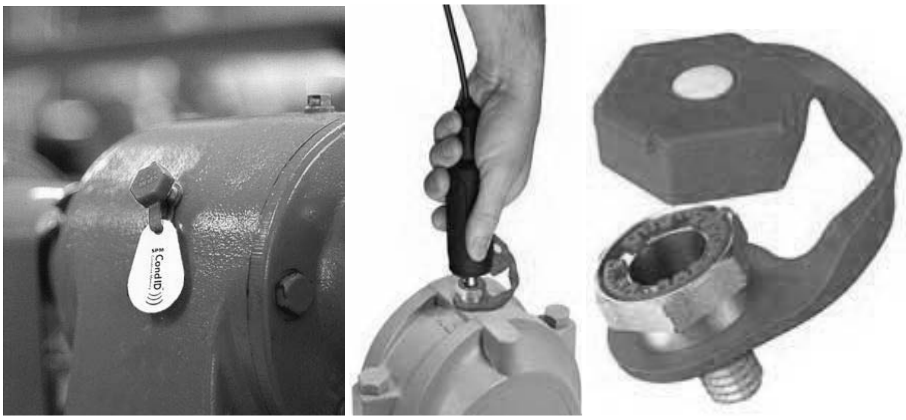


Figure 6-57 - Quick Connect fitting with RFID capability

Table 1 — Effect of mounting on transducer performance

Mounting method	**Effect on transducer performance** (e.g. for a 30 kHz resonant transducer)
Rigid stud mount	No reduction in the resonant frequency of the accelerometer due to its mounting
Isolated stud mount	If a non-conducting rigid material, such as a mica washer, is introduced to prevent ground loops or other influences, the mounted resonance is slightly reduced to about 28 kHz
Stiff cement mount	The resonance is reduced to about 28 kHz
Soft epoxy mount	The resonance is reduced to about 8 kHz
Permanent magnet mount	The resonance is reduced to about 7 kHz
Hand-held	The resonance is reduced to about 2 kHz, but this method is not recommended for measurements above 1 kHz

Figure 6-58 - ISO 13373-1:2002 frequency response limitations

Mechanical transmission path

There must be a good mechanical transmission path between the source of the vibration (the bearings) and the location of the sensor. Many things vibrate on the machine however it is essential to choose the point with the shortest path between the source and sensor.

You must be sure not to mount the sensor on a piece of structure that is itself excited by the machine vibration. Fan covers, coupling guards, motor cooling fins, and other structures are not suitable locations for sensors. These objects can rattle and resonate, and they can block high frequencies from reaching the sensor. Plus, if they are too far from the bearings, the high frequencies that indicate bearing wear will be lost.

Measurement locations

Safety first! It must be safe to access the test point: there must be no risk of becoming entangled, there must be no risk of becoming burned, you should not have to lean over moving or rotating parts, you should be safe from fumes etc.

General guidelines:

1. If the distance between the bearings on a machine component is less than 30 inches (75 cm) – measure only one bearing on that component. If you have a choice, measure the thrust bearing or fixed bearing.

2. Measure at least one bearing on each machine component (i.e. motor, pump, gearbox etc.). Faults like misalignment are detected by comparing data from both sides of a coupling.

3. Different mechanical faults will appear in different measurement locations and measurement axes. The goal is to take the appropriate tests to detect the most common faults. Begin by understanding the machine and its failure modes, then determine how and where to test the machine.

It is necessary to understand the machine being monitored. Vital information should be gathered to determine the frequency ranges of the forcing frequencies. Typically the speed and load are determined from the motor nameplate data. Determine the bearing type and physical details if possible.

Try to gather the motor information such as the number of rotor bars, if possible.

If the machine is not direct driven, gather information regarding sheave sizes, the number of teeth on sprockets, and the number of teeth on any gears. Calculate all shaft turning speeds.

The goal is to determine all possible Forcing Frequencies for the machine.

A sample form is shown below for gathering the information. When one is completed for each machine to be measured and then compiled in a reference book it provides a very useful resource for the analyst.

MACHINE ____________	**PLANT** ____________	**AREA** ____________
UNIT # ____________	**MID** ____________	**AVERAGES** ____________
Prepared By	**DATE**	

TEST RPMs and OPERATING CONDITIONS		**Variable Speed?** YES NO **VERTICAL HORIZONTAL**
Driver ____________	**Intermediate Shaft**	**Driven** ____________
Reference RPM ________	**SPEED RATIO:** ________	**Overhung?** Yes
Ornt: (_)_______ (_)_______	**Ornt:** (_)_______ (_)_______	**Ornt:** (_)_______ (_) _______
Ranges: Low____X High____X	**Coupling Type:** ________	
Mfr: ____________	**Mfr:** ____________	**Mfr:** ____________
Serial #: ____________	**Serial #:** ____________	**Serial #:** ____________
Model: ____________	**Type:** ____________	**Model:** ____________
Type: ____________	**RPM:** ____________	**Type:** ____________
RPM: ____________	**Gear Mesh 1:** Driver ______	**RPM:** ____________
Passing Elements	Driven ______	**Passing Elements**
Bars: ____________	**Gear Mesh 2:** Driver ______	
Slots: ____________	Driven ______	
Poles: ____________	**#Belts** ____________	
Other: ____________	**#Coupling Elements**	
Other: ____________	**Other:** ____________	**Other:** ____________
Brg (name/type)**:** ________	**Brg** (name/type)**:** ________	**Brg** (name/type)**:** ________
Brg (name/type)**:** ________	**Brg** (name/type)**:** ________	**Brg** (name/type)**:** ________
COMMENTS		
Sketch (include mounting block locations, direction of notch, orientations and bearings)		

Figure 6-59

Measurement axes

Triaxial data is best if you have software that can “screen” through the data so you don’t need to analyze each plot manually. The next best axis is vertical or horizontal per bearing (as per the distance between bearings guideline) and one axial reading per component. Some mechanical faults appear in the axial direction. If we do not take data in this axis we may miss the fault. The majority of mechanical faults show up in the vertical or horizontal axis, so don't bother just taking axial readings.

It is important to know where we want to test the machine in order to detect the faults of interest. Unfortunately, we cannot always test the bearings we want to test due to safety issues, motor covers and cowlings, and accessibility.

Often we can find one test point on a bearing, in one axis, but not test points in the other two axes. Using a triaxial sensor or a mounting block allows us to collect all three axes from one point.

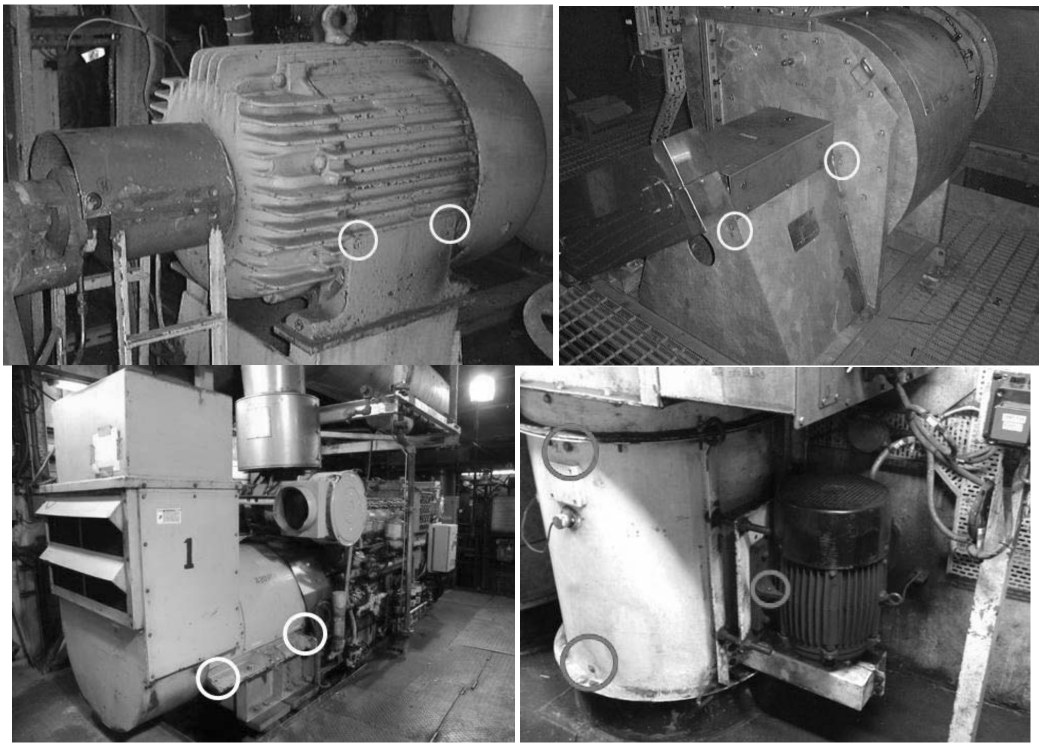


Figure 6-60 - Poor measurement locations

Sensor location suggestions

For motor close-coupled pumps and fans:

- If the motor is < 40 HP (30 kW), measure at the motor driven end (MDE).
- If the motor is > 40 HP (30 kW), also measure at the motor free end (MFE).
- As a rule of thumb, if the distance between the MDE and MFE is > 30 inches (75 cm), test on both bearings

For coupled motors:

- Preferred location of the pad is on the MFE.
- If the motor has cooling fans and finned housing, the next preferred location of the pad will be at the MDE

For overhung coupled pumps and fans:

- Preferred location of the point is on the bearing housing.
- If the design includes a squirreled cage fan supported by pillow block bearings, the preferred location of the sensors are on the MFE and each of the pillow blocks.
- If the design includes a vertical, overhung coupled pump, the preferred sensor locations would be on the MFE and the pump thrust bearing (pump driven end – PDE)
- If the design includes a vertical, coupled, double suction pump, then the sensors will be located on the MFE and the lower pump bearing housing (pump free end – PFE). Note: If the pump capacity is > 3,000 GPM (10,000 LPM), you should take measurements on both the upper and lower bearings

For coupled double suction pumps and fans:

- If the design includes a horizontal, coupled pump, the preferred location of the pads are at the MFE and PFE. Note: If the pump capacity is > 2,000 GPM (7,000 LPM), you should take measurements on the PDE as well

For belt driven machines:

- The preferred location of the test point will be at the MDE where the belt sheave is located close to the motor and at the driven end where the belt sheave is closest

The number of sensors placed on the machine depends in large part on the type and size of component, and on the accessibility. It also depends upon the type of coupling between components: belt, gear, direct drive, flexible coupling, and so on.

It is best to err on the side of too much data. You can always trim back the number of test locations once you have the program up and running, and you can determine if there are any redundancies in the data collected.

Naming conventions

It goes without saying that when you collect vibration readings on a route you must test the correct bearing, on the correct machine in the correct axis. However if you are not sure which machine is “unit 2” or which bearing is “number 2” or what the “V” means then you are likely to make a mistake.

Spend time with the program manager to learn the naming scheme used in your plant, and to understand which machine is which. The convention identifies the position and axis and sometimes also the machine type. We would recommend that you place name tags on points, or mark the point names on the bearings. That will reduce the risk of incorrect point identification.

Many naming schemes number the bearing locations (if there are two back-to-back bearings it is still considered one location). The points are numbered in the direction of power flow; from the "free end" of the motor to the "free end" of the driven component.

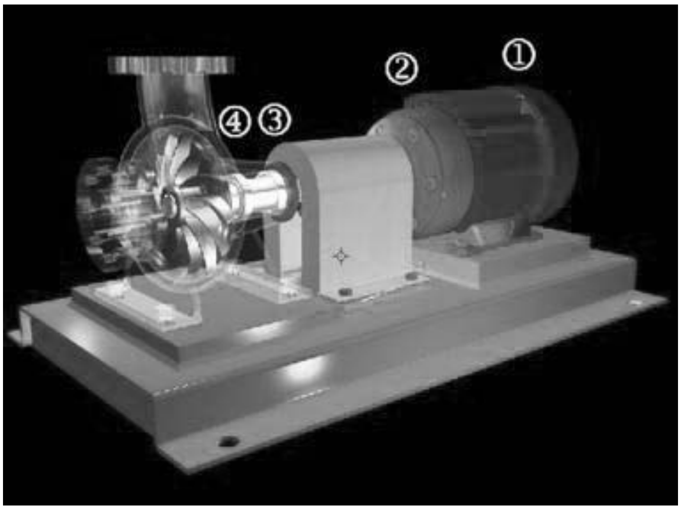


Figure 6-61

Another method uses a combination of an identifier for the component (M:Motor, P:Pump, G:Gearbox, F:Fan (or air handling unit), C:Compressor, and T:Turbine and the designator for inboard "I" and outboard "O".

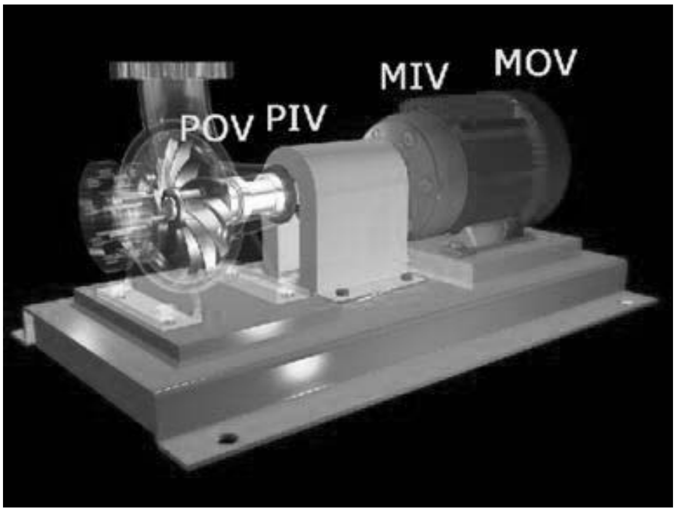


Figure 6-62

Another method identifies the point on the component as either **d**rive-**e**nd "DE" or **n**on-**d**rive-**e**nd "NDE" (also called "free end").

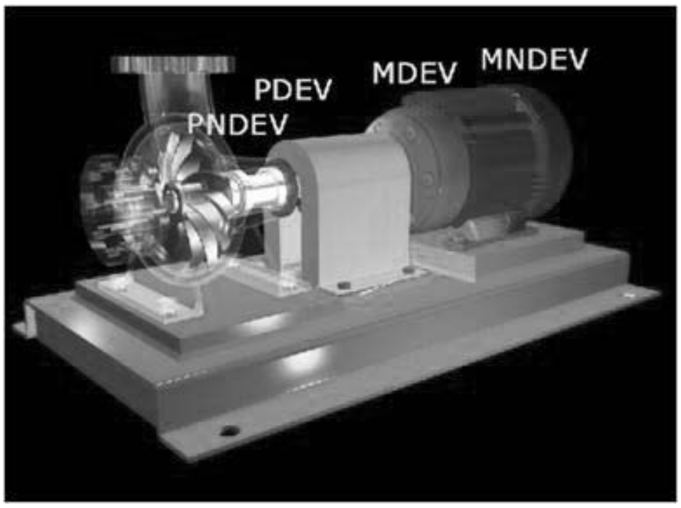


Figure 6-63

And then we must add the axis information. We typically use V:Vertical, H:Horizontal, and A:Axial. Some systems also use the terms radial "R" and tangential "T", however for a horizontal machine Radial = Vertical, and Tangential = Horizontal.

For vertical machines, there is still an axial measurement (although it is in the vertical direction), and vertical and horizontal do not make sense. So we use Axial, Radial and Tangential. Alternatively, there is a convention which makes the vertical axis in the same axis as the discharge pipe, and horizontal axis 90 degrees around the machine in a clockwise direction (looking down on the machine).

There are some systems that use special point identifiers: barcodes, id tags, and other methods. These systems take the guesswork out of data collection. You just connect the sensor, (maybe swipe the barcode), and then the data collector knows exactly where you are. You can take the measurements in any order, and you are assured of the data coming from the correct location/axis.

ISO 13373-1:2002 MIMOSA convention

The convention defines location, transducer type and orientation separately. Six definitions are then combined into an unambiguous, fourteen character (no spaces), measurement location identification.

Definition	Length	Example
Component part (shaft, auxiliary gearbox, roll, etc.)	four alphanumeric characters	See D.2.2 XXXX if unknown
Location (housing number designation)	three digits	001 to 999
Transducer type code	two letters	See Table D.2 XX if unknown
Angular orientation	three digits	000 to 360, XXX if unknown
Transducer axis orientation	one letter	See Table D.3 X if unknown
Direction of motion	one letter	See D.2.7

Figure 6-64 - MIMOSA convention rules

Example: SFTA003AC090RN - Shaft A, bearing housing number 3, single-axis accelerometer positioned 90° counterclockwise from zero, mounted radially, normal motion.

The transducer type is designated by a two-letter abbreviation according to Figure 6-65.

Code	Transducer type	Code	Transducer type
AC	Single-axis accelerometer	PD	Dynamic pressure
AV	Single-axis accelerometer with internal integration	PS	Static pressure
AT	Triaxial accelerometer	SG	Strain gauge
CR	Current	TC	Temperature-thermocouple
DP	Displacement probe	TR	Resistance temperature detector (RTD)
DR	Displacement probe used as a phase reference	TT	Torque transducer
LT	LVDT (linear voltage differential transformer)	TO	Torsional transducer
MP	Magnetic pick-up (shaft speed/phase reference)	VL	Velocity transducer
MI	Microphone	VT	Voltage
OP	Optical transducer	OT	Other

Figure 6-65 - Transducer type abbreviation listing

The angular position of a vibration transducer is measured from a zero reference located at 3 o'clock when viewed at position number 001, looking into the machine. The angle increases counterclockwise (regardless of the direction of shaft rotation) in the plane of shaft rotation from 0° to 360°.

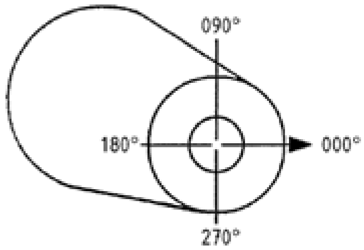


Figure 6-66 - Angular convention: horizontal

The zero reference is located in the direction of flow with angular position measured counterclockwise in the plane of shaft rotation when viewed from the top (position 001) looking down (see Figure 6-67). The zero reference on machines that reverse flow (e.g. pump storage units) is established for operation as a generator.

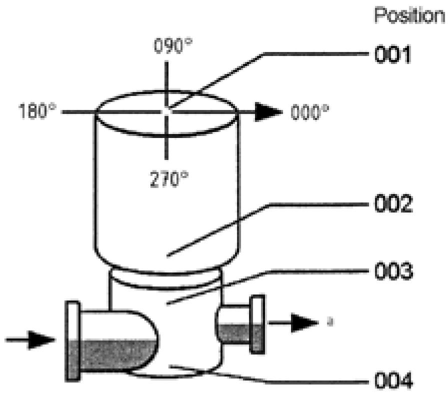


Figure 6-67 - Angular convention: vertical

A single letter defines the direction of the transducer sensitive axis (see Table D.3). This portion of the identification provides unique descriptive information when the transducer sensitive axis does not coincide with the radial defined in D.2.5 (see Figure D.7: XXXAC135H, XXXAC090T, XXXAC315A). It is redundant when the sensitive axis coincides with the defined radial.

Code	Direction	Description
R	radial	transducer sensitive axis perpendicular to and passes through the shaft axis
A	axial	transducer sensitive axis parallel to the shaft axis
T	tangential	transducer sensitive axis perpendicular to a radial in the plane of shaft rotation
H	horizontal	transducer sensitive axis located at 000° or 180° only
V	vertical	transducer sensitive axis located at 090° or 270° only

Figure 6-68 - Transducer axis orientation reference code

The final character in the measurement location identification code is either an N (normal) or R (reverse) to identify transducers mounted in opposition where machine motion in one direction results in positive motion in one transducer (N, normal) and negative motion (R, reverse) in the other.

Axial transducers mounted in opposite directions at the two ends of a machine are the primary example. Axial machine motion towards the reference end is normally designated as positive. The axial transducer closest to the reference end of the machine, position 001, will be designated as normal (N) when mounted so that positive motion towards the transducer produces a positive signal output. Likewise, motion towards the reference end will produce a negative signal from the axial transducer at the opposite end, which is then designated R (reverse).

The angular orientation defines the direction of motion for radially mounted transducers. Therefore, a default of N (normal) should be utilized for transducers mounted radially.

Collecting Good Data

MEASUREMENT PROCEDURE

On a routine basis each machine will be visited and vibration measurements collected. Elsewhere in this training course we have discussed how often to take measurements (monthly, quarterly, etc.), and what type of measurements to take (overall readings, waveforms, envelope/demodulated spectrum, etc.). For now we need to develop the measurement procedure.

Figure 6-69

Earlier the concept of repeatability was discussed. The sensor must be mounted on the machines in the same way every time so that the only reason for change in the vibration measurements will be a change in the condition.

What if it was running at a different speed, or load? The vibration measurements are affected by these parameters, and others. Our task is to attempt to define a repeatable and controllable set of test conditions.

A measurement procedure must be in place and it must be followed for each measurement, every time.

The data MUST be repeatable.

But there may be situations beyond anyone's control. What if the speed or load cannot be controlled? There are some solutions that can help. Depending on the software used, options may be available to help compensate for speed or load changes. However, if not, there are two other solutions.

1. Define a set of Measurement Conditions and collect data only when the speed/load are well within the defined range.

2. Create multiple machines in the database, one for each speed range or load range. This provides the benefit of comparing the machine to similar conditions.

It just may be possible to coordinate with Operations to adjust the speed or load to fall into the range required for the measurement. Cooperation with Operations is always a good policy and they often have information that can lead to a quicker diagnosis of problems, too.

Figure 6-70

Besides the wrong speed or load, other things can go wrong, too. Anyone who has a lot of experience collecting can tell you that they have collected data on machines that were not running, or collected data on the wrong measurement point, or even the wrong machine. Those things do happen.

Any measurement made and stored in the database MUST be stored at the correct point. It must be the correct machine, and the correct measurement point direction.

One solution that helps in this is to use Bar Codes, or RFID tags or connectors that locate the matching point on the route. The use of barcode labels and keyed quick connect pads can greatly reduce this source of error, and generally make navigating the plant much easier - and more repeatable.

BE OBSERVANT

Just take a moment to stop, look and listen. Look for oil leaks. Listen for unusual noises. Feel for unusually hot bearings. And if possible, talk to local operators who may have specific information.

If something of interest is observed, most data collectors enable you to enter a note or a quick-"notecode". It takes time, but it is incredibly valuable information.

Figure 6-71

One of the greatest benefits to operating a vibration monitoring program is that someone is visiting the machine regularly, making these observations. It's surprising what one might find.

Some things to look for include:

- Checking that the machine is operating under the correct conditions.
- Check that it appears to be operating normally.
- Check for General Maintenance: cracks, loose/broken bolts, leaks, clogged vents, grout cracks or other foundation problems.
- Check the machine as a whole:
    - Does it sound or smell different?
    - Do the bearings feel hot?
    - Are pipes or structure vibrating excessively?

Record these observations. It may be useful later.

These observations made in the "Note codes" of the data collector normally store them with the machine data. These are then included in the Measurement Exception Report. This is always a useful tool for the analyst.

FOLLOWING THE ROUTE

In this Category II course it is expected that you are experienced in collecting vibration data. However, here are some helpful tips:

Make sure the surface is clean from dirt and grit. Carry a rag and wipe the surface before collecting the data.

Figure 6-72

If a flat magnet is used, the surface should have been prepared so that it is even and flat. There are surfacing tools that are available for this. The addition of a lubricant between the flat sensor and machine can enhance the measurement.

CHECKING AND REVIEWING THE DATA

On many analyzers it is possible to get a sneak preview of the data before committing to the measurement. This may be in the form of the overall reading or even a waveform or spectrum. Monitoring the data prior to collecting the data for the point shows how the machine is varying due to load or speed changes. If there are fluctuations, attempt to collect the data at the peak of the cycle.

Figure 6-73

Check the spectrum for a Ski slope or transients. After collecting the data for the measurement point this is something that should become normal habit. This can normally be done while the data is being collected if the analyzer is set for live display of the spectrum.

If the person collecting the data is familiar with the machine, then the data should be checked now for to see if levels have increased. If something doesn't look quite right, recheck the sensor mounting and the operating conditions. If it is warranted, then perform additional measurements for better analysis. Measurements that can be made include:

- High resolution data – collect a higher resolution spectrum to include the region that is questionable. Store this data along with the route data.
- Phase Data – take along an additional accelerometer and cable. This can be used to perform Cross-Channel Phase measurements. Record the readings in notes or on a notepad.

These additional measurements should only be performed by qualified personnel.

REPEATABILITY IS ESSENTIAL

It cannot be stressed strongly enough how important it is to collect the data in such a way that the only thing that changes from one test to the next is the machine's condition. If not, as soon as a change is detected, a retest of the machine will be required because of a lack of confidence in the data. The fault may be misdiagnosed and time will be wasted, and you may generally embarrass yourself.

Recognizing Bad Data

There are some rules of thumb for collecting data. One basic one is to view the data as it is collected. Most analyzers have an option for displaying the data as it is collected. Use this mode and watch the screen as it is collected. This is the time to check the data for problems, not after it has been uploaded into the computer.

Besides checking for machinery condition problems the data should be checked to see that it seems reasonable, that the data is good data. There are indications in data that are typical recognizable patterns of bad data. Bad data can be caused by a bad cable or cable connections, a sensor fault, mounting conditions, settling time, or even cable movement. A few concepts must be clearly understood regarding the way transducers work and the way they are powered.

Earlier in this chapter the internal construction of accelerometers was discussed and the way they are powered. In that section we learned that there are a few reasons why the output of a sensor may become unstable. First, a sensor has a built-in amplifier, and in some cases an integrating circuit. When power is applied to the transducer the signal will ring, becoming unstable while the circuit "settles". Any data collected during this settling time will have a time waveform which shows a varying DC offset, and the spectrum will have a "ski slope".

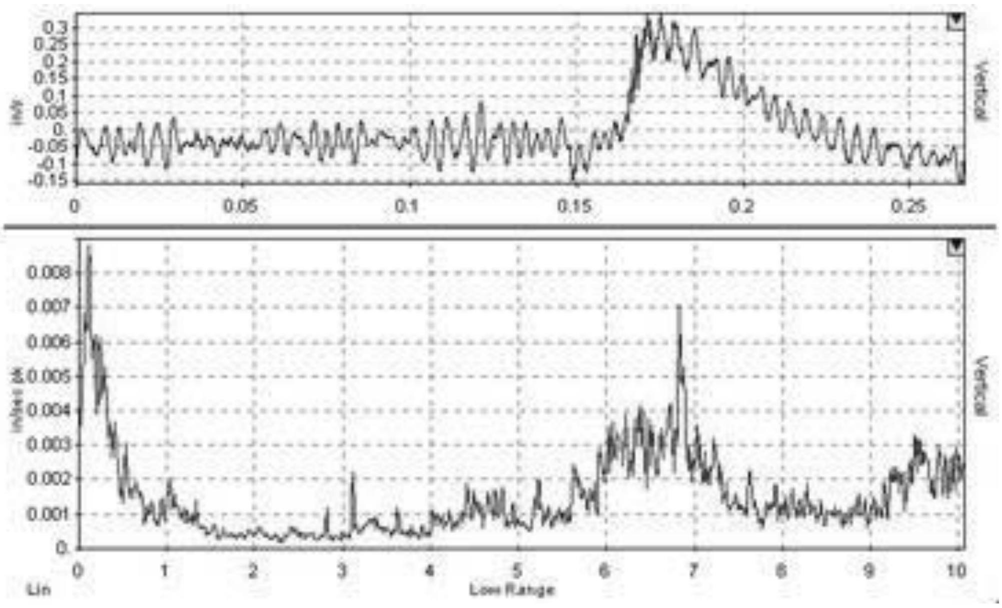


Figure 6-74 - Bad data due to insufficient Transducer Settling Time for the sensor

If this phenomenon is observed, the "settling time" setting in the software must be increased. This controls the settling time in the analyzer. Note that this applies only to ICP transducers (internally amplified.) This setting is also available in the analyzer.

Thermal Transients

Thermal Transients cause similar ski slopes when the sensor is moved from a hot surface to a cold surface or vice versa. The thermal shock causes the sensor output to ring. Allow the sensor to change temperature before beginning the measurement.

Figure 6-75

Mechanical Shock

A similar situation occurs when the sensor is bumped or thumped down as is common when using a magnet mount. A higher voltage is produced and the output will "ring". Again the spectrum will have the characteristic ski slope and the waveform may have a transient or surge evident. See Figure 6-76. A solution is to prevent the sensor from being bumped during the measurement and to place it gently by setting down one edge of the magnet and rolling the sensor into position.

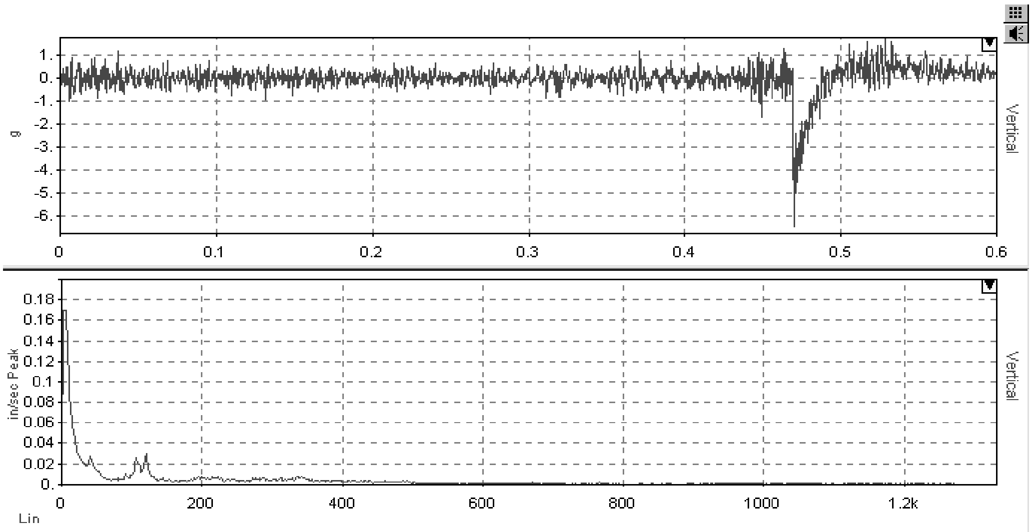


Figure 6-76 - Mechanical shock produces a ski slope in spectrum and a transient in the waveform

Sensor Overload

High amplitude vibration (sensor overload) can "saturate" the amplifier generating a ski slope in the spectrum due to intermodulation distortion or "washover distortion."

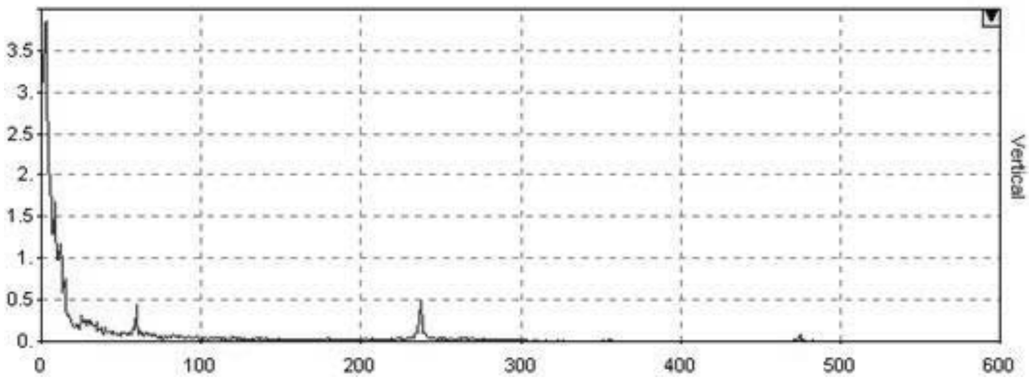


Figure 6-77 - High amplitude ski slope due to "sensor overload"

The excessive vibration can come from surrounding machinery or even excessive cavitation. The amplitude may be excessive.

The saturation can occur in the integration circuit in piezovelocity sensors.

If this continues regularly on a machine, an isolating washer between the magnet and the machine can be used to damp high frequency vibration.

POOR HIGH FREQUENCY DATA

If the high frequency content of the measurement is "missing" or is reduced (this is only seen if data is compared to previous measurements), then it may be that the sensor was not mounted correctly.

If the contact surface is dirty then the frequency response will be reduced - the high frequency content will not be transmitted through to the transducer, which of course means that it will be missing from the measurement.

Figure 6-78

Improve the data collection techniques, ensuring the surface is smooth and clean before the test. If target pads are used, consider purchasing caps that keep debris off the surface.

Loose Mounting and Unexpected Harmonics

Loose mounting can cause the sensor to rattle generating harmonics in the waveform. It is typically caused when the sensor did not make a strong contact with the machine. Ensure the sensor is firmly in place.

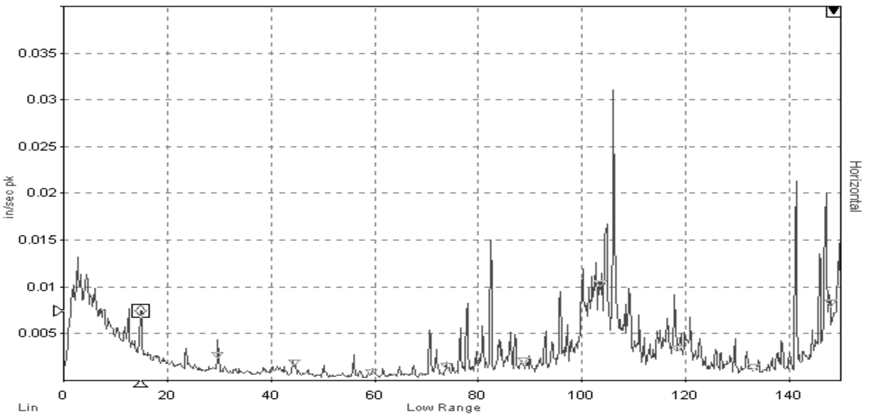


Figure 6-79 - Ski slope and unexpected harmonics in the spectrum due to loose sensor mounting.

POOR SETUP

If the input range is very high and the measured voltage is very low, a "chunky" time waveform may result which produces a ski slope in the spectrum. This may occur due to the sensor being bumped in the beginning of the measurement and the circuit auto-ranges to the maximum amplitude. Retake the data after allowing the sensor to settle.

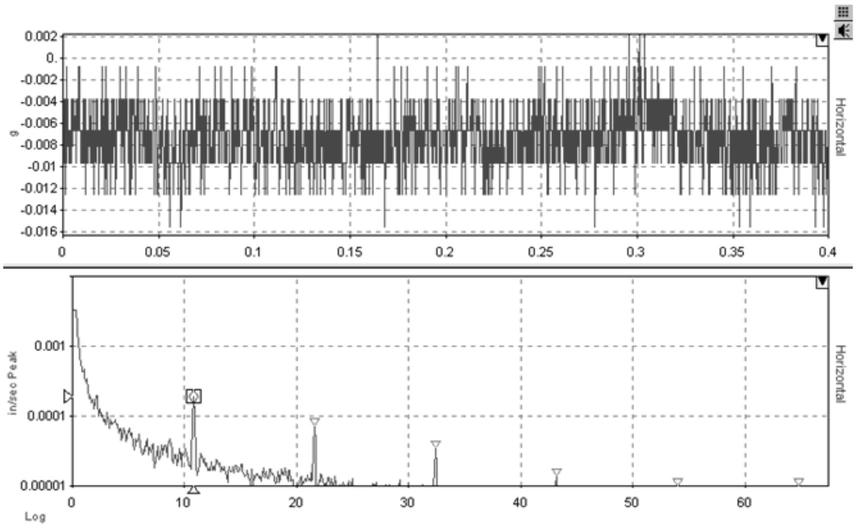


Figure 6-80 - Chunky time waveform due to poor sensor setup

This spectrum is typical of the ski slope due to bad cable connection, or mounting issues. Note the overall alarm was triggered.

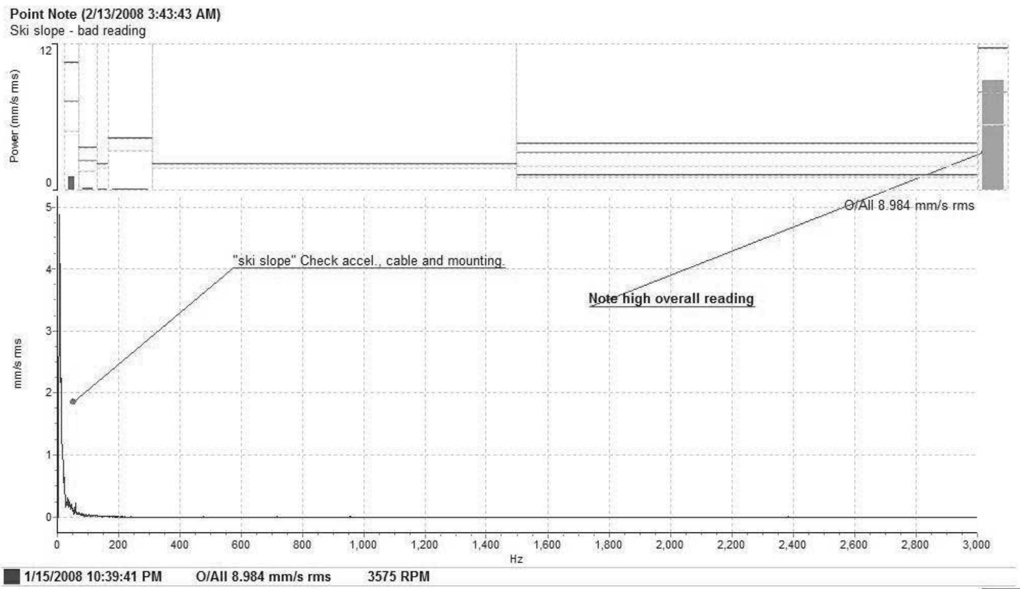


Figure 6-81 - This spectrum shows the ski slope and the alarm triggered.

ICP SENSOR PROBLEMS

There are problems that can occur with internally amplified (ICP) accelerometers.

As described in the accelerometer section, these transducers are powered by adding a DC bias voltage (of between 8 and 14VDC) to the leads of the transducer. The dynamic AC signal is then superimposed on that bias voltage, resulting in a swing from 0 VDC to the limit of the power supply, typically between 18 and 30 VDC. The data collector then removes the bias voltage, leaving the dynamic signal.

By observing the signal level, and checking the bias voltage (when a fault is suspected), it is possible to detect a range of sensor and cabling faults. The analyzer should check for the following conditions:

- Open Bias Fault: Supply Voltage 18-30 volts
- Short Bias Fault: 0 volts
- Damaged Sensor: Low Bias, High Bias
- Erratic Bias and Time Waveform
- Truncated Time Waveform: Sensor Overload

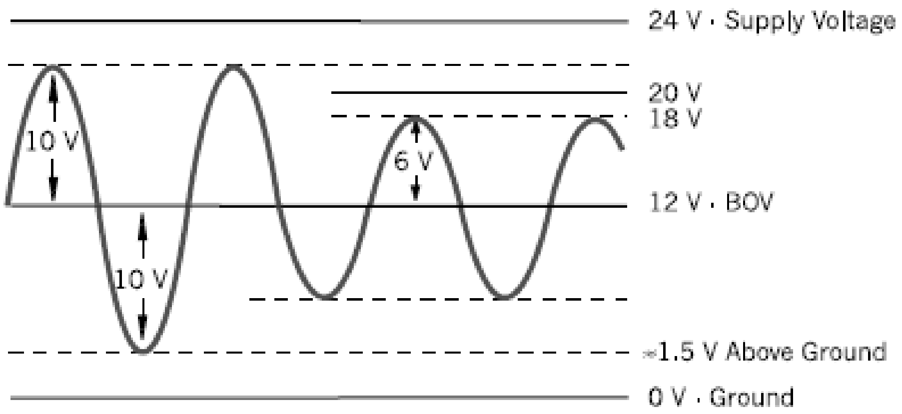


Figure 6-82 - from Wilcoxon, a transducer company. BOV is Bias Output Voltage

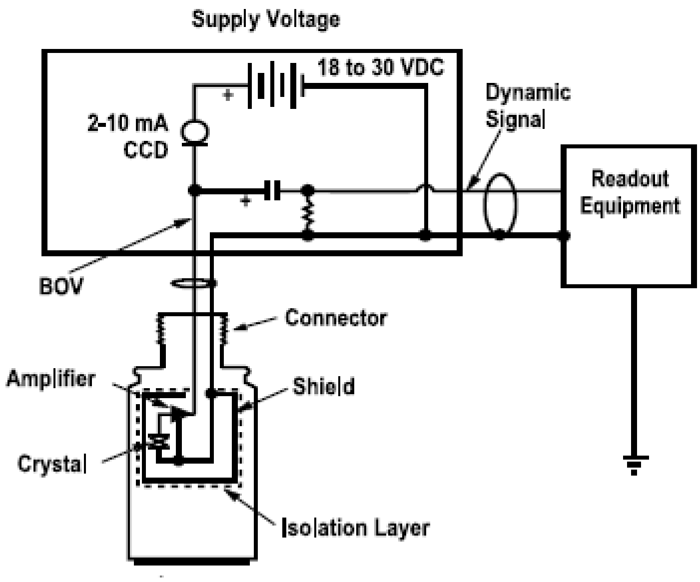


Figure 6-83 - Schematic of sensor and data collector - courtesy Wilcoxin

HARMONICS OF AC LINE FREQUENCY

Another source that wreak havoc on data influence from any line frequency. For this reason it is good to avoid collecting data from the side of the motor that the electrical junction box is fastened to.

Figure 6-84

Wilcoxin, a manufacturer of sensors and data collection components describes it well. "Line Frequency Harmonics in Spectrum Harmonics of AC line power frequency usually indicate interference from motors, power lines and other emissive equipment. First ensure that the sensor shield is grounded (at one end only!). If the shielding is good, check the cable routing. Avoid running the cable alongside high voltage power lines and only cross power lines at right angles. For example, if a power cable is 440 Volts and the vibration signals from the sensor are at the milli and microvolt levels, any cross talk can severely corrupt the data."

Figure 6-85

BOV	Spectrum	Time Waveform	Fault Condition	Action
0	No signal	No signal	No power or cable/connector short	• Test/turn on power • Test cable isolation • Repair/replace cable
2.5 - 5V	No signal	No signal	Damaged amplifier	• Replace sensor
10 – 14V Stable	High low frequency "ski slope"	High amplitude high frequency noise	High frequency overload (steam release, air leak, cavitation etc)	• Repair steam leak/dump • Use less sensitive sensor • Place rubber pad under sensor
10 – 14V Stable	Very high low frequency "ski slope" no high frequency signal	Choppy	Damaged amplifier	• Replace sensor
10 – 14V Stable	Good signal strong 50/60Hz	50/60Hz	Inadequate shielding	• Connect/ground cable
10 – 14V Stable	High low frequency noise	High frequency spikes	ESD arcing impacts	• Reroute cable • Use less sensitive sensor • Place rubber pad under sensor
10 – 14V Stable	High low frequency noise	Jumpy/choppy	Intermittent connection	• Repair connection
18 – 30V	No signal	No signal	Reversed powering	• Reverse leads
18 – 30V	No signal/weak 50/60Hz	No signal	Open cable connections	• Repair connection

Table 6-1 - ICP sensor Diagnostic Chart

Measurement recommendations

ISO 13373-1: 2002 provides some guidelines as to measurement type, axis and location.

B.1 Machine details

For each machine being monitored, the following information should be recorded:

— unique machine identifier (e.g. equipment code);

— machine type (e.g. motor, generator, turbine, compressor, pump, fan);

— power source (e.g. electric, steam, gas, reciprocating internal combustion (RIC), diesel, hydraulic);

— rated speed (e.g. r/min or Hz);

— rated power (e.g. kW);

— configuration (e.g. direct, belt or shaft driven);

— machine support (e.g. rigid or resiliently mounted);

— shaft coupling (e.g. rigid or flexible).

The following additional information may also be recorded:

— function (e.g. driver or driven).

Figure 6-86

B.2 Measurements

For each measuring system, the following information should be recorded:

— date and time (including time zone) of measurement;

— instrument type;

— transducer type[1)] (e.g. eddy current, velocity, accelerometer);

— transducer method of attachment (e.g. probe, magnet, stud, adhesive);

— measurement location[1)], orientation[1)] (e.g. description or code);

— measurement value (e.g. numeric quantity);

— measurement unit[1)] (e.g. µm or mm/s or m/s^2);

— measurement units qualifier (e.g. peak, peak-to-peak, r.m.s.);

— measurement type (e.g. overall, amplitude over time, spectrum);

— FFT or other processing (e.g. filter, number of lines, number of averages).

Figure 6-87

The following additional information may also be recorded:

— speed during measurement (e.g. r/min or Hz);

— power during measurement (e.g. kW);

— other significant operating parameters (e.g. temperature, pressure);

— calibration requirement, type and date of last or next required calibration.

Figure 6-88

Measuring Phase

As a review of phase… there are two types of phase:

- Absolute phase – a phase reading at running speed
- Relative phase – the phase difference between two points

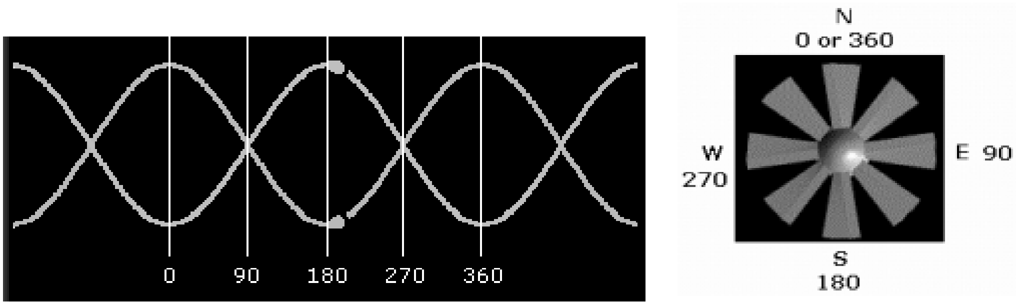


Figure 6-89 - Relative phase is a phase difference between two points

Absolute phase readings are taken with a phase reference – often a once-per- tachometer. Readings can be taken at a number of points and compared to understand relative movement. They are used for balancing.

Absolute Phase reading is made at running speed only although methods do exist to get phase at 2X, 3X, 4X, etc.

Figure 6-90

Relative phase readings are taken between two points on a machine, using a two channel data collector and two sensors. Phase at all frequencies is available

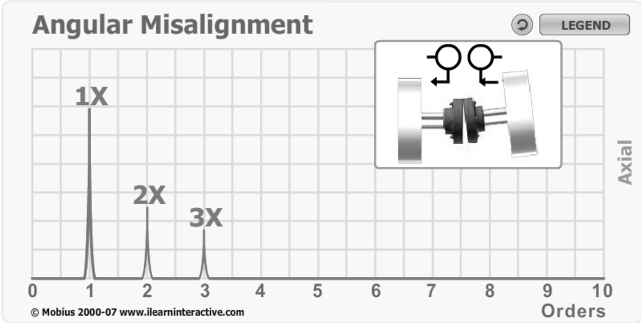


Figure 6-91

Relative phase is used to help in understanding the movement of the machine. Is the machine out-of-phase across the coupling? Normally the relationship looked for is in-phase (zero degree difference) or 180 degrees out of phase.

PHASE AND THE SPECTRUM

In normal routine data collection, whenever a spectrum is derived from the waveform, a by-product is phase data at every line of the spectrum. These phase values are relative to the beginning of the time record used to compute the FFT. But there is no way of knowing when that time record was captured. So the phase values are meaningless and are discarded.

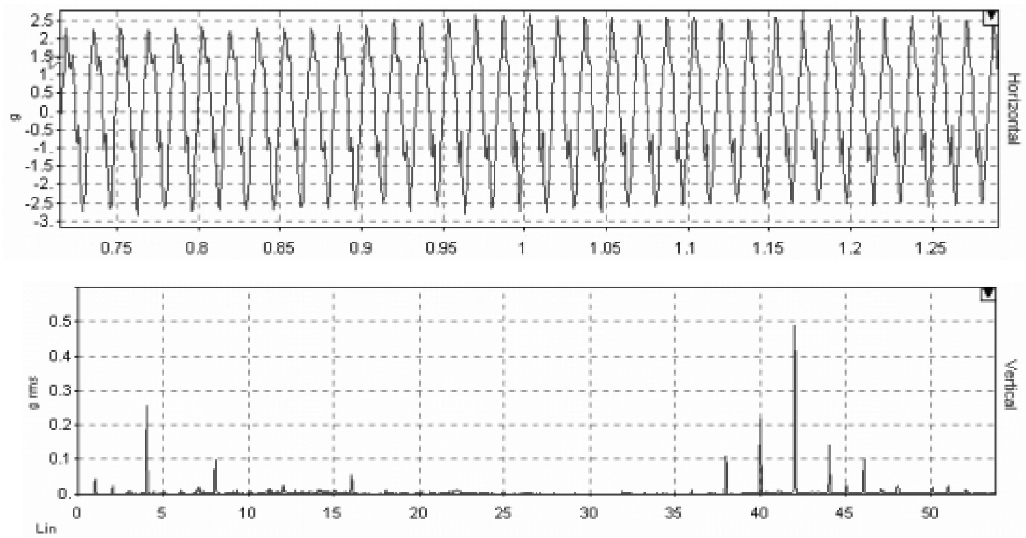


Figure 6-92 - Phase values are calculated for every line in the spectrum, but are discarded.

In order to have value there must be a way to control the start of the time record. Or if two time records were collected from different sources simultaneously, then the phase readings would be relevant and useful.

Most data collectors measure phase. It is usually done via a tach pulse. The tach input may be labeled “external trigger” or “phase”.

Figure 6-93 - the tach input may be labeled trigger or phase.

There are a number of options for a tachometer reference.

A photo tachometer requires reflective tape on the shaft. This triggers the photo tach each revolution and produces a square wave with a frequency of the machine speed.

Figure 6-94 - A photo tach

Instead of a photo tach, a laser head can be used.

A non-contact probe can be used on the keyway to produce a once per rev pulse.

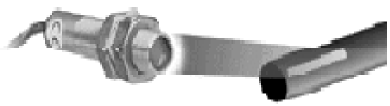


Figure 6-95 - A laser tach

Absolute Phase setup

Figure 6-96 - A typical setup with laser tach and sensor

STROBES

A strobe (stroboscope) with a tracking filter can also be used to obtain a phase reference. However, if there is not a tracking filter, it is not recommended for phase reference because it does not follow the machine speed accurately enough. It is imperative that the reference does not change from measurement to measurement.

Figure 6-97

What happens inside the data collector?

When set up correctly the data collector will begin each time record with the trigger signal. When the FFT is performed, the phase value at the running speed frequency (based on the trigger signal) is extracted. The phase reading is therefore relative to the trigger reference.

The phase reading can be used for balancing as well as for relative phase measurements.

To compare phase readings between points on the machine, the photo tach must not move, but move the sensor to another position and record the reading.

Example:

1. Set up the tachometer and place it on the drive end of the motor in the axial direction to see how it is moving end to end. The reading may be 23 degrees. Is that good or bad? It really does not matter. It is the reference reading. Record that value.

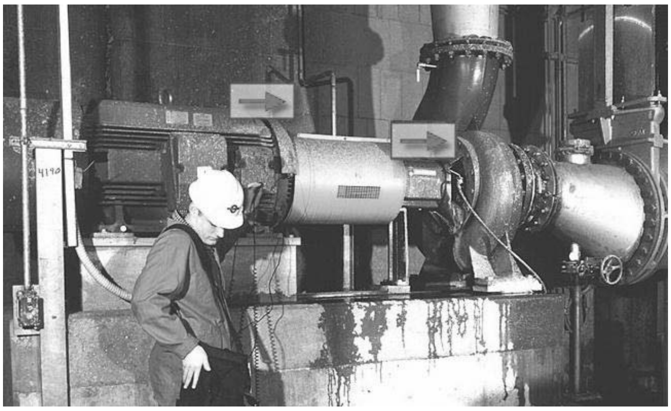


Figure 6-98 - Adjust axial readings for sensor direction

2. Take a reading on the inboard bearing of the pump in the axial direction. This time it reads 28 degrees. But wait. The sensors must be pointed the same way for axial measurements. In this case the reading was taken with the sensor pointed in the opposite direction from the motor reading. So the reading must be adjusted by 180 degrees.

3. Adjust the reading by 180 degrees. The adjusted reading is 208 degrees. So the two points are ~180 degrees out of phase which indicates an alignment problem.

TWO-CHANNEL RELATIVE PHASE

The absolute phase method involved controlling the beginning of the time record by synchronizing it with the rotation of the shaft.

But how does that differ using a multi-channel data collector or analyzer instead of a tach trigger? Now the time record for both channels starts at the same arbitrary time. So comparing the phase readings at 1X (in fact at any frequency), gives a measure of the relative phase between the sources of the two vibration signals. This is called a **"Cross-channel" phase** measurement.

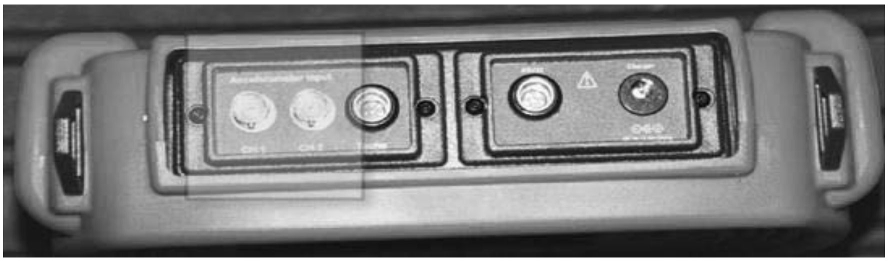


Figure 6-99 - 2 channels can be used to measure relative phase - Cross Channel phase.

To make the same measurement on the motor –pump that was made using Absolute Phase requires a different setup.

Setting up for Cross-Channel phase measurement requires two sensors. Place one on the motor drive end in the axial direction. Place one on the pump drive end in the axial direction. The reading at the running speed will be approximately185 degrees.

OPERATING DEFLECTION SHAPE ANALYSIS: ODS

Absolute Phase measurements can be used to build a picture of "whole machine vibration" to see how it is moving. The results can be plugged into software that then animates the readings to show how the machine or structure is moving.

It makes it very clear whether the machine is rocking from side to side or up and down.

The process requires placing a sensor at an arbitrary point on the machine, which must be left in place as a reference point for all measurements. The other sensor is moved from point to point and the readings are recorded.

This provides a phase angle relative to the arbitrary point.

ODS is covered in detail in the "Natural Frequencies and Resonances" chapter.

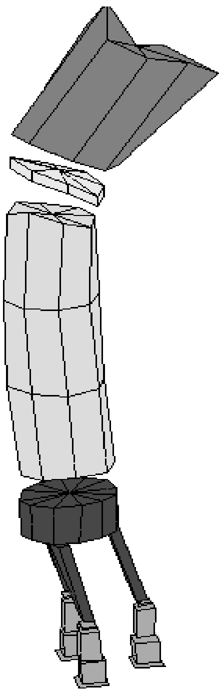


Figure 6-100 - Operating Deflection Shape –ODS

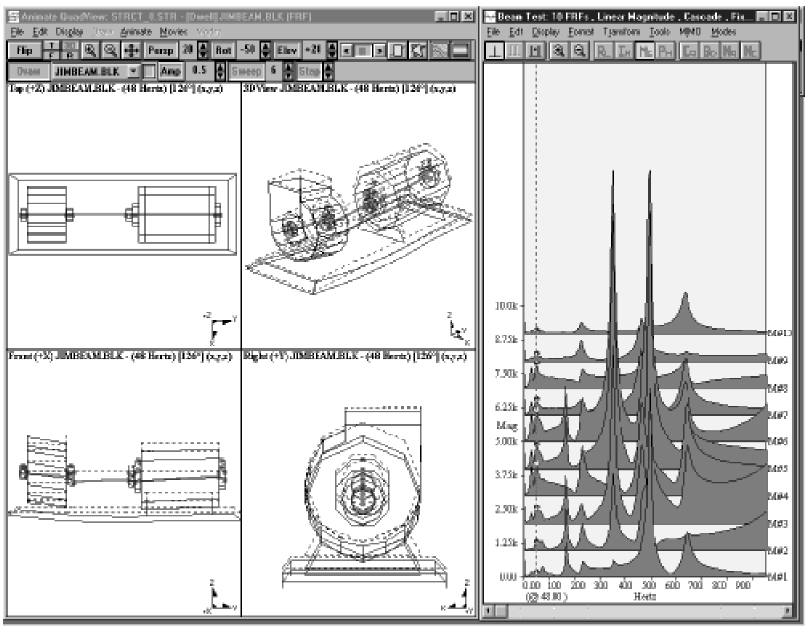


Figure 6-101 - Typical ODS program on motor and fan

Chapter 7
Trending

Objectives:

- Describe Overall trends
- Describe two methods to trend using spectra
- Describe the value of trending bands vs. overall values
- Describe the accuracy of forecasting future levels
- Compare like machines

Trending is very important in condition monitoring: We are very interested in the current state, but we are more interested in how parameters change over time.

Trend plots show data has changed over time and how fast it is changing. This can help pinpoint when action is required.

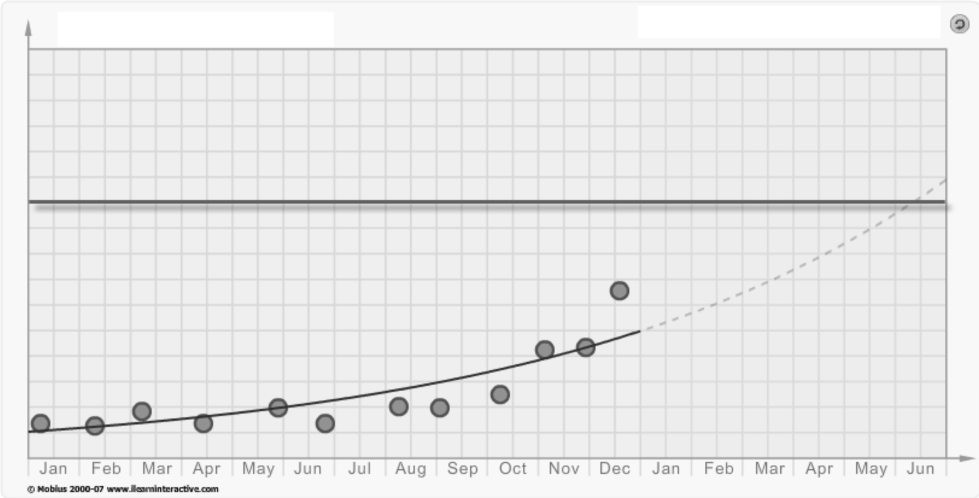


Figure 7-1

Trend plots typically show alarm limits and how the data points are in reference to it.

There are typically two types of trends; trends of measured data and trends of computed values.

Trends of measured data may include overall level readings, process data such as temperature and pressure. It may include high frequency bearing measurements such as Shock Pulse, HFD, PeakVue, etc. Figure 7-2 is a trend of the Overall values.

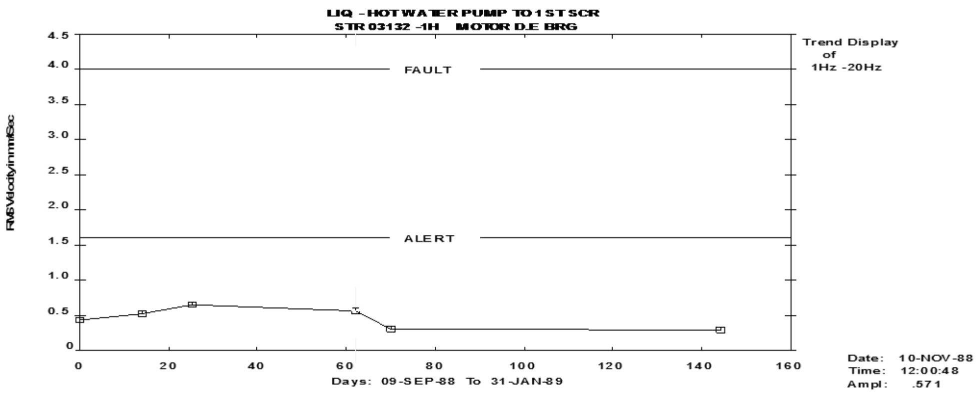


Figure 7-2 - Trend of measure Overall values

Trends of computed data typically come from the spectral bands (or analysis parameter sets) established for the measurement point. The data from the 1x band, 2x band and others can be displayed separately or together to see a complete picture of what is changing in the machine.

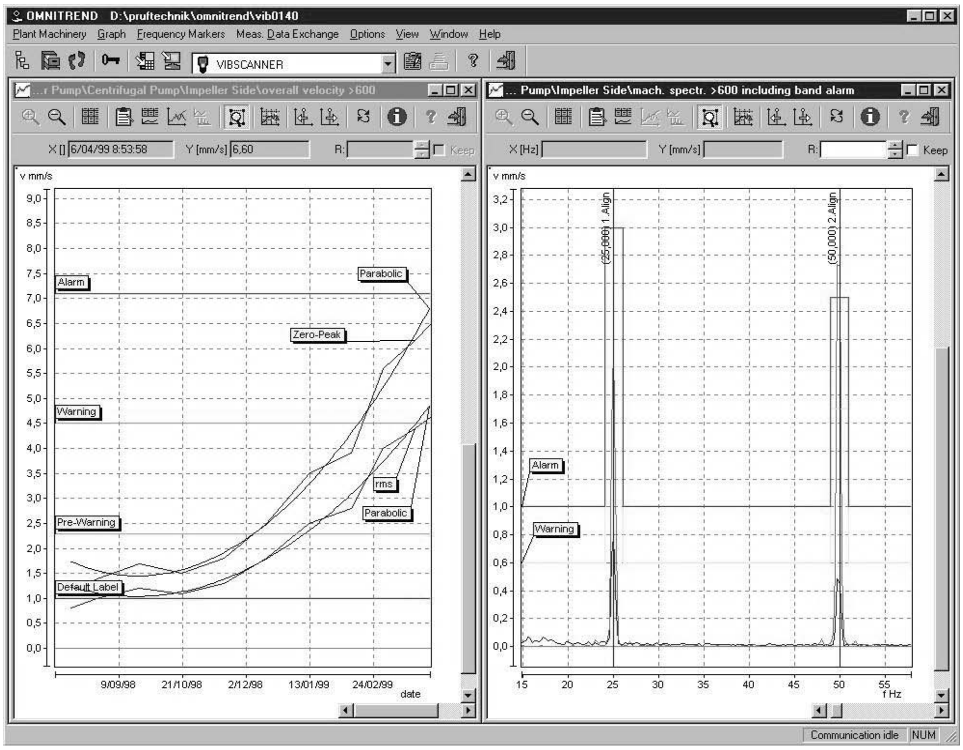


Figure 7-3 - Trends of computed data from spectra

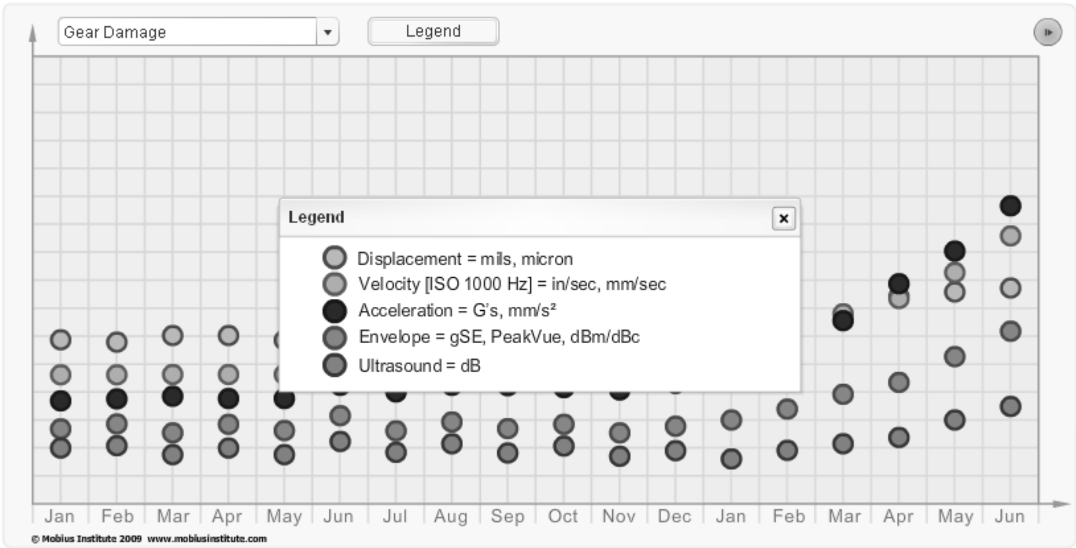


Figure 7-4 - Trending using displacement, velocity, acceleration, high frequency techniques (Shock Pulse, ultrasound, etc.)

INTERPRETING THE TRENDS

The first thing to look for is the change in level or amplitude, and how that level is relative to preset alarm limits.

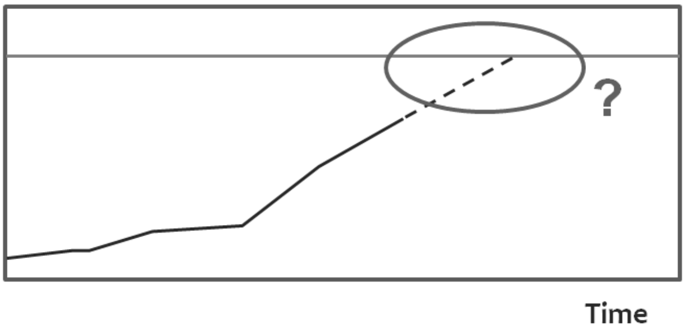


Figure 7-5 - How long until it reaches an alarm?

Figure 7-5 shows requires attention to determine the source of this upward trend.

When a trend is basically flat, then there is a fairly high confidence level that nothing is changing. Don't spend time on further analysis.

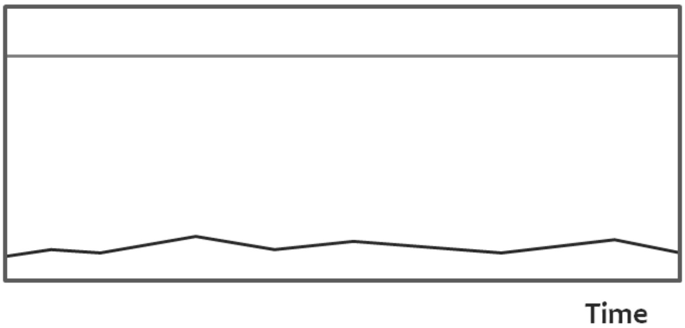


Figure 7-6

However, if a trend shows an increase in level over time, and the levels reach or exceed the alarm limits, then it requires attention.

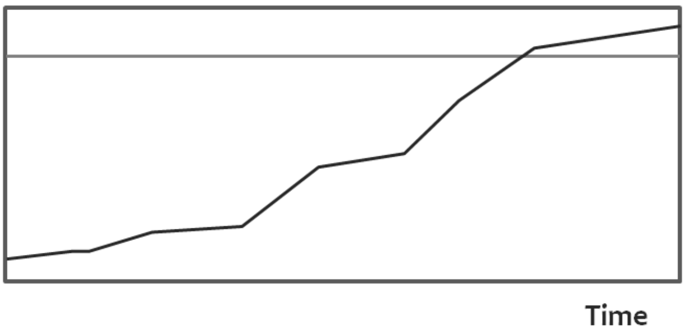


Figure 7-7

If the alarm limits have not been exceeded, look at how quickly the trend is increasing in level and attempt to estimate when the alarm limit will be reached.

TRENDING LIMITATIONS

Remember though that the alarm limits are not perfect. They will only ever provide an indication of severity. It is up to the analyst to first assess how useful these limits may be (were they created statistically on 4 samples or 24 samples? Were they just default limits applied by the software system, or were they set by an experienced analyst, and so on).

Estimating the period of time until the trend will exceed the alarm limit, and generally assessing the rate of change in the levels only serves to indicate how severe the problem is now, and how much time you may have before action is required.

The same is true if the levels have exceeded the alarm limits. After reviewing the accuracy of the alarm limits, you have to assess the overall condition.

NOTE: No single trend that the author is aware of is good enough to be used independently in the decision making process. Trend data only serves to help paint a picture. The rest of the picture is provided by other trend data, data such as time waveforms and spectra, data from other points on the machine, and knowledge of the machine and process. And then production demands, parts availability, and so many other factors come in to the decision making process.

FORECASTING

Some software attempts to predict future values. The data is "curve fit" with "exponential", "parabolic" or other equation. It only provides an estimate of future values assuming the trend continues as it has for the last several weeks.

Figure 7-8 shows the trend plots of the Peak and rms values of the parameter. The additional line on each plot is the **parabolic curve** to extrapolate the change rate into the future.

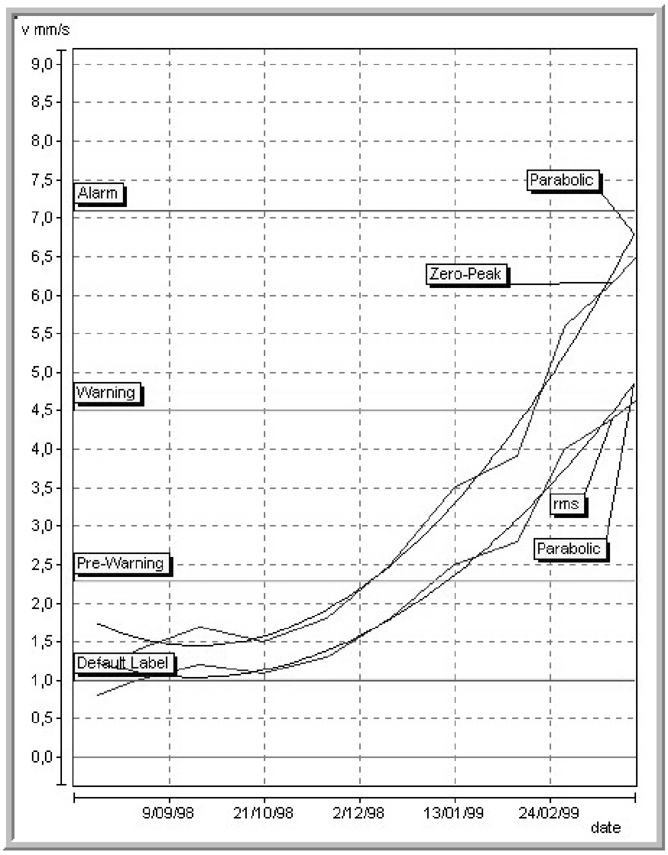


Figure 7-8 - Trend curves and the calculated future curve using a Parabolic curve.

Trending is not the answer, but it does indicate whether a problem is developing, and it can reveal that a problem existed in the past. If you keep good records, you can see how high the vibration/temperature (or whatever is being trended) reached in the past, and you will hopefully remember what the condition was of the machine (how bad were the bearings, how bad was the misalignment, etc.)

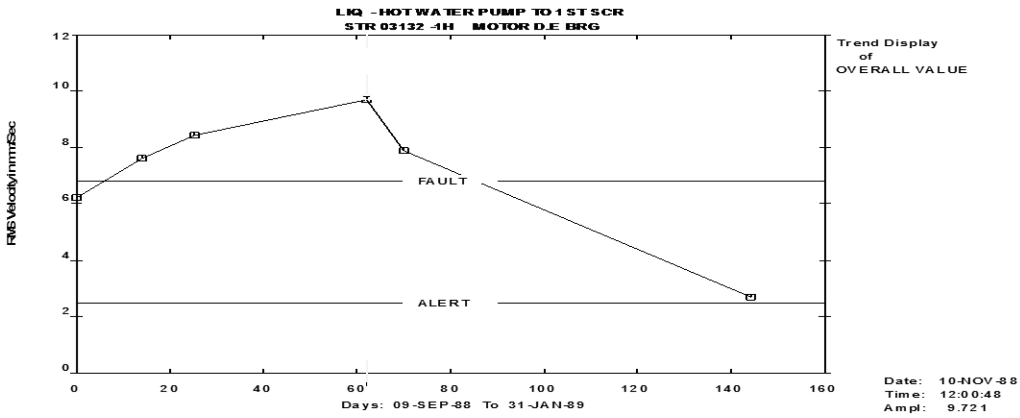


Figure 7-9

Chapter 8

Natural Frequencies and Resonances

Objectives:

- Define Natural Frequency and Resonance
- Describe how to identify resonance in spectra
- Describe resonance and its effect on the vibration
- Describe three special tests for resonance
- Describe three changes that can be made to affect resonance conditions in a plant

Introduction

Every single machine and structure in every plant has “natural frequencies”. If machines and structures are designed correctly, these natural frequencies will not affect the operation or reliability of the machines. In reality, however, a wide variety of fault conditions are either caused by, or are affected by “natural frequencies” – many believe up to 50%.

It really is important to understand what they are, how to detect them, and how to correct them.

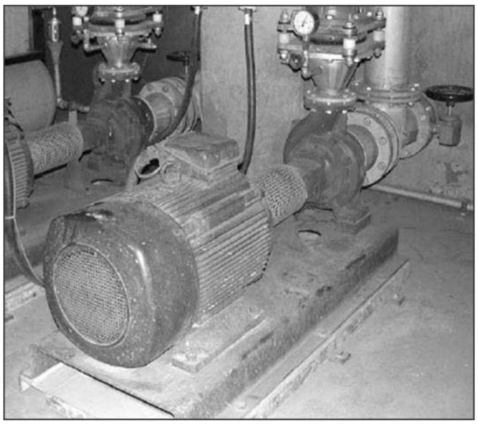


Figure 8-1 - All machines and structures in the plant have Natural Frequencies.

A classic example of a natural frequency is a bell. If a bell it “resonates” at its natural frequency. In actual fact it has many natural frequencies, but one dominant natural frequency is the sound you hear. When the natural frequency is excited, it is called a resonant frequency. The resonance is the act of exciting the natural frequency.

There are two interesting things about the bell. First, under normal conditions it does not resonate. While it is hanging at the top of the church spire it is quiet. The natural frequencies exist, as they do in all structures, but until they are excited they lay dormant.

The second interesting fact about a bell is that if there are two identical bells hanging side by side and one of the bells was struck, the vibration at the natural frequency would excite the second bell, so that it would begin resonating.

Figure 8-2

Plant Resonances

The pipes, foundations, and rotating machinery in every plant have natural frequencies. If designed well, the natural frequencies are not excited (much). However, if a machine happened to be mounted on a structure that had a natural frequency equal to the speed of the motor, the vibration levels would grow considerably – there would be a resonance.

Resonances amplify vibration. The measured vibration levels may be 3 to 50 times higher than they would be normally! So, instead of vibrating at 0.5 mm/sec, for example, the machine could vibrate at up to 25 mm/sec. The potential for structural failure or a catastrophic failure of the machine is high.

Figure 8-3 - All structures and machines in every plant have natural frequencies.

By definition, the natural frequency is: "The frequency of free vibration of a system. The frequency at which an undamped system with a single degree of freedom will oscillate upon momentary displacement from its rest position."

In simple terms, if energy could be injected at all frequencies into a structure, it will vibrate at its natural frequencies. When a natural frequency is excited, the structure resonates, and the vibration amplitudes are amplified. So the stress can be increased up to 100 times (as compared to the stress at a frequency higher or lower than the natural frequency).

Critical Speed

The term critical speed is typically used regarding very large rotors such as large steam turbines. These are flexible rotors. As the rotor approaches its natural frequency it will begin to flex. When the machine RPM coincides with the first mode of vibration, that speed is called the "critical speed."

Figure 8-4 - Large, flexible rotors have Critical Speeds.

It is often defined more simply as the speed that excites a resonance. Vibration increases dramatically near and at the critical speed. As these machines are run up to speed they must run through the critical speeds quickly to avoid catastrophic damage.

The study of critical speeds is complex and beyond the scope of this course. The bottom line is that the machine should not be operated within 20% of a critical speed.

Why are resonances important?

When a natural frequency is excited, the structure resonates. The amplitude of vibration will increase significantly, thus the stresses on the machine increase significantly. The increased stress reduces the life of machine components and structures. Welds crack, metal fatigues, bearings fail, and worse.

Vibration readings will be higher in amplitude in the frequency range affected by resonance, up to 50 times higher. It may lead to a misdiagnosis of the fault.

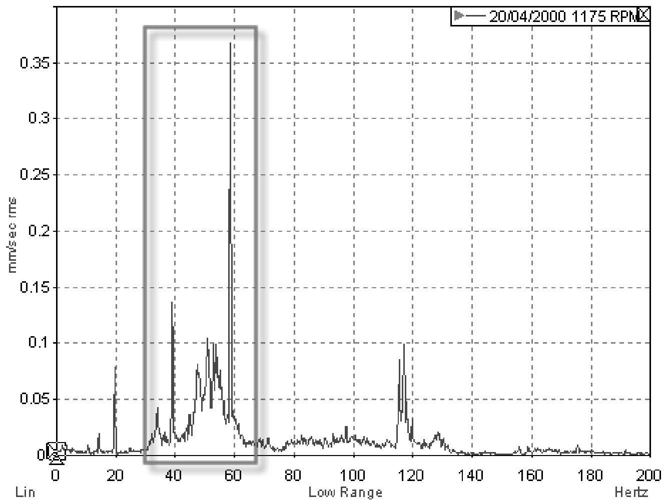


Figure 8-5 - The mound of energy surrounding the peaks in the box is a due to resonance.

In Figure 8-5, the mound of energy surrounding the peaks in the 30 to 70Hz range is due to resonance. Most of that mound is the amplified noise floor.

Natural frequencies

Example: Cantilevered Beam

Imagine a metal bar sticking out of a wall. If there were no machines around, i.e. nothing to excite the natural frequencies, the bar would not vibrate.

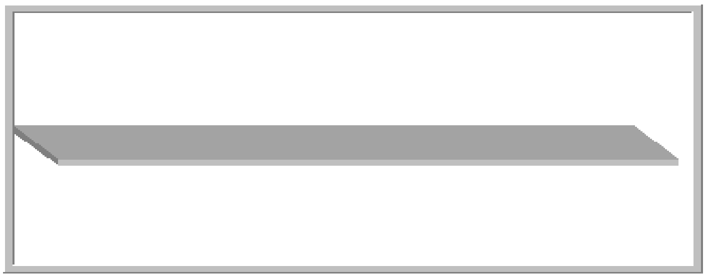


Figure 8-6 - the cantilevered beam will not vibrate unless excited

But let's imagine we had a variable speed motor that could be mounted on the bar. An out-of-balance mass is placed on the rotor, and we run it at a very low frequency – 0.5 Hz for example. (For the sake of the example, we will assume the vibration level generated by the motor would be constant at all speeds, and that significant vibration is only generated at the frequency corresponding to the running speed of the motor.)

Depending upon the design of the bar (its mass, stiffness and damping), we probably will not excite the natural frequency when the motor is running at 0.5 Hz. If we measure the vibration along the bar we will measure the vibration as a result of the motor.

But if the speed of the motor is increased, and we continue to measure the vibration, two things will be noticed:

- The vibration levels will begin to increase (beyond that generated by the motor), and
- The vibration level will be much higher at the free end of the bar.

A certain speed of rotation (excitation frequency) would create the highest level of vibration. That is the point at which the frequency of rotation corresponds with the natural frequency of "the first mode of vibration". The beam is in resonance.

Figure 8-7 - There is a point where the beam vibrates the most. The beam is in resonance at this point.

As speed is increased, the amplitude level of vibration would gradually drop to a point where the level is again fairly uniform along the beam. But as the speed is increased, the excitation frequency would begin to approach the second mode – the vibration would again be amplified – but this time the pattern along the bar would be different.

At one point along the beam, there is no vibration. If you ran your finger along the bar you would feel that at that point, called a node, there is no vibration.

This is the second mode of vibration. There is one node.

Figure 8-8 - Cantilevered bar bending in its second mode.

As the speed of the exciter motor is increased the vibration will reduce again as the exciter frequency no longer coincides with a natural frequency of the beam.

Continuing to increase the speed of the exciter motor moves into the beam's third mode of vibration. This time there will be two nodes where there is essentially no movement. The wire frame in Figure 8-9 shows the extent of the movement. The two lines show the uppermost and lowest movement. The two circles are the nodes which don't move.

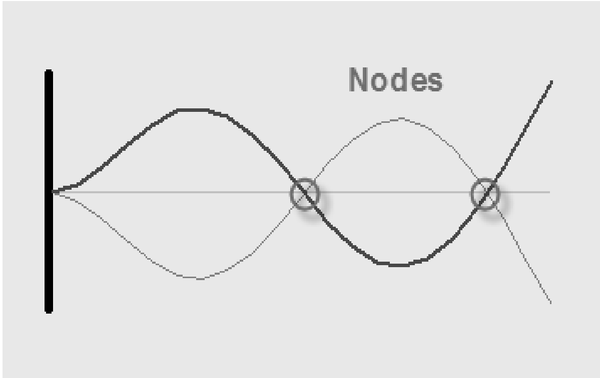


Figure 8-9 - The third mode of vibration has two nodes.

Note that the speed of the exciter motor does not have to match exactly the natural frequency. But it begins to excite the natural frequency as it approaches it. A good rule of thumb is to keep excitation frequencies (such as running speed) at least 20% away from the natural frequency.

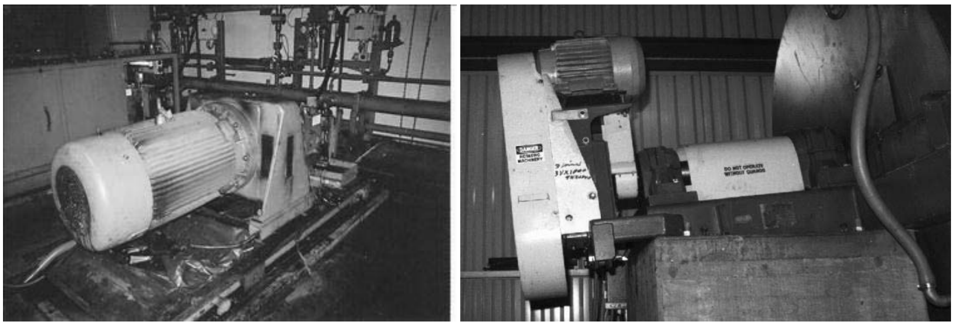


Figure 8-10 - Cantilevered motors in a plant

If this fictional bar was actually a cantilevered motor or support for a motor in a plant, and the motor speed excited one of the natural frequencies of the support, the structure would resonate. This would make the vibration levels on the motor far higher normal.

If the motor /fan mounted on a cantilevered structure is well balanced, precision aligned; if the bearings are in perfect condition, and there are no flow or wear problems, the vibration levels would be fine because the natural frequencies are not excited. It does not resonate.

But if the machine did develop a fault, unbalance, misalignment, etc., and the forcing frequency coincided with a natural frequency, it would now excite the corresponding mode and the machine/structure would resonate and the vibration levels would be amplified.

Describing the Natural Frequency

There is a way we can characterize the natural frequencies of a structure. Later we will discuss some practical ways to gather this information, but for the moment we will consider the scientific definitions.

In this graph, called a Bodé Diagram, the magnitude and phase change with frequency. The example shown here is for a model of our cantilevered beam, and only includes the first three "modes".

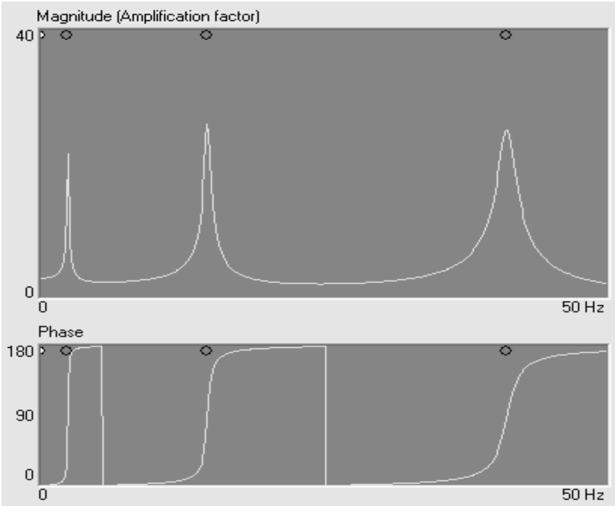


Figure 8-11 - The top graph is the linear amplification of the structure. The bottom graph is the Phase Lag graph.

The top graph is a "frequency response function" (FRF), or more simply, a graph of the linear amplification; the amount that a unit input signal at a given signal will be amplified.

The lower graph is "phase lag"; it indicates how phase will change as a function of frequency.

There are a number of interesting points to note:

1. **There are three modes:** the three "peaks" relate to the three modes.

2. **Phase changes around the modes:** the graph of phase shows that as speed approaches the natural frequency, the phase changes 90°. As it goes past the natural frequency, the phase changes a further 90°. So, by definition, the phase must change 180° for it to be a resonance.

Note: the graph scale is 0 to 180°. Rather than plotting values of 181° for example, 180° is subtracted and plotted as 1° - the graph wraps around. So the vertical lines are a product of this graphing method (which is commonly used), not the actual data.

Notice that the peaks are not sharp peaks. They have some width to them. Therefore, even if the excitation frequency (from the motor speed, blade pass frequency, etc.) does not exactly match the natural frequency, there will be amplification.

The shape of the "peak" relates to "damping". The first mode is "lightly damped". A frequency must be quite close to the natural frequency to excite this mode. The third peak is quite different in shape. It is more heavily damped. It is very wide and a forcing frequency does not have to be very close to excite this mode.

This information can be displayed as a polar plot, called a Nyquist Diagram. It plots Magnitude vs. phase with phase as a vector. A circle indicates the presence of a mode. See Figure 8-12.

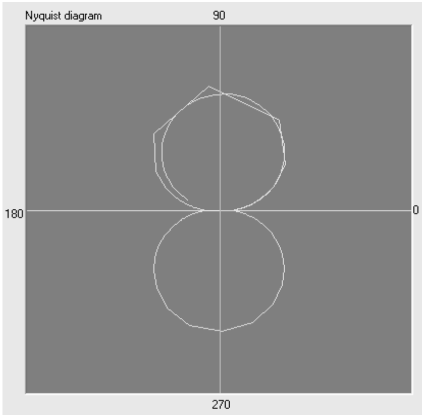


Figure 8-12 - A Polar Plot of Magnitude and Phase. A circle indicates a mode.

In the following charts the shape of the frequency response function is affected by different values of damping. A value of 0.05 is very lightly damped, and a value of 1 is very heavily damped.

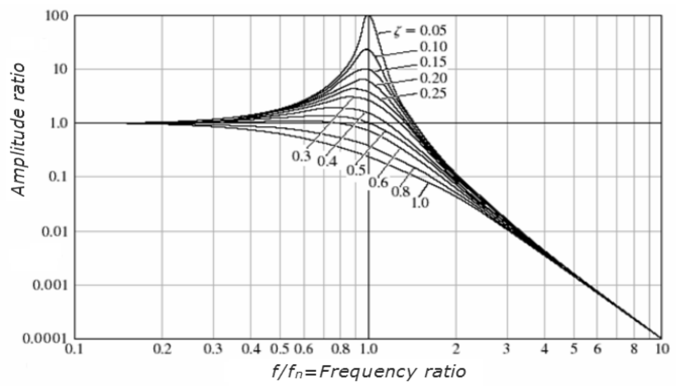


Figure 8-13 - The Amplitude of the response is dependent on damping.

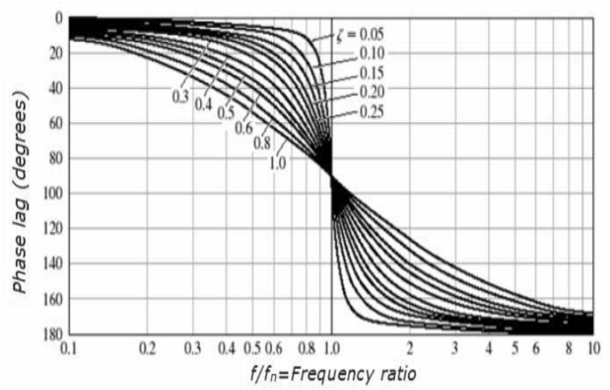


Figure 8-14 - Damping affects Phase Lag

If the damping is increased the amplification is reduced significantly.

Notice how the peaks are now somewhat wider than they were. If there is no damping, the peaks will be narrow and sharp.

The Phase change is also spread over a wider frequency range with the added damping. Notice the difference in the steepness of the phase curves of each of the three modes. The phase of the first peak has the steepest curve indicating it is less damped than the others.

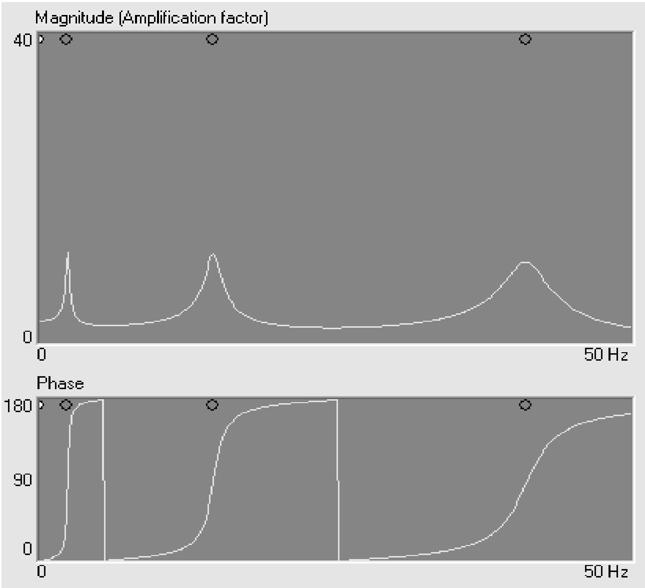


Figure 8-15 - Increasing damping decreases the amplitude and widens the peaks.

The resonance frequency can be moved either higher or lower by modifying two aspects of the structure. Those two aspects are mass and stiffness.

Figure 8-16 shows the effect of adding mass. The resonances are moved to a lower frequency range.

Large, more massive structures have mode shapes with lower natural frequencies.

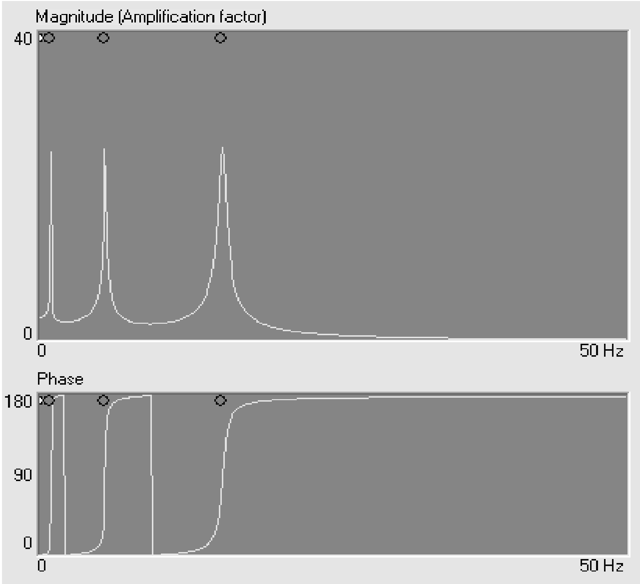


Figure 8-16 - Adding mass moves the resonances to a lower frequency range.

Increasing the stiffness moves the natural frequency to a higher range. In Figure 8-17 notice how the third mode has moved off the graph. So increasing the stiffness of a structure increases the frequency of the modes.

It is similar to tightening a guitar string, as it is tightened the pitch moves to a higher range.

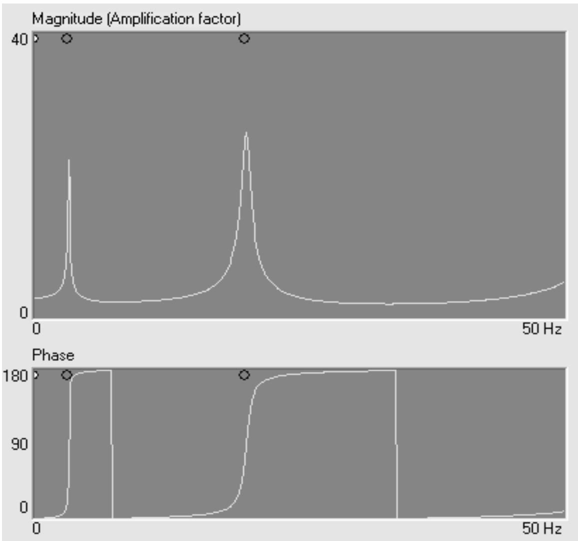


Figure 8-17 - Increasing stiffness increases the natural frequencies. Notice how the third mode has moved off the graph.

This example used one frequency source, a motor, to excite the natural frequency of the structure. In the world of machines there are many sources of vibration at many frequencies. If a machine generates a signal that corresponds with a natural frequency, then the mode will be excited, and the vibration will be amplified.

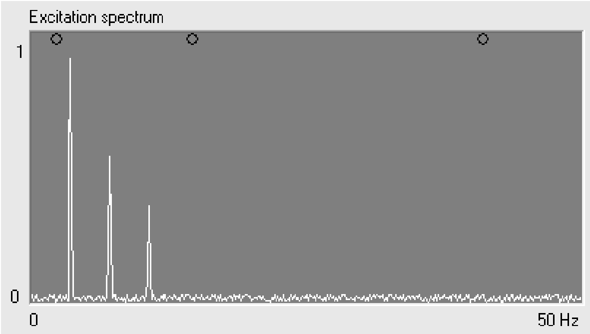


Figure 8-18 - Spectrum with some peaks and a lot of noise

If the same cantilevered beam is exposed to the frequencies from a source that produces the spectrum in Figure 8-18 , any vibration that coincides with the natural frequencies will be amplified. There is a lot of noise in the spectrum.

Figure 8-19 shows the result of adding the spectral frequencies to the system. None of the spectral peaks matched any of the modes, but the noise floor is raised substantially in all three modes.

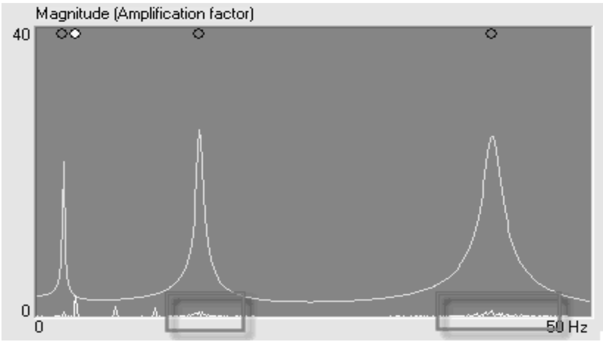


Figure 8-19 - The peaks of the spectrum did not match any modes of vibration, but the noise is amplified.

But if the 2X peak is close to the second mode it will be amplified and the structure will resonate. See Figure 8-20.

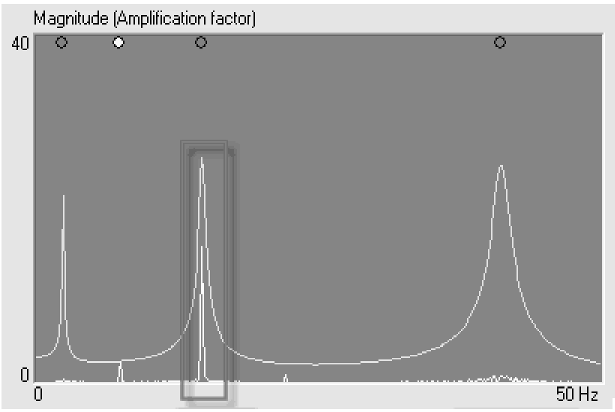


Figure 8-20 - The 2x peak now is same frequency as the second mode and its amplitude is increased.

Detecting Resonance Problems

Without doing any special tests, there are two basic ways to tell if you have resonance problems: unusual failures, and tell-tale signs in the spectrum.

Unusual failures:

When there are machinery failures or structural failures that seem to be as a result of fatiguing and there is not any other explanation, then consider resonance as a possible cause. Structures should last a very long time, and fatigue failures should only occur after MANY years of service.

Typical failures as a result of resonance include:

- Broken welds
- Cracked and leaking pipes
- Premature machine failure
- Broken or cracked shafts
- Foundation cracks

Four Tell-tale signs in the spectrum:

1. Unusually high peaks in the spectrum – peaks are amplified

2. High vibration levels in one direction / axis but not in another

3. Areas in the spectrum where the noise floor and any peaks in the vicinity seem to have been raised. These are referred to as "haystacks" and "humps"

4. Peaks that change amplitude when machine speed changes. Resonance is only excited under certain conditions.

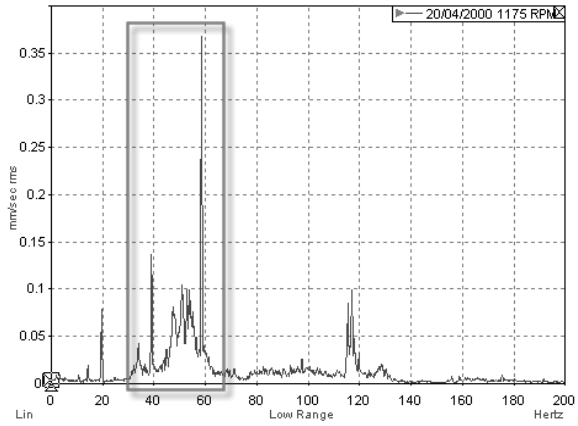


Figure 8-21 - The modes amplify the frequencies in its amplification range. The region from 30 - 70 Hz is typical of Haystacks or Humps. The 3x peak is amplified.

Because resonances amplify vibration, peaks in the spectrum that may normally be quite low (at the blade pass frequency, or gear mesh frequency, etc.) can increase dramatically. When there is a high peak, remember to consider the possibility that it is being amplified by a resonance.

On variable speed machines, you may observe that when the machine is operating at its highest speed, the blade pass frequency may coincide with a natural frequency, so the machine resonates, and the BP frequency peaks will increase in amplitude. However, when it is operating at a lower speed, the BP frequency would be below the natural frequency, so the amplitude would drop to “normal” levels, and the machine will not be in resonance.

Also, when studying the structure more closely, notice that the mass, stiffness and damping can be very different vertically, horizontally and axially. So, there may be a resonance in the horizontal direction (i.e. it is vibrating from side-to-side), but not so vertically (there will be natural frequencies, but they may not be excited). In the case of vertical pumps, however, resonances can quite often exist radially, but not at the same frequency vertically (axially along the machine).

Figure 8-22 - Vertical pumps may have resonances in the radial direction

Testing for resonance

If a machine is suspected of resonating, but a better understanding is needed before attempting correction, then there are a few tests that can be performed to show where the natural frequencies are located and provide an indication of how the machine/structure is moving.

The best way to understand resonances is to excite them. As discussed earlier, multiple natural frequencies exist in all structures. When they are excited is it is simpler to find where they are. But how is that done?

Also, remember that machines do move in three axes. Unless it is already suspected that the problem is predominantly in one direction (for example the motor/pump is rocking from side to side), then these tests should be performed while collecting data in two or three axes.

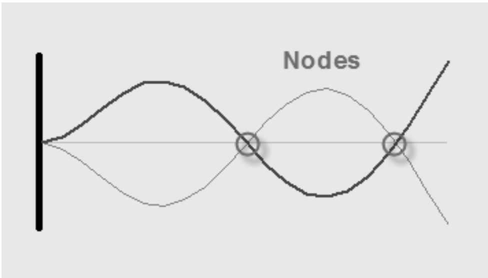


Figure 8-23 - There will be no vibration at the nodal points

Also, remember that "nodes" were discussed earlier. When performing these tests, be careful not to measure vibration at a nodal point, as there will be no vibration to measure. One can often tell where the nodes are located by along the structure. The same can be done with an accelerometer while watching the collector in real-time mode.

METHOD ONE – CHANGE THE RUNNING SPEED

If it is possible to change the speed of a machine, set it at its highest speed and slowly reduce the speed while watching the vibration levels in the spectrum. If possible, use the peak-hold averaging function with many averages so that only maximum

In Figure 8-24 we were able to sweep across a broad range of speeds. The results are very useful.

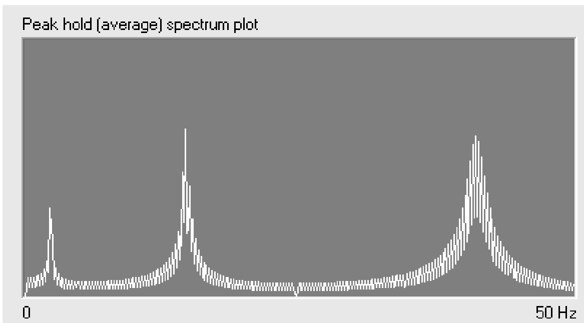


Figure 8-24 - A Peak-hold spectrum as a machine is adjusted through the speed ranges

In Figure 8-25, the speed could not be changed over such a wide frequency range. While it reveals the presence of a natural frequency, more information would be useful.

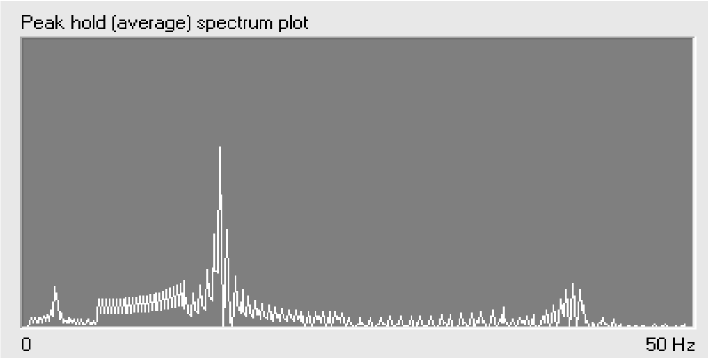


Figure 8-25 - The speed could not be changed over such a wide frequency range, more information would be better.

METHOD TWO – THE BUMP TEST

Striking a machine interjects energy at all frequencies. Soft mallets inject more energy at lower frequencies. A metal bar injects more energy at higher frequencies. The size of the striking object should be adjusted for the size of machine or structure. A piece of hardwood timber works well, but care should be taken to avoid damaging the machine or structure.

How and Why it works - When a machine is struck with a large piece of hardwood timber, energy is injected into the structure at all frequencies. The spectrum of a single impulse does not have any peaks, but instead, the floor across the entire frequency range is raised. That means that any natural frequencies will be excited and their amplitude will be amplified, producing humps or peaks in the spectrum (depending on the amount of damping.)

A bump test can be done while the machine is running although best results can normally be obtained while the machine is off. Typically the vibration resulting from the strike is very short lived and must be captured. Here are some recommended settings.

- Set the analyzer to Peak-hold averaging mode
- Low resolution data is ok (400 lines or less)
- Select a higher frequency range – (1000 Hz)
- Set it for 20 averages
- If possible, set an appropriate gain setting and turn off the auto-ranging feature. If the auto ranging is left on, additional strikes will have to be made to allow the analyzer to adjust the gain.
- Some analyzers have a Pre-trigger feature. Setting this to 30 or 40 % moves the impulse closer to the center of the time window and therefore it is not affected by the windowing.

Start data collection and immediately strike the machine. Strike the machine several times, making sure to strike only once in an average. The machine does not have to be struck during every average. If auto-ranging is on, the gain may readjust if too much delay occurs between strikes.

When the pre-trigger option is used, the minimum pulse amplitude can be entered. The analyzer waits until it sees at least that amount of energy before collecting an average. In this case usually only 4 averages are sufficient

Special hammers are made for doing bump tests. A dead-blow hammer with lead shot inside should not be used because the lead shot produces additional frequencies in the results. Some hammers, called force hammers, are instrumented with an accelerometer built in to them. They can be used to measure how much energy is imparted into the machine and correlate that to the results. Specialty hammers for Bump Test range in size from a few ounces to a 10 lb sledge hammer for larger structures.

The Bump Test Results will have raised areas corresponding to resonances. If the machine was running, the normal spectral peaks will still be present. The bump test on the cantilevered beam produced the results shown in Figure 8-26.

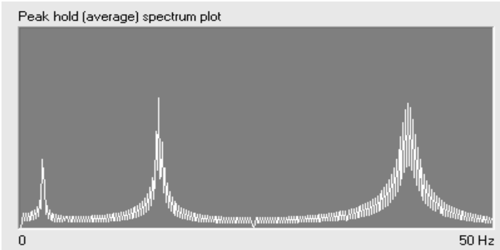


Figure 8-26 - Bump Test results on the cantilevered beam.

METHOD THREE – THE RUN-UP AND COAST-DOWN TEST

These two tests are very similar. The purpose is to catch many measurements during the time the machine is started and run up to speed, or while the machine is coasting down after being shut off. The running speed or other Forcing Frequency in the machine will excite resonances and the data will be recorded in the form of a Peak Hold spectrum or multiple spectra (displayed in a waterfall plot.)

If the speed of the machine can be controlled during these tests, then it will be more successful. If the machine starts up too quickly or slows too quickly, the resonance may not be excited as it passes through that frequency range or the analyzer may miss the event.

The setup for the analyzer is similar to the Bump Test. Set it up for quick measurements opting for lower resolution. Peak Hold can be used but do not use pre-triggering. It is even better to take a series of spectra quickly so they can be displayed in waterfall format which gives a 3D effect.

A typical result is shown in Figure 8-27 which was done on the cantilevered beam. There are three angled lines starting in the lower left corner and extending to the top right resembling a mountain range or a ridgeline. These are the three peaks in the spectrum progressing in frequency as the speed changes. As the 1x, 2x, and 3x pass through the natural frequencies they are amplified, producing the mountain peaks.

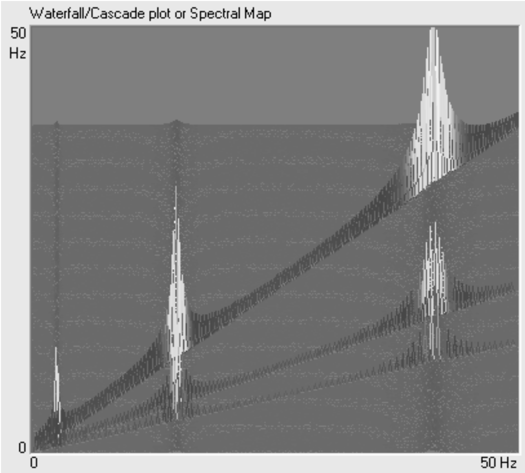


Figure 8-27 - Waterfall plot of coast-down of cantilevered beam.

Figure 8-28 highlights 3 regions in the waterfall plot that the Natural Frequencies, and therefore resonances occur. These are the same regions revealed in Bump Tests that indicate the first three modes of vibration.

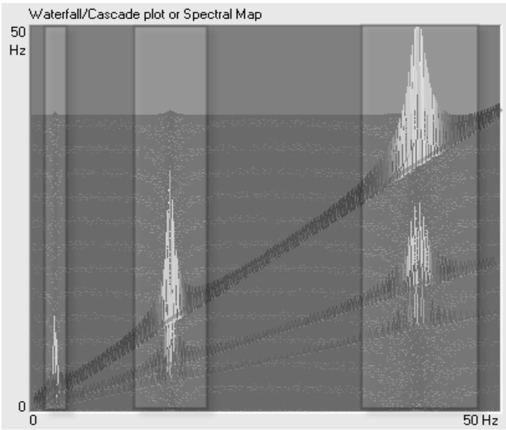


Figure 8-28 - The highlighted regions are the 3 frequency ranges of the first three modes of vibration.

Another display of the same information is the Campbell Diagram which is used by some advanced software packages. The circle radius is proportional to the amplitude. The color intensity scale is sometimes used without the circles.

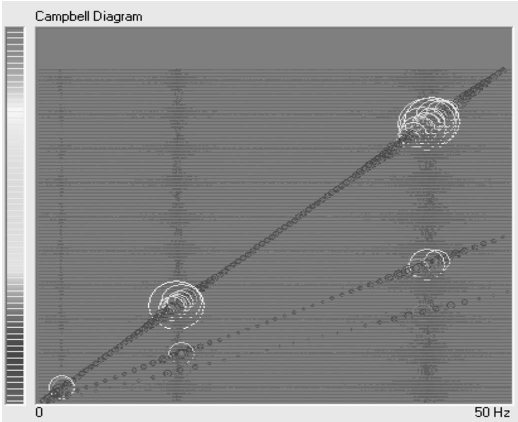


Figure 8-29 - Campbell diagram shows amplitude with radiated circles

METHOD FOUR – ORDER TRACKING

Some analyzers support Order Tracking which tracks amplitude and phase during a run-up or coast-down. It uses a tach signal to give a once per rev signal so the analyzer knows the exact shaft turning speed. It extracts the amplitude and phase data as the speed is changed.

The benefit of this is that the data can be displayed in a Bode Diagram. The amplitude data will resemble the outline of the 1x peak in Figure 8-30. It is only tracking the 1x peak although some sophisticated software has the option of selecting a multiple of the tach signal.

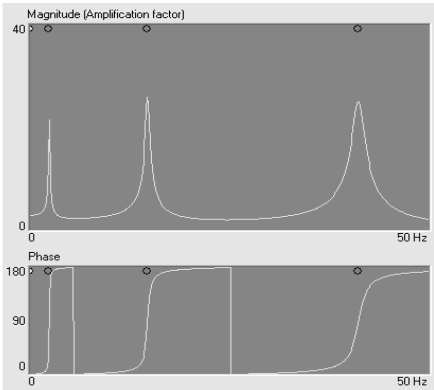


Figure 8-30 - Bode plot of coast-down of 1x using Order Tracking.

EXTERNAL EXCITATION

There are at least two ways to excite a structure that does not have the ability to run-up or coast-down, such as a framework, building, foundation, etc. The first way is to use a speed controlled motor with an unbalance weight added to it. The motor should be clamped, bolted, or strapped to the structure securely. When the motor speed is increased the unbalance weight imparts a vibration at 1x of the motor. A vibration sensor can be placed on the structure and the analyzer used as in a normal run-up or coast-down. Peak Hold averaging can be used or collect a waterfall plot. Typically a 1/20th to ½ hp motor is sufficient to excite most machines and structures.

"Shakers" are commercially available which perform the same function in a much more controlled and precise way. It is possible to slowly change the frequency of excitation and set a magnitude of vibration that suits the application.

Figure 8-31 - Shaker to be used to excite natural frequencies

METHOD FIVE - ODS AND MODAL ANALYSIS

In all of the tests thus far, the natural frequencies have been excited so that they resonate, and then a data collector was used to view a spectrum. The spectrum shows the tell-tale "haystacks" which indicate the presence of the resonances.

But to correctly diagnose a resonance, and to fully understand the motion of the machine (the mode shape(s)), it is ideal to be able to visualize the whole machine/structure vibrating. This information not only helps to confirm the presence of a resonance, but it enables us to plan the modifications required to change the structure so that the resonances are not excited.

For example, in the cantilevered bar example, when the bar is measured at the end there is high vibration at three different frequencies – but what are the mode shapes? In actual fact, the three modes look as follows.

Figure 8-32 - The three mode shapes of the cantilevered bar

As the name suggests, ODS (Operating Deflection Shape) analysis does not require the machine to be stopped. One reference accelerometer is placed at a fixed position on the machine, and another accelerometer is moved from position to position while cross-channel measurements are taken. Amplitude and phase can be computed at all frequencies.

The data is extracted via special software and then plotted to animate the modes. The phase and magnitude readings are then all relative to the reference point, i.e. it shows how each test point is moving relative to that reference.

ODS EXAMPLE

The following example is from the *Emerson Process Management/Asset Optimization Division* paper titled “Operational Deflection Shape and Modal Analysis Testing To Solve Resonance Problems” by Tony DeMatteo.

- The pumps are Morris two-vane, vertical, centrifugal pumps.
    - New 800HP, US induction motors, on a variable frequency drive
    - 700-890 RPM
- The pump is bolted to the floor and the motor is supported on top of a tube covering a 15 foot long drive shaft
- Modal, ODS and FEA analysis was performed

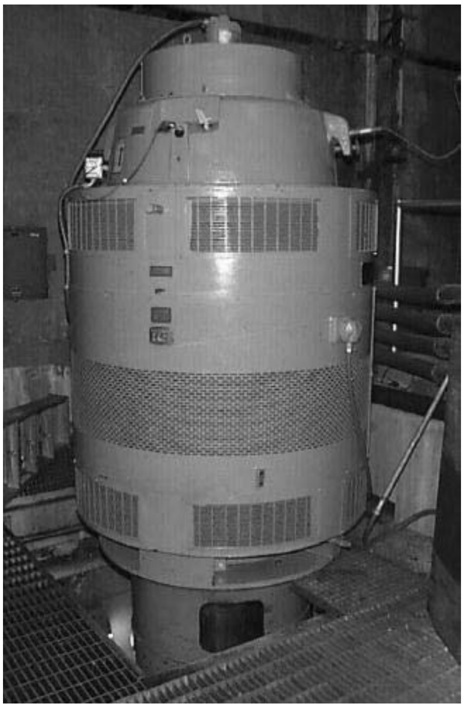


Figure 8-33 - 33 800 HP vertical pump used for ODS

Pumps 1-4 are sewage pumps. The pumps are Morris two-vane, vertical, centrifugal pumps. The drive motors on pumps 1 & 2 are new, 800 HP, U.S. induction Motors, on a variable frequency drive. The motors on pumps 3 & 4 are older Reliance motors with a liquid rheostat speed control. The operating speed range of the pumps is 700-890 rpm. The pump is bolted to the floor and the motor is supported on top of a tube covering a 15 foot long drive shaft.

Figure 8-34 - Pump is mounted to floor and connected to motor via 15 ft. long drive shaft. Motor sits on top of the tube covering the shaft.

A model was created to describe the motor, base plate, tube, pumps, and legs. Measurements are collected wherever two lines intersect. See Figure 8-35.

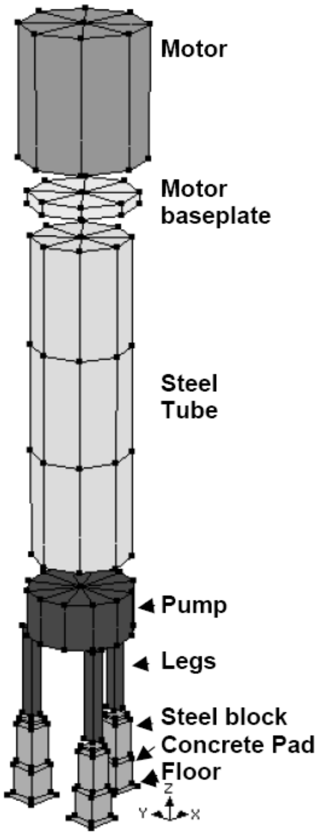


Figure 8-35 - model of pump

ODS RESULT….

A lot of data was required but it is clear how the structure is moving at this natural frequency. It was swaying side to side at a frequency of 30 Hz.

From this information three things can be confirmed:

- A resonance does exist
- It is at 30 Hz.
- Where stiffness should be added to increase the natural frequency to a point that it does not coincide with any forcing frequencies.

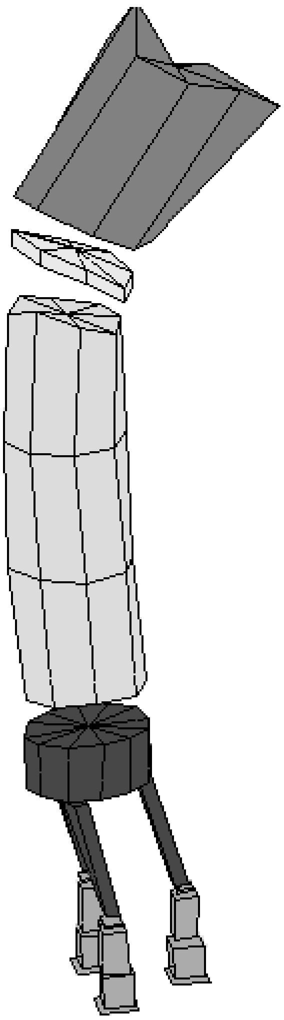


Figure 8-36 - ODS of the vertical pump

MODAL ANALYSIS

The goals of Modal Analysis are the same as ODS. It involves stopping the machine, using a two channel data collector (or spectrum analyzer), using a specially instrumented impact hammer (with a built-in force transducer) and striking the machine while measuring the response with an accelerometer at each of the test points.

By computing frequency response functions (comparing what went in to the structure to its measured response), the magnitude and phase values at any frequency can be extracted.

The hammer used to impact the machine has an accelerometer built in to it. The force of the strike is measured through one channel of the analyzer.

Another channel of the analyzer is used to measure the response of the machine to the impact. Phase and amplitude can be measured at any frequency. Special software extracts amplitude and phase at the natural frequencies. Then the structure can be animated.

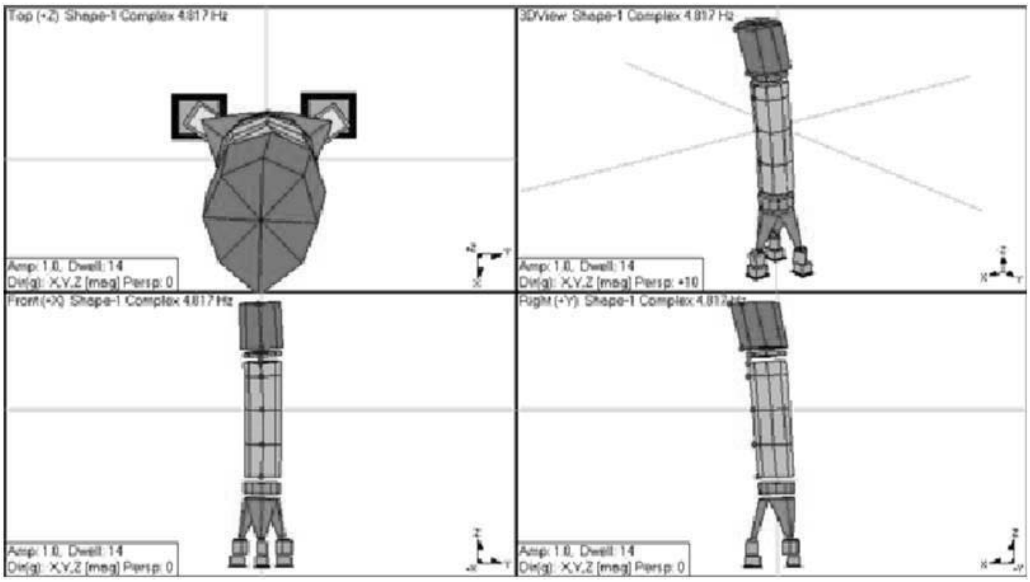


Figure 8-37 - software enables drawing and animating the machine.

Once the amplitude and phase data is captured, it is possible to construct a picture of the machine and structural movement. The animation makes it very clear how the machine is moving and where the structure needs stiffening.

METHOD SIX - FINITE ELEMENT ANALYSIS

Finite Element Analysis goes one step further than modal analysis. It models the structure mathematically. It models the materials, the studs, braces, welds and so forth, computing their mode shapes.

FEA models the effects on the machine for a particular size brace when mounted in one place or another. It shows where the natural frequencies move to before any modifications are actually put into place. This helps the right decisions to be made before attempting modifications.

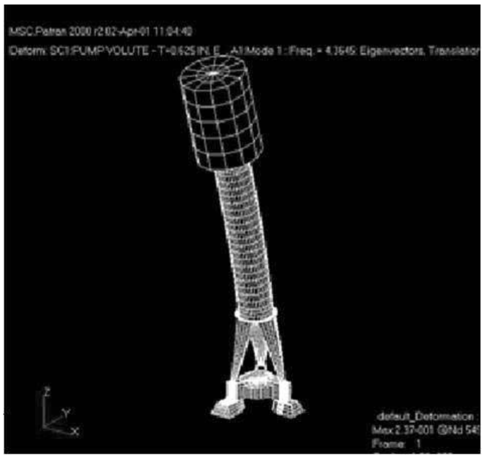


Figure 8-38 - Finite Element Analysis - FEA

METHOD SEVEN - SIMPLE GRAPHING METHOD

There is no doubt that modal analysis and ODS in combination with powerful software will provide the best picture of where the natural frequencies lie, and the shape of the modes. But there is a simpler way.

Creating a basic chart of your machine/structure on a piece of graph paper, and then take vibration readings at each point and plot them on the chart, provides a quick indication of the shape of the resonant mode.

For example, if the foundation of a motor-pump set (a fabricated base) is 2 meters long, then ten measurement points, 20 cm apart could be defined. Take vibration readings at each point, and place a point on the graph in proportion to the measured amplitude. If the maximum expected vibration level was 10 mm/sec, the maximum graph scale would be 10 mm/s. (Displacement would be a better unit, but velocity will still work.) Figure 8-39 is a simple example:

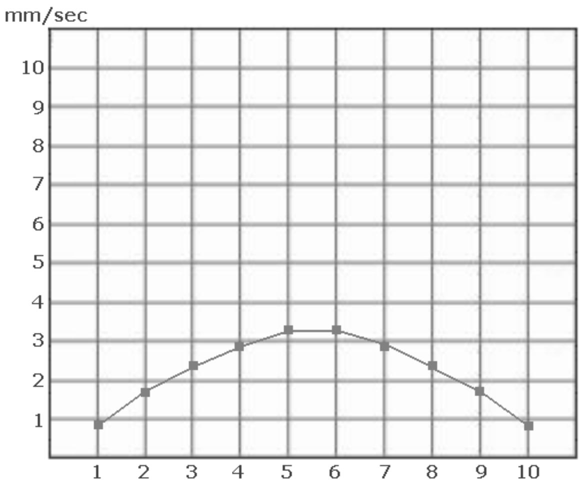


Figure 8-39 - A simple plot of vibration amplitude along a machine foundation.

This example shows that the levels are highest in the middle of the baseplate. If we assume that all of the readings are in phase, we can hypothesize that the motor-pump in the middle of the skid is basically bouncing up and down.

Add phase to the picture:

A step further is to collect phase data to know for sure how it is vibrating. But there must be a phase reference. Given that the machine is running, depending upon your data collector's capabilities, either place a reference accelerometer on a point on the machine and perform a cross-channel measurement (as described for the ODS test), or use a tachometer on the shaft of the motor or pump. Given that the vibration being measured on the foundation (or machine) is generated by the motor/pump, this is a valid reference to use.

Use phase alone:

As a third suggestion, use only the reference accelerometer or tachometer signal to collect phase readings around the machine, just to see what points are in phase and out-of-phase. This gives a very good indication of whether the machine is bouncing up and down, swaying from side to side, rocking up and down, twisting around the center, or some other simple shape.

Correcting Resonance Problems

Once the situation is analyzed and you can see how the machine is moving, it is time to consider altering the structure so that the forcing frequencies of the machine or structure no longer coincide with the natural frequencies, or so that the resonances are so heavily damped that they do not amplify the vibration to such an extent.

A temporary solution is to change the speed of the machine so that it no longer excites the resonance. For example, if the vane pass frequency of a pump matched a natural frequency, then reduce the speed of the machine by 15% so that the mode was no longer excited. Of course, in many situations this will not be possible.

Typically, however, the aim is to change the stiffness so that the natural frequency is increased (or decreased) to a point where it is a minimum of 15% away from the forcing frequency. For example, if the vane pass frequency is 8880 CPM (148 Hz) and it is exciting a natural frequency, then attempt to increase the stiffness of the machine so that the natural frequency is a minimum of 148 * 1.15 or 171 Hz. 15% is a minimum rule of thumb – some people recommend 20%.

In brief, the typical approach is to stiffen structures with additional braces and support structure. But how much is required?

There are two basic approaches: if you have time and expertise, create a "finite element model" of the machine and structure to model the effect of adding braces and/or stiffening. Or apply some practical ideas - try to assess why the resonance exists, and make structural changes that you believe will correct the problem.

It is beyond this paper to discuss finite element analysis (FEA), except to say that it is possible to create a mathematical approximation of the structure which enables you to study its various mode shapes. Then model the effect of adding braces, isolators, etc. until the desired effect is achieved. This is a very challenging area.

EXAMPLES OF CHANGES MADE TO STRUCTURES

The next few pages give illustrations of resonance corrections. This material is based on a paper given Juergen Twrdek at VANZ 2000. The modifications were made at the Wieland Werke AG plants in Ulm and Vöhringen.

These are practical solutions to common problems that may generate ideas for solutions at your plant.

EXAMPLE 1 – SOLUTIONS FOR SHEET STEEL BASES FOR PUMPS

In this common example we have a simple bent sheet steel base for pumps. It is common for these foundations to have natural frequencies in the 10 to 35 Hz range – right around the running speed.

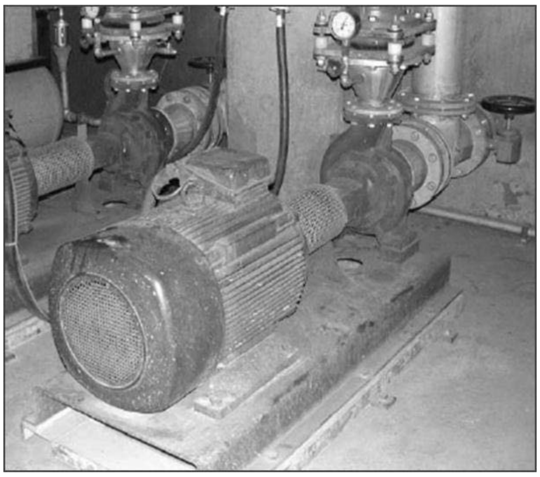


Figure 8-40 - Bent steel bases for pumps can have problem resonances.

One solution is to fill the bases with concrete and put them on separate foundations.

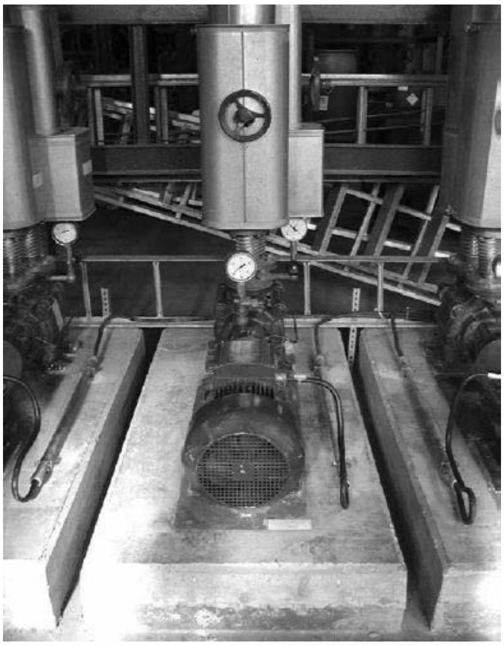


Figure 8-41 - One solution - isolation + concrete

EXAMPLE 2 – SOLUTION FOR OVERHUNG PUMPS

Overhung pumps without adequate support can have a rocking resonance, with high vibration vertically and possibly axially. The solution is quite simple – add a cradle support under the flange. This greatly stiffens the machine vertically, thus the resonance was no longer a problem.

The support does not have to be welded in place or bolted to the floor; it can be created so that the height can be set to match the application perfectly.

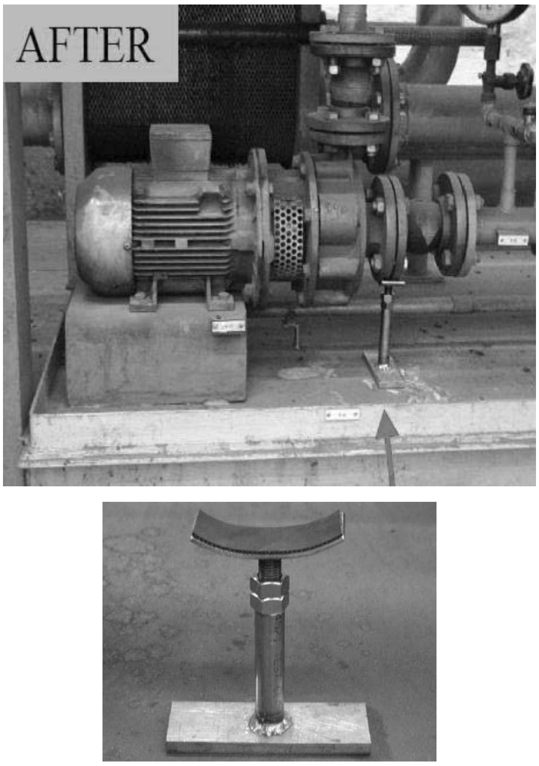


Figure 8-42 - Install a prop under the unsupported pump

EXAMPLE 3 – SOLUTION FOR HORIZONTAL FLEXIBILITY

A great many machines have high vibration in the horizontal axis due to flexibility in that direction. By increasing the stiffness we reduce vibration levels.

This metal base had a resonance in the horizontal direction. After welding the braces and the piece inside, there was no longer a problem.

Four triangular pieces of steel have been added to the outside of the baseplate, and one inside. The stiffness was increased dramatically, and the vibration levels were reduced.

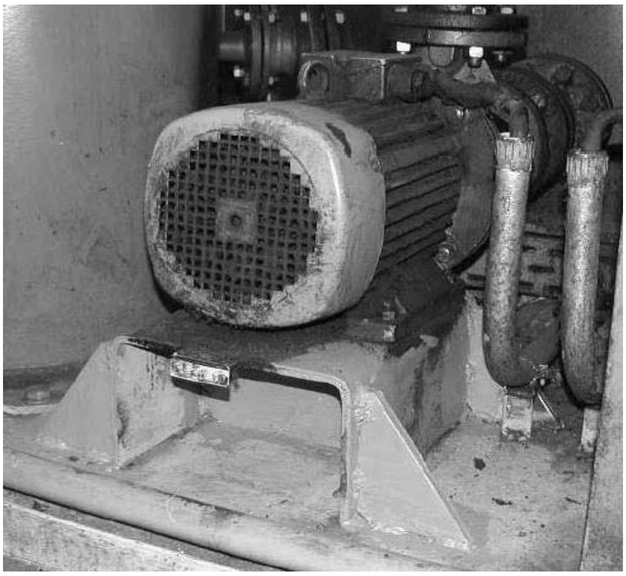


Figure 8-43 - Metal base stiffened by adding braces

EXAMPLE 4 – SOLUTION FOR BASE RESONANCE – OVERHUNG PUMP

As a general comment, there are many overhung pumps with inadequate stiffness on the coupling side and little to no support on the coupling side. They are mounted on foundations with natural frequencies in their operating speed range.

In this example, the machine speed is just below 50 Hz. It was believed that the foundations had a resonance in the operating range of the machine. Bump test data showed there was a resonance at approximately 100 Hz or at 2x (it moved to 159 Hz after modifications). The machine needed to be stiffened to increase the natural frequency away from any of the machine forcing frequencies.

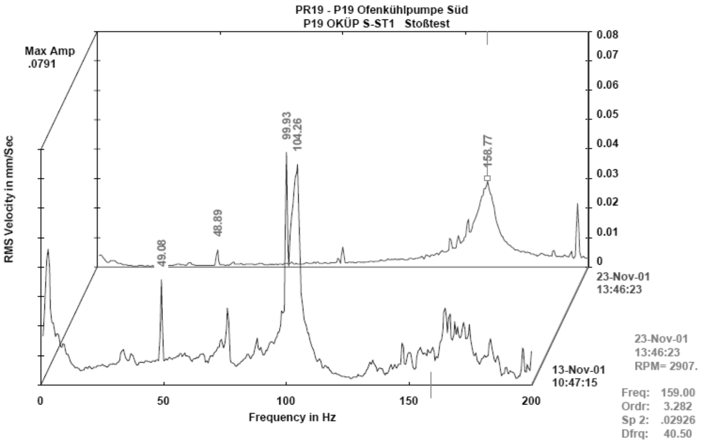


Figure 8-44 - Bump tests on the base plate indicate a resonance at 100 Hz which coincides with 2x and 2 times line frequency.

Three modifications were tried:

1. Adding braces to the foundation

2. Adding support to the motor

3. Adding support to the pump

The change that had the greatest effect was adding support to the pump.

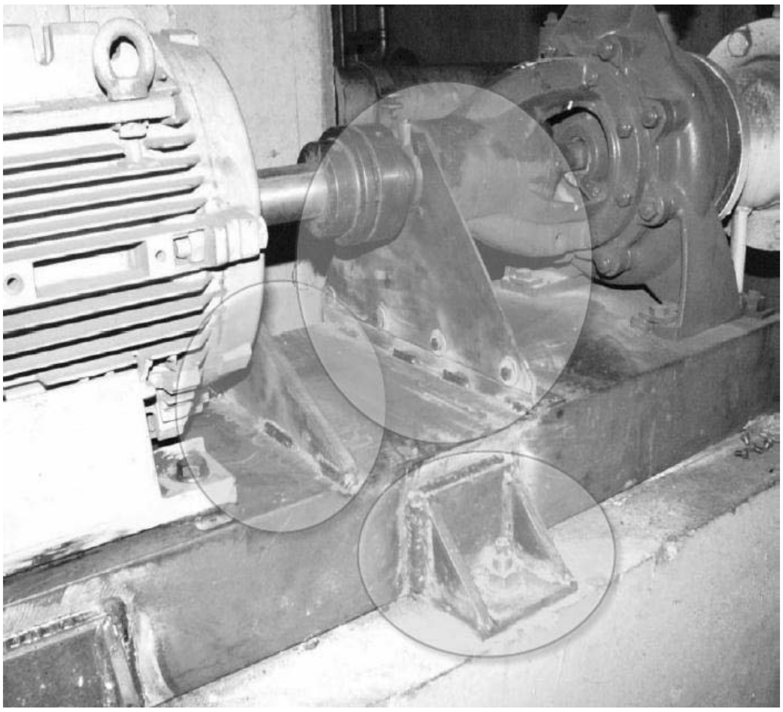


Figure 8-45 - Adding support to the pump had the greatest effect.

The comparison spectra before and after the modifications show a 6:1 improvement. The peak at 100 Hz prior to the changes was 5.829 mm/sec. After changes, the peak is less than 1 mm/sec.

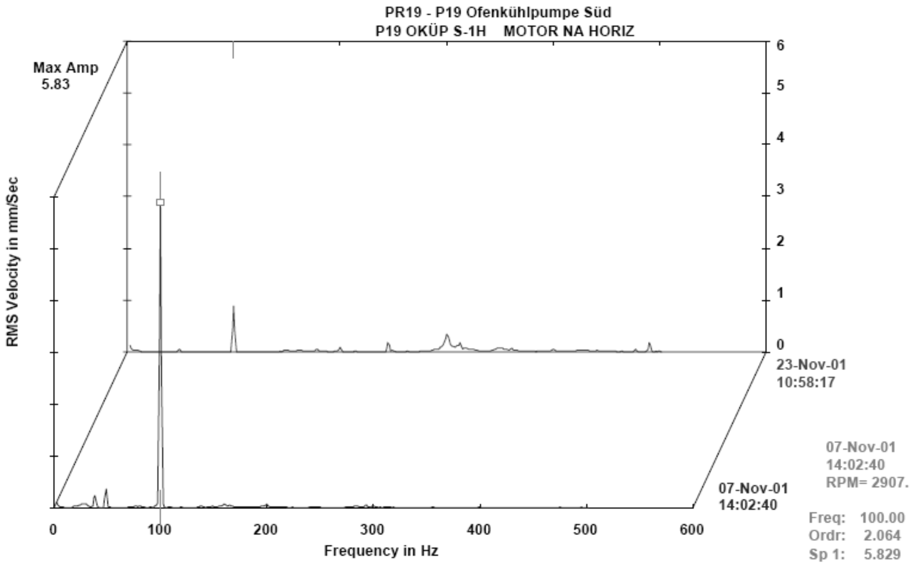


Figure 8-46 - The peak at 100 Hz reduced to approximately 1/6 of Pre-change readings.

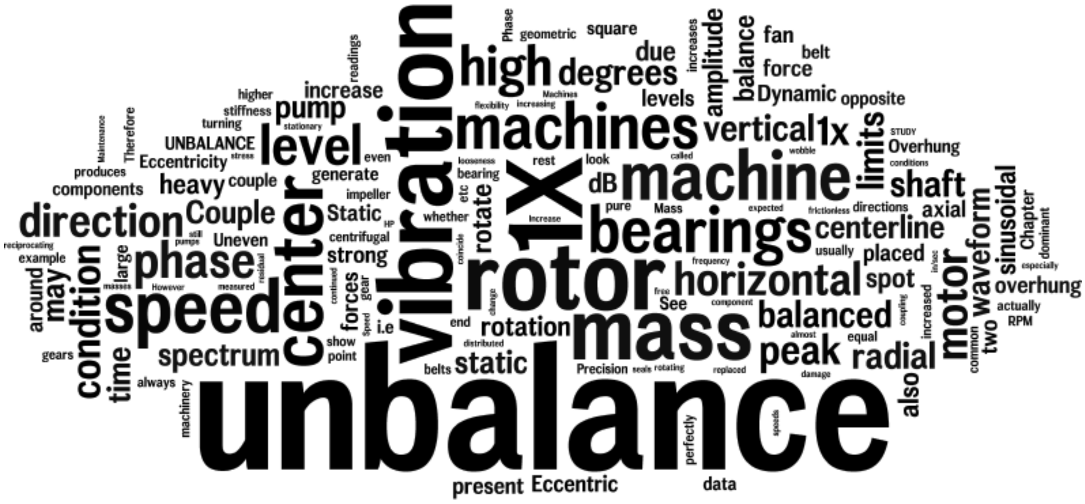
IN CONCLUSION:

A few ordinary tests can remove some of the mystery of problem machines by providing insightful information regarding Natural Frequencies and Resonances.

Peak and Phase measurements can be used in software programs to model the mode shapes. Simple diagram can provide the information necessary in many instances.

Utilizing the principles of Mass, Stiffness, and Damping, changes can be made to the machine / structure to move resonant frequencies so they are no longer troublesome.

Some changes can be fairly simple. The easiest changes usually increase the stiffness.



Chapter 9
Diagnosing Unbalance

Objectives:

- Diagnose Static Unbalance in waveform and spectra
- Diagnose Couple Unbalance in waveform and spectra
- Describe the phase conditions for Static, Couple and Dynamic Unbalance

Unbalance

There are two types of unbalance:

- Static or Force Unbalance
- Couple Unbalance

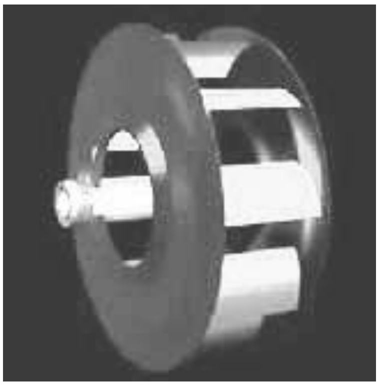


Figure 9-1

Mass Unbalance

Unbalance is a condition where a shaft's geometric centerline and mass centerline do not coincide.

"Unbalance – A shaft's geometric centerline and mass centerline do not coincide"

Another way to describe it is that the center of mass does not lie on the axis of rotation. In other words there is a heavy spot somewhere along the shaft.

Understanding Unbalance

In a perfectly balanced rotor, the center of mass is the same as the center of rotation. The "center of mass" is the point about which the mass is evenly distributed.

- The "center of geometry" is the line through the shaft and bearings
- The "center of mass" is the point about which the mass is evenly distributed

However, if the center of mass is not the same as the center of rotation, the rotor is not balanced. In the example in Figure 9-2, a mass was added and it moved the center of mass away from the center of rotation.

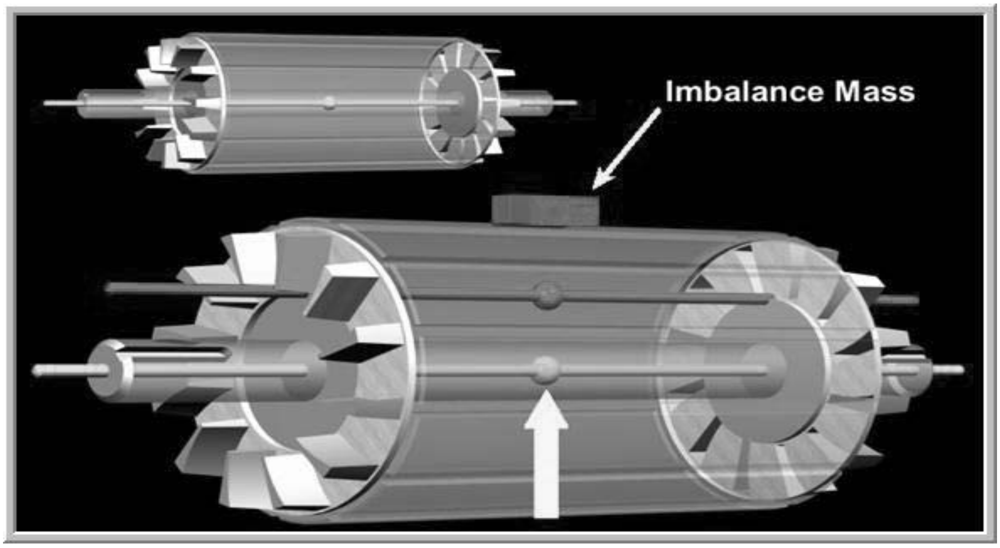


Figure 9-2 - Mass Unbalance - the center of mass is not the center of rotation

If the rotor is placed on two knife-edges, it would rotate and come to rest when the mass, or heavy spot, is on the bottom. This is known as "**static unbalance**", or "**force unbalance**".

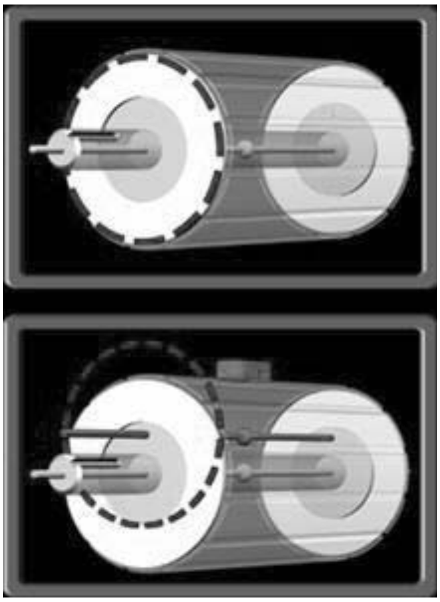


Figure 9-3 - the rotor wants to rotate around the center of mass but is forced to rotate around the shaft

If the rotor could spin in space, it would actually rotate around the center of mass. See the dashed circles in Fig12-7. By forcing it to rotate around the shaft (within the bearings) those bearings, and the rest of the machine, are placed under a great deal of stress.

Causes of Unbalance

There are a number of reasons why a machine may not be in balance. Of course, if a machine was not originally correctly balanced, due to poor training or the lack of the appropriate equipment (and time), then naturally the machine will remain out of balance and damage to seals and bearings may occur.

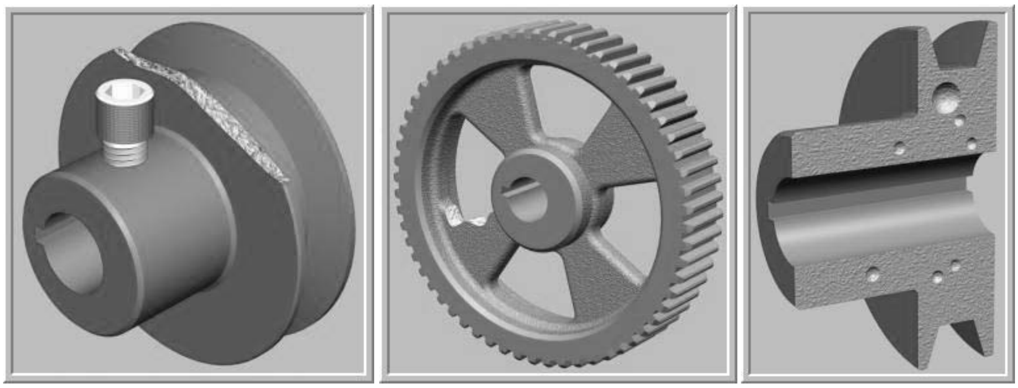


Figure 9-4

There are many causes of unbalance. They include:

- Damaged components
- Manufacturing defects
- Uneven dirt accumulation on fans
- Lack of homogeneity in material, especially in castings. i.e. bubbles, porous sections, blow holes
- Difference in dimensions of mating parts. (i.e. the shaft and bore)
- Eccentric rotor
- Cracked rotor
- Roller deflection (in paper mill rolls)
- Machining errors
- Uneven mass distribution in electrical windings
- Uneven corrosion or erosion of rotors
- Missing balance weights
- Incorrect key
- Uneven or excessive heating

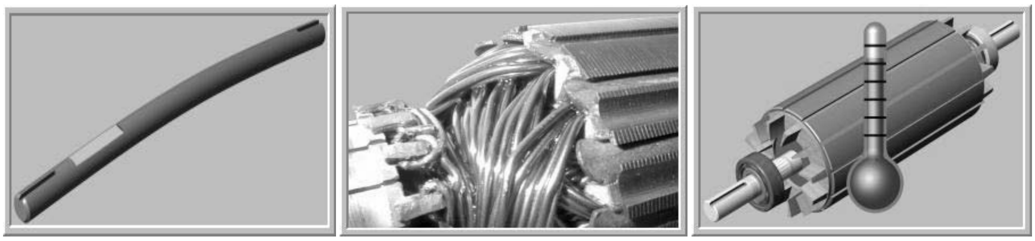


Figure 9-5

ASSESSING THE SEVERITY

The acceptable 1X vibration level is dependent on the size and speed of the machine. Table 9-1 lists the level of vibration for machines operating in the 1500-3600 RPM speed range.

The size of the machine also affects the vibration limits that would be used. As a general rule: for small machines, reduce these limits 4 dB (x 0.63); for large, low speed machines, increase the limits by 4 dB (x 1.6); for large, high speed machines, and reciprocating machines, increase the limits by 8 dB (x 2.5).

Operating speeds ~1500-3600 RPM

Reduce limits by 4 dB (x0.63) for slower machines

Increase limits by 4 dB (x1.6) for large high speed machines

Increase limits by 8 dB (x2.5) for reciprocating machines

1X Vibration level			**Diagnosis**	**Repair priority**
in/sec pk	**mm/s rms**	**VdB (US)**		
<0.134	<2.5	<108	Slight Unbalance	No Recommendation
0.134-0.28	2.5-5.0	108-114	Moderate Unbalance	Desirable
0.28-0.88	5-15.8	114-124	Serious Unbalance	Important
>0.88	>15.8	>124	Extreme Unbalance	Mandatory

Table 9-1 - Guidelines for unbalance limits

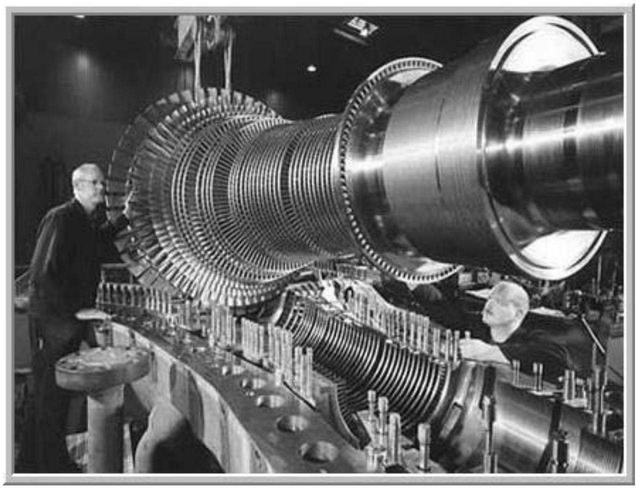


Figure 9-6

The heavy spot on the rotor produces a centrifugal force on the bearings when it rotates, and this force varies smoothly over each revolution of the rotor.

Unbalance forces:

- Put stress on bearings and seals
- Excite resonances
- Exacerbate looseness problems

These "secondary" fault conditions are reduced or eliminated when Precision Maintenance is practiced. (See Chapter One on Maintenance Practices.) Precision balanced machines are far more reliable.

Precision balancing is even more important for high speed machinery because forces generated due to unbalance are much higher at higher speeds. (See Chapter Two)

Assessing the severity of unbalance

ISO standards can be used to assess the severity.

ISO 1940

Mechanical Vibration – Balance quality requirements for rotors in a constant (rigid) state.

Part 1: Specification and verification of balance tolerances

Part 2: Balance errors

Specifications for rotors in a constant (rigid) state:

- Balance tolerance
- Necessary number of correction planes
- Methods for verifying the residual unbalance

Recommendations are also given concerning the balance quality requirements for rotors in a constant (rigid) state, according to their machinery type and maximum service speed

ISO 7919

Mechanical vibration of non-reciprocating machines – measurements on rotating shafts and evaluation criteria.

Part 1: General guidelines

Part 2: Land-based steam turbines and generators in excess of 50 MV with normal operating speeds of 1500 r/min, 1800 r/min, 3000 r/min and 3600 r/min

Part 3: Coupled industrial machines

Part 4: Gas turbine driven sets

Part 5: Machine sets in hydraulic power generating and pumping plants

ISO 10816

Mechanical vibration – Evaluation of machine vibration by measurements on **non-rotating parts**

Provides guidelines for measurements and evaluation criteria for a variety of machine types:

Part 1 - General guidelines

Part 2 - Land-based steam turbines and generators in excess of 50 MW with normal operating speeds of 1500 r/min,
1800 r/min, 3000 r/min and 3600 r/min

Part 3 - Industrial machines with nominal power above 15kW and nominal speeds between 120 r/min and 15000 r/min when measured in situ

Part 4 - Gas turbine sets with fluid film bearings

Part 5 - Machine sets in hydraulic power generating and pumping plants

Part 6 - Reciprocating machines with power ratings above
100 kW

ISO 14694: 2003

This International Standard gives specifications for vibration and balance limits of all fans for all applications except those designed solely for air circulation. ISO 14694 describes the measurement locations, vibration amplitudes, and satisfactory/ alarm/shutdown limits.

Table 1 — Fan-application categories

Application	Examples	Limits of driver power kW	Fan-application category, BV
Residential	Ceiling fans, attic fans, window AC	⩽ 0,15	BV-1
		> 0,15	BV-2
HVAC and agricultural	Building ventilation and air conditioning; commercial systems	⩽ 3,7	BV-2
		> 3,7	BV-3
Industrial process and power generation, etc.	Baghouse, scrubber, mine, conveying, boilers, combustion air, pollution control, wind tunnels	⩽ 300	BV-3
		> 300	See ISO 10816-3
Transportation and marine	Locomotive, trucks, automobiles	⩽ 15	BV-3
		> 15	BV-4
Transit/tunnel	Subway emergency ventilation, tunnel fans, garage ventilation, Tunnel Jet Fans	⩽ 75	BV-3
		> 75	BV-4
		none	BV-4
Petrochemical process	Hazardous gases, process fans	⩽ 37	BV-3
		> 37	BV-4
Computer-chip manufacture	Clean rooms	none	BV-5

NOTE 1 This standard is limited to fans below approximately 300 kW. For fans above this power refer to ISO 10816-3. However, a commercially available standard electric motor may be rated at up to 355 kW (following an R20 series as specified in ISO 10816-1). Such fans will be accepted in accordance with this International Standard.

NOTE 2 This Table does not apply to the large diameter (typically 2 800 mm to 12 500 mm diameter) lightweight low-speed axial flow fans used in air-cooled heat exchangers, cooling towers, etc. The balance quality requirements for these fans shall be G 16 and the fan-application category shall be BV-3.

Figure 9-7

Table 4 — Vibration-levels limit for test in manufacturer's work-shop

Fan application category	Rigidly mounted mm/s		Flexibly mounted mm/s	
	Peak	r.m.s.	Peak	r.m.s.
BV-1	12,7	9,0	15,2	11,2
BV-2	5,1	3,5	7,6	5,6
BV-3	3,8	2,8	5,1	3,5
BV-4	2,5	1,8	3,8	2,8
BV-5	2,0	1,4	2,5	1,8

NOTE 1 Refer to Annex A for conversion of velocity units to displacement or acceleration units for filter-in readings.

NOTE 2 The r.m.s. values given in this Table are preferred. They are rounded to a R20 series as specified in ISO 10816-1. Peak values are widely used in North America. Being made up of a number of sinusoidal wave forms, these do not necessarily have an exact mathematical relationship with the r.m.s. values. They may also depend to some extent on the instrument used.

NOTE 3 The values in this Table refer to the design duty of the fan and its design rotational speed and with any inlet guide vanes "full-open". Values at partial load conditions should be agreed between the manufacturer and user, but should not exceed 1,6 times the values given.

Figure 9-8

Table 5 — Seismic vibration limits for tests conducted *in situ*

Condition	Fan-application category	Rigidly mounted mm/s		Flexibly mounted mm/s	
		Peak	r.m.s.	Peak	r.m.s.
Start-up	BV-1	14,0	10	15,2	11,2
	BV-2	7,6	5,6	12,7	9,0
	BV-3	6,4	4,5	8,8	6,3
	BV-4	4,1	2,8	6,4	4,5
	BV-5	2,5	1,8	4,1	2,8
Alarm	BV-1	15,2	10,6	19,1	14,0
	BV-2	12,7	9,0	19,1	14,0
	BV-3	10,2	7,1	16,5	11,8
	BV-4	6,4	4,5	10,2	7,1
	BV-5	5,7	4,0	7,6	5,6
Shutdown	BV-1	Note 1	Note 1	Note 1	Note 1
	BV-2	Note 1	Note 1	Note 1	Note 1
	BV-3	12,7	9,0	17,8	12,5
	BV-4	10,2	7,1	15,2	11,2
	BV-5	7,6	5,6	10,2	7,1

NOTE 1 Shutdown levels for fans in fan-application grades BV-1 and BV-2 should be established based on historical data.

NOTE 2 The r.m.s. values given in this Table are preferred. They are rounded to a R20 series as specified in ISO 10816-1. Peak values are widely used in North America. Being made up of a number of sinusoidal wave forms, these do not necessarily have an exact mathematical relationship with the r.m.s. values. They may also depend to some extent on the instrument used.

Figure 9-9

Diagnosing Mass Unbalance

An unbalanced rotor will generate vibration at the frequency of shaft turning speed due to the centrifugal force of the unbalance mass. This was seen in the "Principles of Vibration" chapter with the examples of a coin placed on a fan blade.

Therefore it is expected that a machine with an unbalance condition will generate a sinusoidal sine wave and a corresponding dominant peak in the spectrum at shaft turning speed (1x).

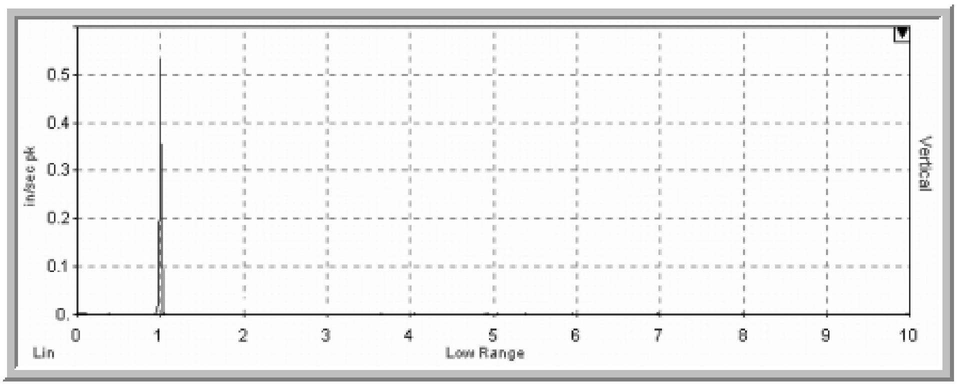


Figure 9-10 - Unbalance generates a peak at 1x in radial directions.

Typically other sources of vibration are also present so the waveform does not have a pure sine wave but is often very sinusoidal. See Figure 9-11. The other sources of vibration could be looseness, misalignment, bearings, etc.

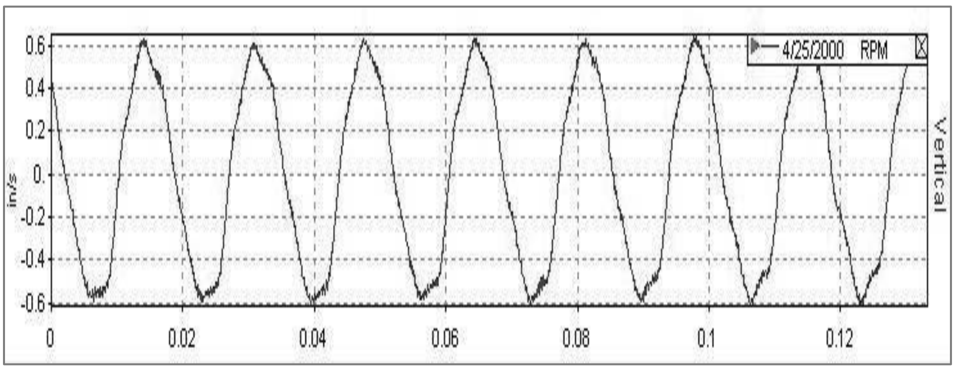


Figure 9-11 - Unbalance waveform is predominantly sinusoidal

There is almost always some residual unbalance, so there is almost always a 1x peak.

If the spectrum is dominated by 1x and the amplitude is high, suspect mass unbalance.

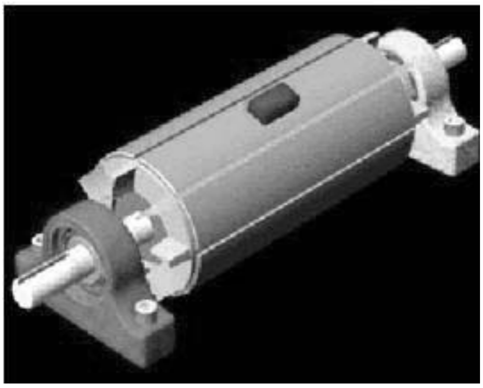


Figure 9-12

Every rotor (fan, pump, etc.) will have some residual unbalance - nothing is perfectly balanced. As a result, there will be a peak at 1X, and if the rest of the machine is "quiet", the 1X peak may still dominate the spectrum, and the time waveform may look sinusoidal. Therefore it has to be determined whether the unbalance actually represents a problem based on the amplitude levels.

VIBRATION ANALYSIS OF UNBALANCE

The time waveform should be sinusoidal and there ought to be a large 1X peak in the spectrum. This vibration characteristic is in the radial directions - i.e. vertical and horizontal.

The measured vibration level at 1X depends on the stiffness of the machine mounting as well as the amount of unbalance, with spring-mounted machines showing more 1X than solidly mounted machines for the same degree of unbalance.

The vertical and horizontal 1X levels should be compared. The more nearly equal they are, the more likely that unbalance is the cause. In any case, the direction in which the machine has the least stiffness will be the direction of the highest 1X level. The horizontal vibration will therefore typically be higher than vertical.

VERIFYING UNBALANCE

To confirm whether the motor or pump is out of balance run the motor uncoupled. If the 1x level is still high, the problem is the motor; otherwise it is the pump. If the speed can be changed, then the amplitude should go up in proportion to the square of the speed. (If the speed is doubled, the vibration should increase by a factor of 4.)

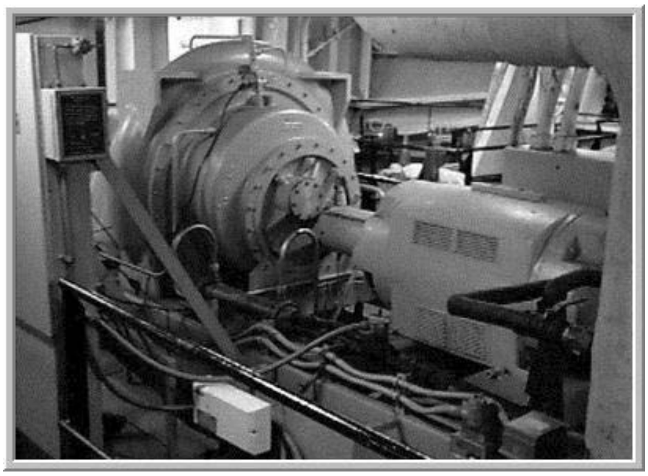


Figure 9-13

Static Unbalance

Static or Force Unbalance is the simplest type of unbalance and is equivalent to a heavy spot at a single point in the rotor. This is called a static unbalance because it will show up even if the rotor is not turning. When the rotor is placed on frictionless knife edges so that it is free to turn, it will rotate so that the heavy spot is at the lowest position. The rotor has an unbalance even in a stationary condition.

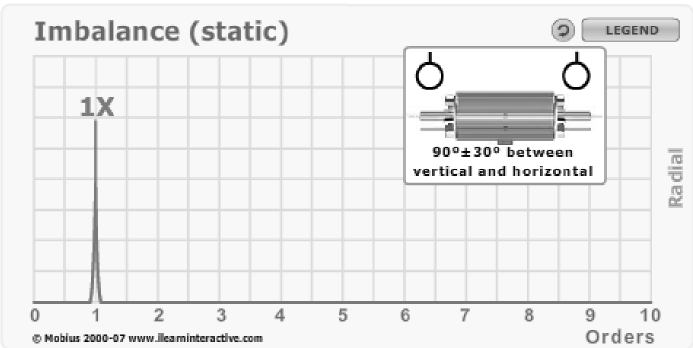


Figure 9-14

When rotating, the static unbalance results in 1X forces on both bearings of the rotor, and the forces on both bearings are always in the same direction. The vibration signals from them are "in phase" with each other.

The phase difference from vertical to horizontal is 90 degrees.

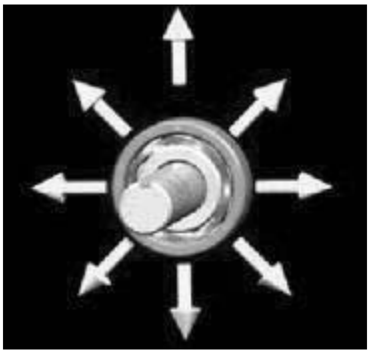


Figure 9-15 - the force is radial

A pure static unbalance produces a strong 1X peak in the vibration spectrum, the amplitude of which is proportional to the severity of the unbalance and the square of the RPM. The relative levels of the 1X vibration at the bearings depend on the location of the heavy spot along the rotor.

Couple Unbalance

Couple Unbalance is a condition when there are two equal unbalance masses on the opposite ends of the rotor, but 180 degrees opposite each other.

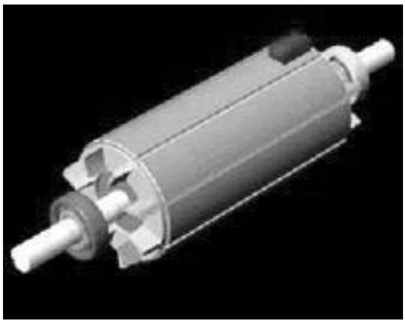


Figure 9-16 - Couple unbalance has 2 equal masses at opposite ends 180 degrees apart.

A rotor with couple unbalance may be statically balanced (it may seem to be perfectly balanced in a stationary condition when placed in frictionless bearings). But when rotated, it produces centrifugal forces on the bearings, and they will be of opposite phase.

The vibration spectrum will look the same as static unbalance; only a phase measurement will help distinguish between static and couple unbalance.

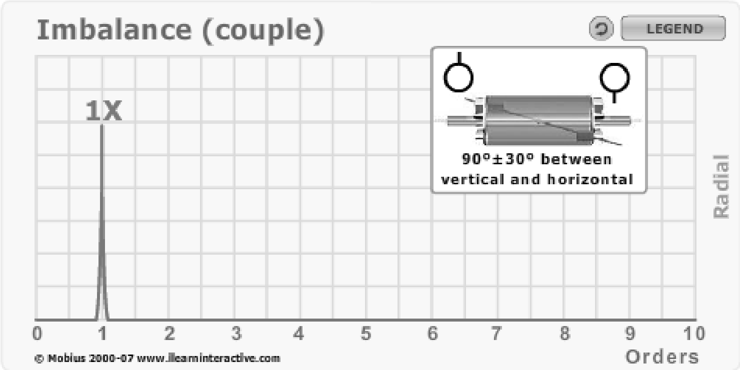


Figure 9-17 - Couple unbalance phase readings end to end are 180 degrees out of phase.

The unbalance may be stronger in the horizontal position than the vertical due to increased flexibility in that direction.

There can be some axial vibration at 1x but it is usually moderate in comparison to the radial directions.

Dynamic Unbalance

In common practice, a pure couple unbalance is seldom found in plant machinery. Instead it is a combination of static and couple unbalance. This condition is called Dynamic Unbalance.

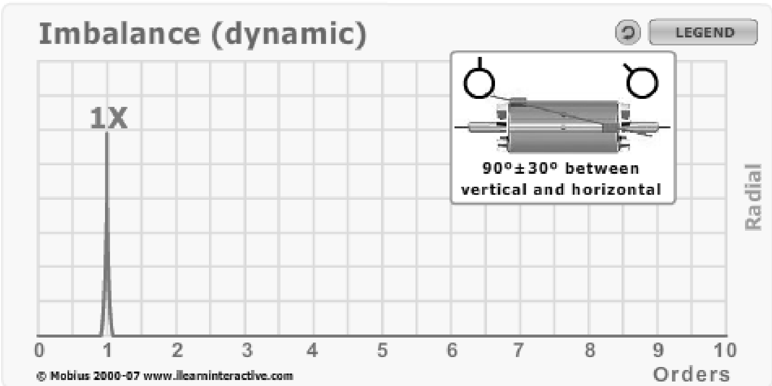


Figure 9-18 - Dynamic Unbalance has high 1x. Phase readings end-to-end will not be in phase nor 180 degrees out of phase.

Dynamic unbalance cannot be corrected in one plane.

Vertical Machines

Vertical machines, such as pumps, are usually cantilevered from their foundation, and they usually show maximum 1X levels at the free end of the motor regardless of which component is actually out of balance.

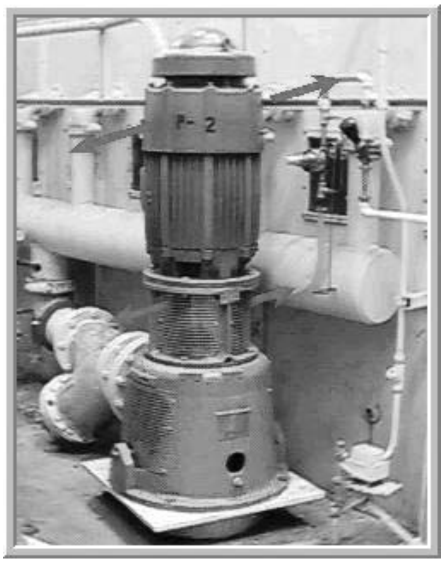


Figure 9-19

The spectrum again will show a strong 1X peak when measured in the radial direction (horizontal or tangential).

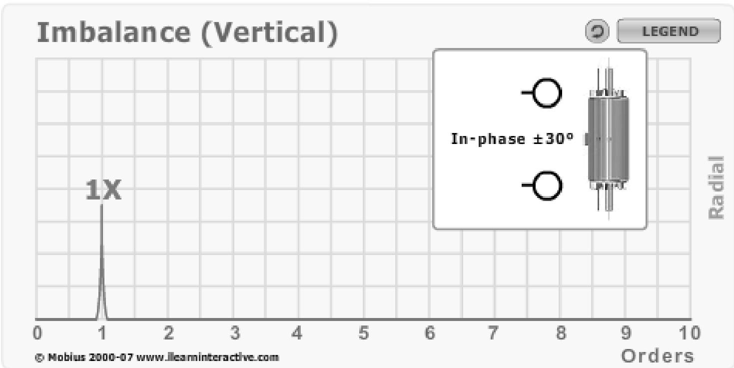


Figure 9-20

Unbalance in Overhung Machines

Overhung pumps and fans are common in industry. Examine the rotating machinery closely to ensure that you know whether a component is in fact overhung or supported on both sides by bearings. In an overhung or cantilevered machine, a high 1X vibration level is present, however this time it will be present in the axial direction as well as in vertical and horizontal. Measurements should be taken from the bearing closest to the overhung impeller or fan blades.

The high 1X in axial is present because the unbalance creates a bending moment on the shaft, causing the bearing housing to move axially.

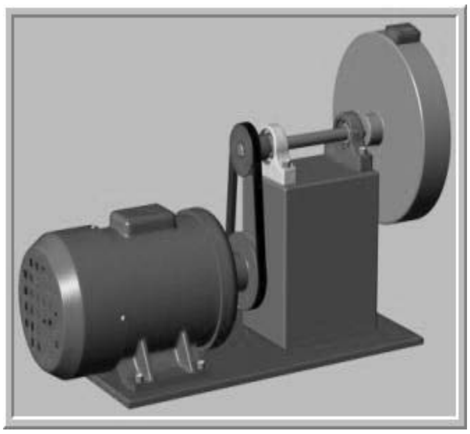


Figure 9-21 - Overhung machine configuration

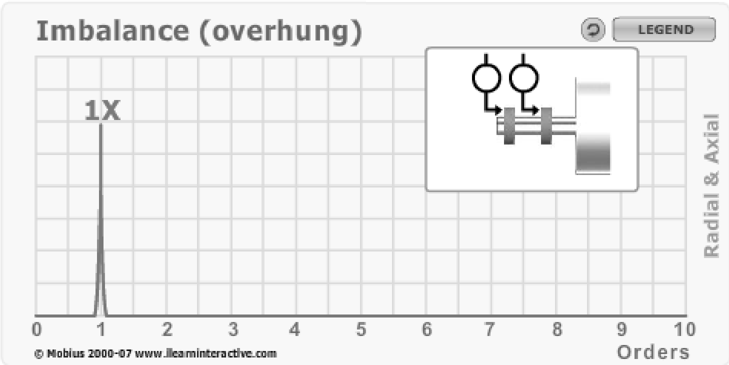


Figure 9-22 - Unbalance in overhung machine

Be sure to collect data close to the impeller vanes or fan blades. Phase data on these two bearings in the axial direction should be in phase.

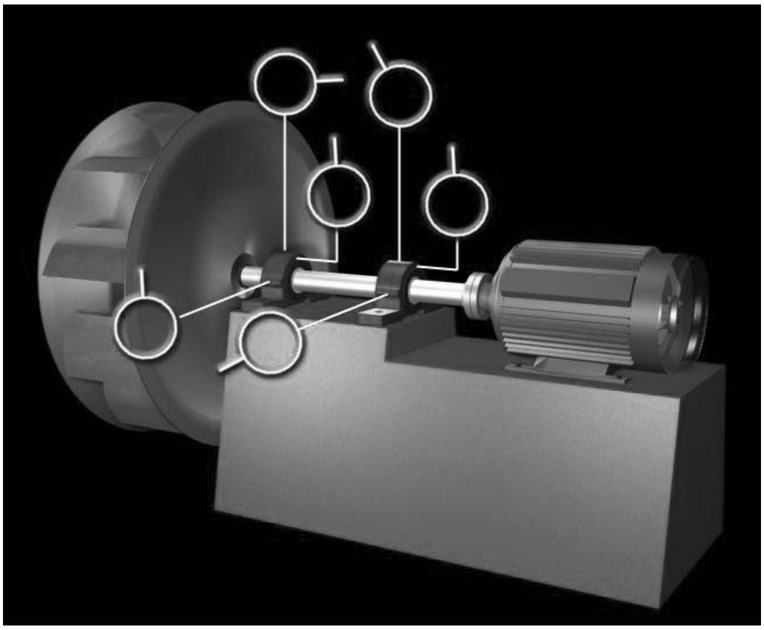


Figure 9-23

Case Study: Ash Hopper Sluice Pump

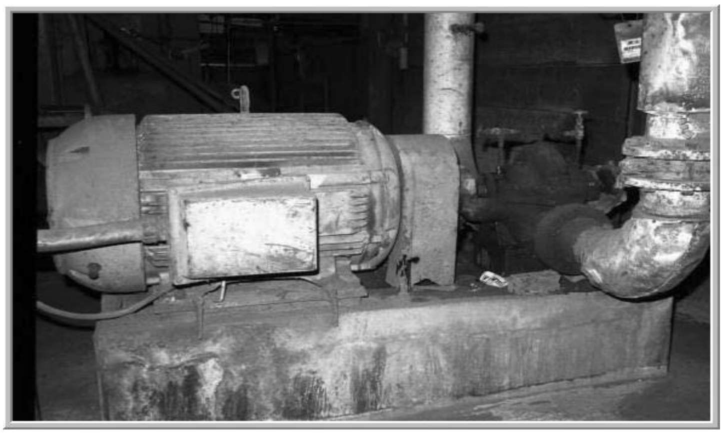


Figure 9-24

- Speed = 3600 CPM
- 150 HP motor
- Centrifugal-supported
- 245 GPM @ 385' head
- Paraflex coupling

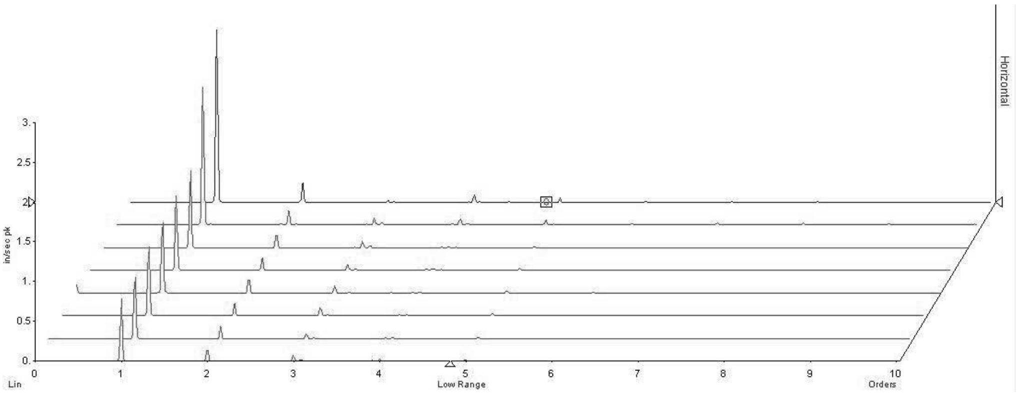


Figure 9-25 - 1x increased over time. It is 2.17 in/sec.

The vibration level at the 1X frequency (2.17 in. / sec.) has been increasing over a period of time. This could be because of uneven buildup of ash on the pump impeller, or erosion/corrosion of the pump impeller.

It is also possible that continued damage to the feet/foundation have caused the machine to become more flexible horizontally. However given that the vertical level has also increased, it would be suggested that it is the out of balance condition that is responsible for the increase in amplitude, and not in a continued weakness of the structure changing the horizontal stiffness (i.e. affecting the horizontal flexibility).

Case Study: Black Liquor DIL #2

- Speed = 1750 CPM
- 20 HP motor, AC
- Centrifugal pump
- Impeller 10.125"
- Overhung pump
- Pump vanes 6
- 400 GPM @ 112' head
- Paraflex coupling

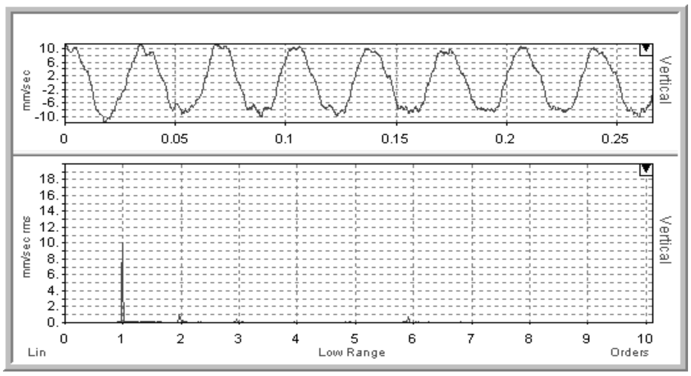


Figure 9-26

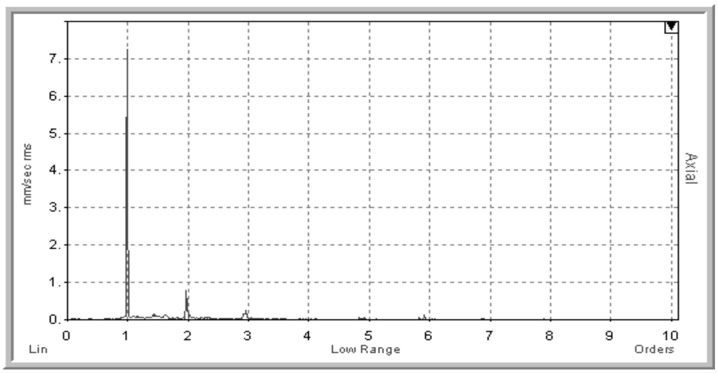


Figure 9-27

This data is from a "Black liquor pump" at a paper mill.

There is high 1X in the vertical direction and the time waveform looks very sinusoidal (periodic). The "squiggles" are due to the 2X and 6X components also present.

A quick look at the horizontal data shows that the level at 1X is also dominant and quite high.

In the axial direction, we again see a dominant and high level at 1X.

So, this is a classic example of unbalance on a machine with an overhung rotor. The motor levels were substantially lower, which is to be expected.

Eccentricity

Eccentricity occurs when the center of rotation is offset from the geometric centerline of a sheave (pulley), gear, bearing, or rotor. It is being discussed now because the symptoms are very similar to unbalance.

Shaft centerline is not coincident with the rotational centerline

The object (pulley, gear, etc.) will “wobble” around the false center, producing a strong radial vibration.

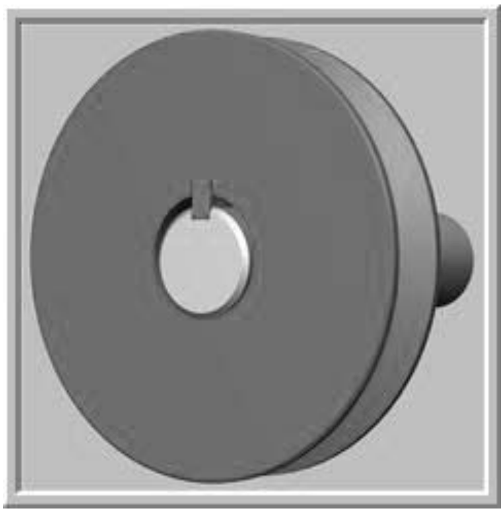


Figure 9-28

Eccentric sheaves/gears will generate strong 1X radial components, especially in the direction parallel to the belts. This condition is very common, and mimics unbalance.

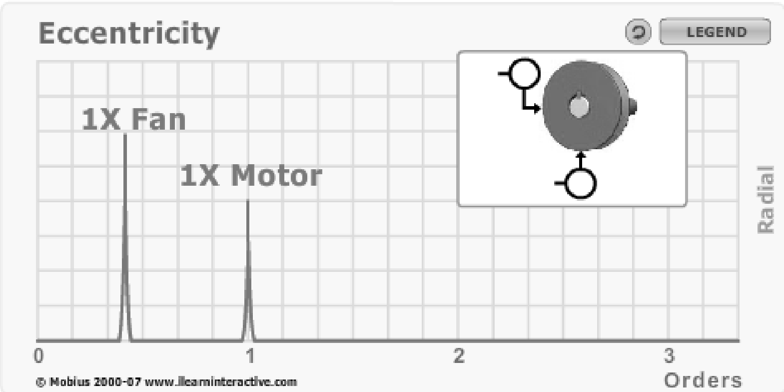


Figure 9-29 - Eccentric gears or sheaves generate a strong 1x radial component.

In belt driven machines, there will be a high 1X vibration level on both components (motor and fan for example), however due to the change in speed, these will be at two different frequencies.

The highest vibration is in line with the belts. Phase readings taken 90 degrees from that will have a phase change of 0 degrees or 180 degrees.

Figure 9-30 - Collect vibration measurements in-line with the belts.

Eccentricity in belt driven machines can be checked by removing the belt(s) and checking again for the 1X peak on the motor.

Note: Eccentric rotors are covered in the chapter on electric motor analysis.

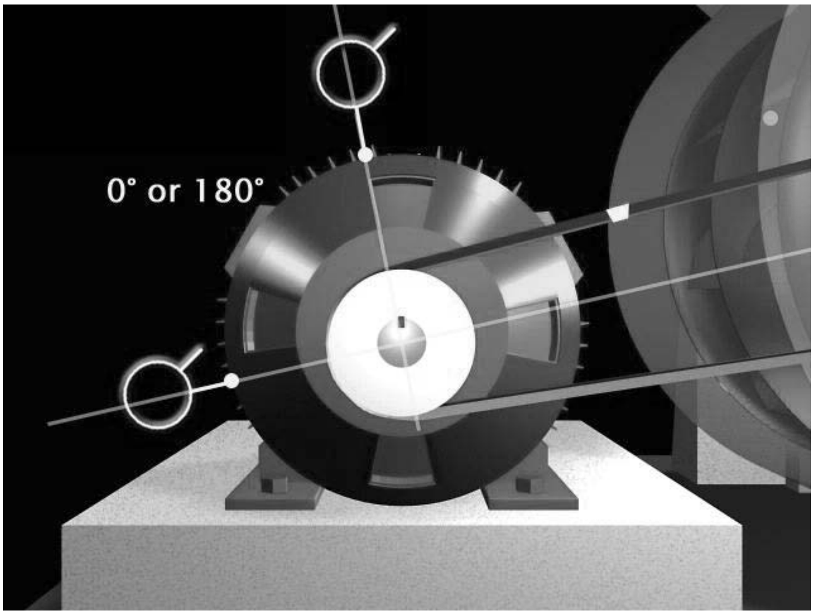


Figure 9-31

The mass unbalance can be balanced out, but the "wobble" cannot be easily removed. The gear or sheave will have to be replaced.

Eccentricity can affect gears, pulleys, belt drives, and electric motors.

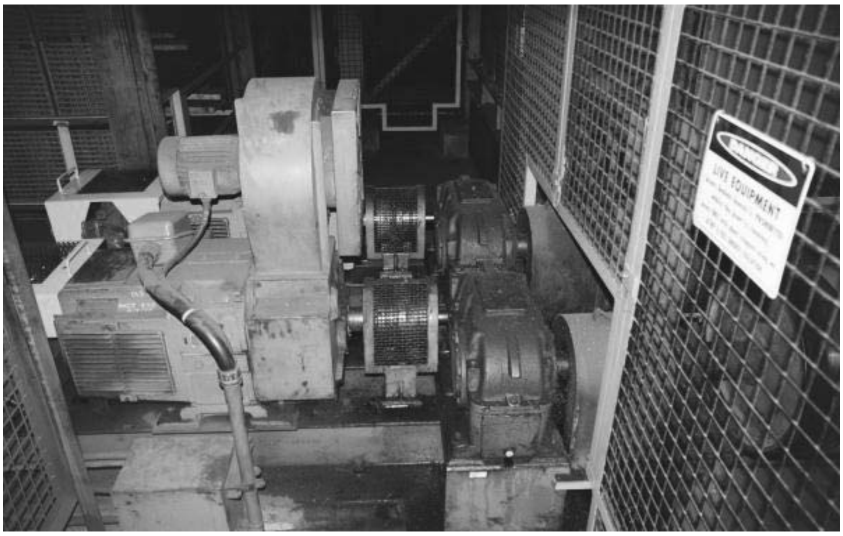


Figure 9-32 - Eccentric sheaves and gears must be replaced.

Chapter 10

Balancing Rotating Machinery

Objectives:

- Understand vectors, vector addition and vector subtraction
- Understand how a single plane balancing works.
- Understand the difference between one plane and two-plane balancing.
- Become familiar with the relevant ISO standards.

Balancing rotating machinery

Balancing rotating machinery is an important part of any reliability program. Balanced machinery run more smoothly and therefore run longer without incident. Modern vibration analyzers and purpose-built balancing instruments have made it easier to balance machines in-situ – that is, without having to remove the rotors and ship them off to a balancing machine.

Like any instrument of this type, it is always important to understand how the instrument works and how to correctly prepare the job and deal with any unusual readings that you may observe. In an ideal world you can take the vibration and phase readings, follow each of the steps, and the machine will be precision balanced. But life is rarely that simple…

The goals of this chapter

This section is intended to provide an introduction to balancing. The aim is not to teach you *everything* you need to know so that you can balance a machine – there is simply not enough time on a detailed course like this. Instead you should:

- Understand vectors, vector addition and vector subtraction
- Understand how a single plane balancing works.
- Understand the difference between one plane and two-plane balancing.
- Become familiar with the relevant ISO standards.

With this base knowledge you should find it far easier to learn the detailed steps required to perform the balance, and you will have a greater appreciation of what is involved if your request that a machine be balanced.

What is balancing?

First, what is balancing? The ISO standard 1940-1973 (E) definition is:

"Balancing is the process of attempting to improve the mass distribution of a body so that it rotates in its bearings without unbalance centrifuge forces."

Those "unbalance centrifuge forces" are destructive. The forces slowly destroy bearings, and can generate vibration that can damage other equipment and processes. All machines should be precision balanced and aligned to reduce these destructive forces – and therefore increase the life of the machine and improve reliability.

The ISO standard 1940-1:2003 definition is:

"Balancing: procedure by which the mass distribution of a rotor is checked and, if necessary, adjusted to ensure that the residual unbalance or the vibration in the journals and/or forces on the

bearings at a frequency corresponding to service speed are within specified limits." [Origin: ISO 1925:2001, definition 4.1]

The goal and the intent are the same – reducing vibration levels to within specified limits. Later in this section we will discuss the ISO standards for balancing and the recommended vibration levels for different applications.

There are two more definitions from ISO 1940-1:2003 that are important:

"Unbalance: condition which exists in a rotor when vibration force or motion is imparted to its bearings as a result or centrifugal forces." [Origin: ISO 1925:2001, definition 3.1]

"Residual unbalance: unbalance of any kind that remains after balancing." [Origin: ISO 1925:2001, definition 3.0]

The ISO standard 1925:2001 is "Mechanical vibration – Balancing – Vocabulary".

Preparing for the balance job - a word of warning

Given the ease of use of balancing software provided with modern analyzers it is conceivable that balancing technicians may jump straight into a balance job without thinking about the machine dynamics – the interplay between the unbalance forces, the rotor and the bearings. Balancing is a difficult, time consuming, and potentially dangerous task. It is essential that you take all precautions before and during a balance job. Follow all lock-out tag-out procedures, and do not make any assumptions about the safety systems installed at the site.

Safety first!

There are four main sources of danger that you should be aware of. They are:

- The machine starting unexpectedly
- Balance weights flying off the rotor
- The general dangers inherent in working in a hazardous environment
- Becoming entangled – you or your equipment – in the machine

Figure 10-1 Follow all lock-out and tag-out procedures

ISO 20806:2009 "Mechanical vibration – criteria and safeguards for the in-situ balancing of medium and large rotors" is a useful standard to be acquainted with. It also includes information on safety

Another 'warning' is to not to underestimate balancing. If everything goes smoothly you may complete a balance job in a couple of hours. However there is a lot that can go wrong. You need to think about what you are doing and not make any assumptions. Balancing programs can make the measurements and calculations easier, however resonances, foundation problems, thermal growth and more can make the balance job quite challenging.

Is the machine out of balance?

Many people fail at the first hurdle. They measure high vibration and assume the machine is out-of-balance. So they attempt to balance the machine. Yet, to their surprise, the balance weights recommended do not reduce the vibration levels as much as expected.

So you should verify that the machine is out of balance before you spend time trying to attempt to balance it! It may sound obvious, but a great deal of time has been lost by people trying to balance a machine that is misaligned, or that has some other fault that presents similar vibration symptoms to unbalance.

It could be that the machine has more than one fault condition. Correct those other fault conditions before you attempt to balance the machine.

Some conditions can make it very difficult to balance a machine. For example, a natural frequency close to the running speed will cause the phase readings to be unsteady. If the speed, phase or amplitude are not steady, then it will be difficult to successfully balance the machine.

The balancing check-list

Before you consider performing an in-situ balance job, you must first consider the following checklist. In order to balance the machine you must check that:

- You can start and stop the machine.
    - The machine will need to be stopped and started in order to add trial weights and the final balance weights.
- It is possible to add balance weights (you will need access to the rotor, fan blades, etc.).
    - Many machines have balance rings, or places on the rotor where balance weights can be added. You must check how the weights will be added (or removed) and that you have suitable balance weights.
- It is possible to gain access to the machine.
    - You will need to be able to access the balance ring or rotor (wherever the balance weights will be added).
- It is possible to control the speed of the machine.
    - It is important that the speed of the machine remains constant during the tests and from one test to the next (i.e. between the original run and each of the trial, final and trim runs).
- The speed, amplitude and phase must be steady during the tests.
    - The speed of the machine, and the amplitude and phase readings, must be constant during the tests. This may mean that you must run the machine for many minutes (sometimes longer than one hour) in order for the machine to be running in a stable, repeatable condition. Thermal transients and load changes must be considered before attempting to balance the machine.
- It is possible to take a phase reading (you will need access to the shaft).
    - You will need to acquire phase readings during the test. This step will be discussed in greater detail, however you must ensure that you can either add reflective tape to the shaft and mount a laser or photo-tachometer, or mount a non-contact probe to detect a keyway or other such physical irregularity.
- You will need the required instrumentation and a balance program/method.
    - Of course, you will also need the instrumentation to measure the amplitude and phase, a calculator or balance program (or polar plotting paper and protractor), and some scales to weigh the balance masses.

Vectors and polar plots

If you wish to perform a single-plane balance with a polar plot, then you must understand vectors. If you wish to understand the balancing process and the effect of adding trial weights, then it helps a great deal if you understand vectors.

Understanding vectors, and knowing how to add and subtract vectors may involve learning some new concepts, but I am sure that this section will make it very clear to you.

Vectors are a combination of an angle and a "scalar" amount. The "scalar" amount could be wind speed, current flow in an ocean, or in the vibration world, a vibration amplitude at the running speed of the machine. For example, if a ship was sailing at 10 knots in a northerly direction, then we can represent that as a vector: 10 knts @ 0°. Readings such as 2 micron @ 45° and 0.5 in/sec @ 125° are vibration vectors.

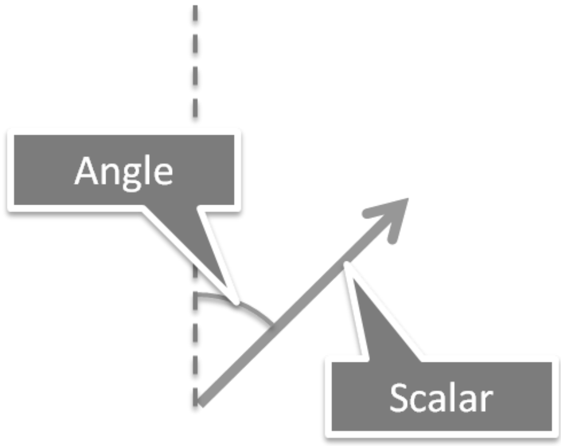


Figure 10-2

Vectors are normally represented on a circular plot called a "polar plot" (Figure 10-3).

The scalar value (amplitude) is represented by the length of the line (arrow), and the angle is drawn with 0° at the top.

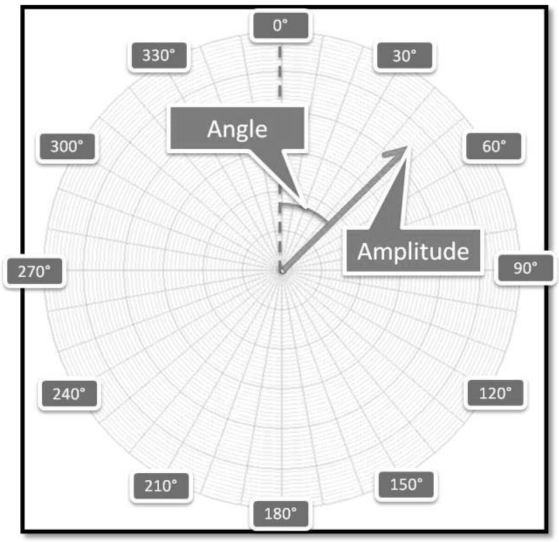


Figure 10-3

The markings on the polar plot make it possible to read angles and lengths, although it can be helpful to also have a ruler and a protractor; especially when it comes to vector addition and subtraction.

The radiating circles represent the amplitude.

Figure 10-4 shows a simple example. We made the radius of the plot equal to 6 mils which means every ring represents 1 mil (units of displacement). The ruler will help us to measure the length of the vectors.

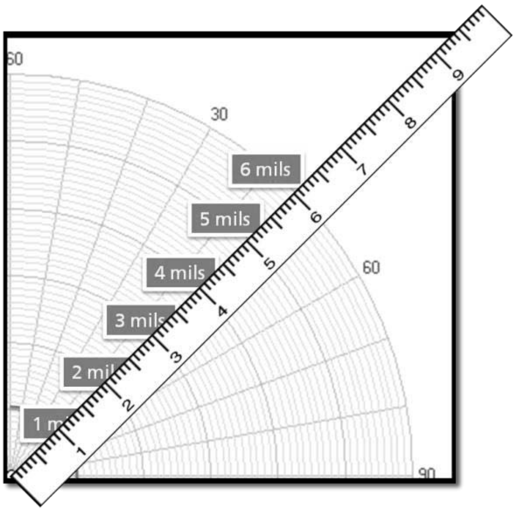


Figure 10-4

If we measured a vibration of 5 mils at 45° then we would draw the vector as shown in Figure 10-5.

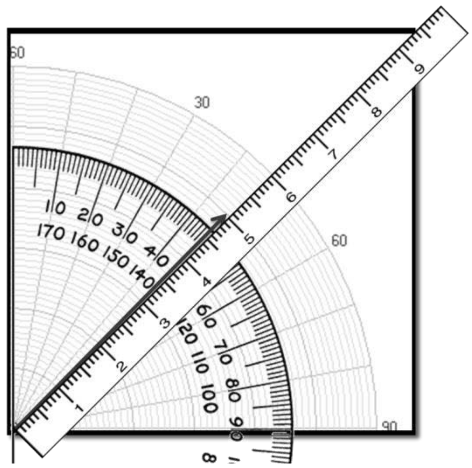


Figure 10-5

To make sure we really understand vectors, let's go through a simple example.

In Figure 10-6 we have a ship traveling at its maximum speed of 10 knots towards the east. The ship is moving due to its engines. There is no wind and no current.

Therefore the ship travels east at 10 knots. As a vector we would describe this as: **10 knts @ 90°**

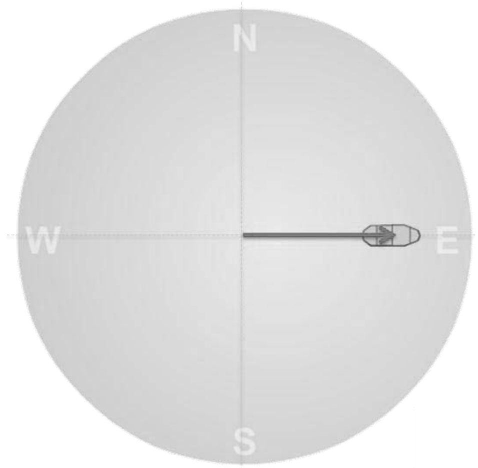


Figure 10-6

Unfortunately for the ship, a wind blows up from the south which has the effect of pushing the ship to the north. The wind is strong enough such that if it turned its engines off the ship would travel at 10 knots to the north.

But if the ship turns on its engines and tries to sail to the east, it would find that it does not travel to the east; instead it travels to the north-east.

We can therefore represent the actual movement of the ship with a different vector, one that points at 45° (Figure 10-7).

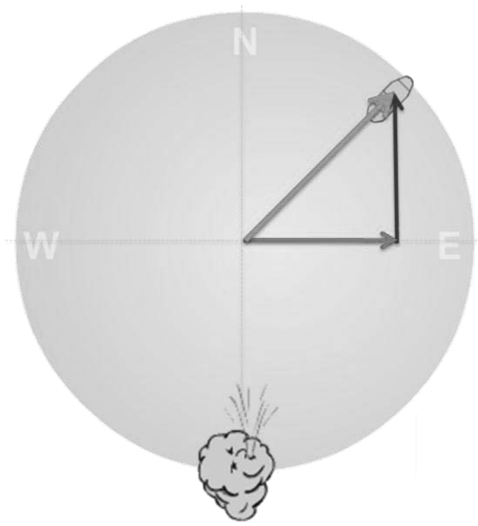


Figure 10-7

If we were to measure the length of that vector we would find that it was
14.14 knots in length. So the good news for the ship is that it is traveling faster than before. The bad news is that it is going in the wrong direction…

We can determine the angle and length mathematically or we can plot the vectors on a polar plot and either use the scale provided or use a ruler and protractor.

Adding vectors

What we have actually just done is to add two vectors. Let's look at that a little more closely.

Adding vectors is a case of moving the tail of one vector to the tip of the other vector.

It does not matter which vector is added first. No matter how you add them the result will be the same (Figure 10-8 center and right).

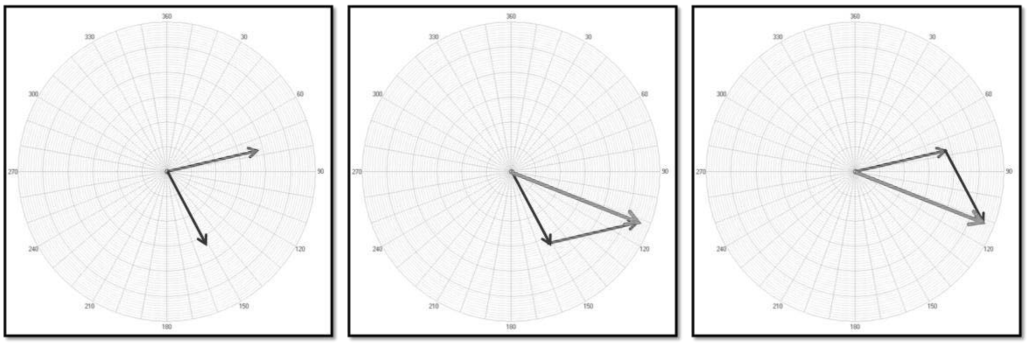


Figure 10-8

And it does not matter how many vectors must be added; the process is the same (Figure 10-9).

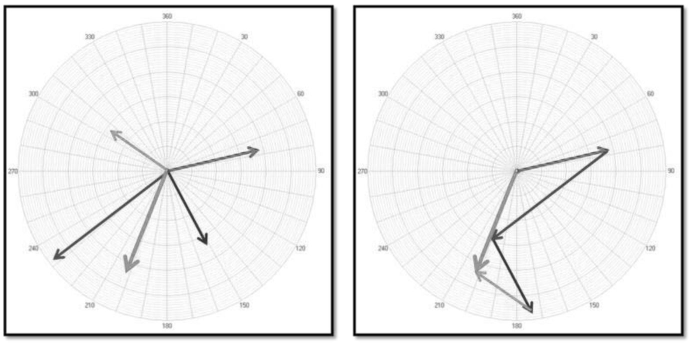


Figure 10-9

Vectors can "easily" be added mathematically. It is a case of breaking each vector into its "X" and "Y" components, and then adding the "X" components and adding the "Y" components.

The length of the vector "Z" can be calculated using Pythagoras' theorem. And the angle can be calculated using ATAN ($\tan^{-1}$), as shown in Figure 10-10).

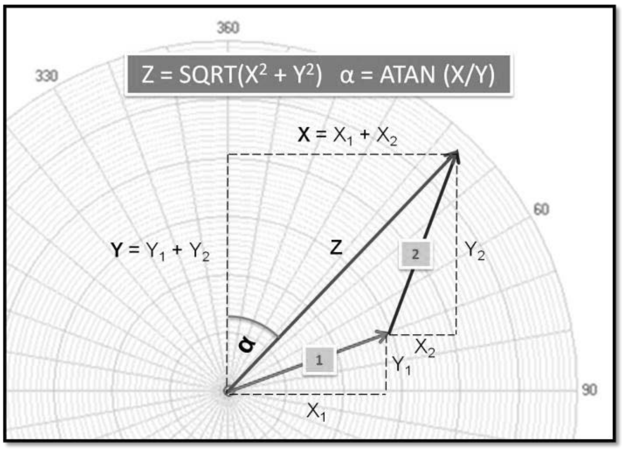
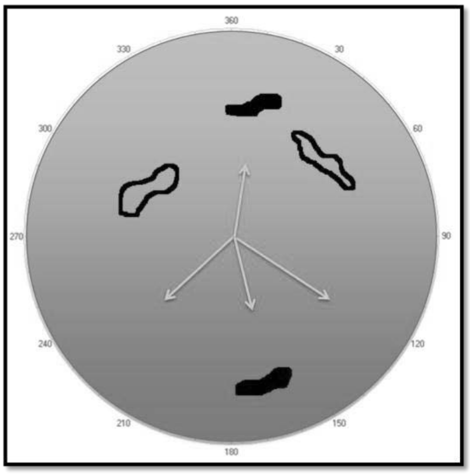


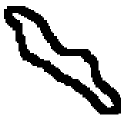
Figure 10-10

It is worth making a very important point right now. If you consider an out-of-balance rotor, the source of unbalance is probably not just one source at precisely one location (like a bolt that has been attached to the rotor in the wrong location).

More than likely there are a number of sources of erosion and material porosity, dirt/grim build-up, and so on.

Each source of mass (or loss of mass) can be represented by a vector (Figure 10-11).





Porosity or erosion: this is a *loss* of mass so the vector points *away* from these points.

Build-up of dirt/grime that cannot be cleaned off. They add mass.

Figure 10-11

And just as before, we can add the vectors together. The red vector is the final vector that we would measure as the unbalance (Figure 10-12).

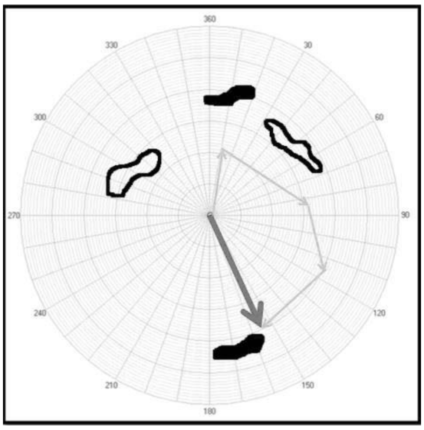


Figure 10-12

Please note that you would not need to add these vectors; this just happens naturally. The point is that each individual source of unbalance contributes to the final unbalance that causes the machine to vibrate.

There is one interesting situation to consider. What would happen if we added the two vectors in Figure 10-13?

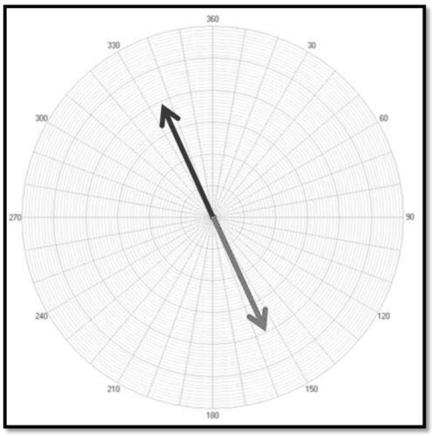


Figure 10-13

Yes, that is correct. Because they have the same length, and their angles are opposite, the result is a vector with zero length!

Why is this relevant? Because that is exactly what we are trying to do when we balance the rotor. If the red vector represents the sum of the sources of unbalance (porosity, erosion, etc.), then we will add a weight that would create the blue vector – it will have an equal and opposite effect.

Subtracting vectors

Subtracting vectors often causes a great deal of confusion. It is a necessary step when performing single-plane balancing with vectors. The confusion is unnecessary.

To subtract vector "A" from vector "B", you simply turn vector "A" around 180° and then add it!

If we have vectors "A" and "B" and we want "A-B" then we flip "B" around by 180° and then add its tail to the tip of vector "A" (Figure 10-14).

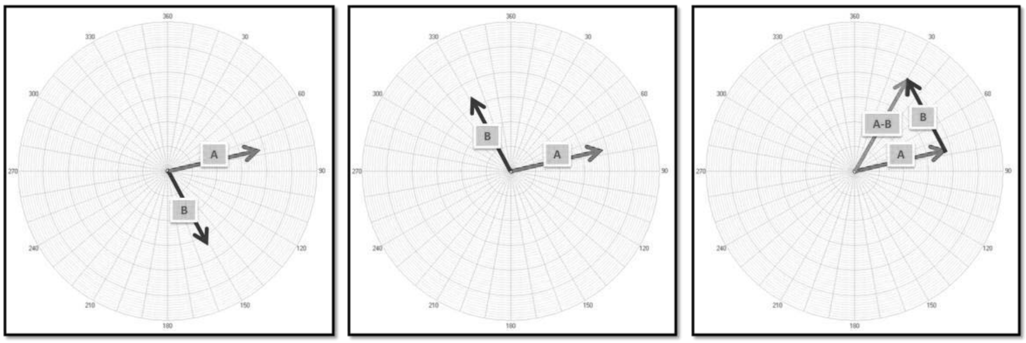


Figure 10-14

Note that subtraction is not like addition: "A-B" is different to "B-A" (sorry if that sounds obvious).

The result is the same vector as last time, rotated 180° (Figure 10-15).

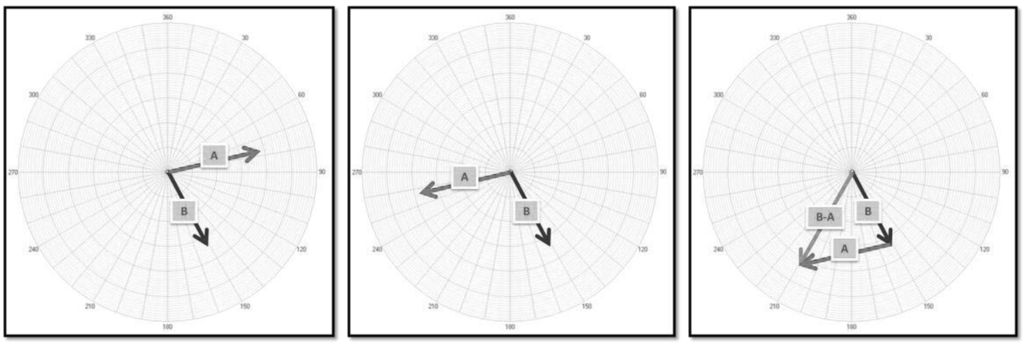


Figure 10-15

Single-plane balancing

In order to perform a single plane balance we need to determine where to place the balance correction weight, and how much it must weigh. Note that more than one weight may need to be added if it is not possible to add the weight at the desired angle (because there is not a blade or balance hole at that angle.

Summary of the single plane method

First we take a vibration measurement on a bearing and record the magnitude and phase at the running speed of the machine (therefore we need a once-per-revolution tachometer reference). We document this information as the "As-found" condition of the machine prior to balancing it.

Next we add a "trial weight" to the rotor, and measure at the same point and record the magnitude and phase at the running speed. We do this to see how the rotor was "influenced" by the addition of the weight. The magnitude and phase should both change by 15% or 15 degrees or more.

Now we know how the magnitude and phase of the vibration at the monitoring point has changed based on the addition of the trial weights. We can then perform a calculation (or sketch it out with vectors on polar graph paper) to determine where we should add weights (and how much they should weigh) to cancel the effect of the out of balance force.

Remember, we do not know why the machine/rotor is out-of-balance – we can't see a weight on the rotor. It is out-of-balance because of poor design/manufacturing/assembly, wear/erosion on blades/vanes, build-up on blades/vanes, incorrect key length, and distortion/bends.

We then add the correction weight and take another measurement to see if the unbalance is within tolerance. Often the unbalance will not be within tolerance on the first attempt, so we use the new readings to determine where a "trim" weight should be added in order to bring the machine into tolerance.

Once the machine has been balanced satisfactorily, the results should be documented.

The procedure just described is used regardless of whether you use a data collector, balance computer, or graph paper. We take initial readings, add trial weights to see how the balance state was "influenced", and then we determine where the final weights should be placed. (Please note that there are other ways to balance a machine, but they are not covered in this course.)

Using vectors

Although you will probably use a balance calculator or your data collector to perform the balancing task, it is important to understand the single-plane vector method. If you understand this procedure, you will be able to cope when your balance job does not go smoothly, and you will find it much easier to understand the two-plane and "influence coefficients" method used by most balance calculators and data collectors.

We will use polar graph paper to record the vibration amplitude (magnitude) and phase. The radial rings represent the magnitude (in this example each ring is 1 mil), and the radial lines are angle (degrees). 360/0 degrees is at the top.

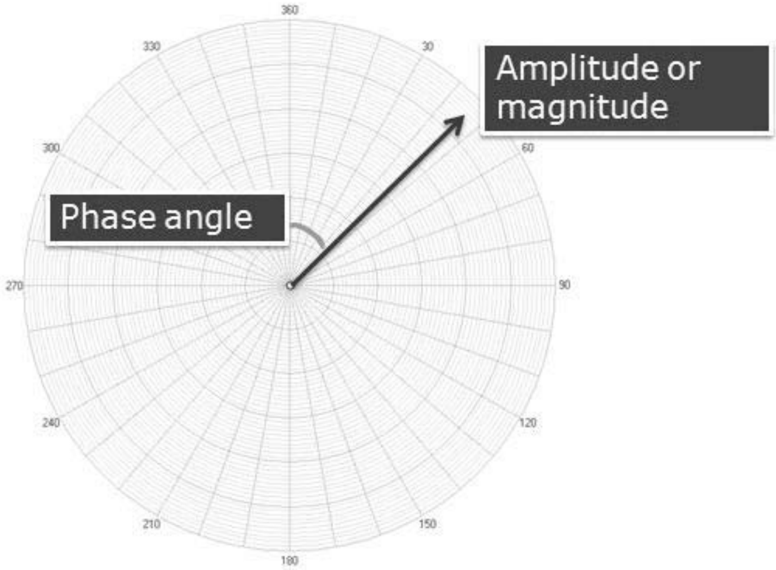


Figure 10-16

When presented this way, the magnitude/phase pairs are called vectors. We draw a line the length of the magnitude at an angle of the phase reading. We can use the format regardless of whether we measure the vibration in units of displacement, velocity or acceleration. The only rule is that we must use the same units (and the same sensors mounted in the same locations) for the entire test.

Measurement setup

The first step is to set up your measuring equipment (data collector) so that you can take a magnitude (i.e. vibration amplitude reading) and phase reading at the running (turning) speed of the machine. The sensor will be positioned so as to measure the highest amplitude, which is normally in the horizontal direction.

Figure 10-17

Original balance run

With the machine is running at normal speed, the magnitude, phase and RPM are recorded. This is called the "original reading". The magnitude and phase are plotted on the polar graph paper. In this example, the magnitude was 5 mils, and the phase reading was 45 degrees. It is labeled "O" for original unbalance.

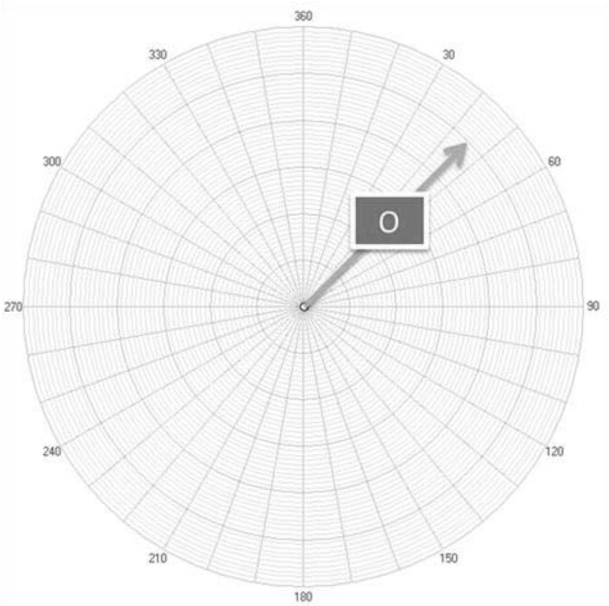


Figure 10-18

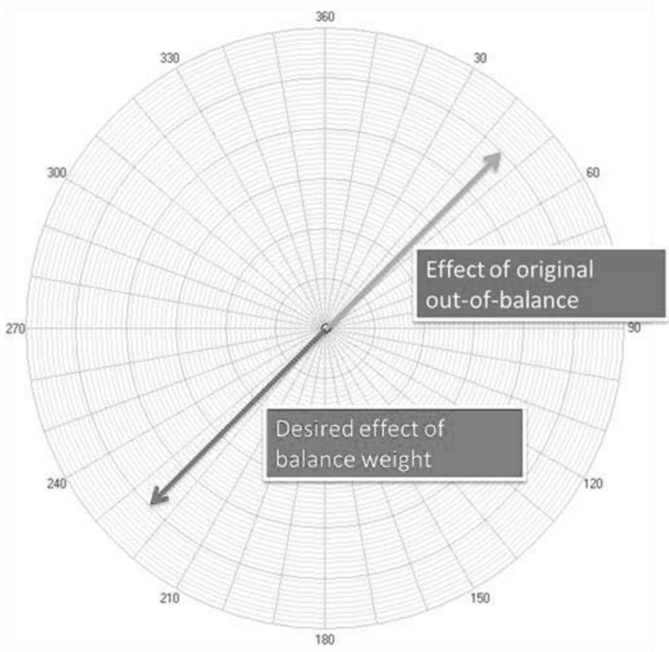


Figure 10-19

Add the trial weight

Then the machine must be stopped and we add a "trial weight". In a moment we will discuss how to calculate the mass of the trial weight, but we are trying to influence the state of balance of the rotor.

For example, if we add 5 grams of weight and we see that the vibration amplitude reduces by 30% and phase angle changes by 30°, we can then calculate how much weight is required, and the location of the weight, so that the unbalance force is cancelled altogether.

If we add a trial weight that is too small, the vibration amplitude and phase will not change very much, and any calculation we perform will be very inaccurate. On the other hand, if the weight is too large, and we happen to place it where the unbalance is greatest, we could create a damaging and potentially unsafe situation.

Balance weights come in a large variety of shapes and sizes. Some are designed so that they are easily added for the "trial run", but they must then be affixed permanently at the conclusion of the balance job.

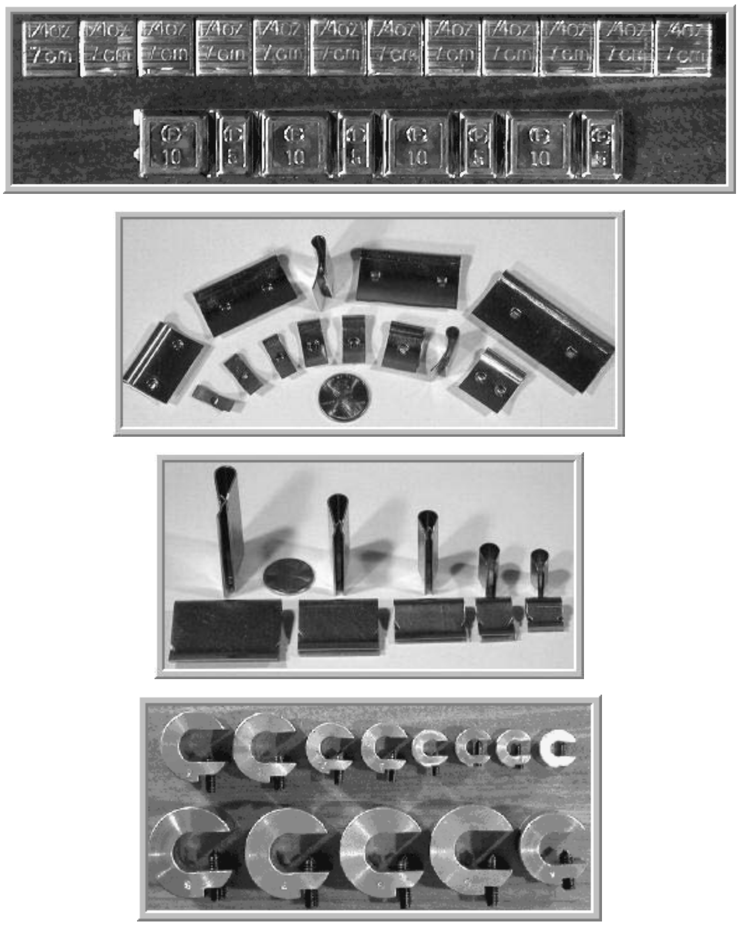


Figure 10-20

It is also very important to have a good set of scales so that you can weigh the trial and final balance weights.

Figure 10-21

Selection of the trial weight is very important. As a general guideline (Wowk), add a weight that produces a force of 10% of the rotor weight. This criterion was originally developed for flexible rotors where the forces may be amplified as the rotor passes through the first critical speed. For rigid rotors, you could safely use 2-3 times this mass.

To calculate the forces produced by the trial weight, you can use these formulas

Imperial:

$$F = 1.77\ x\ W_t\ x\ R\ x\ \left(\frac{RPM}{1000}\right)^2$$

W_t=Trial weight (oz)

R=Radius of trial weight (inches)

Metric:

$$F = 0.01\ x\ W_t\ x\ R\ x\ \left(\frac{RPM}{1000}\right)^2$$

W_t=Trial weight (grams)

R=Radius of trial weight (cm)

As noted, this force should be equal to 10% of the rotor mass, therefore, the “F” in the above equations should be equal to the rotor mass (in pounds or kg respectively) x 0.1.

The following equation can also be used to calculate the mass of the trial weight:

$$W_t = 56{,}375\frac{W_R}{N^2 r}$$

W_T=trial weight (oz)

W_R=Static weight of the rotor (lb)

N=Speed of the rotor (RPM)

r=Radius of trial weight (inches)

For shafts rotating in the range 1200 to 3600 RPM, this equation can be simplified to:

$$W_t = 0.004\frac{W_R}{r}$$

In metric this simplified to:

$$W_t = 30\frac{W_R}{r}$$

W_T=trial weight (grams)

W_R=Static weight of the rotor (kg)

r=Radius of trial weight (mm)

For example, if the machine had a 90 kg rotor, and the balance weights were positioned at a radius of 350 mm, the trial weight would be 30 x 90 / 350 = 7.7 grams

Selecting the position for the trial weight

Many people simply select an arbitrary position for the trial weight; often in line with the tachometer reference. However we can be smarter than that. We can position the weight opposite where we believe the heavy spot is located. If you are using a proximity probe and the proximity probe is positioned in-line with the phase reference (optical-tach, laser-tach or keyphasor), then we can simply place the trial weight opposite the unbalance vector. That is, if we measure a phase angle of 45° we should place the weight at 225° (45+180). If the rotor is rigid, and there are no significant phase lags due to the mechanical system or electronics, then we are bound to reduce the vibration on the first shot. Not only will this help our calculations, but it will ensure a safe trial run – if we happen to place the trial weight in-line with the out-of-balance weight (for example at 45°), then we would cause the vibration levels to rise considerably, and possibly to dangerous levels.

Trial run

With the trial weight carefully installed on the machine, we run it up to the same speed, wait for the reading to settle, and then again record the magnitude and phase at the running speed.

If the magnitude has not changed by 30% or more, and if the angle has not changed by 30 degrees or more, then we will have to stop the machine and replace the trial weight with a larger weight.

Let's assume the new reading was 4 mils and 130 degrees. We add this to the polar plot. This vector represents the vibration level due to the original unbalance and the trial weight, so it is labeled "O+T".

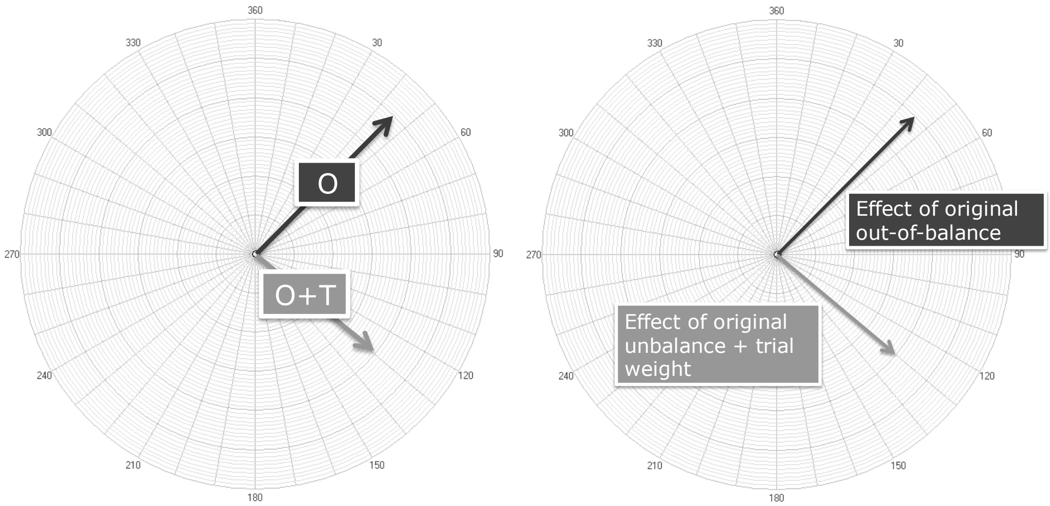


Figure 10-22

Now we draw a line from the end of the original “O” vector to the end of the “O+T” vector. This represents a **subtraction of the two vectors**, leaving just the influence of the trial weight alone, so we mark this vector “T”.

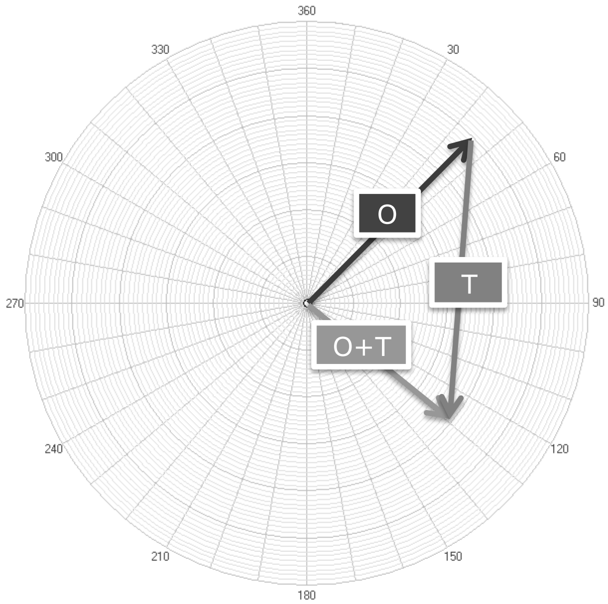


Figure 10-23

The new vector represents the vibration due to the trial weight alone. If we could balance this machine perfectly and then add the trial weight and measure the vibration, we would measure the same amplitude and phase as represented by this vector.

Now we have to measure the length of the “T” line from the graph paper, and the angle between the trial “T” vector and the original “O” vector. The length of the line is 6 cm (or major divisions), so that represents 6 mils. Using a protractor you can measure the angle as 40 degrees CCW (counter-clockwise).

So the solution requires the new mass to be added to the machine at a position 40 degrees counter-clockwise from where you placed the trial weight. But what should it weigh?

The weight is calculated as a ratio of the original vibration level "O" to the vibration due to the influence of the trial weight – our new vector "T" – multiplied by the mass of the trial weight. It is therefore (5 x 5/6) = 4.1 grams.

If the trial weight is removed

Typically one will remove the trial weight and replace it with the final balance weight or correction weight. It is possible to do the calculations if the trial weight is not removed but we will not cover that in this course.

Now, remove the trial weight and add a new 4.1 gram weight 40° from where we placed the trial weight.

But the big question is; in which direction do we measure the 40°; clockwise or counter-clockwise? This is an area where a lot of balance jobs go wrong.

In modern vibration analyzers that use a photo-tach, ***positive*** phase angles are measured ***against*** the direction of rotation.

Therefore:

- If the rotor is rotating **clockwise**, move **opposite** the O+T shift
- If the rotor is rotating **counter-clockwise**, move **with** the O+T shift

In our example in Figure 10-24, the shaft is turning counter-clockwise, so we have to place the weight 40° against rotation, i.e. clockwise.

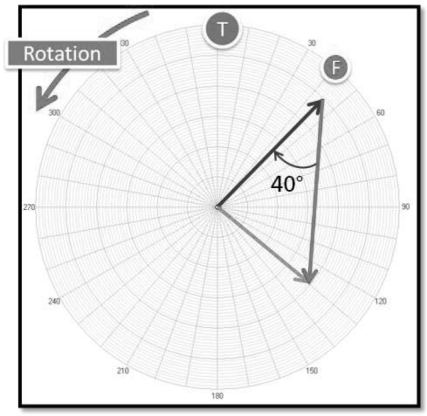


Figure 10-24

Note that if you use a strobe to measure phase, it reports positive phase angles in the direction of rotation.

Therefore:

- If the rotor is rotating **clockwise**, move **with** the O+T shift
- If the rotor is rotating **counter-clockwise**, move **opposite** the O+T shift

Residual unbalance

At this point we know what the unbalance is. We have just been told to add 3.3 grams. If we are adding that weight at a radius of 12 inches, or 300 mm, then our unbalance is
39.6 gr-in or 990 gr-mm.

This is $\mathbf{U_{res}}$ – it is the residual unbalance. We could now look up the standards and determine if that is permissible.

We will discuss this more in a bit.

Trim balance

Now we would add the 4.1 grams (we removed the trial weight) at 40° and run the machine again. We are hoping that the vibration afterwards would be very low. But we do not always find that... We often have vibration that is still a little too high.

In this case we measured 3 mil @ 320° (Figure 10-25).

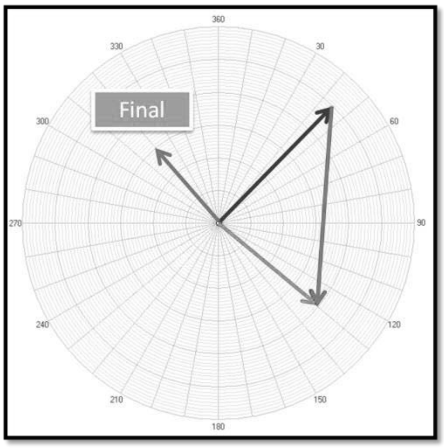


Figure 10-25

We need to perform another step to further reduce the vibration.

This step is called the *trim run*.

If you consider what we have just done, it is very similar to the trial weight process. We measured a vibration and phase, added a weight, and then measured the vibration again. Like the situation we discussed where we left the trial weight on the rotor, in this case we are going to leave the final weight on the rotor.

So now we have to use the vectors to determine a new weight and location.

Now we have re-labeled the plot (Figure 10-26). The “O” original vector is still the original. But the “trial” is actually the vibration measured with the trial weight removed and the final weight of 4.1 grams added.

Note: we will leave the final weight on the rotor, so we are trying to balance out “O+T1” not “O”.

We will label the trim run “T1” (our first trim run) so we have “O+T1” and the vector “T1” is just the effect of adding the 4.1 gram weight.

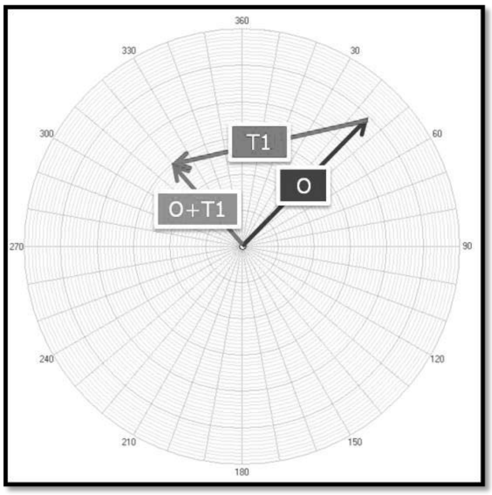


Figure 10-26

Now we are trying to counteract the unbalance force due to the final weight we added to the rotor (Figure 10-27).

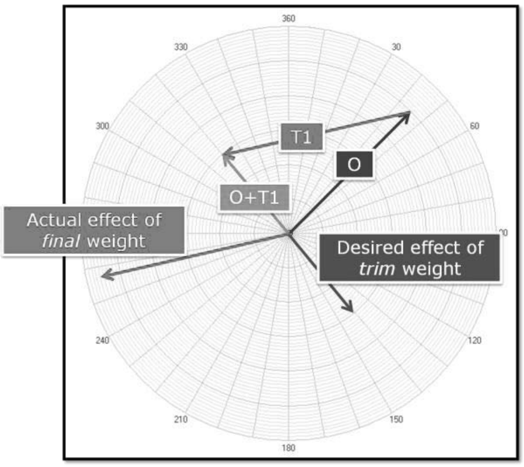


Figure 10-27

You can see that the red vector is too long so we need less weight, and it needs to be moved in the opposite direction by approximately 120°

If we measure the angle we get 116° (Figure 10-28). Measure the lengths of the O+T1 vector (which we know is 3 mils) and T1 vectors and compute the ratio – it tells us how much weight should now be added.

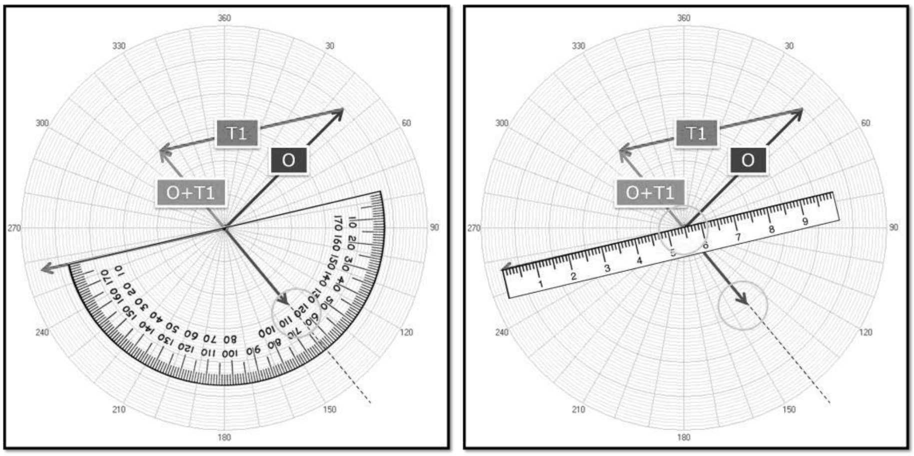


Figure 10-28

The solution is to add the balance weight 116° from where you placed the "final" weight *in the direction of rotation*.

The weight is the ratio:

Final mass x (O+T1)/T1

4.1 x 3.0/5.5 = **1.8 grams**

Note that "4.1" was the final weight we added to the machine; "3.0" was the measured vibration in mils after we added the final weight, and we computed the effect "T1" as 5.5 mils.

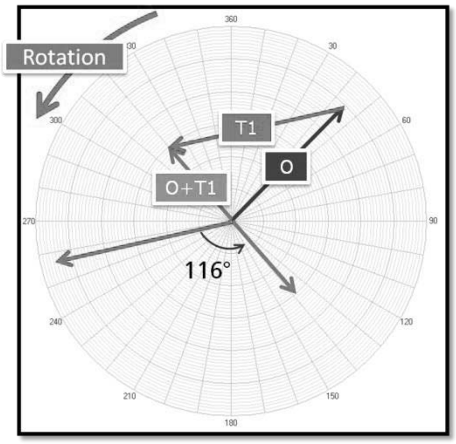


Figure 10-29

Once again we can calculate U_{res} and check if it is less than U_{per} based on G2.5 (or G1.0 or whatever you choose to use).

Of course, we can repeat this process again and again until the vibration amplitude, or U_{res} is low enough.

If you have to repeat it more than a couple of times the rule of thumb is to remove all of the weights and start from scratch. You should also consider other possible causes of your inability to balance the machine. These causes can include the presence of other mechanical faults such as looseness, resonance, misalignment, soft foot etc. Other possibilities include problems with the measurement equipment, phase readings, phase lag etc.

Splitting weights

When you calculate the final weight for the balance job, it must be added to the rotor. In some cases the weight can be added at any angle on the rotor. However some rotors have holes where weights are affixed or blades where weights can be attached. In this situation it may not be possible to add the weight at the precise angle required. The solution is to add two weights to the rotor (on available blades or holes) such that the combination of both weights has the same net effect as a single weight at the desired angle.

For example if the balance solution was to add 6 grams at 75°, yet we have blades at 60° and 120°, we would need to split the weight between the two blades so that the vectorial summation of the two weights is equal to 6 grams at 75°.

Note: The assumption is that the final weights will be placed on the blades at the same radius that the trial weights were added.

The following procedure may be used to determine the correct mass of the weights:

DRAW THE POSITIONS OF THE BLADES AND DESIRED BALANCE WEIGHT ON THE POLAR PLOT

For this example I have drawn the blades, but you simply need to indicate where the two blades (or balance holes) are located. (We have to resolve the 6 gram weight at 75° into two equivalent vectors at the angles of the two blades.)

Add the vector that indicates the desired weight. In our case the mass is 6 grams and the angle is 75°. The weight is drawn at the limit of the polar plot.

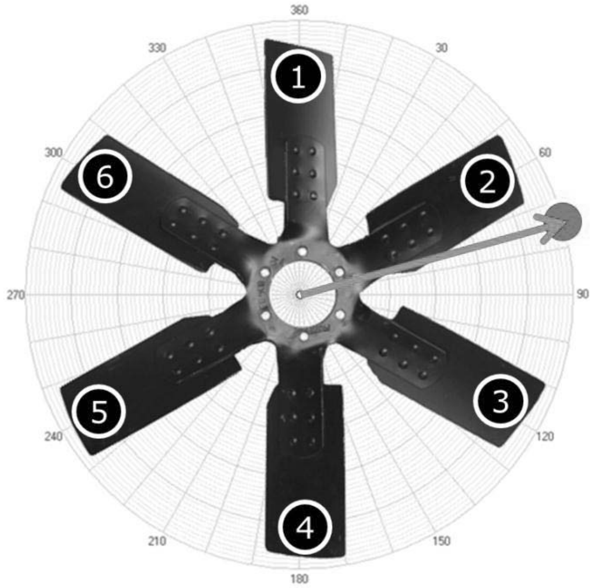


Figure 10-30

DRAW THE EQUIVALENCE VECTORS FOR ONE OF THE BLADES

Draw a line parallel with blade 2 so that it goes through the point representing the balance mass. In this example we drew a line through blade 2 and a parallel line that goes through the weight.

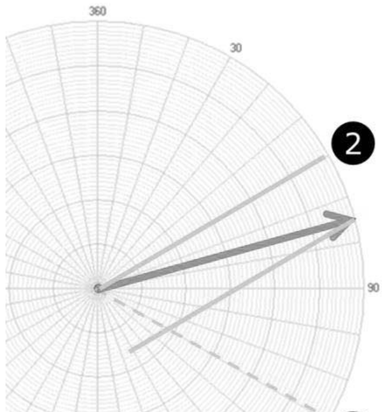


Figure 10-31

DRAW THE EQUIVALENCE VECTORS FOR THE OTHER BLADE

Now we will repeat the process for the second blade. We will draw one line through the blade and a second line parallel with the blade that intersects the balance weight.

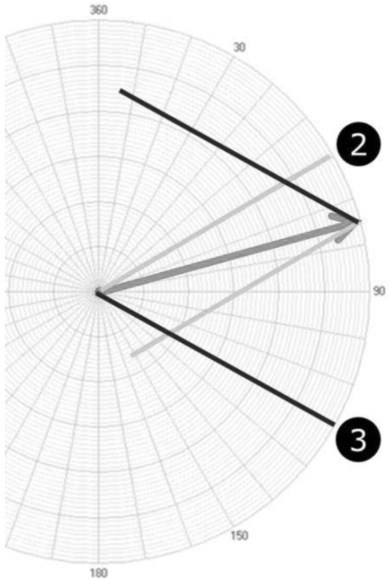


Figure 10-32

ADD VECTORS THROUGH THE BLADES TO THE INTERSECTION OF THE EQUIVALENCE VECTORS

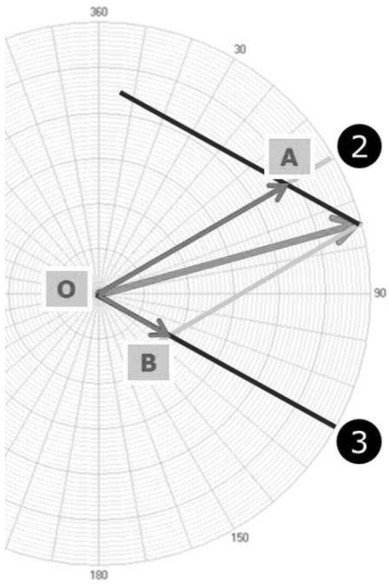


Figure 10-33

DETERMINE THE MASS OF THE EQUIVALENCE WEIGHTS

Now we can compare the length of the two vectors to the length of the vector representing the desired balance mass. You can see that the vector “OA” is longer than the vector “OB”, therefore more mass will be placed on blade 2.

In our example, we have scaled the plot such that each ring is 1 gram. We now look closely at the scale and find that the mass of the weight on blade 2 should 4.9 grams, and the mass on blade 3 should be 1.9 grams.

When 4.9 grams is added to blade 2 and 1.9 grams is added to blade 3, this fan will be balanced; we will have added the equivalent of 6 grams at 75°.

It is very important that you are not confused with the last step. Although we have drawn the OA and OB vectors along the blades, this does not indicate where on the blades the balance weights will be added. They must be added at the same radius as the trial weights. The vectors are simply used to help us compute what the mass of the weights should be.

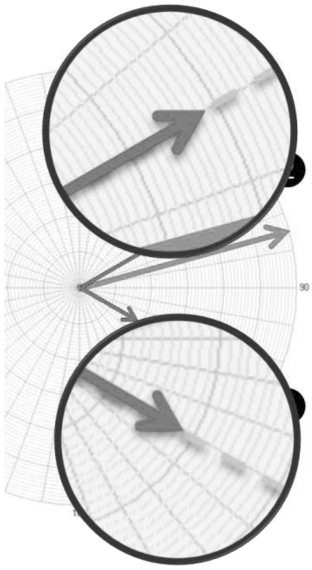


Figure 10-34

Rather than scaling the plot you can just compute the ratio of the lengths of the vectors.

The mass placed on blade 2 is the ratio of O-A to O-X times the mass to be split, and the mass of the weight on blade 3 is O-B to O-X times the mass to be split.

$$Weight\ on\ blade\ 2 = \frac{O-A}{O-X} \times final\ weight$$

$$= \frac{4.9}{6.0} \times 6.0 = 4.9\ grams$$

$$Weight\ on\ blade\ 3 = \frac{O-A}{O-X} \times final\ weight$$

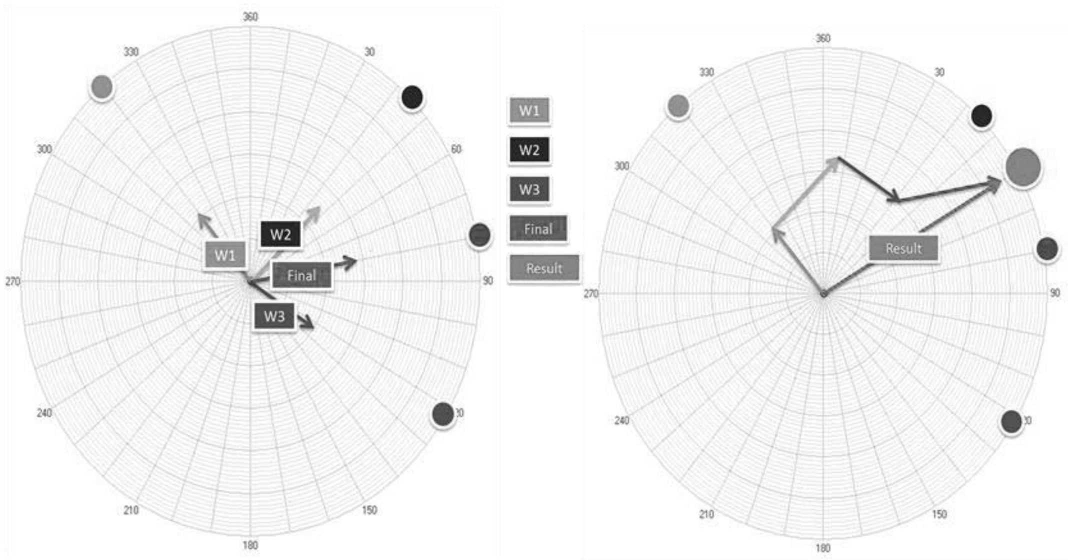
$$= \frac{1.9}{6.0} \times 6.0 = 1.9\ grams$$

In this example, because the plot was scaled correctly, the equations are simple and obvious. But if you simply make the outer ring of the polar plot equal to the mass of your balance weight, then these equations make it very easy to calculate the final weights.

Combining weights

When adding weights to a machine you may find that there are already one or more balance weights on the machine: attached to blades, bolted into holes, etc. There will be situations where it will become necessary to consolidate the existing weights with the weight(s) that must now be added to the machine.

The vector method of summing weights is the reverse of splitting weights which was covered in the previous pages. Instead of having a single weight and splitting it, start with two weights and sum them.



Draw a vector representation of each weight. Draw parallel lines across the tip of the opposite vector. Where these two new parallel lines intersect is the point of the Sum Vector.

Software programs can sum multiple weights at a time and are very useful tools.

There are times when adding correction weights is not permissible for various reasons. But mass can be removed. Most balancing programs have the option in the setup to choose whether to add weights or remove mass. Selecting the 'remove mass' option (if available) forces the analyzer to display the locations where weight should be removed instead of added. There are tables that provide information about weight removed for various diameter and depth of holes on various gauges of metal.

Two-Plane Balancing

Two plane balancing is required when there is dynamic unbalance (i.e. a static component and a couple). Although it is possible to perform a two-plane balance with vectors, it is not often performed that way and we will not discuss that method. Instead we will assume that you have a balancing system or a vibration analyzer with a balancing program.

It is beyond the scope of this course to go into much detail on two plane balancing, but we do feel it is important that you know what it is and when it is necessary.

The first issue we must quickly discuss is whether you *need* to perform a two-plane balance. There are four ways to determine whether a two plane balance is required:

1. You can allow the ISO standards to guide you.
2. You can decide based on a rule of thumb related to the ratio of the length of the rotor to the diameter of the rotor.
3. You can attempt a single plane balance, and if that is unsuccessful, attempt a two-plane balance
4. You can use phase readings to determine how the phase at the two bearings compare.

Rule of thumb

The general rule of thumb for determining whether a two plane balance is required is presented in Table 10-1. It appears in MIL-STD-167A (2005) and in numerous text books.

Rotor characteristics	Speed (RPM)	Type of balance
L/D ≤ 0.5	0 - 1000	Single-plane
	> 1000	Two-plane
L/D > 0.5	0 - 150	Single-plane
	> 150	Two plane

Table 10-1 (The Length and Diameter of the rotor are exclusive of the shaft)

While this method provides a guide, there are many situations where it will not work. It is best to use phase readings to determine if there is a couple component.

Two plane balancing is far more challenging than single-plane balancing. On the face of it, two-plane balancing may seem marginally more difficult as it only requires an additional trial run. While the calculations may be far more difficult, most analyzers can perform them quite easily.

Why is two plane balancing more difficult? The rotor must be rigid. There must be just two bearing supports, and they must be flexible, but not resonant. And the system must be linear. (Wowk)

In the single-plane balancing section we provided a rule of thumb regarding when two-plane balancing must be used instead of single-plane balancing. In reality, the issue is simpler. If there is a strong couple effect, or a strong cross-effect between the two bearings, then you cannot use the single-plane method. As long as the machine meets the requirement above, you may use the two-plane method.

It is therefore recommended that you attempt single-plane balance first. There are times when one end of the rotor has much more amplitude than the other end. Some technicians choose to perform a single plane balance on the rough end first and then do the opposite end. Sometimes bringing the rough end into specs reduces the amplitude at the opposite end enough that it does not have to be balanced.

A benefit of the two plane balance is that all planes of the machine are corrected in one process.

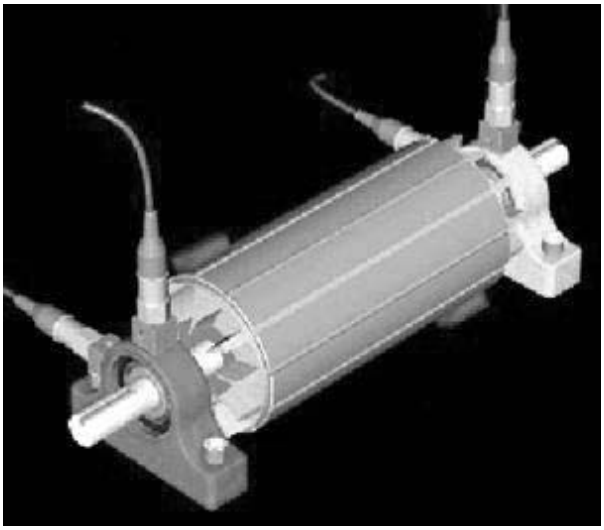


Figure 10-35 A rotor set up for two plane balancing with four sensors

At least one sensor is required for each plane. The sensor is typically mounted in the horizontal plane because there is likely to be greatest movement in the horizontal axis. Most software packages can perform calculations for up to two sensors at each plane.

Two plane balancing procedure

1. The machine is run at normal speed
2. Amplitude and phase is recorded at both points
3. The machine is stopped. A trial weight is added to one plane.
4. The machine is run and readings are taken at both points.

5. The machine is stopped. The first trial weight is removed. A trial weight is added to the other plane.
6. The machine is run and readings are taken at both points.

The original run

After following all of the safety procedures discussed previously, you will take vibration readings on each bearing. These readings indicate the original unbalance state.

It is important to remember to double-check that the machine is out-of-balance and not misaligned or suffering some other fault condition that generated high vibration.

Trial run one

Now the machine is stopped and a trial weight is added to **one** of the balance planes. You must add enough weight to the shaft so that, ideally, the amplitude and phase will both change sufficiently for the balance program (or your vector plots if you wish to do it that way) to be able to accurately determine what influence that mass at that location had on the balance of the shaft. You would hope for a minimum of 15% change in amplitude and/or 15° change in phase angle.

You should therefore perform the trial weight calculations that have been discussed in the single-plane section.

The trial weight can be placed at any angular position on the shaft. You will need to record that information in the balance program. The simplest approach is to place the weight at 0°. Alternatively you can place the weight opposite the high spot.

Trial run two

Now you must remove the first trial weight (some balance programs allow you to leave it on) and add a weight to the second plane.

As just discussed, the balance program will measure the amplitude and phase and determine how much that mass influenced the balance state.

Balance calculation

The two-plane balance programs use an "influence coefficient" method. It is designed to determine the optimum position for a weight on the two balance planes that will minimize the vibration levels. Some programs allow you to optimize the results over different speeds, and they allow you to give greater preference to one end of the machine or the other.

The bottom line is that you will be given a mass and angle for each plane. You must then install those weights, after removing the trial weights, and run the machine again to see if the vibration amplitude is sufficiently low (see the balance grades section).

Trim run

If the amplitudes are not low enough, the balance program will suggest new masses and positions for trim weights. You typically have the choice of either leaving the "final" weights on the machine or removing them before adding the trim weights.

If the system is linear then this method should give you good results.

Balance standards

Balance standards exist and are commonly used to specify a balance job. Using a balance standard does provide repeatability and accountability. Manufacturers and corporations may also have their own criteria for balance quality.

There are two types of standards: standards based on the final amplitude of vibration, and standards that specify the "permissible residual unbalance". The residual unbalance is the unbalance that remains once the balance job is complete.

It is important to have a target when balancing a machine and to understand what balance standards apply to you and your facility. It is also important to document the balance job in terms of the original unbalance, the amount of residual unbalance remaining after the machine has been balanced and the standard used to determine that the balance job was successful.

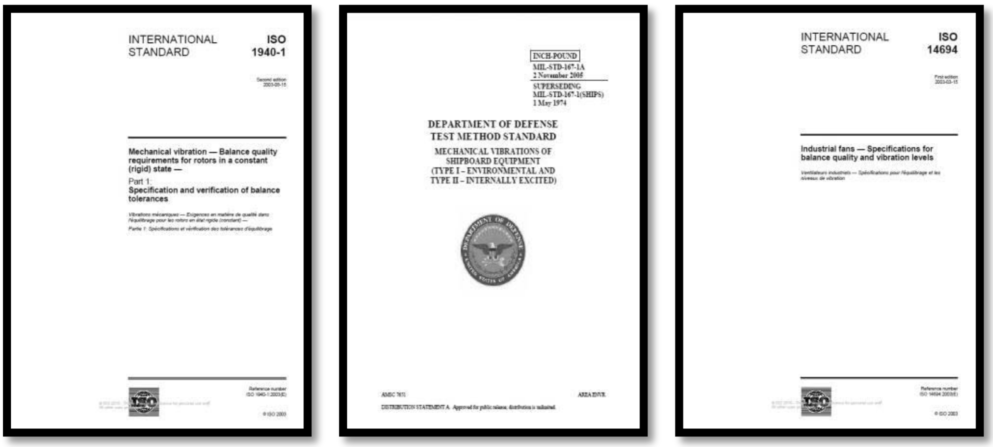


Figure 10-36

Chapter 11

Diagnosing Misalignment

Objective:

- Diagnose Angular Misalignment in waveform and spectra
- Diagnose Offset or Parallel misalignment in waveform and spectra
- Describe the phase conditions for Angular and Offset or Parallel misalignment
- Describe the phase characteristics of misalignment vs. unbalance
- Define Soft Foot
- Explain two effects of soft foot condition
- Identify misalignment in belt driven machinery

Misalignment

Misalignment is the root cause of the majority of machine breakdowns: bearing failures, and damaged seals, shafts and couplings. In fact, it is widely believed that 50% of machine failures are due to misalignment.

Precision alignment significantly increases the life of machines. Misalignment adds load in the form of stresses and forces to the bearings. A 20% increase in load cuts the bearing life in half. Doubling the load reduces the life to 1/7th of its design life.

Bearing life is often described as the L_{10} life factor ≈ 1/ΔRPM x [1/ΔLoad]^3

Misalignment damages seals and bearings. Shafts and couplings can break. The downtime, parts, and labor are very expensive - and avoidable.

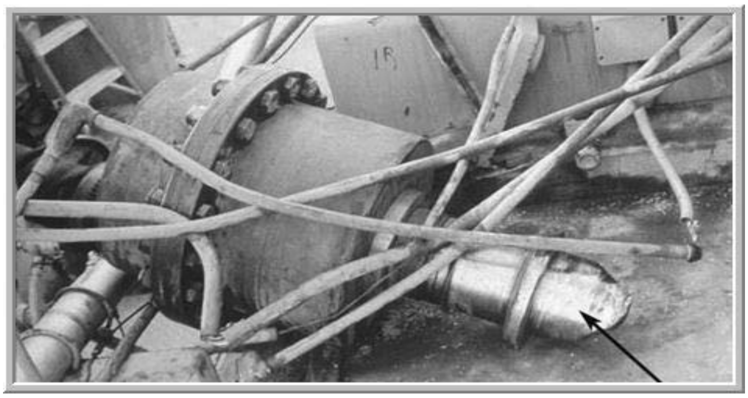


Figure 11-1 - Misalignment is responsible for 50% of machinery failures

Seals are high cost items, often costing up to a third of the total pump cost. Misalignment also causes seals to fail prematurely due to the increased load. Seals do not tolerate misalignment: face rubbing, elevated temperatures, and ingress of contaminants quickly damage expensive components. The life of the seal can be reduced to 30-50% of design life.

If the two shafts of a machine are correctly aligned, there is less stress on the seals, bearings, shafts and couplings of the machine. The machine runs more smoothly, and power consumption can be reduced. All of these factors contribute to increased life, and thus less likelihood of unplanned downtime and catastrophic failure.

Misalignment defined:

"Shafts are misaligned when their rotational centerlines are not collinear when the machines are operating under normal conditions"

Unless special precautions are taken in installation the shafts will not be collinear. There will be misalignment. The shafts will be vertically or horizontally offset from each other and there will be an angle between the two shaft centerlines.

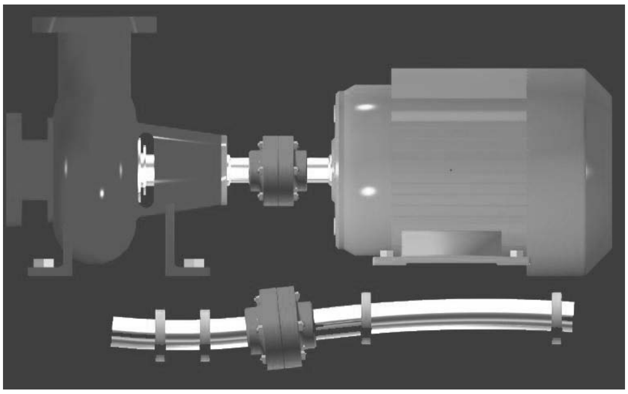


Figure 11-2 - Misalignment forces shafts to bend and flex

Causes of misalignment

There are many causes of misalignment. Usually more than one cause is the source of misalignment found in machinery. Machines should be checked for these possible causes.

- Inaccurate assembly of components, such as motors and pumps.
- Relative position of components shifting after assembly
- Distortion due to forces exerted by piping
- Distortion of flexible supports due to torque
- Temperature induced growth of machine structure
- Coupling face not perpendicular to the shaft axis
- Soft foot, where the machine shifts when hold down bolts are tightened.

The top two machines in Figure 11-3 are viewed from above showing the two types of misalignment. The first is Angular misalignment and the second shows Parallel or Offset Misalignment.

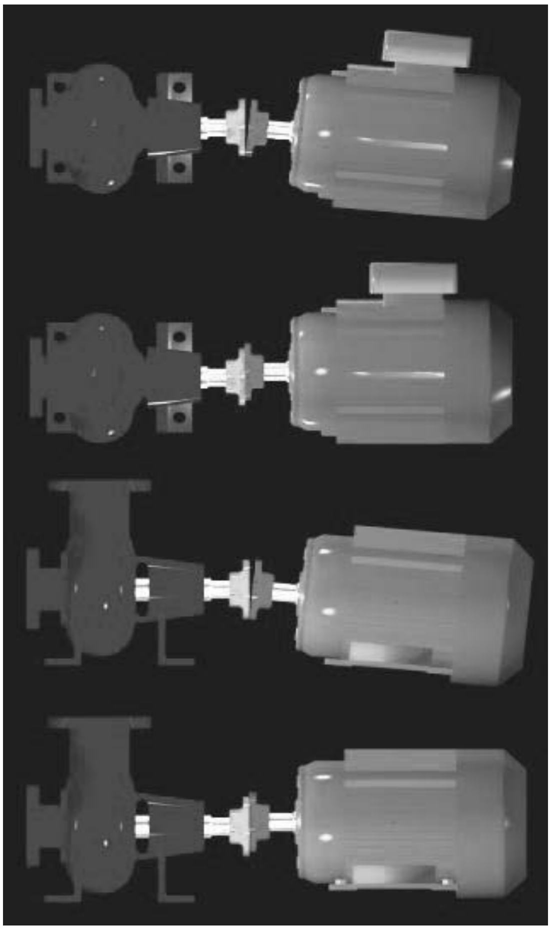


Figure 11-3 - Types of Misalignment

The bottom two illustrations are from the side view of the machine. The first illustrates the Angular Misalignment and the bottom one illustrates Parallel or Offset Misalignment.

The only way to ensure the shafts are collinear is to first make sure the machine has good foundations, there are no soft foot problems, and there are no looseness, runout, or other problems, and then take such measurements (with dial indicators or laser alignment tools) to determine where the shaft centerlines are located relative to one-another. Then typically one of the machine components is moved so that the shaft centerlines are collinear.

Diagnosing Offset (Parallel) Misalignment

When the misaligned shaft centerlines are parallel but not coincident, then the misalignment is said to be parallel (or offset) misalignment.

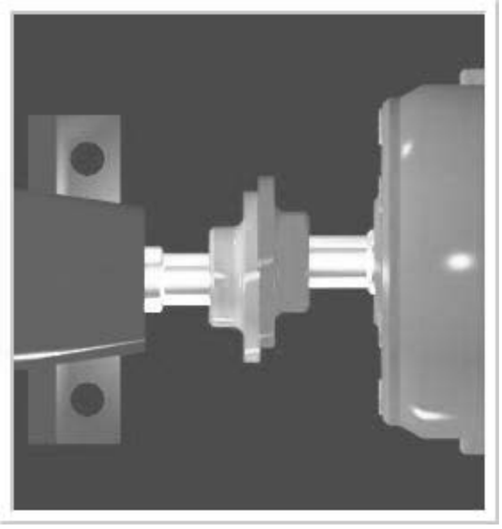


Figure 11-4 - Offset or Parallel misalignment

Parallel misalignment produces both a shear force and bending moment on the coupled end of each shaft.

High vibration levels at 2X as well as 1X are produced in the radial (vertical and horizontal) directions on the bearings on each side of the coupling. Most often the 2X component will be higher than 1X.

Axial 1X and 2X levels will be low for pure parallel misalignment.

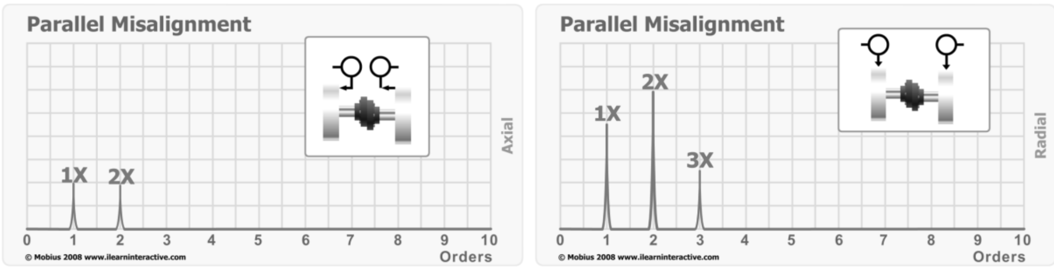


Figure 11-5 - Offset or Parallel misalignment

The vibration is 180° ±30° out of phase across the coupling in the axial direction, and out of phase in the radial direction.

Diagnosing Angular Misalignment

When the misaligned shafts meet at a point but are not parallel, then the misalignment is called angular (or gap) misalignment.

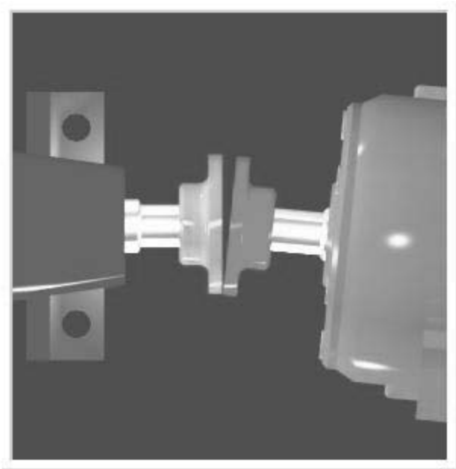


Figure 11-6 - Angular or Gap misalignment

Angular misalignment produces a bending moment on each shaft, and this generates a strong vibration at 1X and some vibration at 2X in the **axial** direction at both bearings.

There will also be fairly strong radial (vertical and horizontal) 1X and 2X levels, however these components will be in phase. See Figure 11-7.

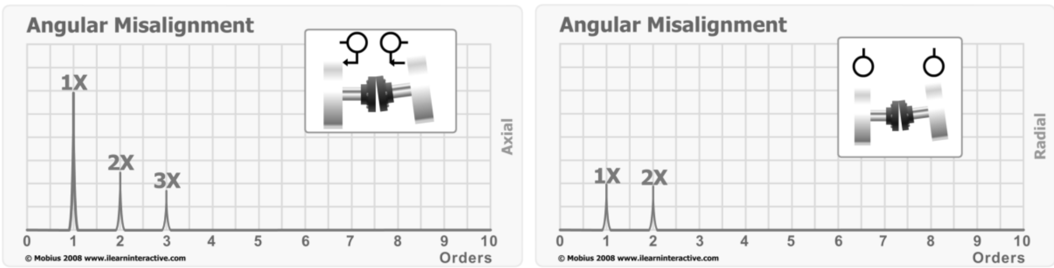


Figure 11-7 - Angular misalignment patterns in Axial and Radial directions.

The vibration is 180 degrees out of phase across the coupling in the axial direction, and in-phase in the radial direction.

Misaligned couplings will usually produce fairly high axial 1X levels at the bearings on the other ends of the shafts as well. This means that you can collect the axial reading on the outboard bearings of the motor or pump, for example, and still detect misalignment.

Almost all misalignment conditions seen in practice are a combination of these two basic types.

Diagnosing Common Misalignment

Most misalignment cases are a combination of parallel and angular misalignment. Diagnosis, as a general rule, is based upon dominant vibration at twice the rotational rate (2X) with increased rotational rate (1X) levels acting in the axial and in either the vertical or horizontal directions.

Flexible coupling problems will add 1X and 2X harmonics. In reality, misalignment produces a variety of symptoms on different machines; each case must be individually diagnosed, based upon an understanding of the causes.

Severe Misalignment

In addition to the 1X and 2X peaks, a strong 3X peak is also often associated with misalignment. (Higher order harmonics are also common when misalignment is more severe.)

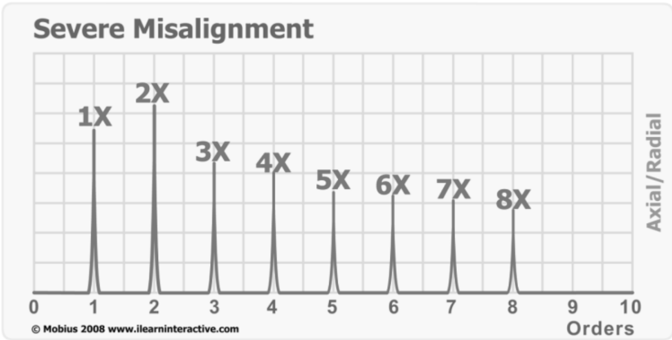


Figure 11-8 - Severe misalignment produces 1x harmonics

It can be confused with looseness, however the harmonics will not be as strong and the noise floor will not be raised.

Misalignment or Unbalance?

One way to distinguish between misalignment and unbalance is to increase the speed of the machine. The vibration level due to unbalance will increase in proportion to the square of the speed, whereas vibration due to misalignment will not change. Of course, this is not a test that can be performed on all machines.

Another test that can be performed is to run a motor uncoupled. If there is still a high 1X, then the motor is out of balance. If the 1X goes away, then either the driven component has the unbalance problem, or it was a misalignment problem. Every little test can provide additional clues.

Figure 11-9 - Misalignment or Unbalance?

Always remember that overhung components will also generate high 1X axial vibration, and a bent shaft can be easily mistaken for misalignment. So think carefully about the machine, and ensure that unbalance and bent shaft is ruled out before making a misalignment call.

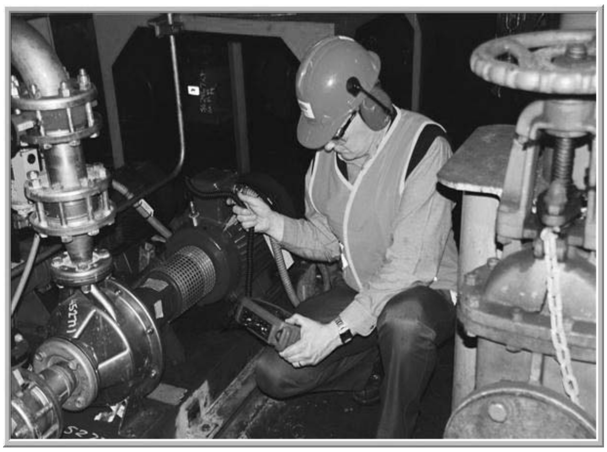


Figure 11-10 - Overhung rotors generate a 1x peak in the axial direction.

Beware of False 2x Peaks

Whenever there is a high 2x peak on a 2 pole motor, first make sure it is not 2x line frequency (120 Hz or 100 Hz).

At first glance this data appears to have a high 2x peak, but is actually a mix of a 2x peak and twice line frequency peak. If it is found that the 2x peak is indeed 120 Hz, investigate the reason for this peak, but also continue investigating misalignment.

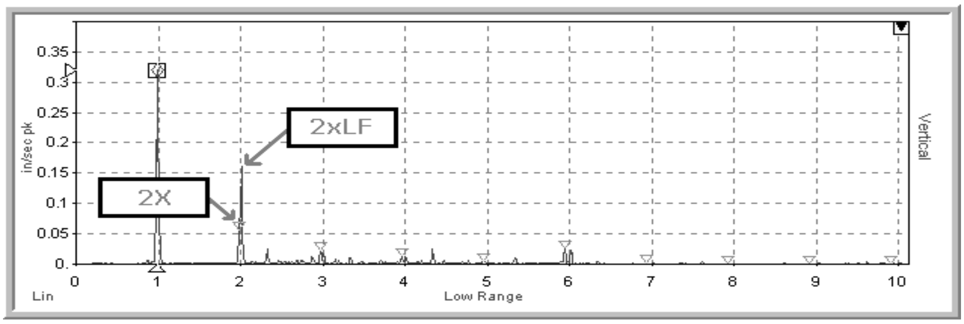


Figure 11-11 - The 2x peak can be twice line frequency in 2 pole motors

Temperature Effects on Misalignment

Due to thermal expansion and contraction, the best alignment of any machine will always occur at only one operating temperature, and hopefully this will be its normal operating temperature. It is imperative that the vibration measurements for misalignment diagnosis be made with the machine at normal operating temperature.

Vibration Analysis of Misalignment

From the description of angular and parallel misalignment, one can see that the 1X, 2X, and 3X frequencies are important, so it is therefore important (as always) to accurately determine the running speed of the machine.

It is also necessary to analyze the vertical, horizontal, and axial data. The axial measurements are very important when attempting to diagnose misalignment.

As noted earlier, both the vertical and horizontal levels can be high, however unlike unbalance, they will not necessarily be equal. In fact, one may be over twice the amplitude of the other.

Be sure and collect an axial reading. Fortunately, it is typically satisfactory to take the axial reading from the outboard bearings (collecting inboard readings can be difficult due to coupling guards and lack of space). Comparison of axial and radial measurements is vital to determining misalignment.

Phase data can always help in the final diagnosis if needed. Remember that phase data differs for parallel and angular misalignment.

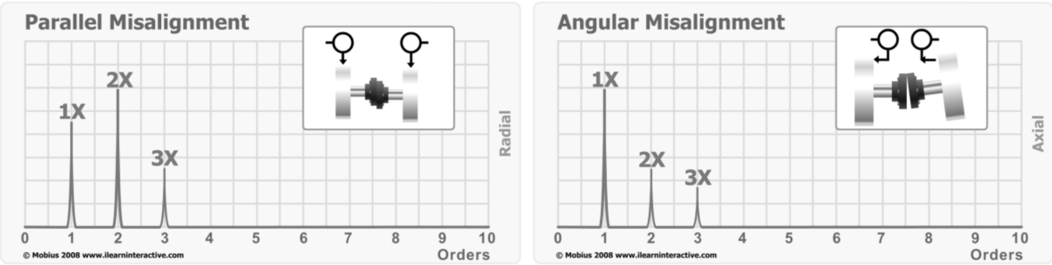


Figure 11-12 - Phase characteristics for Parallel and Angular misalignment.

The time waveform data can be useful for identifying misalignment. When there is binding in the coupling the waveform can exhibit “M & W” shapes. See below.

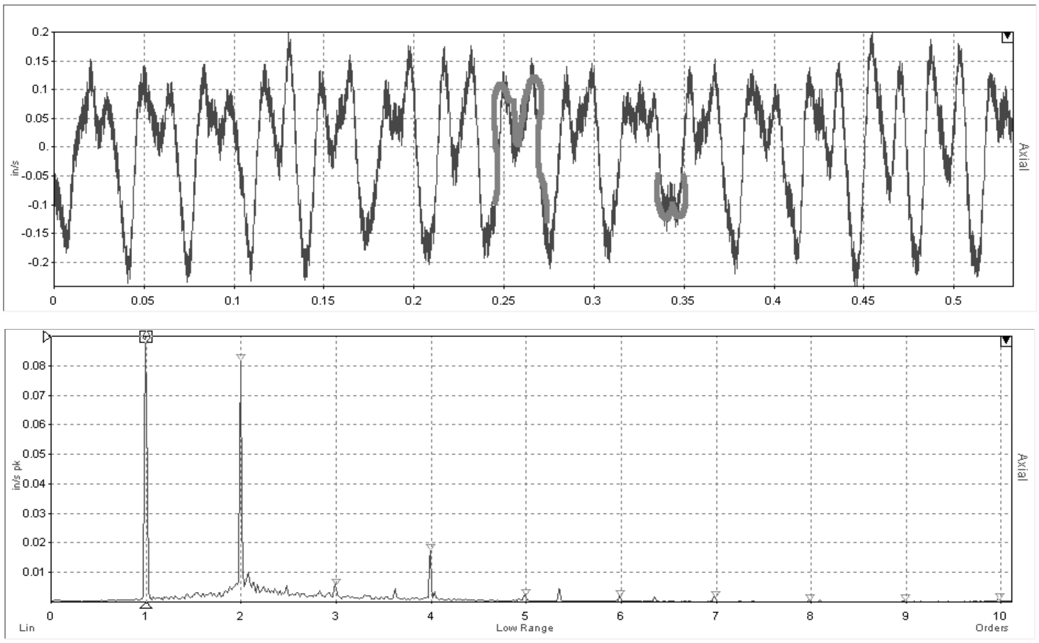


Figure 11-13

Case Study: Cooling Water Pump #2

This machine is a 20 HP AC electric motor driving a centrifugal pump through a flexible coupling. Nominal motor rotation speed is 3550 RPM. There are 6 vanes on the pump impeller.

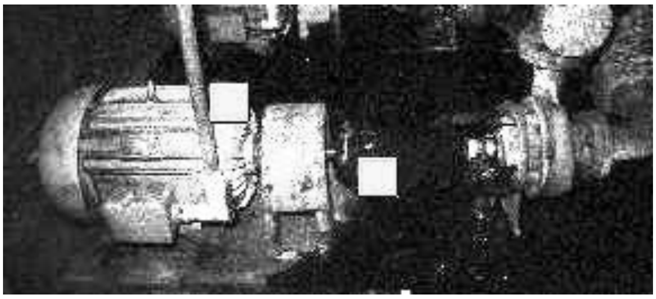


Figure 11-14 - Cooling Water Pump #2

Data from the vertical direction shows a high 1x and moderate to high 2x.

The vertical data from the pump has a high 1X peak, and moderate to high 2X vibration. The high 2X is our first indication.

The Horizontal direction confirms it more. See Figure 11-15

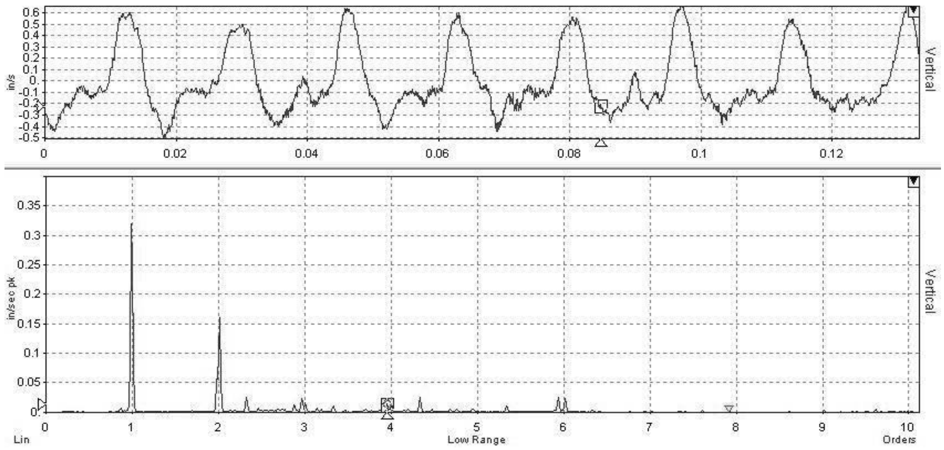


Figure 11-15 - High 1x and 2x in Vertical direction.

Figure 11-16 has a high 1X and 2X, and it is a pure 2X (the harmonic marker sits on top of the 2X peak). There is also a small 3X peak. But what will the axial data reveal?

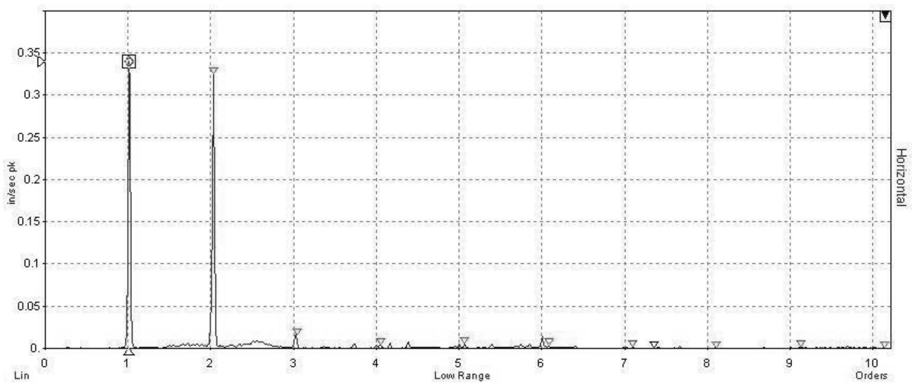


Figure 11-16 - Horizontal has a high 1x and 2x

The axial direction has a high 1x and a moderate 2x.

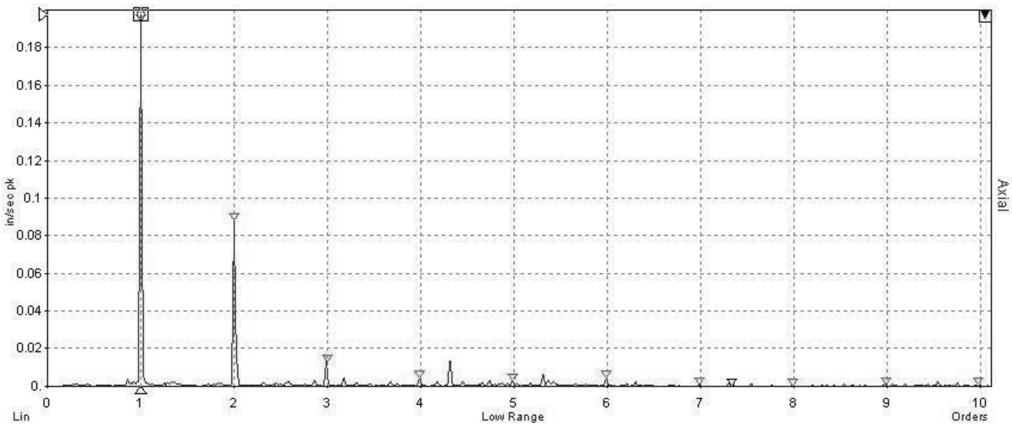


Figure 11-17 - Axial with a high 1x and a moderate 2x peak.

Note that the graph scale is different - the amplitude of the 1X peak is actually lower than the levels observed in the vertical and horizontal axes.

THE DESTRUCTIVE NATURE OF MISALIGNMENT

Inspecting the axial measurement on the motor, the data has many more peaks. Some of them are sidebands around a non-synchronous peak with spacing of 1x turning speed. This is typical of a bearing inner race. This is not really surprising considering the amount of misalignment and the time it has been running out of alignment.

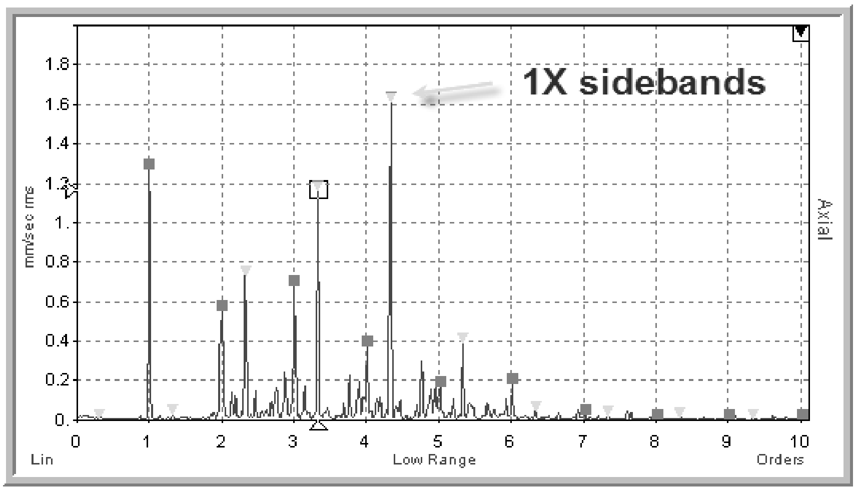


Figure 11-18 - Data from the motor in the axial direction.

Vibration is not always a good indicator of misalignment. Studies have shown that misalignment can exist without the normal patterns in the spectral data. Further, the vibration levels may not increase as misalignment gets worse. The vibration patterns are related to the type of coupling and the speed of operation. But misalignment can still exist and be very destructive. . The solution is to perform precision alignment.

Belt /Pulley Misalignment

If pulleys are misaligned, there will be a high 1X peak in the axial direction, of the motor and/or fan (or other driven component). Data taken from the motor may have a high peak at the fan 1X frequency, and data from the fan may have a large peak at the motor 1X frequency.

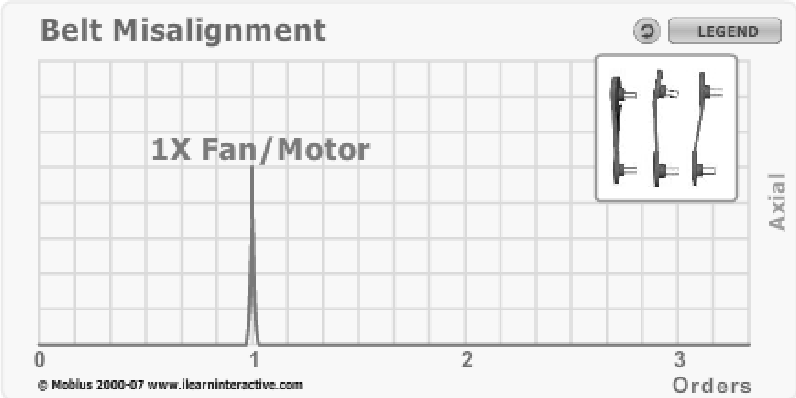


Figure 11-19 - Belt misalignment has a high 1x of opposite component in axial direction

Phase data in axial direction will be 180° out from the driver to the driven component.

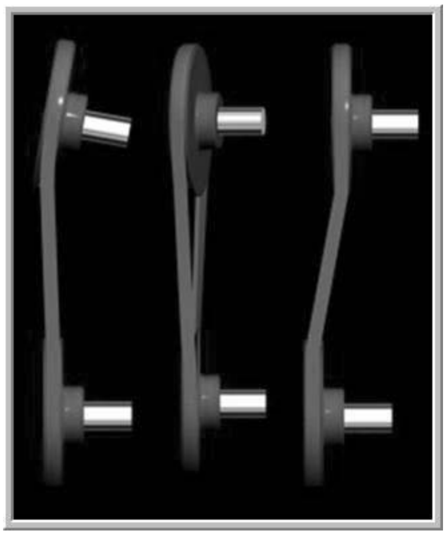


Figure 11-20

Soft Foot

Soft Foot occurs when not all of the feet on a machine are flat on the base. They may not be all the same height or it may be bent.

If a soft foot condition exists, there will be a high 1X peak in the radial direction, and often a 2X and 3X component as well. Because the soft foot condition can deflect or distort the frame of the motor (or pump, etc.) other signs may be present. A motor will exhibit twice line frequency (i.e. 120 Hz or 100 Hz). A pump may have a higher peak at the vane/blade pass frequency due to uneven clearances between the rotating vanes and the diffuser vanes.

Figure 11-21 - Soft foot condition can cause 1x, 2x, and 3x peaks as well as twice line frequency.

Bent Shaft

A bent shaft is commonly caused by uneven heating in the rotor, due to a bad rotor bar. If the bend is permanent, it can be balanced out to bring the vibration levels back in line. Bent shaft is often confused with misalignment.

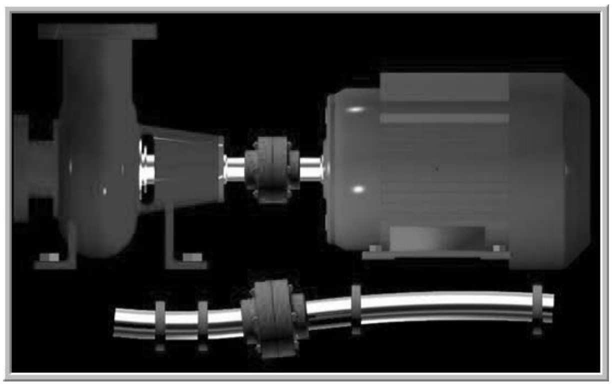


Figure 11-22

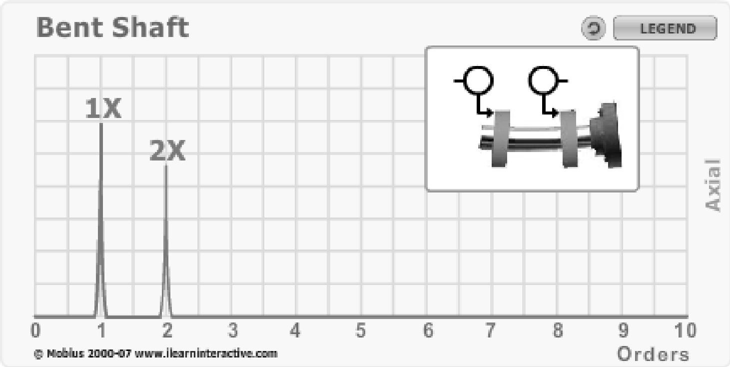


Figure 11-23 - High 1X vibration, phase is 180° out of phase

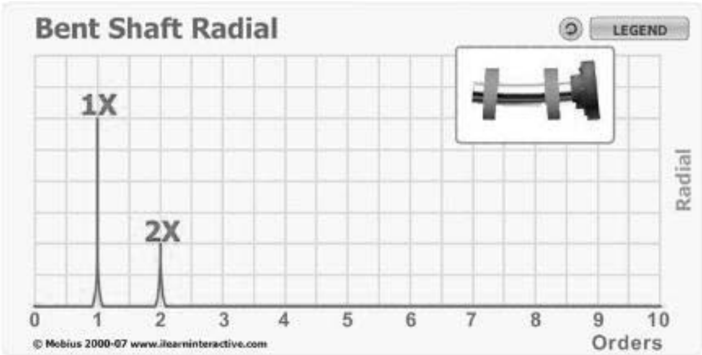


Figure 11-24 - 1X peaks will be present in radial direction

Cocked Bearing

A cocked bearing, which is a form of misalignment, will generate considerable axial vibration. Peaks will often be seen at 1X, 2X, as well as 3X.

Cocked bearing can result from poor installation practices in thermal fitting or press fitting of the bearing.

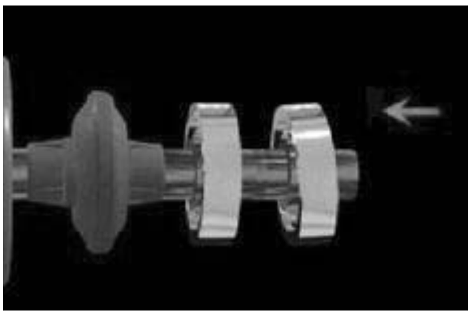


Figure 11-25

A cocked bearing has a strong 1x vibration which can cause it to be confused with misalignment or unbalance in an overhung pump or fan. The presence of a high 2x and 3x in the axial confirm a cocked bearing. The bearing must be re-installed.

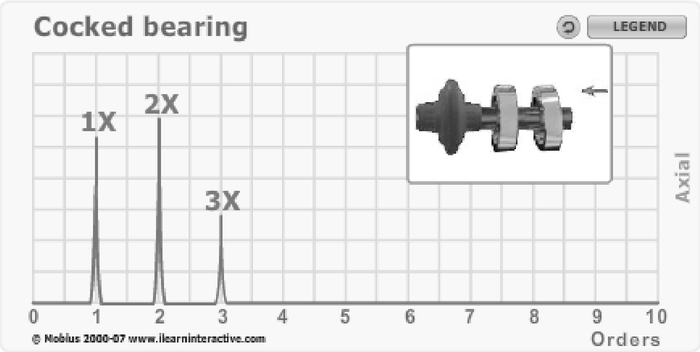


Figure 11-26

The bearing may be cocked on the inner race or outer race and each has its own characteristics. If it is cocked on the inner race, it will wobble with each rotation. If cocked on the outer race it will have an angle relative to a position.

Phase is a good indicator and can be checked in the axial direction comparing readings across the shaft such as side to side or top to bottom. In both cases the phase will be 180° ±30° out from each other.

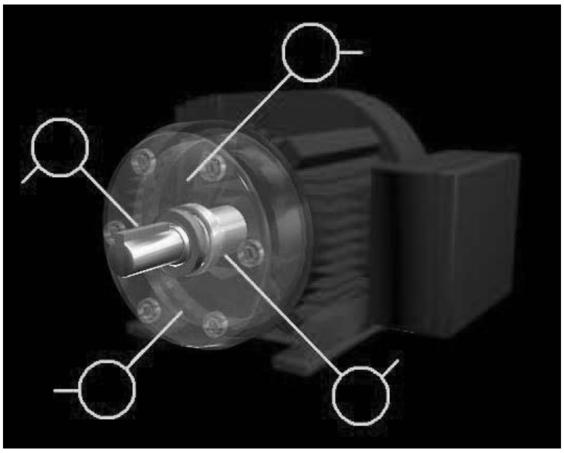


Figure 11-27

Chapter 12

Shaft Alignment

Objectives:

- Understand the importance of precision shaft alignment
- Understand the limitations of dial-indicator alignment
- Understand the importance of pre-alignment checks and soft foot corrections
- Understand reverse-dial and rim-face alignment
- Learn how to move the machine
- Understand thermal growth

Introduction

This chapter of the training course will help you to understand misalignment - understand why it is important to align your machines, and understand how to align your machines.

If you can achieve these goals, and you can put it into practice, your machines will run more smoothly, and your plant will operate more profitably.

There is a lot to learn, but we will take it one small step at a time.

Why is misalignment so important?

We are all under great pressure to increase uptime, reduce costs, and improve product quality. No matter what your role in the organization we can all contribute to these goals. One way you can help is to perform precision shaft alignment on your rotating machinery.

The fact is that misalignment is the root cause of the majority of machine breakdowns: bearing failures, and damaged seals, shafts and couplings. It is widely believed that 50% of machine failures are due to misalignment.

Figure 12-1 The majority of bearing failures are caused by misalignment

Poor lubrication practices, imbalance, resonances and other factors also contribute to machinery failure, but it is widely agreed that misalignment is the major cause.

Bearing damage

Rolling element (anti-friction) bearings are precision components designed to operate with clean lubricant, reasonably constant temperature, and axial and radial forces/loads within design guidelines. When a machine is misaligned, the pre-load and dynamic forces are raised considerably.

Figure 12-2 Misalignment places considerable load on bearings

If you increase the load on a bearing by just 20%, its life is halved. If you double the load on a bearing, you reduce the life to one seventh of its design life. When machines are misaligned, the load on bearings is increased considerably.

When a bearing fails, production may stop and there can be secondary damage, i.e. damage to other parts of the machine due to the bearing failure. The downtime, parts, and labor are very expensive - and avoidable.

Seal damage

Seals are high cost items, often costing up to a third of the total pump cost. Misalignment also causes seals to fail prematurely due to the increased load. Seals do not tolerate misalignment: face rubbing, elevated temperatures, and ingress of contaminants quickly damage expensive components. The life of the seal can be reduced to 30-50% of design life.

Figure 12-3 Seals will fail prematurely due to misalignment

The result is lubricant leakage and other lubrication problems, and in many cases, total seal failure with little or no warning. When the seal fails, production may stop. The seal and bearing will have to be replaced. The total cost of parts, labor and downtime can make this a very expensive failure.

Coupling damage

The impact of misalignment on couplings varies greatly according to the type of coupling used. The forces and friction are detrimental to all couplings, however the signs of damage, and the nature of the damage can vary.

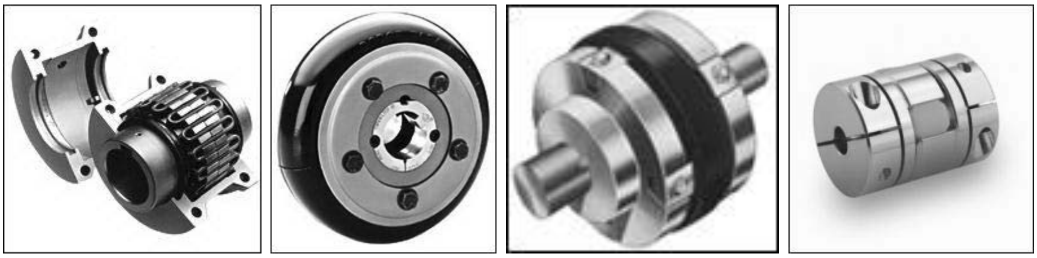


Figure 12-4 Misalignment has different effects on different types of couplings

The rubber or plastic sleeve between the two hubs in flexible couplings can wear - in fact you will often find a small pile of rubber or plastic under the coupling when you remove the coupling guard.

There is a common misconception: "*I use flexible couplings so I do not need to worry about precision alignment*". It is very important to understand that the life of the coupling will be reduced if misaligned, and more importantly, bearings, seals and shafts will still be under increased load and thus likely to fail prematurely.

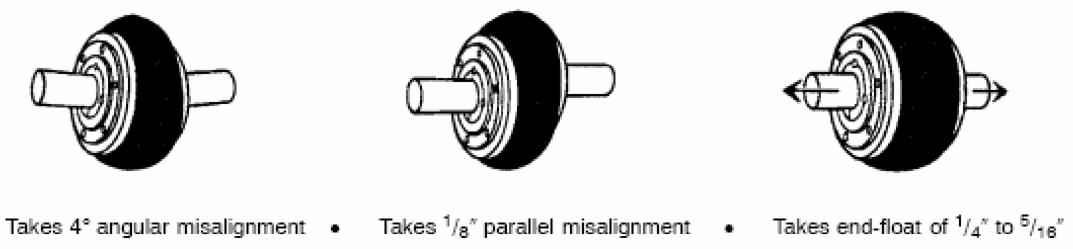


Figure 12-5 Misalignment will reduce the life of flexible couplings

In gear couplings, misalignment results in increased wear on the mating teeth. Under severe misalignment conditions, the load on the teeth will be concentrated to the end of the gear tooth flank. Misalignment can also cause lubrication problems, resulting in metal to metal contact and therefore greatly increased wear.

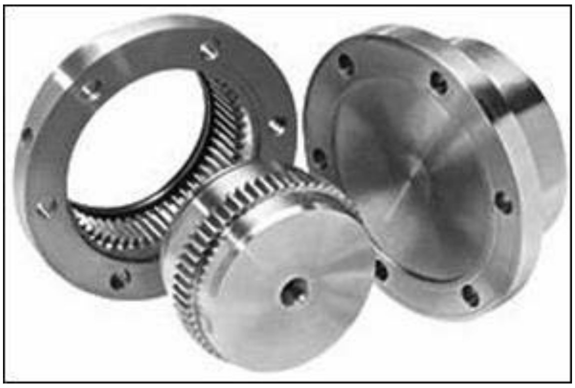


Figure 12-6 Misalignment can increase the wear on gear couplings

If you eliminate misalignment, the machine will provide greater service. Bearings, shafts, seals and couplings will last longer. Unexpected breakdowns cause secondary damage to machines, and the downtime and repair can cost a small fortune.

Vibration

The rotational forces that result from misalignment generate vibration. We have seen how these forces can damage the coupling, seals, shaft and bearing, but the vibration can damage other components - even machines located within close proximity.

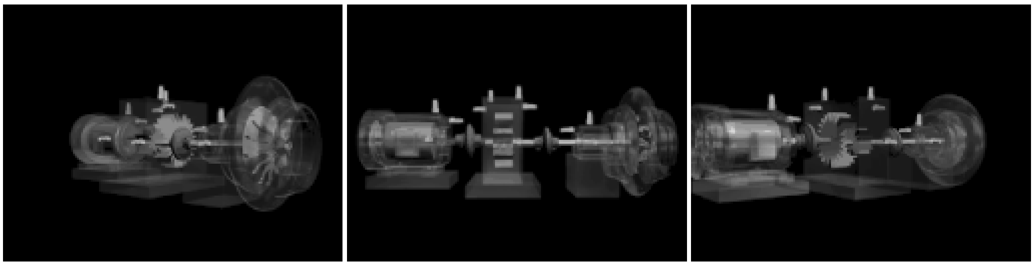


Figure 12-7 Misalignment generates vibration

Have you ever heard of a standby machine that was started, only to quickly fail? This can be due to "brinelling". The bearings in the standby machine are subjected to the vibration of local machines, and that vibration either creates localized wear, or it results in "plastic deformation" of the bearing surface.

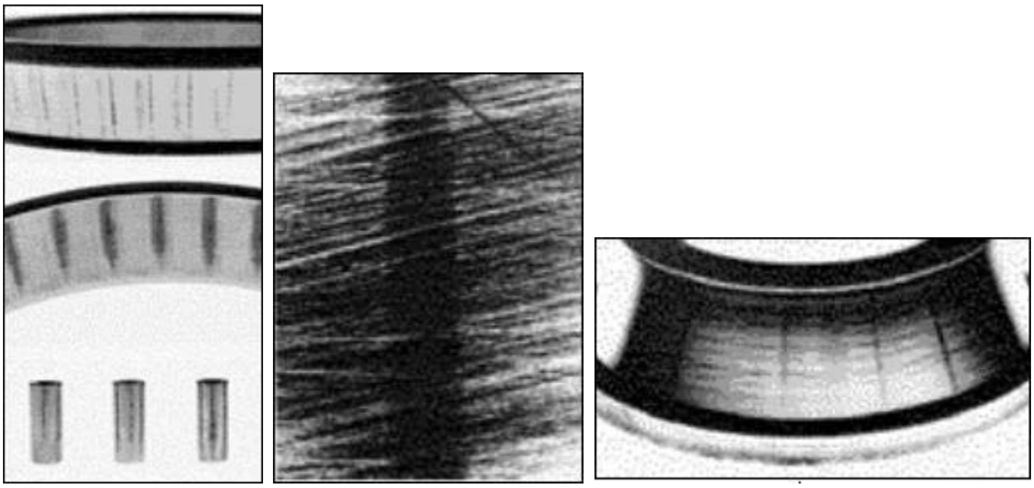


Figure 12-8 Brinelling – bearings are subject to vibration from surrounding machines

Energy consumption

Misaligned machines can also consume more energy. Although studies have produced varying results, it is generally considered that a misaligned machine will consume up to 15% more energy. Large studies have documented savings of between 3% and 8%.

Consider the following equations. If we can reduce the consumption of a 30 HP (22 kW) 460 Volt motor from 36 amps to 32 amps, and we are paying 0.06 per kW Hour then the savings are approximately $1,500.

kW = (Volts x Amps x pf x 1.732)/1000

Annual savings = 8400 x kW x kWH cost

Example kW = (460 x 4 x 0.92 x 1.732)/1000 = 2.931

Example savings = 8400 x 2.931 x 0.06 = $1477

Product quality

Misalignment can also result in reduced product quality in many industries. By reducing vibration levels, and aligning rolls and other items, product quality can be maximized.

Downtime and production capacity

The biggest issue of all is production capacity. If you can increase the reliability of rotating machinery, downtime will be reduced. If you increase the uptime, your plant can increase production which has a major impact on the bottom line.

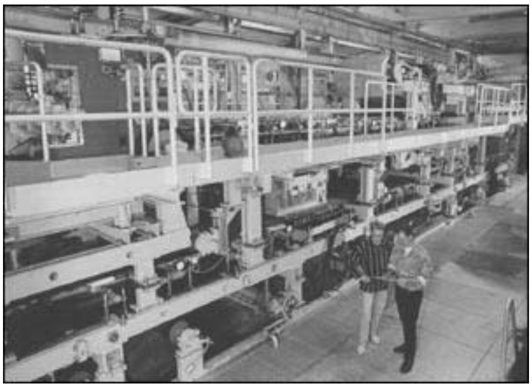


Figure 12-9 Reliability = increased uptime

It is therefore in everyone's best interest to correctly align machines. The extra time and effort required will increase profitability - and job security.

Detecting misalignment

There are two ways to determine if a machine is misaligned: you can look at the maintenance records and alignment records to see if the machine was misaligned, and you can monitor various physical parameters such as vibration and temperature.

First, if a machine has not been "precision" aligned, that is, it was aligned "by eye" or only using a straightedge, then it is very likely that it will be misaligned.

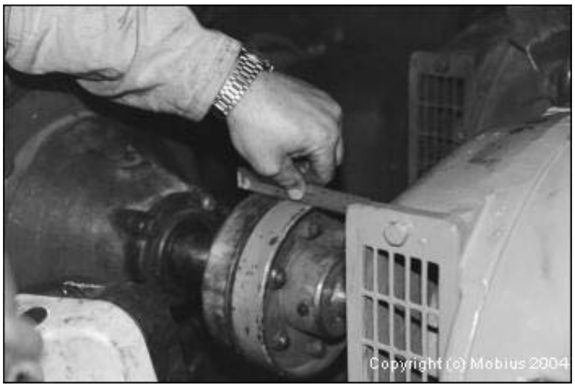


Figure 12-10 Machines will likely be misaligned if precision alignment was not performed

Later we will discuss the alignment procedures, and we will discuss tolerances (which tell us how good the alignment must be to achieve maximum life), but for now it is safe to say that if you are not correctly using dial indicators or a laser alignment system, the machine will not be within tolerance.

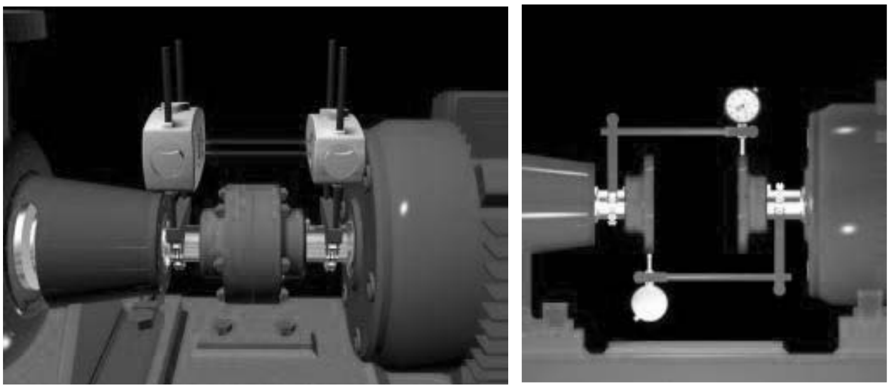


Figure 12-11 Laser alignment (left) and dial indicator (right) systems

We have just described all of the reasons why you should perform shaft alignment: damaged bearings, shafts, seals, and couplings. If you have experienced any of these problems then it is probably due to misalignment.

Other physical signs include loose or broken hold-down bolts, loose shim packs or dowel pins, excessive oil leaks at bearing seals, and loose or broken coupling bolts.

Figure 12-12 Misalignment can be detected via physical signs

You should always perform "root cause analysis" when a machine fails. Root cause analysis is used to determine why a machine failed. It is not good enough to simply repair it and put it back in service - what is to stop the same fault from developing again? Issues like lubrication, imbalance, resonance and misalignment should be considered when a machine fails. And thus when a machine is put back into service, it should be *precision* balanced and *precision* aligned, and the lubrication program should be maintained.

Detecting misalignment

Due to the nature of misalignment, there are at least two physical signs that shafts are not aligned: the machine will vibrate in characteristic ways, and the coupling may get hot which can be detected with non-contact temperature guns or thermal imaging cameras.

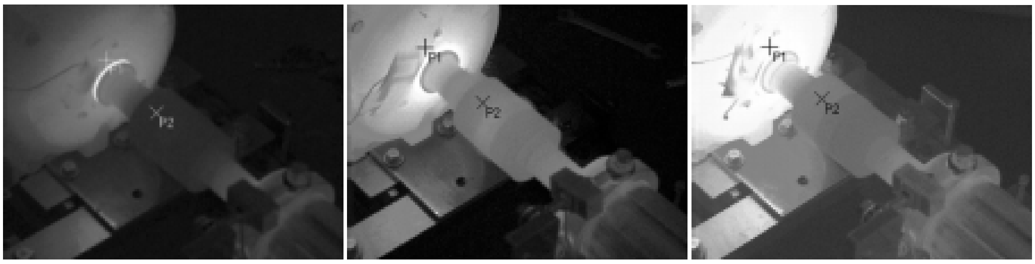


Figure 12-13 Thermal imaging showing heat buildup due to misalignment

Using vibration analysis to detect misalignment

Vibration analysis has been successfully used for many years to detect misalignment. Depending upon the nature and severity of the misalignment (and the nature of the coupling and size/speed of the machine), vibration measurements can help us detect misalignment and assess the severity.

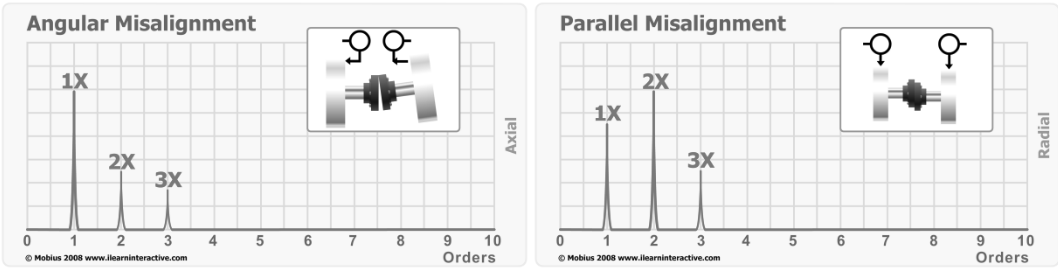


Figure 12-14 Vibration measurements help us detect misalignment

The specific methods used to detect misalignment are covered in the diagnostic section of Category II course, however there are a few quick comments to make:

- The combination of vibration spectra, time waveforms, **and phase readings** give the best indication of misalignment.
- Measurements should be taken in the axial and radial directions. One direction alone is insufficient.
- Simple measurements like rms "overall" readings cannot provide definitive evidence of misalignment.
- High frequency readings (like shock pulse, spike energy, HFD, Peak Vue) cannot be used to detect misalignment.
- Vibration analysis cannot always be used to detect misalignment. Sometimes the vibration levels and patterns do not change when a machine is misaligned.

What is misalignment?

We should take a few minutes to understand what misalignment means so that you can better understand why it is so destructive. A definition:

"Shafts are misaligned when their rotational centerlines are not collinear when the machines are operating under normal conditions."

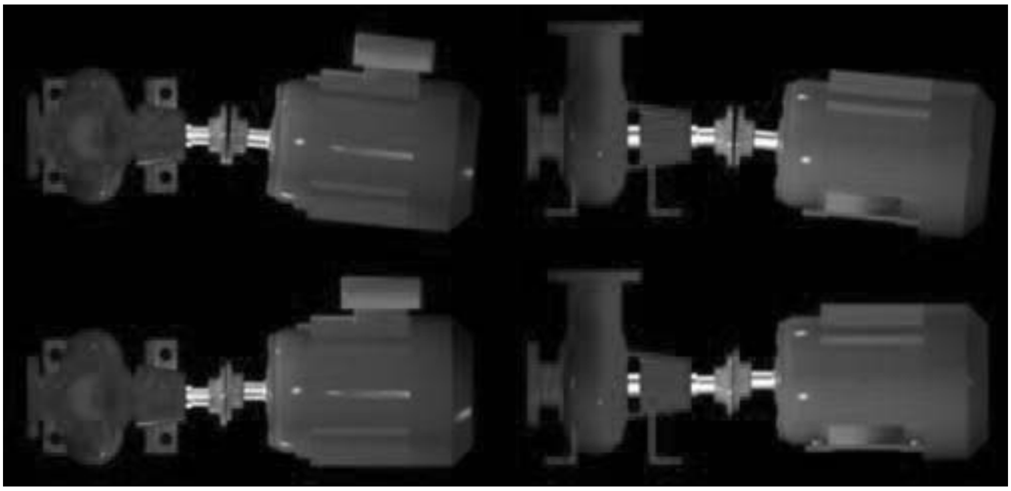


Figure 12-15 Different forms of misalignment

When two machine components, for example a motor and a pump, are assembled and coupled together, there is likely to be some misalignment - in fact, it is guaranteed. The pump will have been attached to its piping, and the motor connected to its conduit, and both will have been bolted to their baseplate.

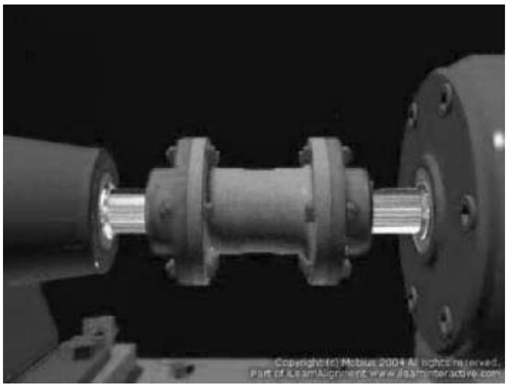


Figure 12-16 Motor and pump coupling

We hope there is no stress involved with the connection to piping and conduit - but there will be. We hope the feet of the machines and the baseplate are true and make perfect, flat contact, but they probably won't. And we hope that the two machines are positioned at the correct height and in a straight line, but that is also highly unlikely.

Sadly, all of these factors, and others, contribute to the fact that unless we take special precautions, machines will not be aligned correctly, and thus we will not get the desired life or performance out of the machine.

A closer look at misalignment

First, let's look more closely at the term misalignment. Our aim is to have the "rotational centerlines" of both shafts in line when the machine is operating. But what is the rotational centerline?

If the pump was uncoupled and you turned its shaft, it would rotate around a straight line. This is the pump's rotational centerline. When it is uncoupled, it is easy to turn the shaft (well, as easy as it will ever be). The same is true for the motor - its shaft also has a rotational centerline. Ideally, the motor and pump (Figure 12-17) will be mounted so that the two rotational centerlines are aligned perfectly - there will be no offset, and no angle between the shafts in either the vertical or horizontal direction. The coupling will bolt together easily, and there will be minimal stress on the bearings, seals or any other components when the shaft is turned. In this case the shafts are said to be "collinear".

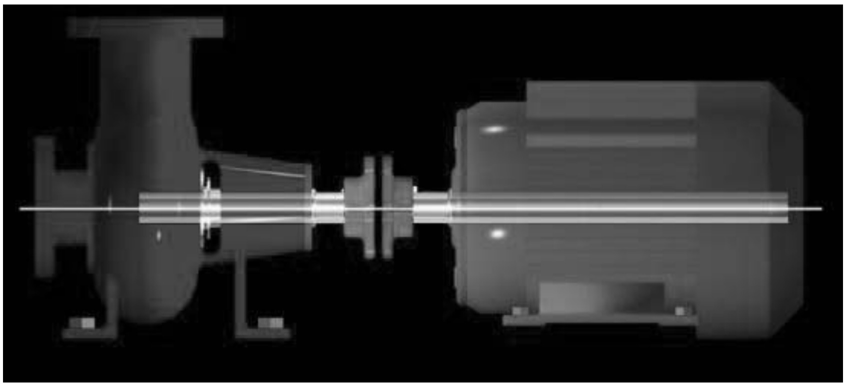


Figure 12-17 Collinear shafts – aligned rotational centerlines

But instead what happens is that the two components do not come together perfectly - perhaps one is a little higher than the other, and a little to one side. It might look okay to the naked eye, so the components are coupled together and the machine is run that way. It "looks" OK, and the machines runs, so what's the problem?

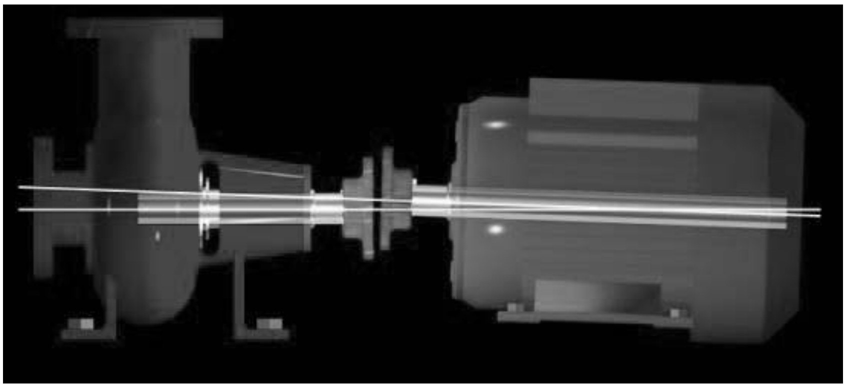


Figure 12-18 Rotational centerlines misaligned

The different forms of misalignment are shown in Figure 12-19.

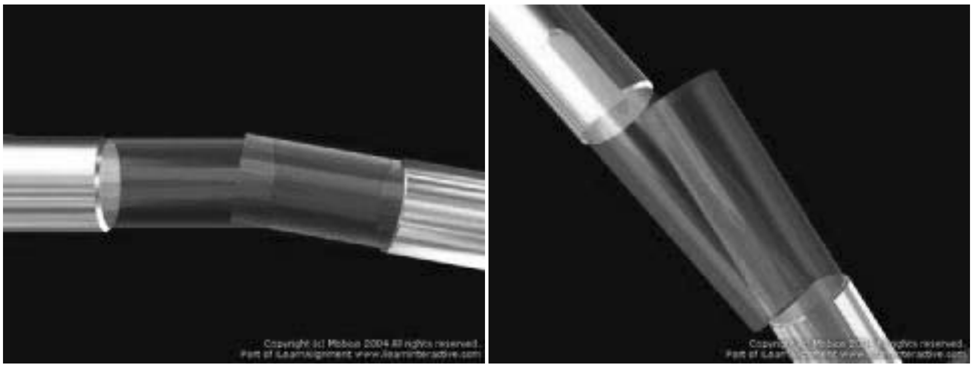


Figure 12-19 Vertical (left) and horizontal (right) offset and angular misalignment

You will see an offset and angle introduced in the vertical direction, and then you will see a different offset and angle introduced in the horizontal direction. The result is two shafts that are misaligned. In actual fact, when the components are coupled together, the coupling will be forced to "give" in order to accommodate the misalignment, and the two shafts may be forced to bend slightly (Figure 12-20).

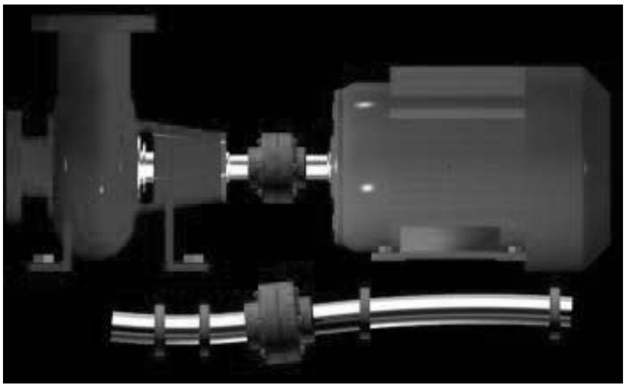


Figure 12-20 Bend in shafts due to misalignment (exaggerated)

Think about what is happening inside this machine - what is happening to the shaft, and the bearings, and the seals, and the coupling? With every rotation the shaft has to flex - it is forced to stay within the bearings. That puts strong radial and axial forces on the coupling, bearings, seals and shaft.

Offset and angular misalignment

Let's have a look at the different types of misalignment.

If one component is higher than the other, or to one side, there will be an offset. This is called "offset misalignment", or sometimes "parallel misalignment".

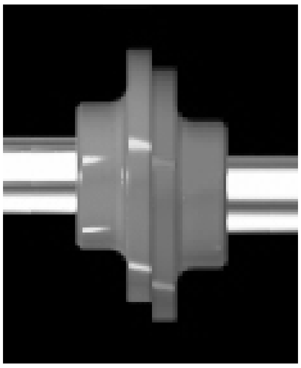


Figure 12-21 Offset or parallel misalignment

It is called "angular misalignment" (or "gap misalignment") when the two shaft centerlines meet at an angle.

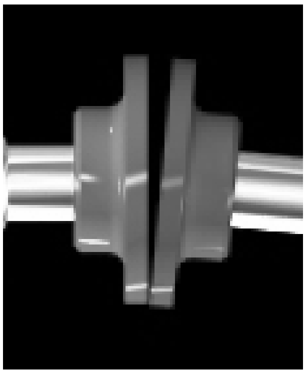


Figure 12-22 Angular or gap misalignment

In reality, shaft misalignment is a combination of both offset and angular - it will almost never be just angular or offset (parallel) misalignment.

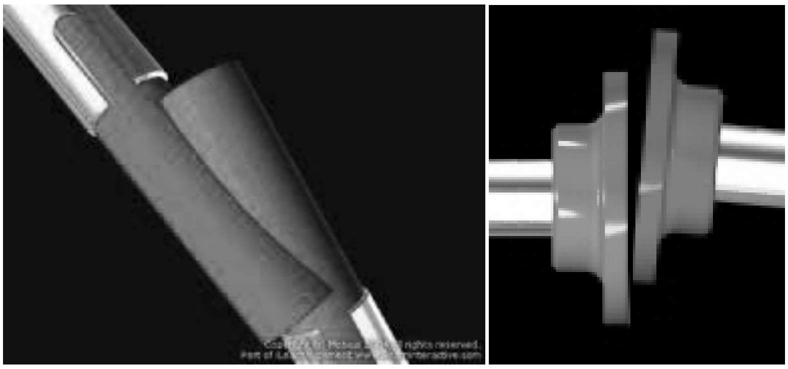


Figure 12-23 Offset and angular misalignment combined

It must be noted that the offset and angle exist both vertically and horizontally, to different degrees. For example, the motor shaft may be offset a little higher but very much to the left of the other shaft, and it may make a large angle vertically, but a small angle horizontally.

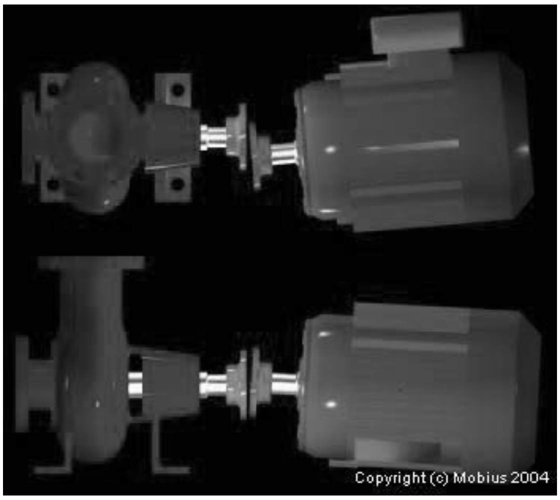


Figure 12-24 Vertical offset and angle (plan view, top) and horizontal offset and angle (side view, bottom)

Visualizing tolerance

You can think of the tolerances as a cone - the motor's shaft centerline can exist anywhere within the cone (Figure 12-25). The cone does not come to a point because we allow a certain amount of offset.

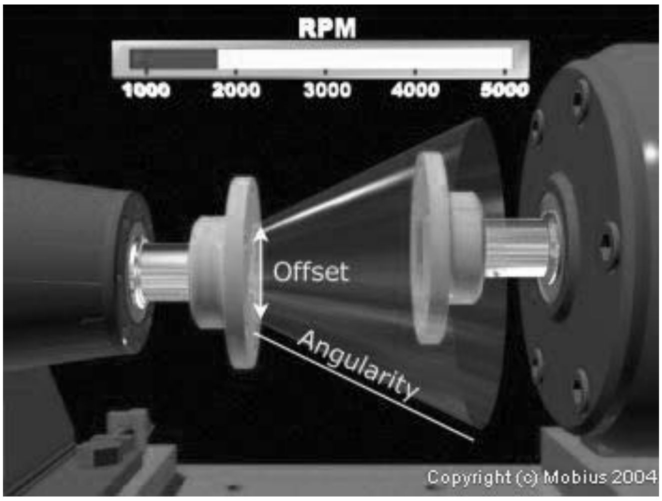


Figure 12-25 Tolerance 'cone' (exaggerated)

Tolerances and speed

While there are a number of factors to consider, the key issue when it comes to the relationship between the misalignment and the damage that can be done to the machine is the machine speed. If the speed is greater, the damage will be greater.

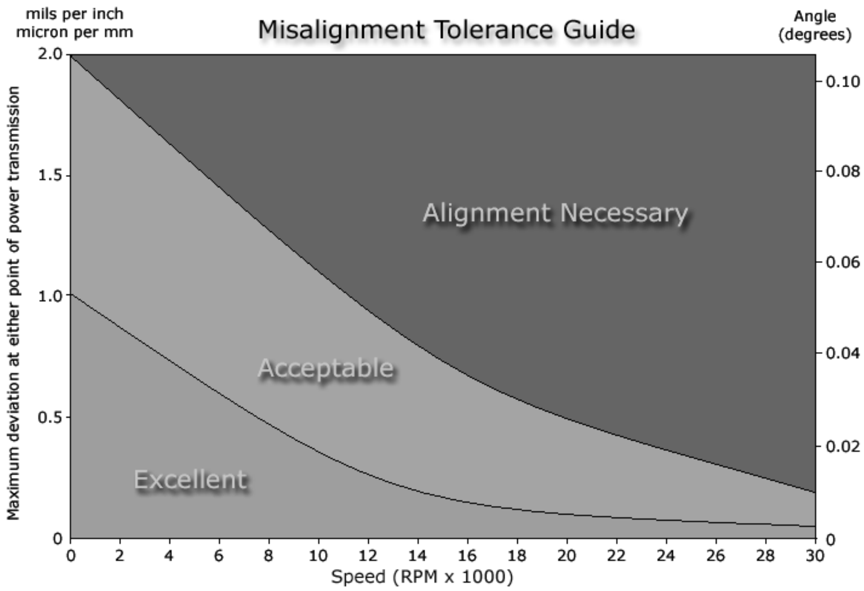


Figure 12-26 Misalignment tolerance guide (From: Shaft Alignment Handbook by John Piotrowski)

For this reason the tolerances need to be tighter on higher speed machines - that is, we allow a smaller offset and smaller angle between the shafts for higher speed machines.

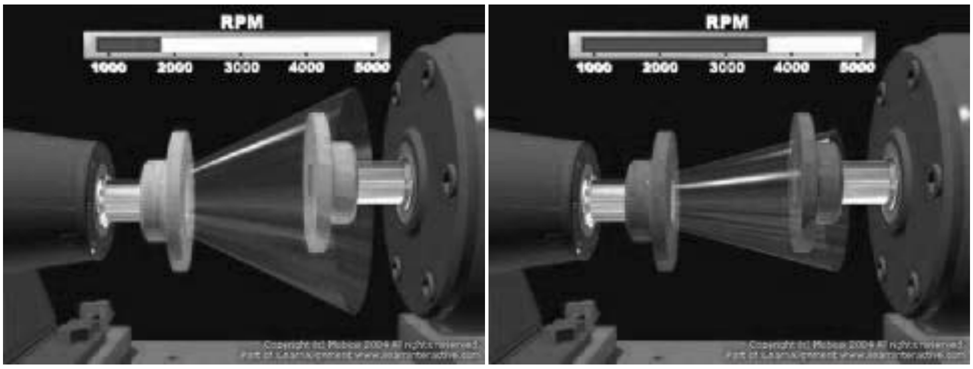


Figure 12-27 Smaller tolerance 'cone' on higher speed machine (right)

In Figure 12-27 we show that we can start with a certain amount of misalignment which is acceptable (within tolerance) for a lower speed machine (1800 RPM), but if the machine was running at 3600 RPM, the allowable offset and angularity is reduced, thus the machine is no longer in tolerance (it turns red).

You can see that the machine has the same amount of misalignment, but because it is running at a higher speed it is now out of tolerance.

Published tolerances

Tolerances are available from a number of sources. The tolerances from the PRUFTECHNIK company have been included here for your reference. You can see that the allowable offset and angularity is reduced for higher speed machines. You can also see that the tolerances have been set as "acceptable" and "excellent". You should always aim for "excellent" tolerance - your machine (and company balance sheet) will thank you for it.

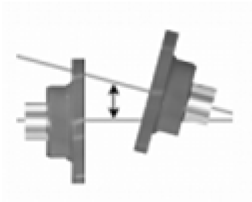
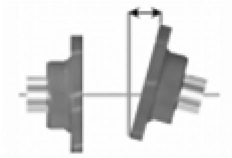
	RPM	inch (mils)		metric (mm)	
		Acceptable	Excellent	Acceptable	Excellent
Short "flexible" couplings					
	600	9.0	5.0		
Offset:	750			0.19	0.09
	900	6.0	3.0		
	1200	4.0	2.5		
	1500			0.09	0.06
	1800	3.0	2.0		
	3000			0.06	0.03
	3600	1.5	1.0		
	6000			0.03	0.02
	7200	1.0	0.5		
Angularity:					
Inch: Gap difference per 10 inch	600	15.0	10.0		
coupling diameter	750			0.13	0.09
Metric: Gap difference per	900	10.0	7.0		
100mm coupling diameter	1200	8.0	5.0		
	1500			0.07	0.05
	1800	5.0	3.0		
	3000			0.04	0.03
	3600	3.0	2.0		
	6000			0.03	0.02
	7200	2.0	1.0		
Soft foot	Any	0.06		2	

Table 12-1 Tolerances from PRUFTECHNIK

Here is the tolerance table for spacer shafts.

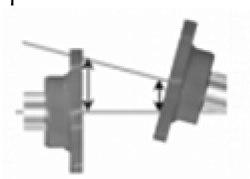
	RPM	inch (mils)		metric (mm)	
		Acceptable	Excellent	Acceptable	Excellent
Spacer shaft and membrane (disc) couplings:					
Inch: Offset per inch spacer	600	3.0	1.8		
shaft	750			0.25	0.15
Metric: Offset per 100mm	900	2.0	1.2		
spacer shaft	1200	1.5	0.9		
	1500			0.12	0.07
	1800	1.0	0.6		
	3000			0.07	0.04
	3600	0.5	0.3		
	6000			0.03	0.02
	7200	0.03	0.02		
Soft foot	Any	0.06		2	

Table 12-2 Spacer shaft tolerances from PRUFTECHNIK

Dynamic movement

When we originally defined misalignment we said:

"Shafts are misaligned when their rotational centerlines are not collinear when the machines are operating under normal conditions."

Did you notice the phrase *"under normal operating conditions"*?

When you take alignment measurements the machine is normally cold, and it is certainly not operating. But when it is started, a number of things happen. Rotational forces, operating pressures, and increased temperature all cause the position of the two rotational centerlines to change! Your machine could go from a state of precision alignment to out-of-tolerance misalignment.

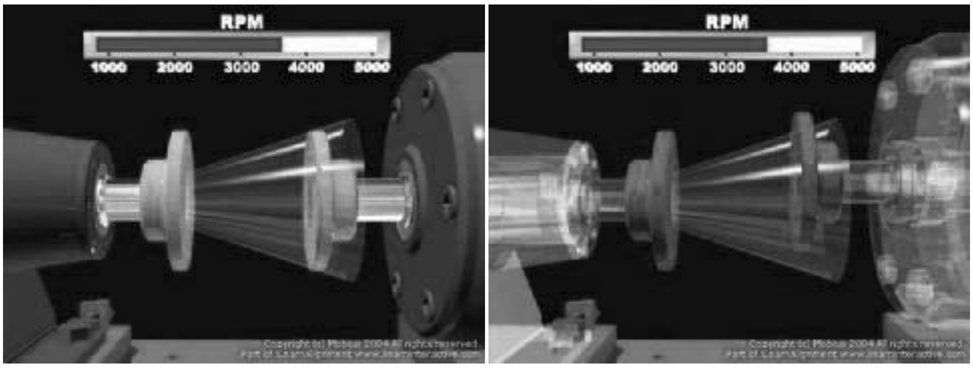


Figure 12-28Misalignment within tolerance when cold (left), but out of alignment when running (right)

In Figure 12-28, the machine has a certain amount of misalignment (exaggerated) which is within tolerance. But as the machine starts running, it heats up and moves out of alignment. All machines will undergo some change - they all increase in temperature, and thus the metal expands. They all experience rotational forces and most undergo changes due to pressure/flow of process fluids/gases. But most of the time we can afford to ignore this effect.

The simplest way to deal with this condition on the physically larger machines that undergo greater temperature changes is to compute the thermal growth and include that information into the alignment targets. Some manufacturers of affected machines will provide this data.

Figure 12-29 Compute thermal growth for larger machines

For example, using calculations that utilize the coefficient of expansion for different types of metals, you may determine that the shaft of the blower may lift 10 mils [0.25 mm], while the shaft of the motor will only lift 5 mils [0.125 mm]. You would therefore align the machine when it is cold so that blower shaft is 5 mils [0.125 mm] below the motor shaft. When it starts the two machines will gradually increase in temperature, and they will move into alignment.

It is also possible to align the machines soon after they have stopped when they are still hot. There are a number of issues to consider, but it is better than doing nothing.

It is also possible to attach laser heads to the bearings of the machine via special brackets so that you can measure exactly how much the two components will move. Measurements are taken when it is cold and again when it is hot. The movement is then factored into the cold alignment targets.

Pre-alignment tasks

One of the most important issues related to shaft alignment is what you do before you measure and correct the alignment. Your preparation of the alignment job is key to your success (and to your safety).

If possible, you should begin your work before the machine comes off-line. You should make sure all of your alignment equipment is ready to go, and that the batteries in the laser alignment equipment (if applicable) are ready for a day's work.

You should also review the maintenance records. Depending upon the nature of the alignment job (i.e. why are you doing an alignment, and what type of machine is being aligned), it is a very good idea to look at:

1. The maintenance records, to see why it should require alignment (unless it is simply the replacement of a motor).

2. Make sure that you have a good selection of shim sizes. They should be clean and straight.

3. Previous notes from the last time you performed an alignment on this machine: did you experience bolt-bound or base-bound problems, were there any other issues?

4. You should determine your alignment targets. Are there any recommendations from the manufacturer? Do you have to compensate for thermal growth?

5. And if possible, you should review the results from the previous alignment job performed on this machine.

Before you approach the machine you must follow all lock-out and tag-out procedures. And you must close all pump valves, etc. to ensure that the shaft cannot begin rotating because of fluid/gas/air flow.

Figure 12-30 Follow all lock-out and tag-out procedures

Collect "as-found" readings

You should record the as-found readings - take a set of alignment readings to show the initial alignment state. This is not necessary if it is a new installation, but it can be very beneficial to be able to show the improvement to the alignment state as a result of your work.

Create a clean work area

You <u>must</u> create a clean work area. Any dirt, grit, burrs, or other debris that gets between the machine and the baseplate, or under/between shims, etc., can cause you all kinds of problems.

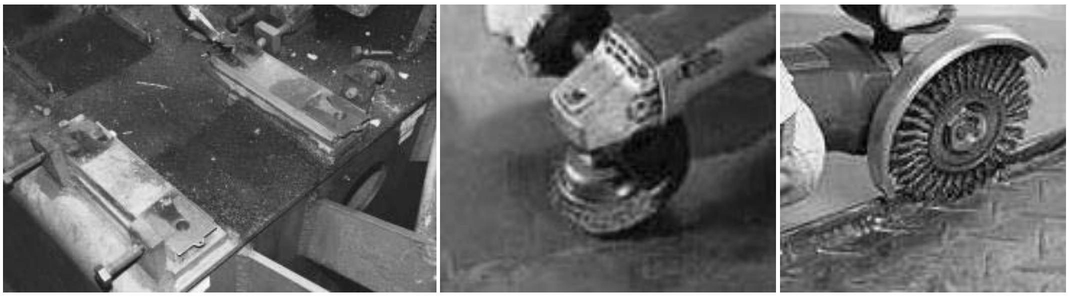


Figure 12-31 Remove dirt, grit, burrs and other debris

Prepare your shims

You should prepare your shims carefully. Remove any shims that are rusted, bent, painted, or dirty (beyond cleaning). Shims can act like small springs under the machine feet which, among other things, can make the alignment process very difficult.

Figure 12-32 Clean shims and remove those that are rusted, bent or painted

Take care of the bolts

You should replace bent, damaged or oversized bolts with new bolts. You should lubricate the bolts, and always use the same torque when tightening the bolts. You should also tighten and loosen the bolts in the same order.

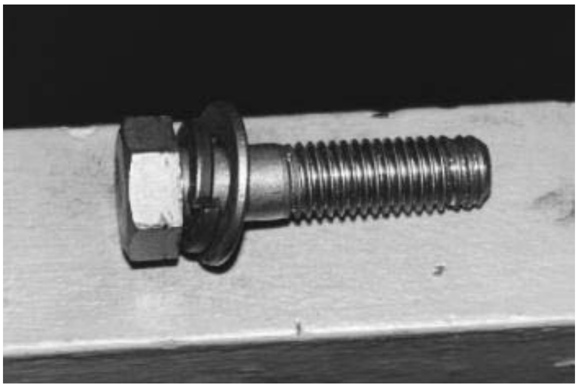


Figure 12-33 Replace bent, damaged or oversized bolts

Prepare the foundations of the machine

You should remove the taper pins and loosen off jacking screws before you begin taking measurements.

You also need to loosen the coupling bolts (in a rigid coupling). If you do not do this, it will be impossible to measure an offset and angle, as the coupling will cause the shaft to bend.

Check the physical condition of the machine

You should check the mechanical "health" of the machine. Rotate the shaft and see if there is any looseness, rubbing or binding. You should check to see if the shaft is bent, and if there is any coupling runout. You should check the coupling for excessive wear and proper fit, and check that the key is the correct length.

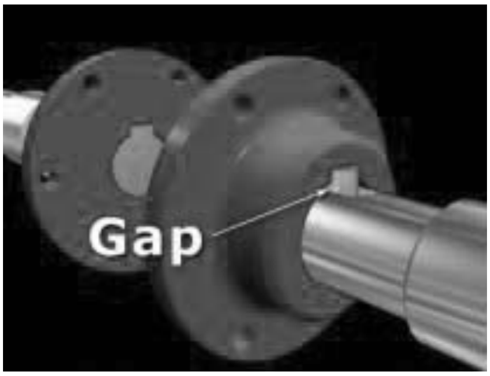


Figure 12-34

You should check for excessive piping strain, conduit strain and other forms of stress placed on the machine. If you loosen the feet and they move more than 0.002" or 0.05 mm then you should make corrections.

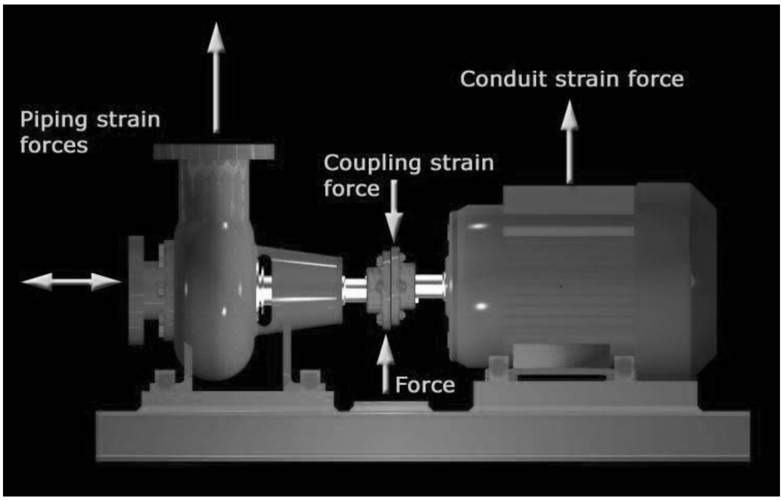


Figure 12-35 Check for excessive stress on the machine

Check and correct soft foot

You should check and correct soft foot. Soft foot is the condition where the feet do not make perfect, flat contact with the baseplate. Some people equate it with a short leg on a chair, rocking back and forth. But there is more to it than that.

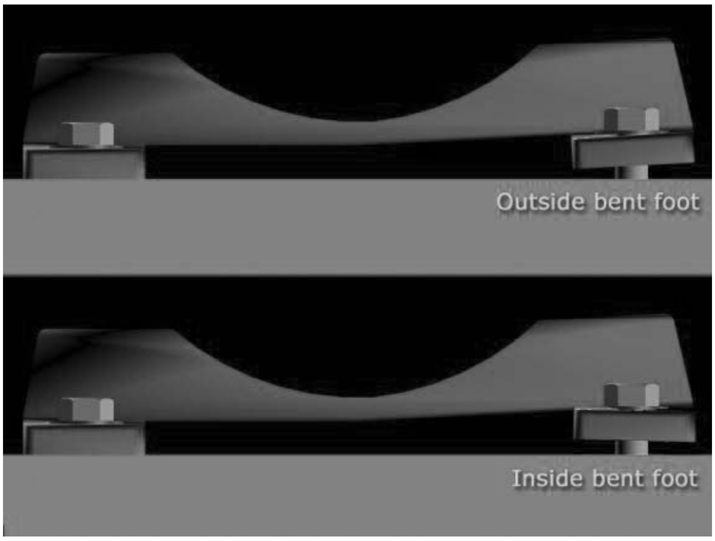


Figure 12-36 Gross soft foot check – obvious gaps under feet

You can start with a "gross soft foot check". Look for any obvious gaps under feet, and shim accordingly. Then you can perform a soft foot check with dial indicators or your laser alignment system.

Figure 12-37 Checking for soft foot using a laser alignment system

This involves loosening each foot, one at a time, and measuring how much the foot lifts. If it lifts more than 0.002" or 0.05 mm, then the soft foot condition must be corrected.

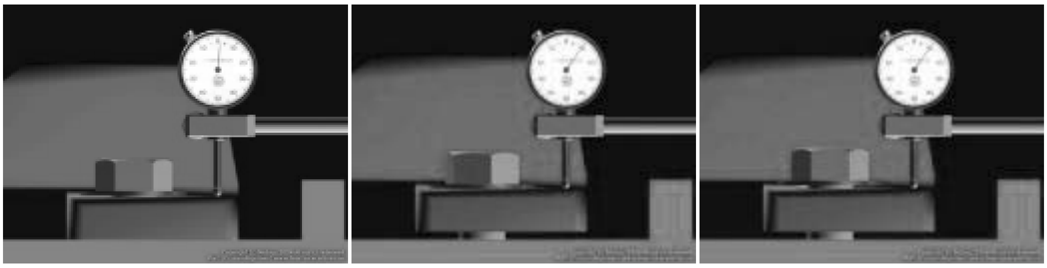


Figure 12-38

Note in Figure 12-38 that as the bolt is loosened the needle indicates that the foot is lifting. But when it reaches 10 the needle stops moving, even though the bolt is still being loosened.

Soft foot can take a number of forms. In summary, if two of the feet diagonally opposite each other have the highest readings then you have rocking soft foot (Figure 12-39). Shims can be placed under those feet. Otherwise the readings may indicate a bent foot condition, a "squishy" foot condition (too many shims or dirty shims under the foot), or the existence of pipe/conduit stress.

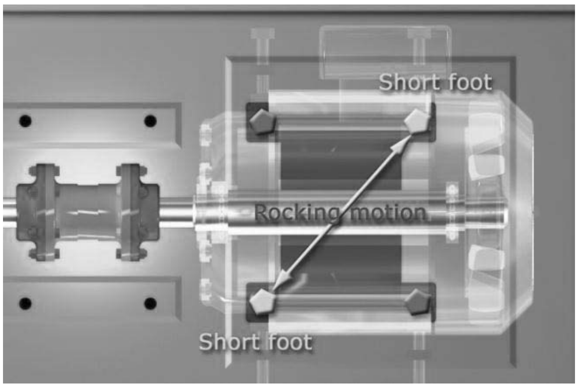


Figure 12-39 Rocking soft foot

Soft foot can make the alignment task very difficult and frustrating, and it can distort the machine frame and bearings - thus reducing the life of the machine. **You must not ignore soft foot.**

Begin the alignment process

Once you have prepared the site, checked the mechanical health of the machine, and corrected the soft foot condition, you are ready to begin the alignment measurements and correction.

Determining the alignment state

We now know what misalignment is, but how do you determine where the rotational centerlines are located so that you can make corrections? If you knew that the shaft of the motor was parallel but higher than the shaft of the pump by 10 mils (0.25 mm), you could lower it by that amount and be done. But how do you measure the offset and angle?

There are three ways to determine the relative positions of the shaft rotational centerlines: by eye, with dial indicators, and with laser systems (there are other methods, but we won't discuss them here).

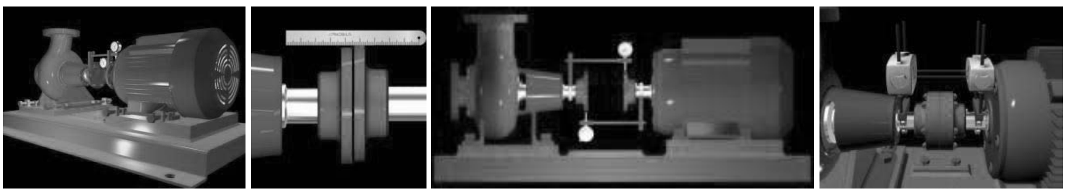


Figure 12-40 Methods of determining shaft rotational centerlines

Using a straightedge or feeler gauge

You can use very rudimentary tools to attempt to determine the relative position of the shafts, but the accuracy is very poor. These methods can be used for an initial "rough alignment", but not for the final alignment.

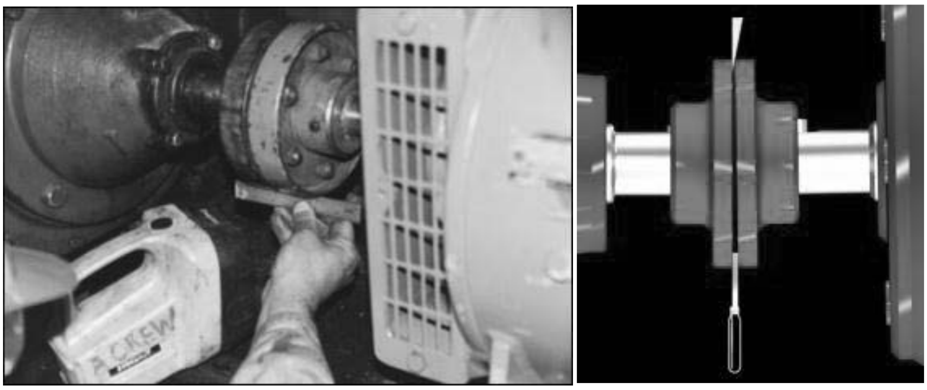


Figure 12-41 “Rough alignment” using rudimentary tools

You can measure the gap in the coupling to determine the angularity. For example, if there is a gap at the bottom of the coupling but not at the top, then you know the motor must slope up and away from the coupling. You can then lower the feet of the motor to close the gap. The shafts will now be parallel.

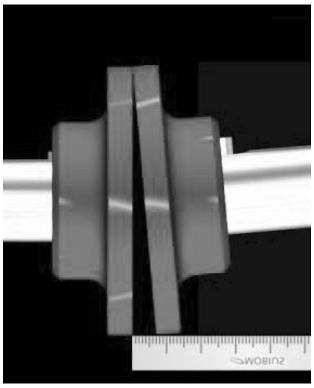


Figure 12-42 Gap at the bottom of the coupling shows the motor is sloping upwards

If you sit the straightedge on the coupling hub you will see a gap between the straightedge and the lower coupling (assuming there is no runout and the couplings are the same diameter). You can measure the gap, or simply move the machine until the gap disappears.

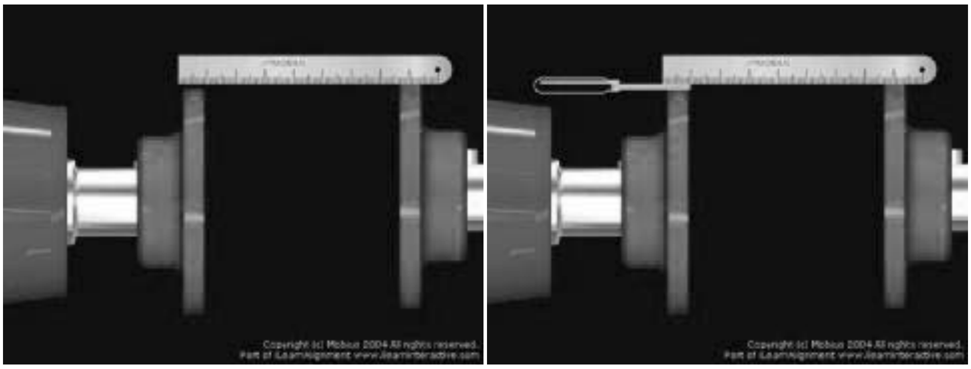


Figure 12-43 Using a straightedge to sight a gap

Again, you will look at the gap on top of the coupling to determine the vertical shimming required, and you will look at the offset at the side of the coupling to determine the feet movements.

It should be noted that we are not actually measuring the position of the shaft "rotational" centerline - we are simply making a rough assessment of the position of the shaft "geometric" centerline. In fact, we are really just aligning the couplings. All of our measurements thus far are taken on the coupling. If the coupling is incorrectly bored, if the shaft is bent, if the coupling is not round, or for a number of other reasons, we will not even be aligning the shaft geometric centerline.

With practice and common sense you can improve the alignment state with a straightedge and feeler gauge, but you will not correct the alignment. It is highly unlikely that the machine will be within tolerance using these methods alone.

Using dial indicators

Dial indicators are commonly used to measure the relative positions of the shaft rotational centerlines. There are a number of configurations that can be used to attach the dials to the shaft, but we will quickly review the two most common methods: the rim and face method and the reverse dial method.

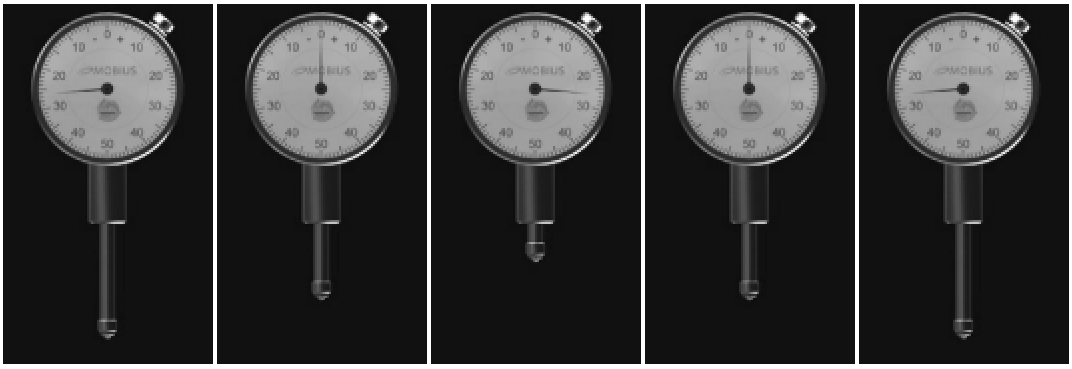


Figure 12-44

Dial indicator limitations

It is important to know that there are a number of limitations with these dial indicator methods that can result in poor alignment results.

Bar sag

One of the biggest issues is "bar sag". When the shaft is rotated from the 12:00 position to the 6:00 position, the bar carrying the rim measurement bends slightly. This adversely affects the rim readings. You <u>must</u> first separately measure the amount the bar will bend, and then compensate the readings.

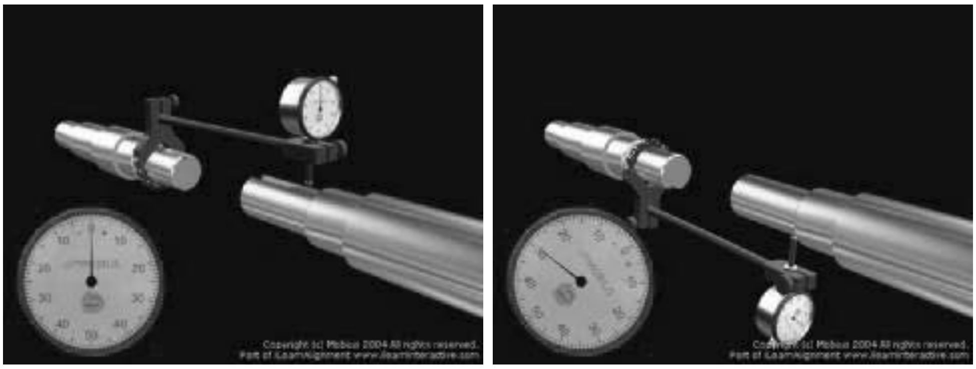


Figure 12-45 Dial indicators showing bar sag at the start of the test and at the end of the test

These two shafts are perfectly aligned yet the dial indicates that there is an offset – this is entirely due to the bar sag.

Reading accuracy

Another issue is reading accuracy. Because you have to take the reading from the dial, you have up to 0.5 mil [0.005 mm] rounding error. It is very difficult to take the reading from the dial, especially in a crowded machine space in poor light when the dial is up-side-down.

Additional problems

There are many other important issues: internal friction/hysteresis, reading errors, play in mechanical linkages, axial shaft play (especially for face readings), and more. And it is also common for people to make mistakes with the calculations and graphical method.

For many years people have been able to do a good job with dial indicators, however laser alignment systems produce more accurate and dependable results (with automated calculations of feet corrections), and the measurements can be taken far more quickly.

The Rim and Face method

The rim and face method requires two measurements to be taken, one on the rim of the coupling (in order to measure the offset), and the other on the face of the coupling (in order to measure the angularity). You can see where this method gets its name!

Commercial systems are available with special brackets that are designed to maximize measurement accuracy. But the basic principle is always the same: one dial measures the face angle, and the other measures offset.

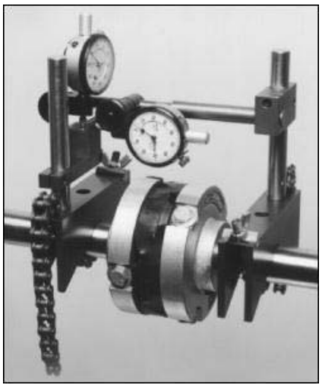


Figure 12-46 Rim and face configuration

Figure 12-47 shows an example of one of the tests in action. Notice that both shafts are rotated, which means that the system is measuring the relative position of the shaft rotational centerlines.

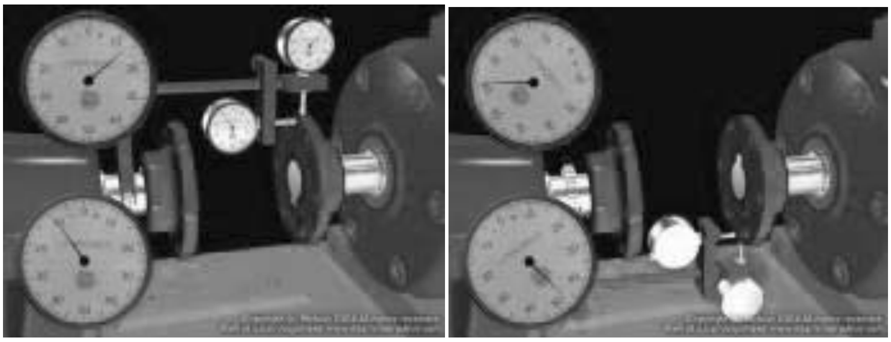


Figure 12-47 Rim and face test

This rim measurement will have revealed whether the motor shaft is higher or lower than the pump shaft (at the coupling). If the motor shaft were lower, the plunger would have been pushed into the dial (as it moved from the 12:00 position to the 6:00 position) which will be recorded on the dial face as a positive reading.

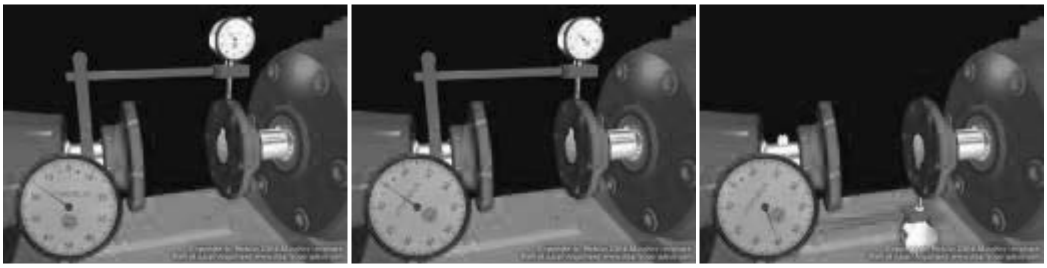


Figure 12-48 Rim measurement

Likewise, the face measurement will tell us if the motor shaft slopes away higher or lower (or parallel) to the pump shaft. If it sloped downward, the plunger would be pushed into the dial as the rig is rotated to the 6:00 position.

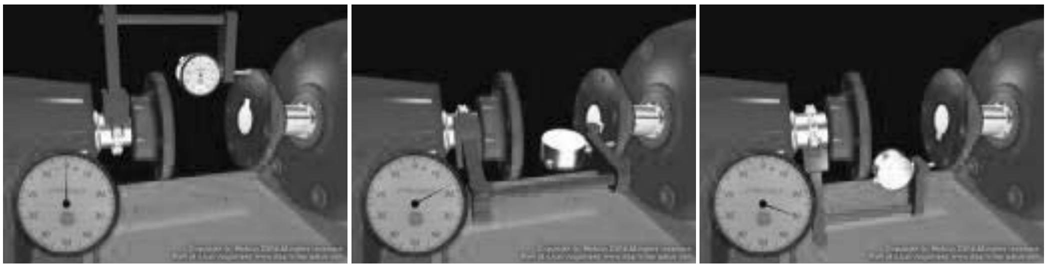


Figure 12-49 Face measurement

Based on the readings we take at the 12:00, 3:00, 6:00 and 9:00 positions, we can determine the offset and angularity, and we can calculate the changes required to the feet height and lateral position in order to make the two shafts collinear.

We can also do this graphically. By drawing the relative positions of the shaft to scale on a piece of graph paper (once in top/plan view, and again in side/elevation view), we can determine how the feet need to be moved.

The Reverse Dial method

The reverse-dial method is the most commonly used dial indicator alignment method used today. In this case we attach two dials to the two coupling rims to take two offset readings.

Again, there are a number of ways that these dials can be set up in order to collect essentially the same information. There are a number of commercial systems that employ this method.

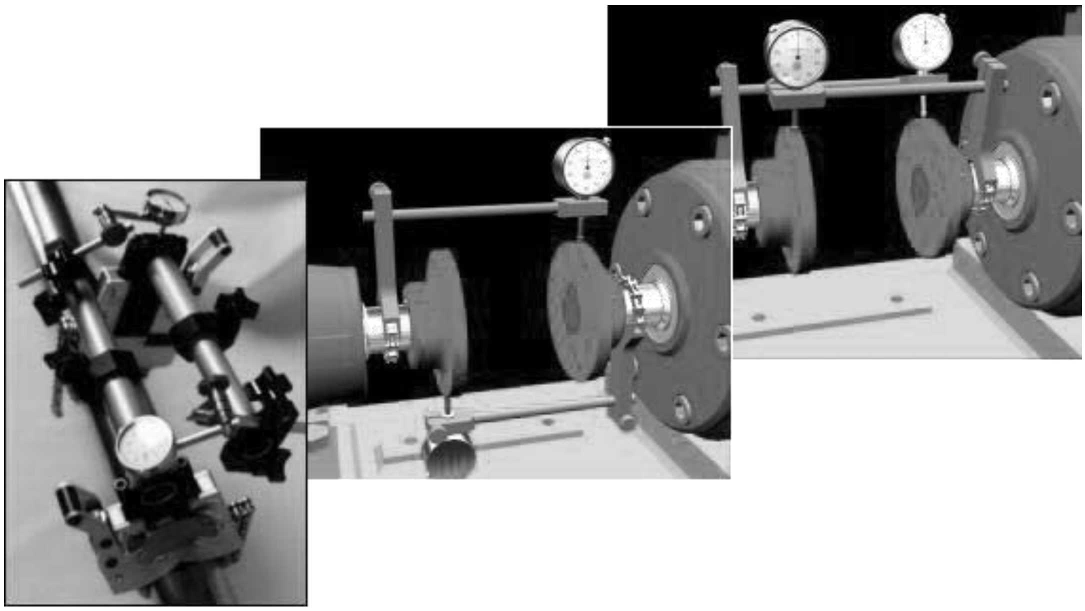


Figure 12-50 Different setups for the reverse dial method

As before, the two shafts are rotated, and the plungers either move into the dial or out of the dial based on the relative position of the shaft rotational centerline at the measurement points.

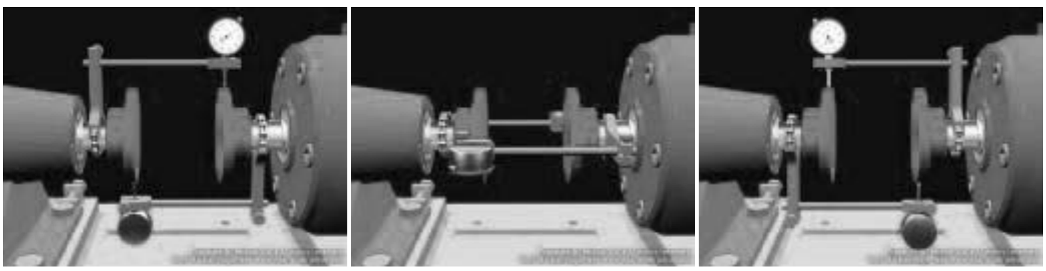


Figure 12-51 Reverse dial test

To determine the relative positions in the vertical direction (for shimming) we start at the 12:00 position and rotate to 6:00 and record the two values.

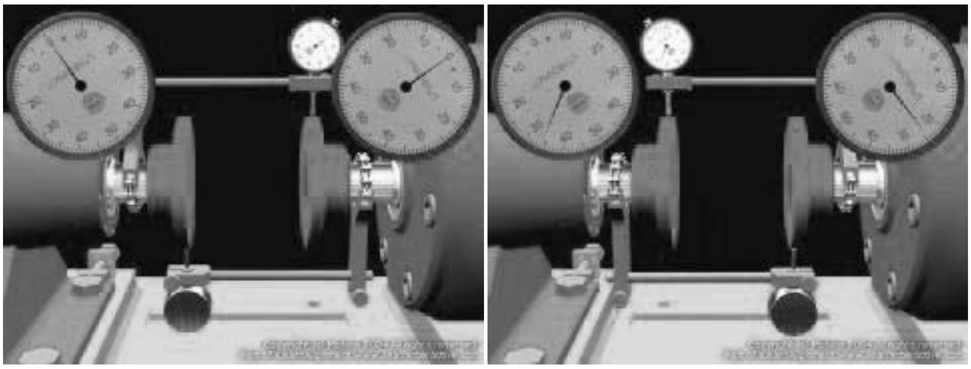


Figure 12-52 Measurements at the 12:00 and 6:00 positions

For horizontal foot movements, we compare the readings at 3:00 and 9:00.

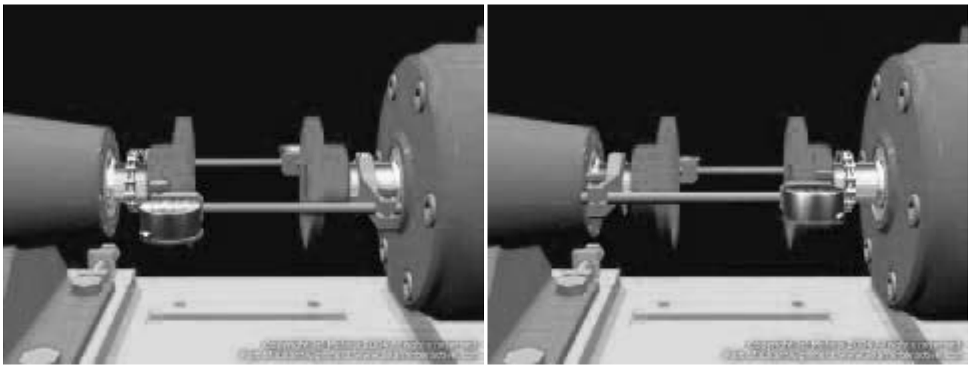


Figure 12-53 Measurements at the 3:00 and 9:00 positions

We can then use these dial readings and perform calculations to determine the offset and angularity at the coupling, and to compute the changes required to the feet height and lateral position in order to make the two shafts collinear.

We can also do this graphically. By drawing the relative positions of the shaft to scale on a piece of graph paper (once in top/plan view, and again in side/elevation view), we can determine how the feet need to be moved.

Laser alignment systems

Laser alignment systems come in many shapes, sizes and colors. They employ a range of technologies, from prisms that reflect the beam, to dual laser transmitters/detectors, and detectors that can assess movement in five axes.

Figure 12-54 Modern laser alignment systems

The two basic components of the laser alignment system are the "emitter" (sometimes called the "transmitter") and "detector" (sometimes called the "receiver").

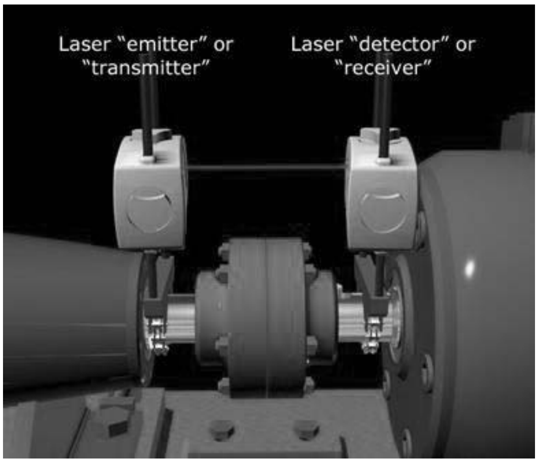


Figure 12-55 Emitter and detector –basic components of a laser alignment system

A number of the systems work in a very similar way to the reverse-dial indicator method: each head has an emitter and detector. The two laser heads are attached to the shaft on either side of the coupling. The laser heads are zeroed, and then the shafts are rotated. The detectors monitor the change in position of the laser - just like watching the plunger being pushed into the dial. In some cases the laser detector can detect movement in two dimensions - "x" and "y".

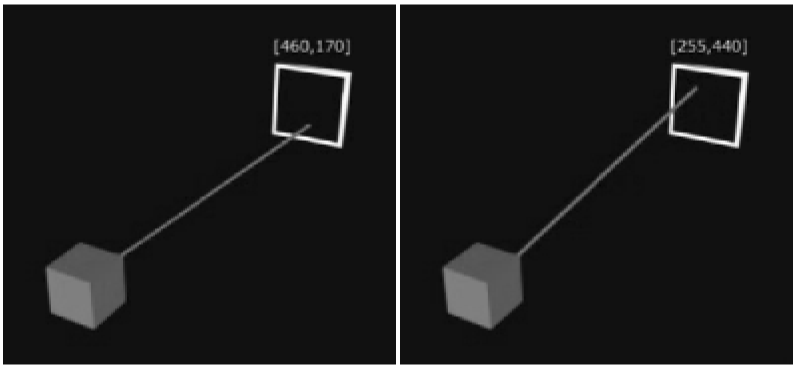


Figure 12-56 Detecting movement in the x and y directions

In other cases, the laser detector is sensitive in a single axis - just like the dial indicator (but FAR more sensitive and accurate).

Regardless of the method used, the shaft is always rotated (if possible) in order to measure the position of each shaft's rotational centerline. Using advanced technology, it is possible to rotate the shaft through only 60 degrees in order to gather enough data.

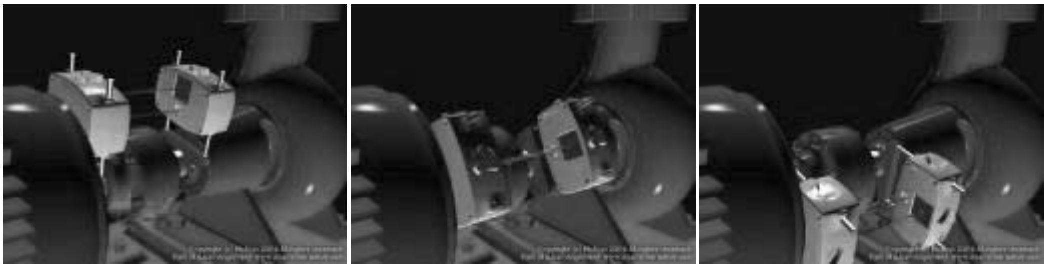


Figure 12-57 Gathering data through a 60 degree shaft rotation

The laser alignment systems are not only extremely accurate, they also come with "computers" that perform all of the calculations. In many cases they show you graphically how the feet should be moved, and can tell you when the alignment is within tolerance.

Laser systems are designed to be used in harsh environments; and they are safe to use - but don't stare into the beam!

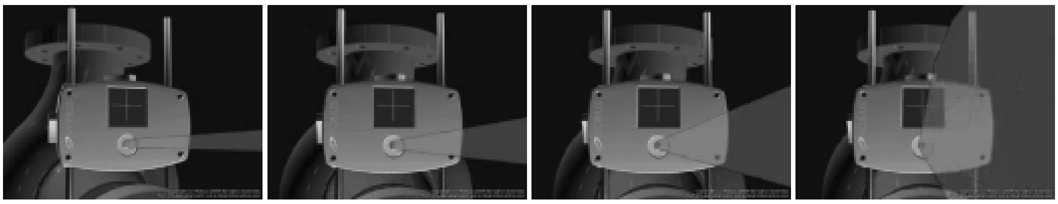


Figure 12-58 Don't stare into the beam!

Moving the machine

When the alignment measurements have been performed, you will determine how the feet should be moved laterally and vertically (by adding or removing shims) so that the rotational centerlines are collinear.

Figure 12-59 Adding a shim

As long as there is not a gross misalignment condition, you should always perform the vertical correction before a horizontal correction. A horizontal correction will not affect the vertical position of the machine, but a vertical correction will always affect the horizontal position.

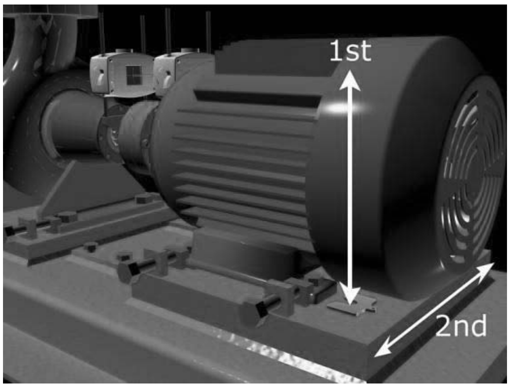


Figure 12-60 Perform the vertical correction first and the horizontal correction second

When a vertical correction is made it is almost impossible not to disturb the horizontal position of the machine. Unless the base plate is severely warped, when you make a horizontal correction the vertical position of the machine should remain constant. If there is gross misalignment (greater than 40 mils or 1 mm), you should make an initial vertical move, and an initial horizontal move, before you make the final vertical and horizontal moves. The initial move is required because unless a base plate is perfectly flat, a large horizontal move will result in a small vertical change.

Moving the machine vertically – shimming

The vertical correction is made by adding or removing shims. You should never leave more than four shims under the foot of a machine. Shims should be clean, straight, and carefully manufactured.

Figure 12-61 Using shims for vertical correction

You may find that you cannot remove enough shims in order to bring the machine into alignment (i.e. it is "base-bound"). In this situation you may need to lift the front feet of the "stationary" component (the pump, for example), as well as the front feet of the "movable" component (the motor) - or you can machine the feet or the baseplate.

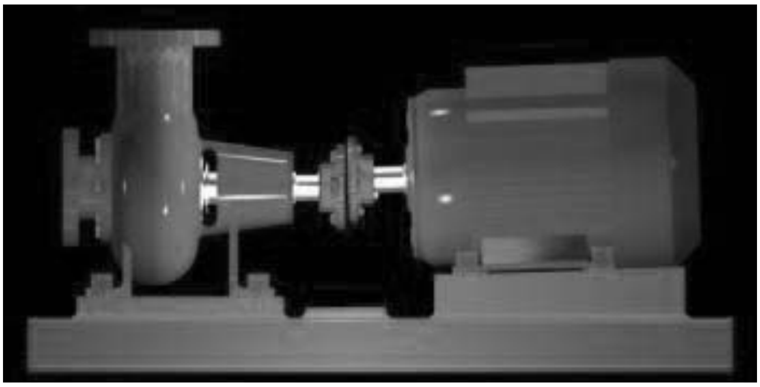


Figure 12-62 “Base-bound” motor

Moving the machine laterally

Although sledge hammers and pieces of 2 x 4 timber can be used to move the machine laterally, it is highly recommended that jacking bolts are used instead.

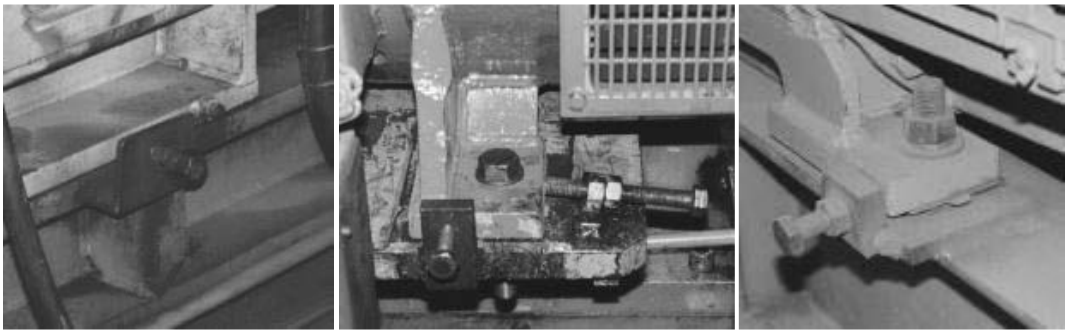


Figure 12-63 Jacking bolts for lateral movement

Laser alignment systems can be used to monitor the movement of the machine so that you can make positional changes until you are in tolerance. Sometimes the movable component cannot be moved far enough to bring it into alignment - it may be "bolt-bound". While you could move the stationary component, you can also consider opening the foot holes, turning-down the bolts, or drilling new holes in the baseplate.

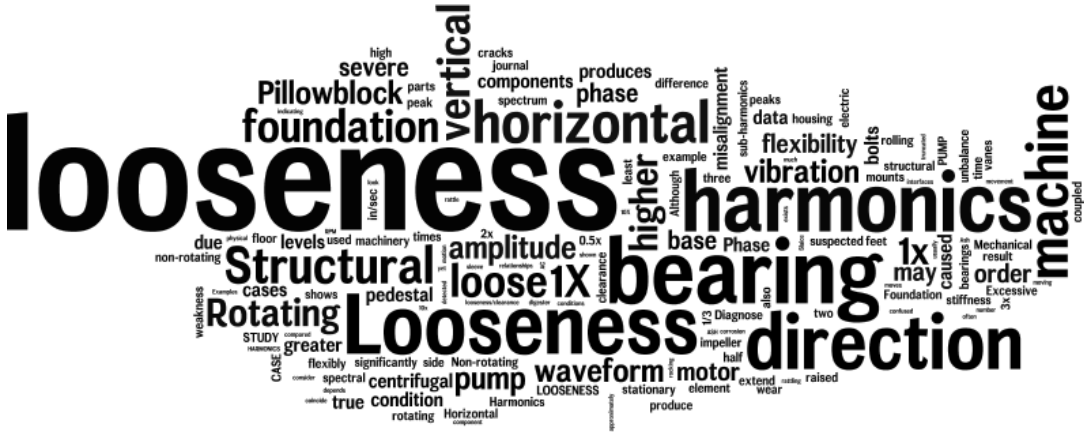
Whenever you tighten the hold-down bolts, you should ensure that they are always tightened to the same torque. It is a good idea to lubricate the bolts before performing the alignment. It is also recommended that you always tighten and loosen the hold-down bolts in the same order.

Figure 12-64 Tighten hold-down bolts to the same torque

After you have moved the machine, and tightened the hold-down bolts, you must always repeat the measurements to check that the machine really is still in tolerance. If you have had to make large changes either vertically or laterally, you should also recheck the soft foot condition.

Conclusion

Precision alignment is critically important to the ongoing reliability, and thus profitability, of your plant. The additional time spent performing the alignment can always be justified. Laser alignment tools make the measurement and movement of the machine easier, however it is essential that you understand the measurements, movements and all of the potential pitfalls in order to ensure that you can successfully align machines under all circumstances.



Chapter 13

Diagnosing Looseness

Objectives:

- Diagnose Rotational Looseness in spectral and waveform data
- Diagnose Structural and Non-rotating looseness in waveform and spectra
- Describe the phase characteristics of looseness conditions

Mechanical Looseness

Mechanical Looseness generates peaks at 1x. There are actually three types of looseness to consider:

- Rotating Looseness
- Structural Looseness
- Non-rotating Looseness

Rotating looseness is caused by excessive clearance between rotating and stationary elements of the machine such as in a bearing, while **non-rotating looseness** is looseness between two normally stationary parts, such as a foot and a foundation, or a bearing housing and a machine.

Structural looseness occurs where there is weakness in the baseplate, foundation or feet, and the machine is able to rock from side to side.

Rotating Looseness

Rotating looseness can occur due to wear in a bearing. Other bearing wear symptoms are first detected, followed by bearing looseness.

Excessive clearance in journal (sleeve) and rolling element bearings (bearing looseness) produces harmonics of 1X that can extend, in some cases, above 10X.

As the looseness condition worsens, the number and amplitude of the harmonics increases. Some peaks will be higher than others when they coincide with structural resonances or other sources of vibration, for example vane pass frequency.

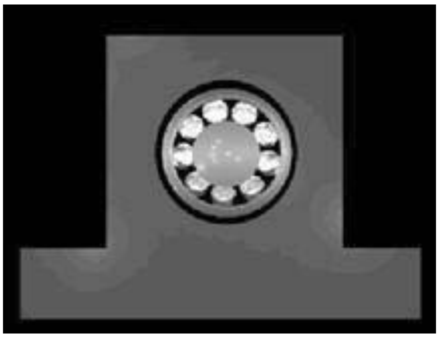


Figure 13-1 - Bearing loose in a housing

Harmonics can extend beyond 10x.

The noise floor can be raised.

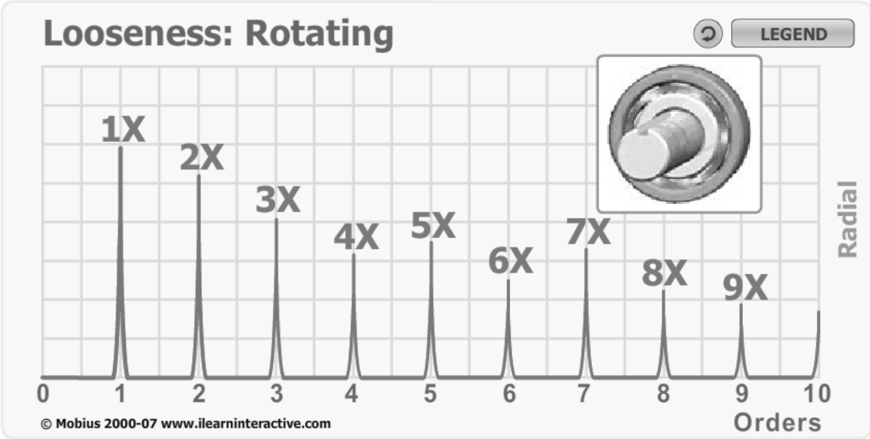


Figure 13-2 - Rotating looseness produces harmonics

It can be confused with severe misalignment except that the raised floor indicates rotating looseness.

Phase relationships that are true for misalignment are not true for looseness. With looseness the phase is erratic, unstable.

Excessive looseness/clearance (in rolling element and journal bearings) can produce harmonics of 0.5X as shown below. They are called half order components or sub harmonics. They can be produced by rubs and severe impacting. Even 1/2 and 1/3 order sub-harmonics are possible.

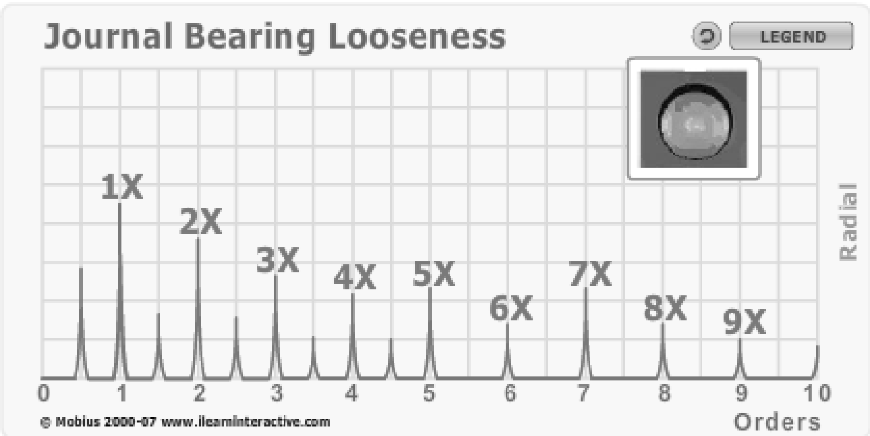


Figure 13-3

Why the harmonics?

Harmonics are present because the vibration is "non-linear." The movement within the bearing is clipped or truncated.

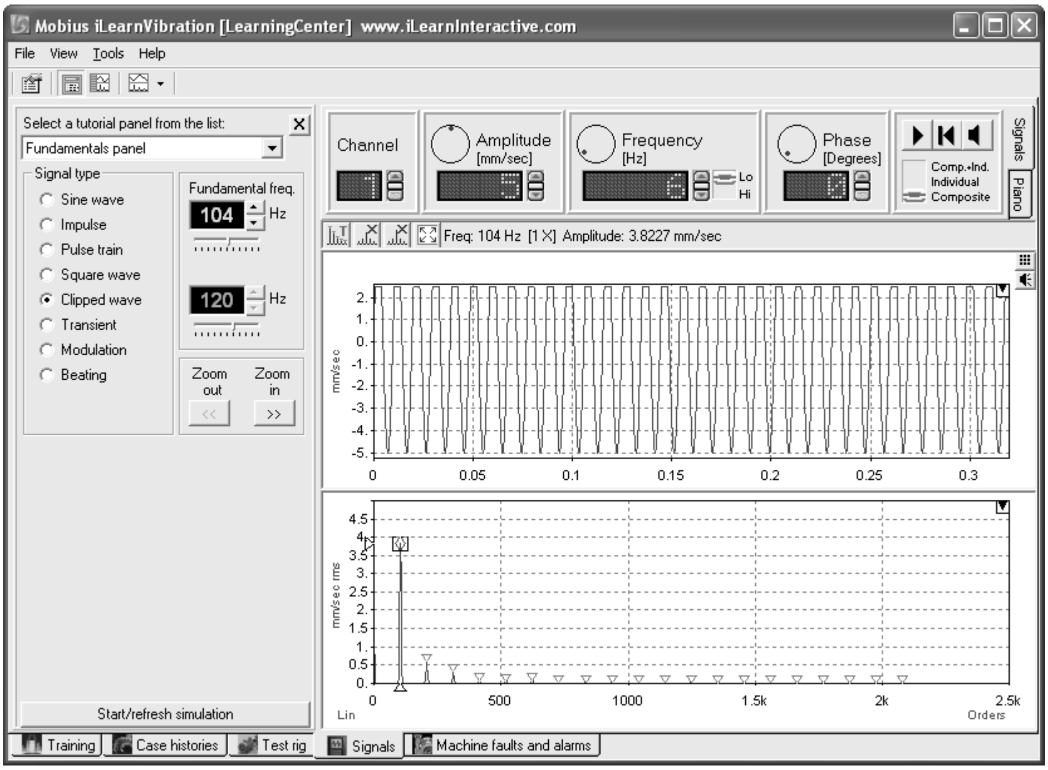


Figure 13-4 - clipping and truncation produce harmonics

Other forms of looseness, or rattling, will also result in harmonics of 1X being observed in the spectrum. Examples include loose cowlings, structural steel just touching a motor, loose guards, etc.

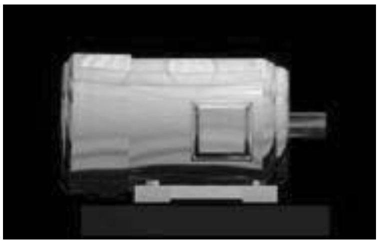


Figure 13-5 - loose parts result in 1x harmonics

Rotating Looseness Example:

A bearing pedestal was so loose that it rattled.

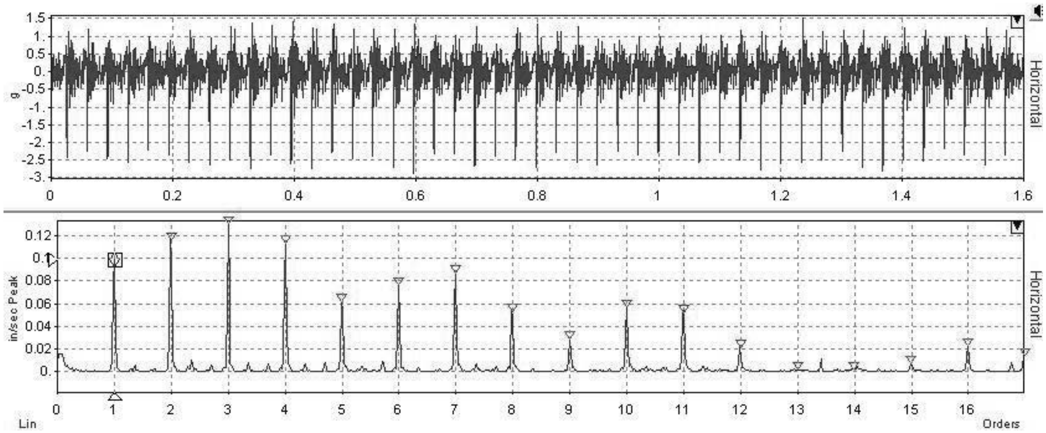


Figure 13-6 - Rotating Looseness with many harmonics. Impacting in the waveform.

The harmonics are obvious in the vertical direction, due to the fact that there is also non-rotating looseness (described next), or foundation flexibility. The levels in the horizontal direction were approximately three times higher.

The value of the time waveform is clearly seen in this example. The transients from the rattle (periodic impacts) are very clear.

The axial direction still has harmonics, but the amplitude levels are well down - 1/20th of the level.

Structural Looseness or Foundation Flexibility

Structural looseness is foundation flexibility caused by weakness or looseness of machinery mounts, base plates, or concrete and grouted interfaces of the internal base itself.

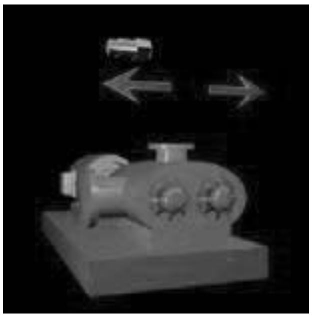


Figure 13-7 - Structural looseness has the highest 1x in the direction of least stiffness.

Looseness between a machine and its foundation will increase the 1X vibration component in the direction of least stiffness. This is usually the horizontal direction, but it depends on the physical layout of the machine.

If the machine has resilient mounts, the vibration will always be greater in the Horizontal direction.

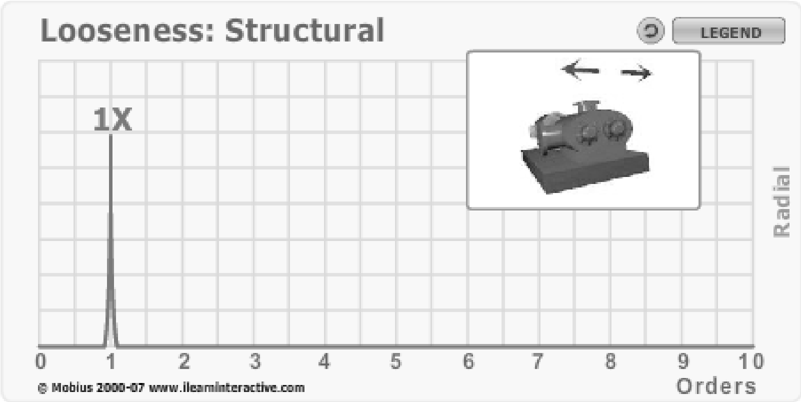


Figure 13-8 - Structural looseness produces 1x peak

Looseness vs. Unbalance

- If 1X horizontal is much greater than 1X vertical, looseness is suspected.
- If 1X horizontal is lower than or equal to 1X vertical, then unbalance is suspected.
- The phase relationship of 90° between vertical and horizontal direction will not be true if it is a looseness condition.

Foundation flexibility or looseness can be caused by loose bolts, corrosion, or cracking of mounting hardware.

Phase Measurements

Phase readings can be compared between the base, hold-down bolts, and the foundation. If there is between 90° and 180° and a significant difference in amplitude, it is an indicator that one surface is moving and the other is not.

Case Study: Ash Hopper Sluice Pump

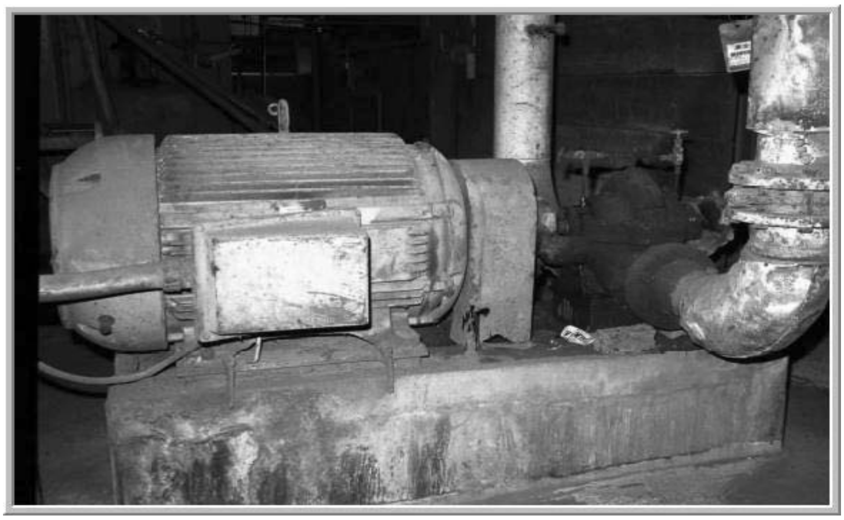


Figure 13-9 - Ash Hopper Sluice Pump

This machine is used to pump ash out of a power boiler. The machine is a 150HP electric motor flexibly coupled to a centrifugal pump with 6 impeller vanes. The pump provides 245 gallons per minute at 385 feet (117 metres) head pressure.

Although there are indications of unbalance, there is significantly higher amplitude in horizontal axis, indicating that foundation flexibility exists.

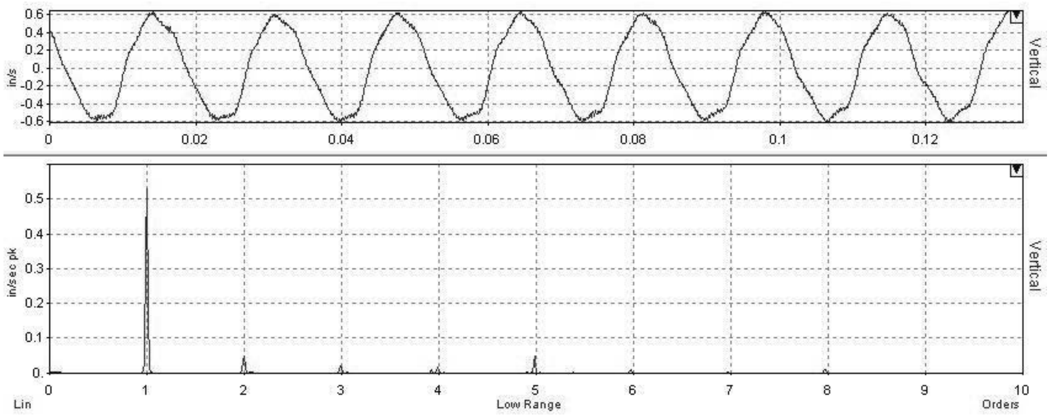


Figure 13-10 - Data from the vertical direction... 0.5 in/sec

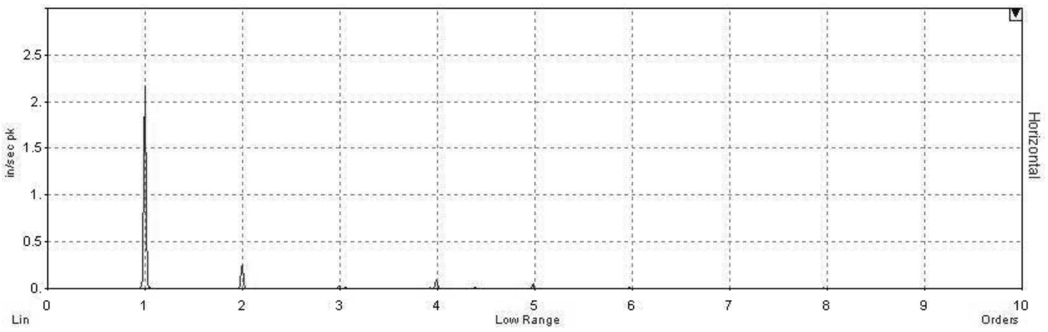


Figure 13-11 - The horizontal direction is significantly higher at 2.1 in/sec.

The Horizontal direction is 4x greater than the Vertical direction. Compare amplitudes in Figure 13-10and Figure 13-11.

Case Study: Cool Liquor Pump #1

This AC electric motor driven centrifugal pump moves digester liquid in a paper manufacturing plant. The nominal running speed of the motor is 3580 RPM. The centrifugal pump has four vanes on the impeller. The two machine components are flexibly coupled.

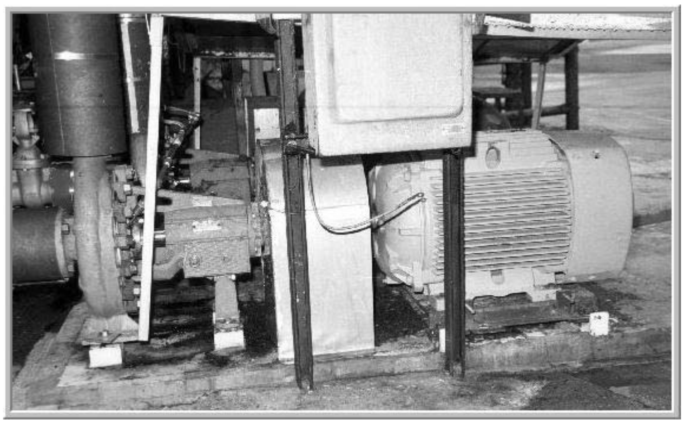


Figure 13-12

Although the amplitude levels are not high as yet, the data certainly shows a strong series of harmonics, and the time waveform exhibits random vibration.

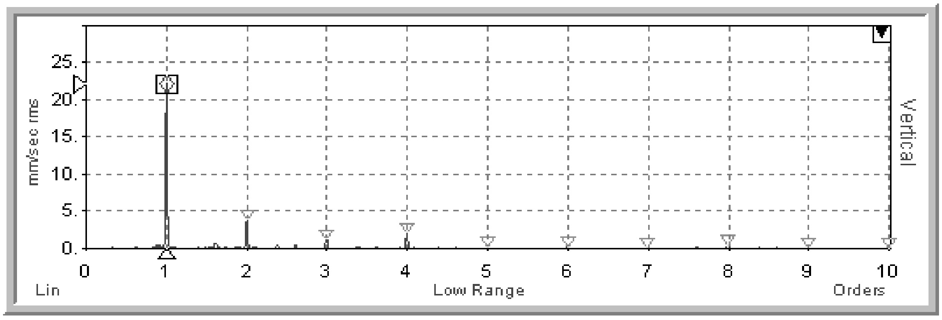


Figure 13-13 - The vertical is very high, but the horizontal is even higher

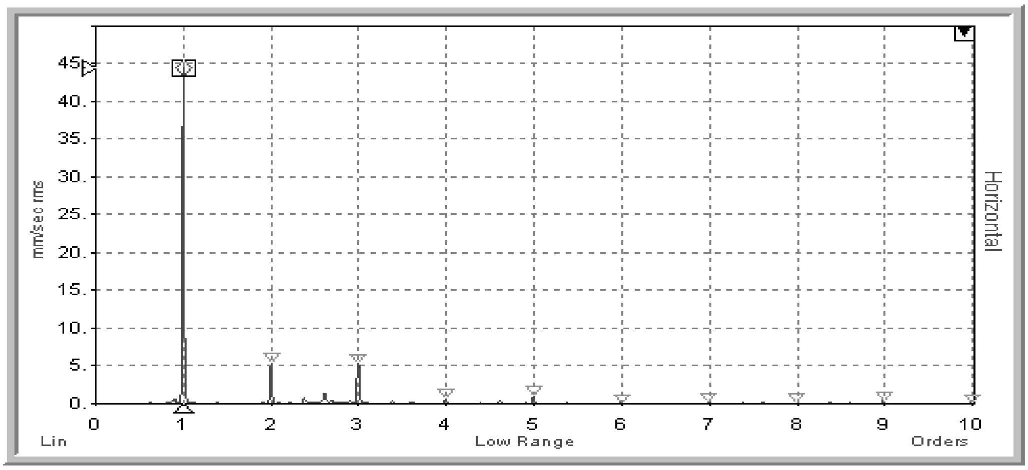


Figure 13-14 - The horizontal is 1.8 times higher than the vertical

Loose Pedestal Bearings (Pillowblock)

Structural Looseness in a pillowblock bearing shows a different characteristic in the spectral data.

Figure 13-15 - Pillowblock bearing

Looseness can be due to cracks in the frame structure or bearing pedestal, loose bearing pedestal bolts, or faulty machinery isolators.

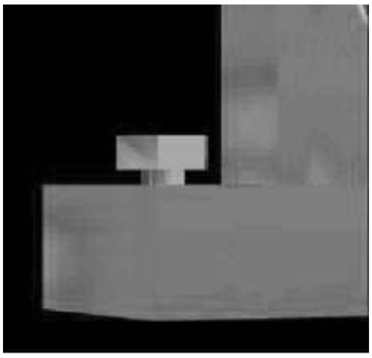


Figure 13-16 - Looseness in Pillowblock bearing

The spectrum may look a little like misalignment but there may be a rocking motion. There are components of 1x, 2x, and 3x, but often no more harmonics. In more severe cases it may have half order peak. (0.5x). When cracks exist there may be ¼, 1/3, and ½ order sub-harmonics.

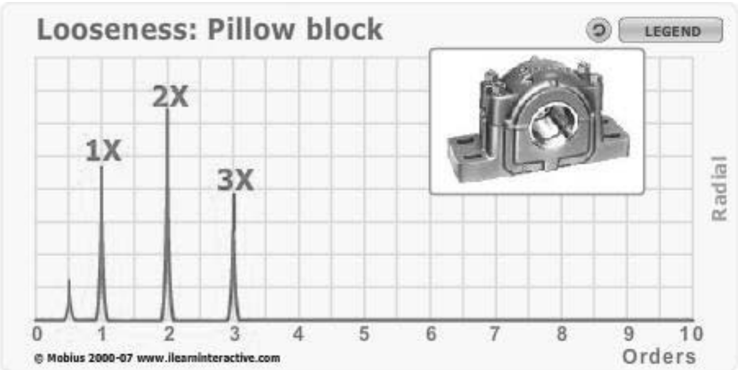


Figure 13-17 - Pillowblock has 1x, 2x, 3x and 0.5x in severe cases.

Phase can again be used to verify this condition. There will be a 180 degrees phase difference between the bearing and the base.

Chapter 14

Rolling Element Bearing Analysis

Objectives:

- Discuss the root causes of bearing failure
- Describe bearing forcing frequencies and their relationship to bearing geometry
- Describe the progression of bearing wear in terms of the velocity spectrum
- Explain demodulation / enveloping and high frequency tests for bearing wear
- Describe the failure progression of typical rolling element bearings.

Rolling Element Bearings

In this chapter we are going to view the many vibration analysis techniques that can be applied to diagnosing rolling element bearing faults, and we will discuss the four stages of bearing wear. This chapter will cover spectrum analysis, time waveform analysis, enveloping (and demodulation) techniques. Additional high frequency analysis techniques: such as Shock Pulse, Spike Energy, PeakVue, and acoustic emission will be briefly mentioned.

After completing this section you should understand how to apply these techniques so that you can confidently detect bearing faults from the earliest stage, and confidently tracks a bearings condition through the four stages of wear.

Reliability

In this chapter we will learn how to detect faults and monitor their condition; however it is important to first review what should be done in order to extend the life of the bearing. As a vibration analyst you may consider that it is your job simply to detect the fault. That should be only part of the job description. If bearings were correctly specified, transported, stored, installed, lubricated and operated, then they would provide reliable service for many years. Sadly, almost all of the aforementioned factors are not taken care of – as a result bearings only reach 10% of their design life time.

When bearings fail we may experience downtime, secondary damage, additional labor costs (overtime), and even injury to staff. We wish to avoid these unnecessary expenses.

But the question is, what can you do about the problem? What are you willing to do about the problem?

There are basically two paths we can take – and we should take both paths. We can do what we can to ensure that we get the longest life out of the bearing, and we can monitor the bearing so that we can take the most appropriate action before it fails.

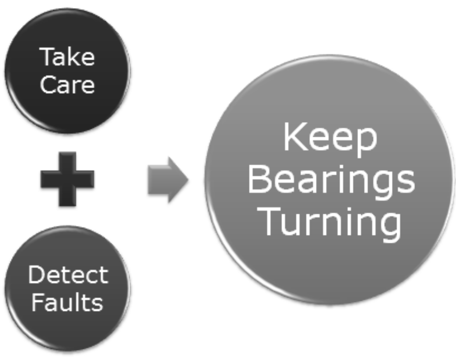


Figure 14-1

Poor design, poor installation practices, and poor maintenance leads to reduced bearing life.

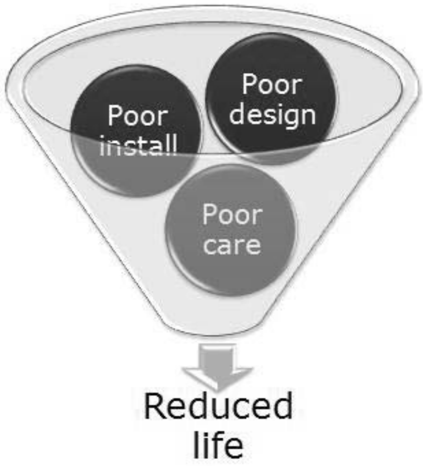


Figure 14-2

As stated earlier, and as generally agreed in the industry, less than 10% of bearings reach their design lifetime. The following graphic highlights the main reason why this is true.

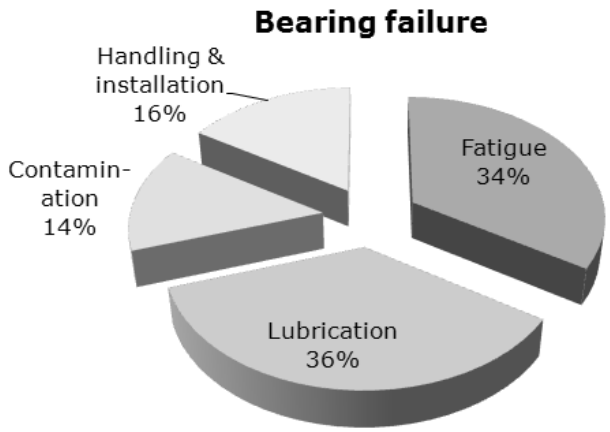


Figure 14-3

FATIGUE: 34%

- Normal or Pure Fatigue
  - With ideal design, installation, lubrication, and contamination, a bearing can be selected for infinite fatigue life
- Parasitic Loads
  - These loads, above and beyond design loads, reduce the fatigue life of a bearing
- Added loads decrease bearing life by a CUBED function (approximation, varies with bearing type)
- A 10% load increase from misalignment reduces the calculated bearing life by one-third!

LUBRICATION: 36%

- Is it really a lubrication failure?
- Separate the lubricant type from the lubrication process
    - Too Much
    - Too Little
    - Wrong Type
- Evidence of Improper Diagnosis and Analysis
    - Changing lubricant type to solve a lubrication problem
    - Failure to understand the application and environment

CONTAMINATION: 14%

- Bearing life theories are based on a contaminant-free bearing and lubricant
- Sources of Contamination
- Seal types
- Poor Maintenance Practice
    - Pressure washing
    - Installation techniques
- Often, the better the seal, the more sensitive to fits, tolerances, and shaft deflections
- What are the true root causes of contamination?

POOR HANDLING & INSTALLATION: 16%

- Perhaps the most avoidable of all bearing failure sources
    - Fits and tolerances are critical
    - Proper Tools and Skills are Essential
- Storage, Transportation
    - Before and after installation
- Failure mechanisms are created before the shaft makes its first revolution

The Proactive Approach

- Correct bearing for the application
- Proper bearing installation techniques
- Proactive Skills for balance, assembly, alignment
- Lubrication regime
- Storage, shipping, handling
- Proper operation
- Time to do all of these jobs right
- Training, Training, Training

It is clear that if the bearings are properly transported, stored, specified, installed, and operated, and if they are correctly lubricated, then they would have the best chance at remaining in operation for the longest time possible.

Condition monitoring

Condition monitoring programs have the potential to detect incipient bearing faults and to reveal the root causes of the bearing failures. Sadly, many programs do not take this extra step. Many programs can be described as follows: they collect data, analyze the vibration, then detect the faults (hopefully) and report their condition.

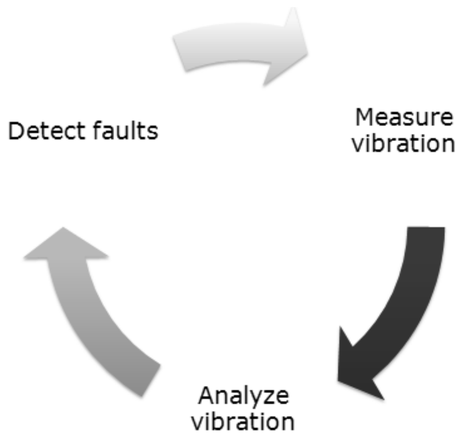


Figure 14-4

The focus is on detecting the bearing fault before it fails, recommending their replacement, and getting back to the measurement/analysis task. In many programs, conditions such as unbalance, misalignment, and resonance are not treated as seriously (often action is not taken when the conditions are reported). As vibration analysts, we can do more. We can employ additional technologies so that we have a complete picture of the health of the machine:

- Vibration analysis and acoustic emission
- Oil analysis and wear particle analysis
- Infrared analysis

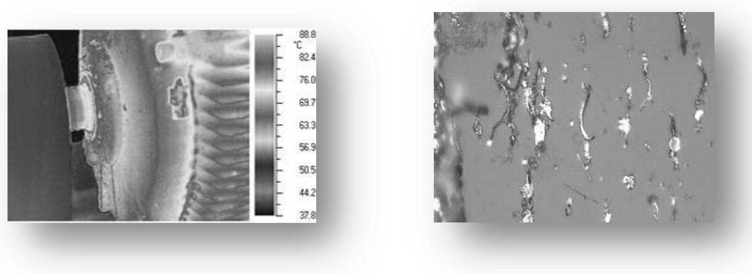


Figure 14-5

The Ideal Condition Monitoring Program

The ideal program would be well designed. Machines would be tested before they were accepted into stores after repair or purchase. Machines would be precision balanced and aligned. Resonance would be brought under control. Machines would be properly lubricated. When you diagnose faults and make your recommendation, not only would you provide a clear, actionable report, but you would verify your diagnosis and perform root cause failure analysis so that the fault would be less likely to occur again.

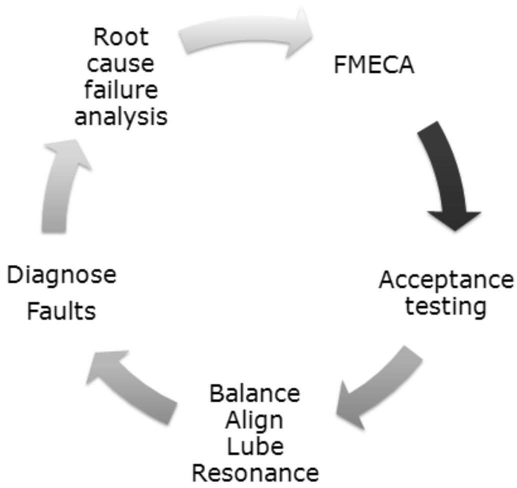


Figure 14-6

All of these steps are taken in the "world class" condition monitoring programs. The focus is on reliability; not just bearing fault detection.

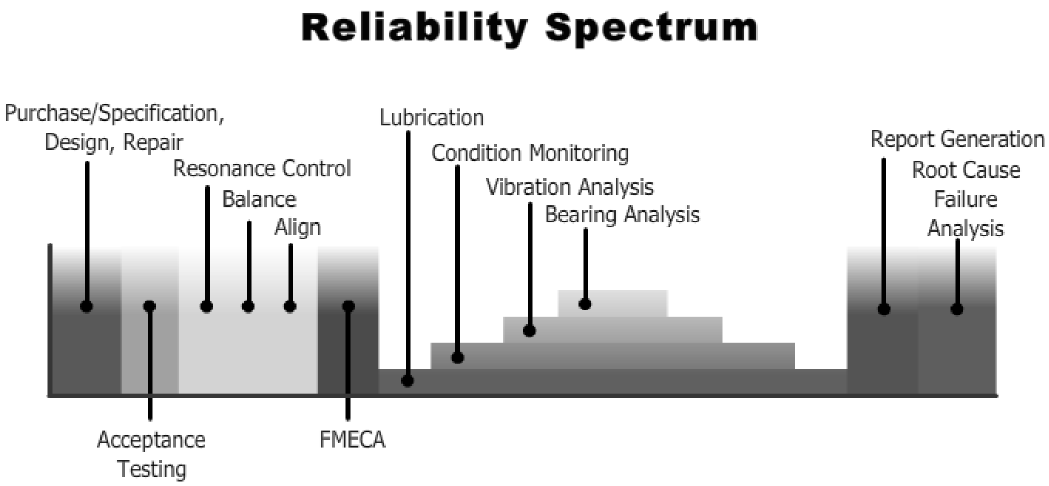


Figure 14-7

Bearings and lubrication

The lubricant between the rolling elements and the raceway is under tremendous pressure, especially in the load zone. In fact, the lubricant acts more like a solid than a liquid at this interface. The pressure is so great that there is deformation in both the rolling element and the metal of the inner and outer race. It is this constant, repeating pressure that ultimately causes the metal to fail.

Let's focus for a moment on the interface between the rolling element and the race.

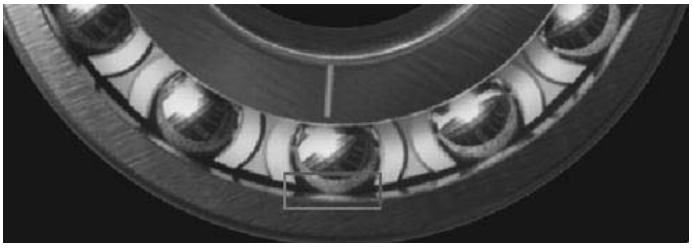


Figure 14-8

If we were to look at the surface of the bearing and race under a powerful microscope, we would see that the bearing is not perfectly smooth.

There are high-points and low-points on each surface. The lubricant's job is to keep those surfaces apart. But we are talking about very close tolerances. The gap between the two surfaces is very small: 0.5 μ m or 20 μ inches. The diameter of a human hair is 40 times as large as the gap!

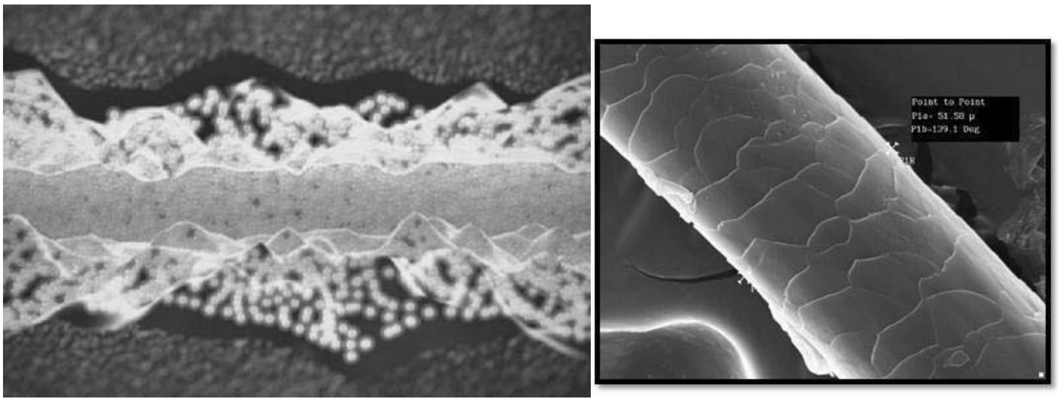


Figure 14-9

If the bearing is poorly lubricated, those surfaces will come closer together, and the high-points from each surface will come into contact. This contact is random, but the forces involved will both generate noise, and increase the level of wear.

Lubrication is very important. The volume of lubricant must be correct and it must have the properties intended for the application - the viscosity and other properties are designed to keep the surfaces apart and thus reduce friction and wear, and of course it should not be contaminated.

Acoustic Emission (Ultrasound)

Acoustic emission or ultrasound technology can be used to grease bearings in an intelligent way. First of all, ultrasound technology can tell you if the bearing needs grease. If the bearing does need grease, you can listen to the ultrasound as you grease it to ensure that you do not over grease it. As mentioned in the last section, over greasing is a major cause of bearing failure and ultrasound technology can help you address this root cause.

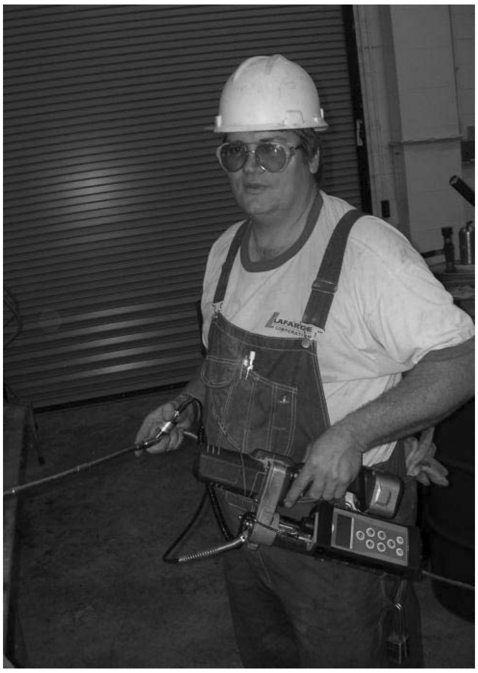


Figure 14-10 an ultrasound device integrated into a grease gun

What is ultrasound?

Ultrasound refers to frequencies above 20,000 Hz, which is the limit of human hearing. In other words, ultrasound refers to frequencies higher than what a healthy human ear can hear. You may ask how we can hear ultrasound if it is a higher frequency than the human ear can hear. The answer to this question is a technique called heterodyning. Essentially the frequencies are shifted down to a range in which we can hear them without altering other characteristics of the sound itself.

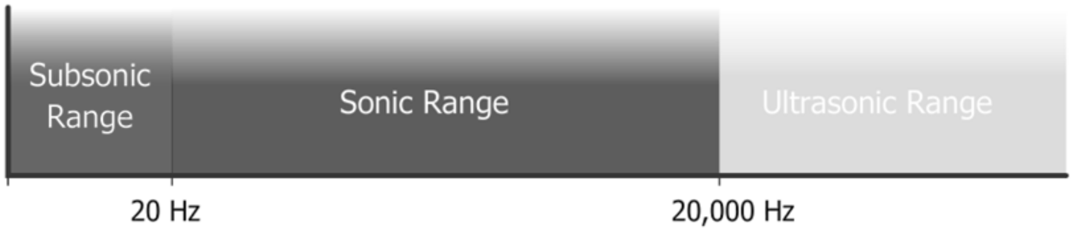


Figure 14-11 Ultrasound frequencies

Ultrasound: Qualitative

You can learn a lot from listening.

A low, muffled smooth noise, and stable readings, indicates a normal functioning bearing.

A high-pitched sound or a rushing sound indicates a lack of lubrication, an overload, or a rotating speed beyond the specification of the bearing.

A crackling noise, associated with unstable (or random) readings, indicates faults, wear or loose metal particles in the lubricant.

Figure 14-12 Listening to ultrasound

Friction generates sound in the ultrasonic range that can be detected by these devices. When there is not enough lubricant in the bearing, the bearing surfaces will begin to make contact with each other and this can be heard.

Ultrasound: Quantitative

Ultrasound can also be measured quantitatively as a dB value. Therefore, while greasing a bearing and listening to the ultrasound, one can also observe changes in the dB level of the ultrasound. Although you should get training in this technique before implementing it widely in your plant, we will describe here the general approach to using ultrasound to manage bearing greasing in a condition monitoring program. Here are the steps involved:

- Listen while you grease the bearing
- Watch dB value on ultrasound meter
- dB value will decrease as you add grease
- When there is enough grease dB value will rise. This means there is pressure building up in bearing
- Stop greasing – document dB level
- Use this level as a baseline
- Periodically measure dB levels on a route
- Use the following guide to determine if more grease is required

Essentially what you are doing is listening to the ultrasound and monitoring the dB value to determine when the bearing is greased correctly. When correctly greases the dB value is recorded and saved as a baseline. In the future one can simply measure this dB value on a route to see if it has increased. If it has then it means it is time to grease the bearing once again.

Below are some guidelines as to what increases in dB levels to look for once you have created a baseline for a bearing.

- Comparing readings to the baseline reading*
    - 8 dB increase indicates pre-failure or lack of lubrication
    - 12 dB increase indicates beginning of the failure mode
    - 16 dB increase indicates advanced failure condition
    - 35-50 dB increase warns of catastrophic failure

*these values come from a NASA study that was made available to us from UE systems

Bearing geometry and vibration

Thanks to the geometry of the bearing, we can calculate the defect frequencies that various faults will produce. This tells us where to look for these frequencies in our vibration data. Bearing defect frequencies can be calculated with formulas based on the bearing geometry or we can look them up in a database. More importantly however, bearing defects create unique general patterns that we can learn to recognize. Once we know how to recognize these patterns we no longer need to depend on the calculated frequencies which means we also do not need to know specifically which bearing is in our machine.

Defect frequency calculations

There are four key frequencies generated by rolling element bearings:

1. The Ball Pass Frequency – Outer race (BPFO). This is the rate at which a ball or roller passes a point on the outer race of the bearing. If there was damage on the outer race, we would expect to observe a periodic "pulse" of vibration at this rate.

2. The Ball Pass Frequency – Inner race (BPFI). This is the rate at which a ball or roller passes a point on the inner race of the bearing. If there was damage on the inner race, we would expect to observe a periodic "pulse" of vibration at this rate.

3. The Ball Spin Frequency (BSF). This is the angular frequency of the rolling elements. If there was a point of damage on one of the rolling elements, we would expect to observe a periodic "pulse" of vibration at this rate. In fact, because that point of damage may strike the inner race and outer race for each rotation, we may observe higher levels of vibration at twice this frequency.

4. The Fundamental Train Frequency (FTF). This frequency, also called the cage frequency, is the rotation rate of the bearing cage. It is the time it takes for a rolling element to complete one trip around the bearing. We do not always measure vibration at this frequency, but it is observed in other ways (we see sidebands of the FTF).

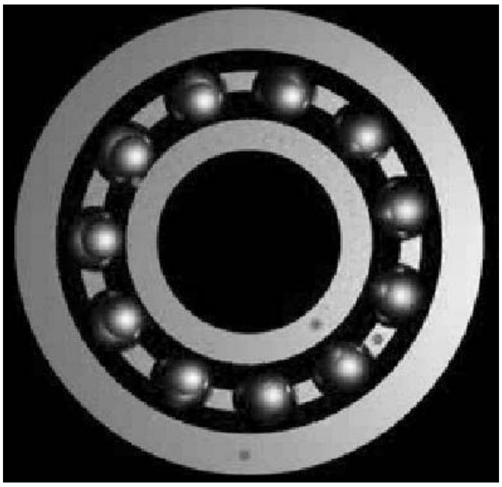


Figure 14-13

All of the following equations are based on the following dimensions.

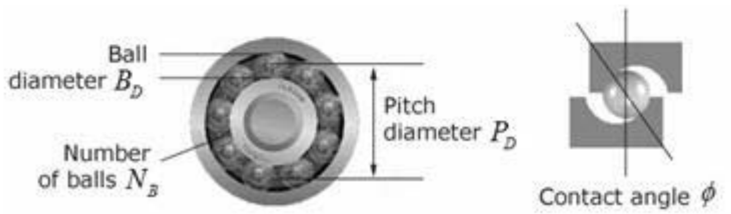


Figure 14-14

Ball Pass Frequency Outer race (BPFO)

We can compute the Ball Pass Frequency Outer race using the formula below. The calculation is independent of whether the inner race or outer race is rotating.

$$BPFO = \frac{N_B}{2}\left[1 - \left(\frac{B_D}{P_D}\right) Cos(\emptyset)\right]$$

This frequency will be non-synchronous (non-integer), and it will typically be in the range 4.2X-11.3X

This frequency can also be estimated as follows:

$$BPFO \approx \frac{N_B}{2} - 1.2$$

This will be the most commonly observed defect in rolling element bearings (on horizontal machines). The point at the bottom of the outer race is constantly taking the full load of the machine's mass and dynamic forces.

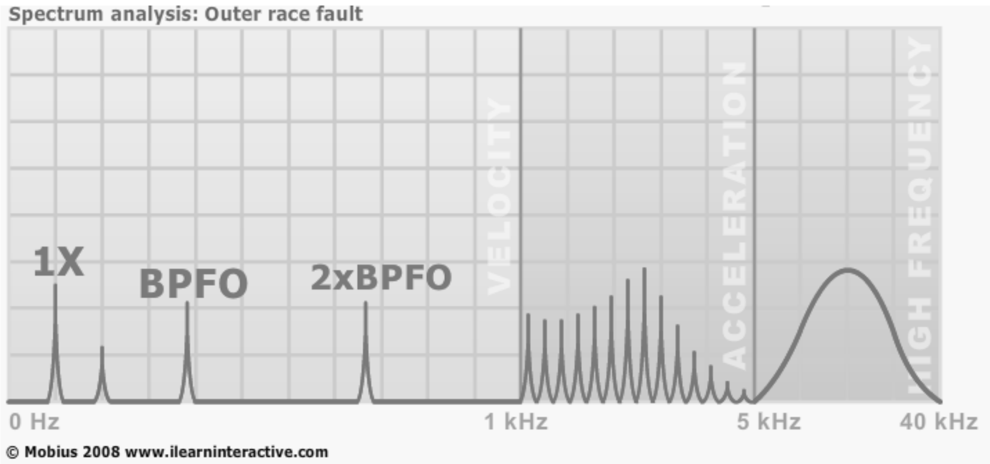


Figure 14-15

Ball Pass Frequency Inner race (BPFI)

The following formula can be used to calculate the Ball Pass Frequency Inner race (BPFI). This frequency is independent of whether the inner race is rotating or the outer race is rotating.

$$BPFI = \frac{N_B}{2}\left[1 + \left(\frac{B_D}{P_D}\right) Cos(\emptyset)\right]$$

This frequency will be non-synchronous (non-integer), and it will typically be in the range 6.3X-13.8X.

Note that this formula can be simplified to a much simpler calculation. If you know the number of rolling elements you can estimate the BPFI.

$$BPFI \approx \frac{N_B}{2} + 1.2$$

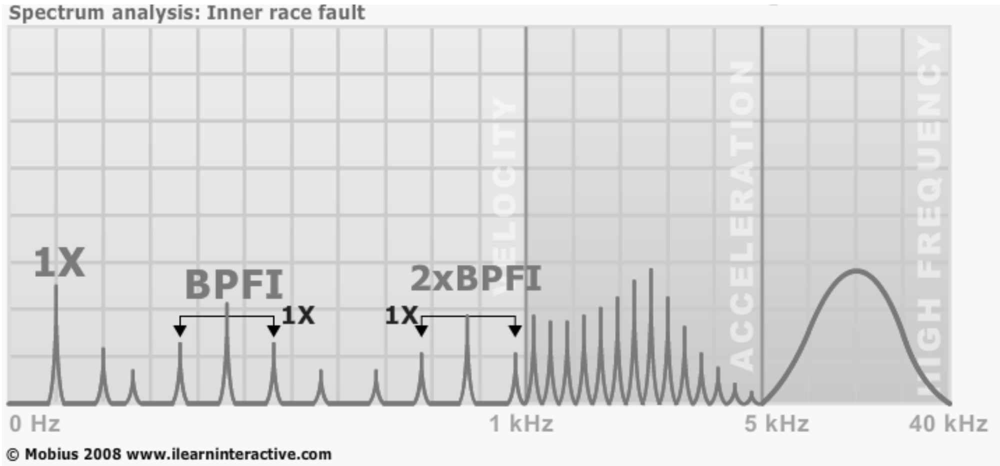


Figure 14-16

Because the defect on the inner race rotates around with the shaft, it also moves in and out of the load zone (which would be at the bottom of the bearing in a horizontal machine). In other words, the defect on the race will hit against the balls harder when the defect is in the load zone, and more softly when it is out of the load zone. This change in the forces is repeated each time the shaft goes around. Another way of saying this is that the inner race defect frequency is modulated by the shaft rate. In the spectrum this results in shaft rate sidebands around the inner race defect frequency.

What would happen if the inner race of the bearing was fixed but the outer race rotated (like in a car wheel)? The outer race defect would be modulated in this case and would have shaft rate sidebands around it. The inner race defect would not have shaft rate sidebands.

Fundamental train frequency (FTF)

The Fundamental Train Frequency, or cage frequency, can be calculated as shown below. The frequency is calculated differently depending upon whether the inner race is rotating (the most common situation) or if the outer race is rotating (for example on some conveyors and on wheel bearings).

Inner race rotating:

$$FTF = \frac{1}{2}\left[1 - \left(\frac{B_D}{P_D}\right) Cos(\emptyset)\right]$$

$$FTF \approx \frac{1}{2} - \left(\frac{1.2}{N_B}\right)$$

The FTF is commonly between 0.33X and 0.48X – however with certain thrust bearings, the contact angle is 90 degrees, and therefore the FTF will be 0.5X.

Outer race rotating:

$$FTF = \frac{1}{2}\left[1 + \left(\frac{B_D}{P_D}\right) Cos(\emptyset)\right]$$

$$FTF \approx \frac{1}{2} + \left(\frac{1.2}{N_B}\right)$$

The FTF is commonly between 0.52X and 0.67X (or 0.5X for the thrust bearings mentioned above).

The Ball Spin Frequency (BSF)

The ball spin frequency (BSF) can be calculated as shown below. The ball spin frequency (the angular frequency of the rolling elements) can be calculated the same way regardless of whether the inner race is rotating or the outer race is rotating.

$$BSF = \frac{P_D}{2B_D}\left[1 - \left(\frac{B_D}{P_D}\right)^2 Cos(\emptyset)^2\right]$$

This frequency will be non-synchronous (non-integer), and it will typically be in the range 1.9X – 5.9X.

As mentioned previously, the calculation computes the actual rotating frequency of the rolling elements; i.e. the number of times the rolling elements rotate per second (or minute). It is common to observe higher vibration amplitudes at twice these frequencies because a spall on the rolling element may come in contact with both the inner race and inner race per rotation – thus there are two events per rotation of the rolling element.

(In fact, these calculations determine the ratio of the ball spin frequency to the turning speed of the shaft – to compute the BSF in Hz or CPM it must be multiplied by the turning speed of the shaft.)

This calculation can be simplified as follows to provide an estimate of the frequency when only the number of rolling elements is known.

$$BSF \approx \frac{1}{2}\left[\frac{N_B}{2} - \frac{1.2}{N_B}\right]$$

This is important to know in the event that you observe a peak (or peak with harmonics) that you suspect may be a bearing defect frequency. If you can estimate the number of rolling elements you can repeat the calculation ± 2 rolling elements and see if one of the calculations comes close to the frequency you observe.

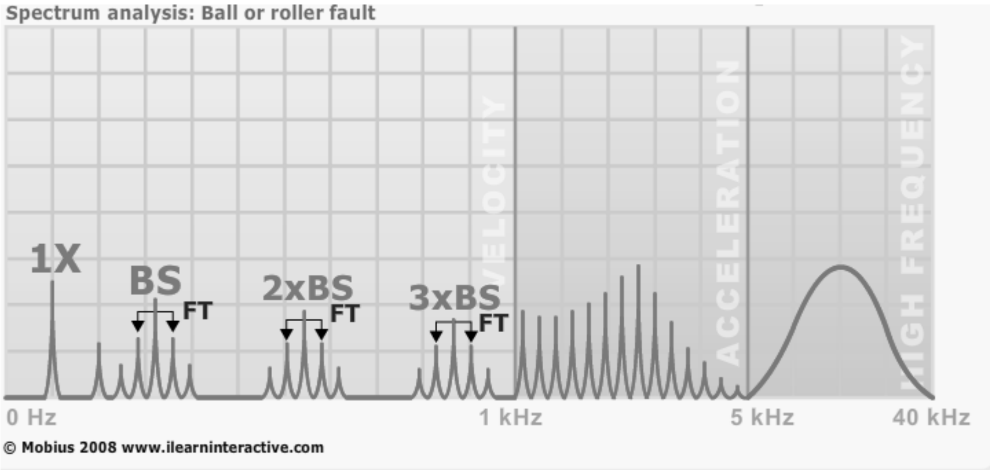


Figure 14-17

The balls or rollers travel around in the cage, which rotates at the cage rate (FTF). A particular ball with a defect on it will therefore travel in and out of the load zone in the bearing each time the cage goes around. In other words, the ball spin defect frequency will be modulated by the cage rate. In the vibration spectrum we will therefore expect to see FTF sidebands around BSF.

Bearing defect frequency tips

Please note that the defect frequencies are not exact. Due to slippage, skidding, wear, and imperfect bearing details (i.e. the dimensions may not be perfectly accurate), the frequencies may be off by a small amount.

When we do not have the part number of the bearing, or we suspect that the information we have is incorrect, then we can still estimate the defect frequencies. There are a few "rules of thumb" that you may find useful to know.

RULE OF THUMB #1

First, if you add the BPFI to BPFO then you can compute the number of rolling elements. For example, if BPFO is 3.2X and BPFI is 4.8X then we know that the bearing has 8 rolling elements.

This is particularly useful to know because if we observe two series of peaks that we suspect are bearing defect frequencies, one BPFI and the other BPFO, then we can add them together to see if they add up to a whole number. If they do then your confidence about the origin of the vibration will increase.

RULE OF THUMB #2

Second, if you know how many rolling elements are in the bearing then you can estimate the defect frequencies as described in the earlier formulas, or more simply as follows:

$$BPFO \sim 0.4 \times \#Balls$$

$$BPFI \sim 0.6 \times \#Balls$$

This generalization should only be applied for bearings with 8 to 12 balls/rollers.

RULE OF THUMB #3

If we do know the bearing part number (and a bearing database) we may still find that there are no peaks in the spectrum that line up with the frequencies in the table. This may mean that the bearing is OK, or the bearing information is incorrect. If are peaks at frequencies that are "close by" that appear to be bearing defect frequencies (there are harmonics, they are non-synchronous, and there are FTF or 1X sidebands) then it would be useful if we could confirm that they are in fact defect frequencies. Well, there is a way to do just that:

$$BPFO\left(\frac{\#B}{R_{N\pm1,2,3..n}}\right) = BPFO \pm FTF \times n$$

Based on the defect frequency equations provided earlier, we can learn that the difference between the BPFO from a bearing with 13 rolling elements and the "same bearing" with 14 rolling elements is equal to the cage rate (FTF) of the bearing.

For example, if we look in this table of bearing information for a 22312 bearing, we can calculate the difference between the defect frequencies when different numbers of rolling elements are used. If we believed that the bearing had 13 elements however the peaks in the spectrum did not line up, we can add (or subtract) FTF (0.409) to the quoted frequencies to estimate the BPFO, BPFI and BSF for bearings with 12, 14, 15 and 16 rolling elements. We can perform the calculations and compare them to the results provided in the table. As you can see below, they correlate quite well.

Example using 22312 with 13 rolling elements. Looking at the table below we can see that the FTF is 0.409. We can now look at the other lines in the table for different numbers of rolling elements. We can compute the estimate and compare to the table.

- n=14 » BPFO = 5.312 + 1 x 0.409 = 5.721

- n=15 » BPFO = 5.312 + 2 x 0.409 = 6.13
- n=16 » BPFO = 5.312 + 3 x 0.409 = 6.539

BEARING TYPE	#B/R	FTF	BSF	BPFO	BPFI
22312	15	.413	2.697	6.192	8.808
22312	16	.411	2.665	6.584	9.416
22312	16	.411	2.665	6.584	9.416
22312	16	.411	2.665	6.584	9.416
22312	16	.411	2.665	6.584	9.416
22312	16	.411	2.665	6.584	9.416
22312	14	.409	2.570	5.721	8.279
22312	14	.409	2.570	5.721	8.279
22312	13	.409	2.570	5.312	7.688
22312	14	.402	2.385	5.626	8.375
22312	16	.411	2.665	6.584	9.416

Figure 14-18

RULE OF THUMB #4

It is also helpful to know that bearings of the same dimension series have very similar defect frequencies. The second and third digit of the bearing number represents the width and diameter of the bearing. So if you are unsure of the exact bearing number, but believe you know the dimension series, then you can look up the bearing database to get an estimate of the defect frequencies.

This is not an exact science, but it can help a great deal if you are not confident that the changes in the spectrum you are observing are related to a bearing with increasing wear.

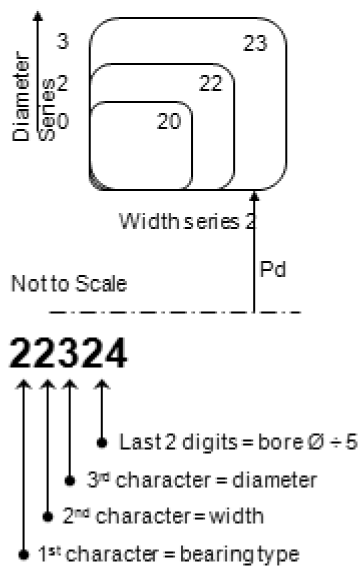


Figure 14-19

To aid in this process we can recognize that certain types/series of bearings are commonly used in the same applications. The first digit of the bearing relates to the type of bearing, as shown below.

Figure 14-20

As a guide only, the following table shows the type of bearing commonly used in the listed applications.

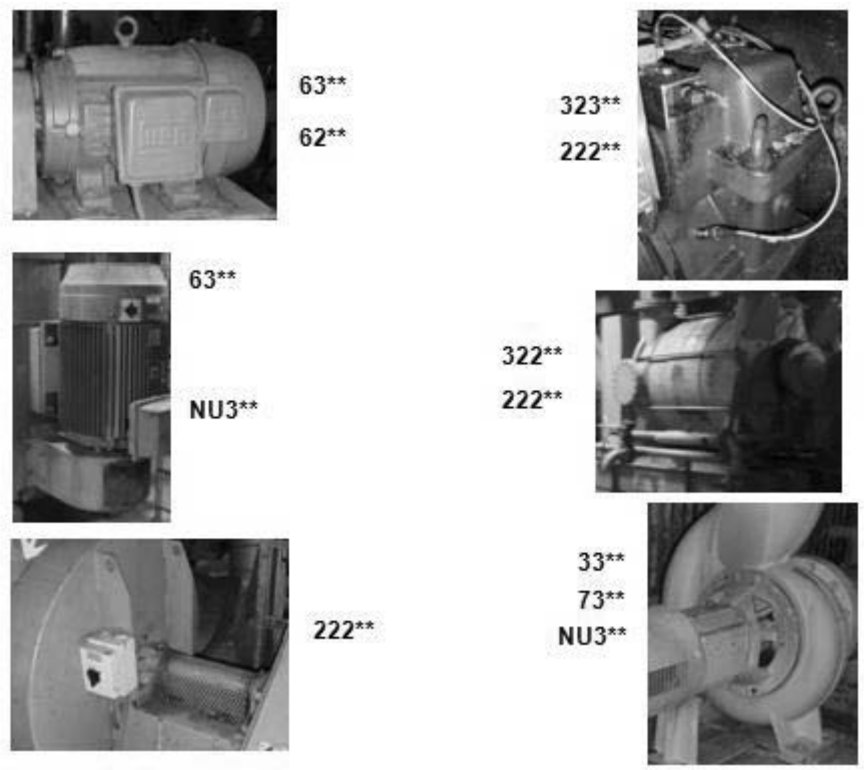


Figure 14-21

Again, it must be stressed that this is not an exact guide; instead it is provided to help you make an "educated guess" about the approximate defect frequencies. If you know the bearing type and/or application and/or number of rolling elements, and you have a bearing database, then this information should help you be more successful detecting bearing fault conditions.

Bearing defect frequency summary

There is another key observation we can make; all of the defect frequencies are non-synchronous – that is, they are non-integer multiples of the shaft turning frequency. We will soon learn in detail that bearing faults (wear, spalls, cracks) will generate harmonics in the spectrum, and there will often be 1X sidebands (inner race fault), FTF (<0.5X) sidebands (ball/roller damage), or no sidebands (outer race fault). If the outer race is spinning the BPFI will ***not*** have sidebands, BPFO ***will*** have shaft rate sidebands.

Therefore we can detect bearing faults without knowing the bearing number with a fair degree of confidence. Coupled with the rules of thumb that we have just covered, we can be quite confident of detecting faults without a great deal of supporting information.

Vibration – The complete picture

It is important to understand the nature of the vibration generated by rolling element bearings. We just mentioned the high frequency vibration that is generated, but as we will learn in greater detail, we can break this vibration up into four components:

1. Friction: There will always be some friction between surfaces, regardless of how good the lubricant may be. As the lubricant degrades (too much, too little, lubricant additive pack degradation, or contamination), friction will increase. Frictional vibration is non-periodic. Friction generates broadband noise – vibration at a wide range of frequencies.

2. Stress wave: If we have metal-to-metal contact we will generate very high frequency, very short duration pulses. These pulses, or shock waves, may be random in nature or periodic – it depends upon the reason for the metal-to-metal contact.

3. Periodic vibration: When an even occurs periodically, at sufficient amplitude to be detected, the vibration will be present in the time record and a peak will appear in the spectrum. Common examples are the vibration related to unbalance or the vane pass rate, however when there is damage on the inner race of a bearing, for example, a periodic source of vibration will also be generated.

4. Resonance: When surfaces impact they generate vibration, *and* the structure and machine components (and the attached accelerometer) will resonate as a result. The natural frequency may be quite high when dealing with vibration from bearings. We can detect the vibration from those resonances because the vibration is amplified and because the

frequencies are distinct from the much higher amplitude vibration that masks the lower frequency vibration generated by the bearing.

This vibration can be detected using a number of techniques, summarized below:

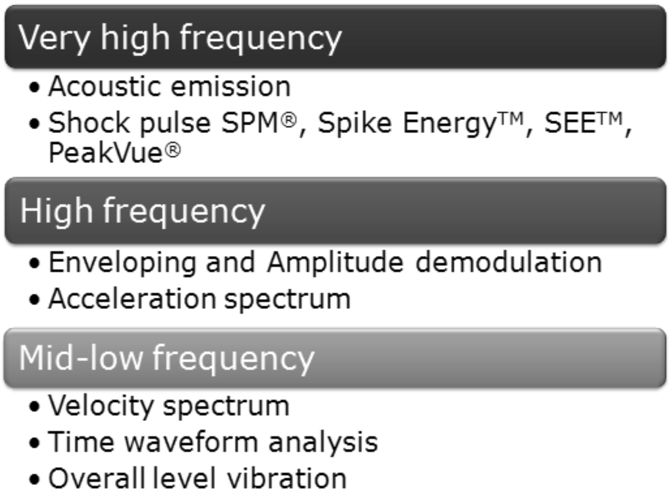


Figure 14-22

The vibration generated due to poor lubrication or minor bearing damage is very high in frequency, but low in amplitude. The frequency generated will be over 20 kHz in the earliest stage – some systems look at much higher frequencies to detect the first signs of bearing wear. We need techniques to remove the low frequency, high amplitude vibration in order to detect this source of vibration. Techniques such as acoustic emission, Shock Pulse, Spike Energy, and PeakVue are designed to do just that.

Enveloping and demodulation are also designed to monitor high frequency vibration and detect first stage bearing wear, however some systems are not designed to monitor such high frequencies, and many systems capable of detecting these high frequencies are not set up to do so.

As the fault develops the vibration generated is lower in frequency. The high frequencies are still generated, however now lower frequency vibration (over 2 kHz) is generated. Time waveform analysis and spectra (in units of acceleration) can be used to detect the fault and enveloping/demodulation can be used.

With increased wear comes higher amplitude, lower frequency vibration. Now the conventional velocity spectra will display signs of bearing wear.

And finally, as the fault develops further, and damage becomes more severe, low frequency noise is generated, and some of the characteristic patterns we expect to see begin to change to vibration patterns normally associated with looseness. At this stage we may not measure the high frequency vibration; there is too much damage.

Stage One Bearing Faults

The following points summarize the characteristics observed when a bearing is in Stage One:

- Sub-surface damage only
  - Friction and minor impacts
- Very high frequency vibration
  - Friction: greater than 20 kHz in earliest stage
  - You can't hear it (without assistance)
  - 'Noise' due to inadequate lubrication
  - Very low levels
  - *Very* short duration impacts
- 'Stress waves' or 'Shock pulses'
  - 1 kHz » 15 kHz
- Traditional vibration analysis techniques are inadequate at this stage.

The following graphic provides a summary of the change you will see in the spectrum. Note that the velocity spectrum will not show any sign of the fault condition.

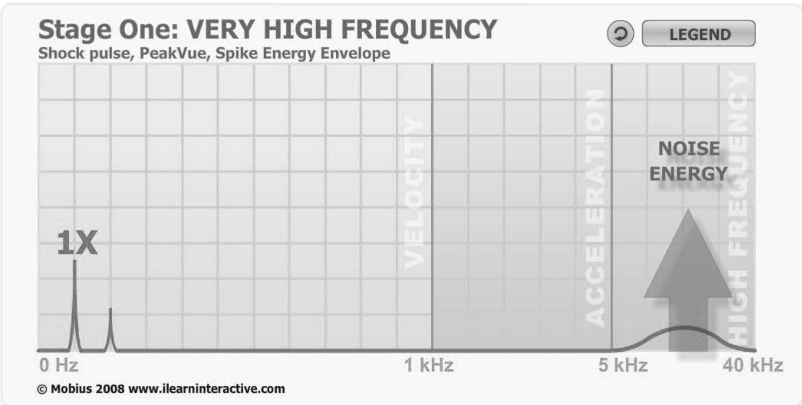


Figure 14-23

The effectiveness of each bearing detection technique is summarized below:

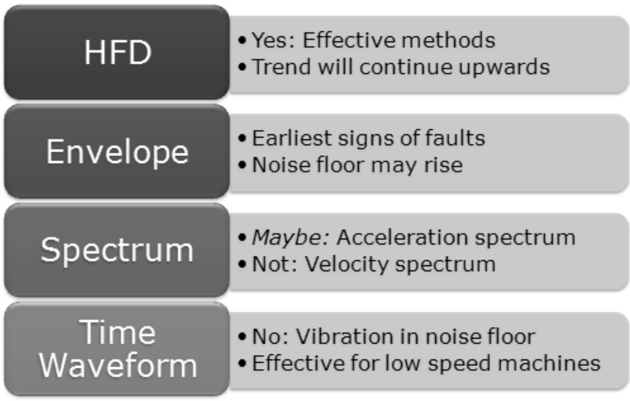


Figure 14-24

The following table summarizes the possible actions you could take at this stage:

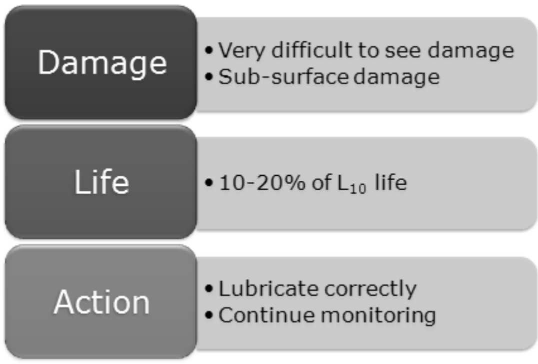


Figure 14-25

Stage Two Bearing Faults

The following points summarize the characteristics observed when a bearing is in Stage Two:

- Sub-surface damage only
  - Friction and minor impacts
- Very high frequency vibration continues to increase in amplitude.
- Vibration significant enough to excite ‘resonances’.
  - Structural, bearing, and sensor
- Envelope (demodulation) spectrum should show signs
  - Defect frequencies present in spectrum
  - Velocity spectrum still won’t indicate fault. Acceleration spectrum should indicate fault.

Figure 14-26

The following graphic provides a summary of the change you will see in the spectrum. Note that the velocity spectrum will not show any sign of the fault condition.

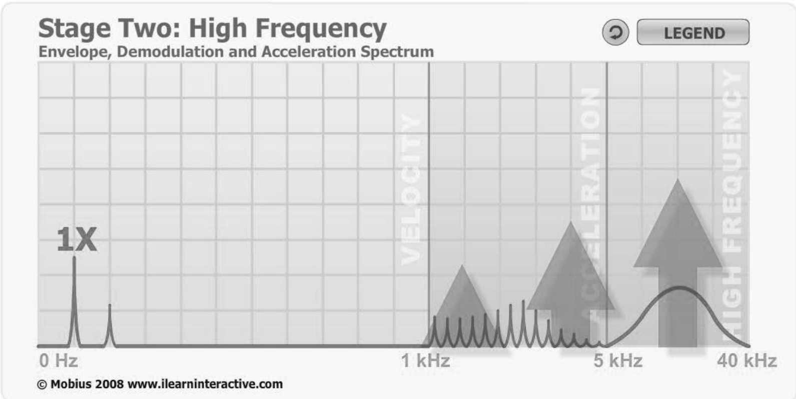


Figure 14-27

The effectiveness of each bearing detection technique is summarized below:

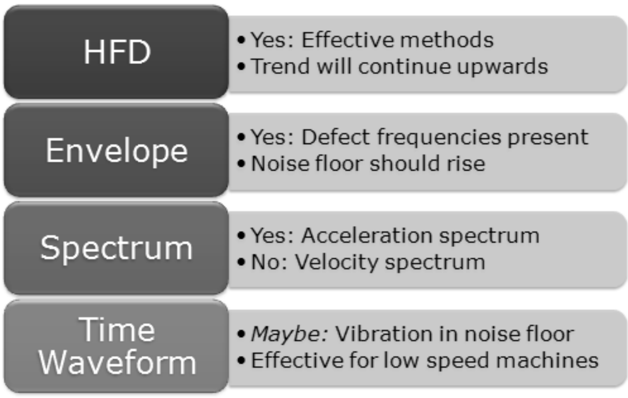


Figure 14-28

The following table summarizes the possible actions you could take at this stage:

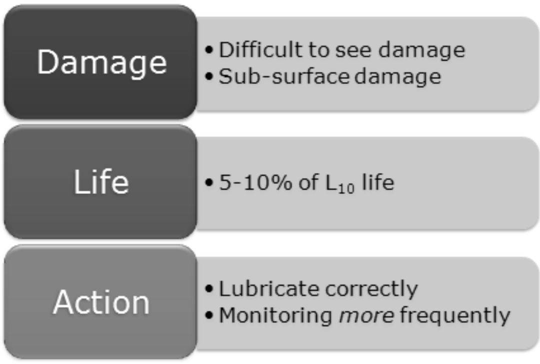


Figure 14-29

Before we discuss stage three and four bearing faults, we will discuss the high frequency vibration analysis techniques.

Demodulation/enveloping

Demodulation and enveloping are two names that refer to the same technique. These techniques take the high frequency vibration that we observe as bearings fail and make it available as low frequency vibration that we can analyze.

In order to understand what is happening, consider ball or roller impacting a spall in the outer race. As we have seen, the roller will pass or hit the spall at a certain rate as dictated by the bearing fault frequency calculations. This gives us the repetition rate of the impacting and this is what we look for in the normal velocity spectrum. Perhaps there are 3.4 hits impacts per revolution of the shaft, the frequency we look for is 3.4x.

Now consider that each impact actually generates a high frequency vibration or sound. You can think of it as the sound the bearing makes when the roller hits the spall. This sound will also happen repetitively each time the spall is hit by a ball. In brief, what demodulation techniques do is listen for this high frequency "sound" occurring and then using a bunch of filters, the technique tells us the repetition rate of the impacting – which is the bearing frequency.

There are a number of steps in the process.

Step one: High-pass or Band-pass Filter

We are working with a vibration signal that is dominated by low frequency, high amplitude vibration that comes from the unbalance, misalignment and other conditions. The vibration signal also contains a very high frequency, very low amplitude signal. It is very difficult to detect this signal without special processing. The following is an example, however the bearing vibration is GREATLY amplified.

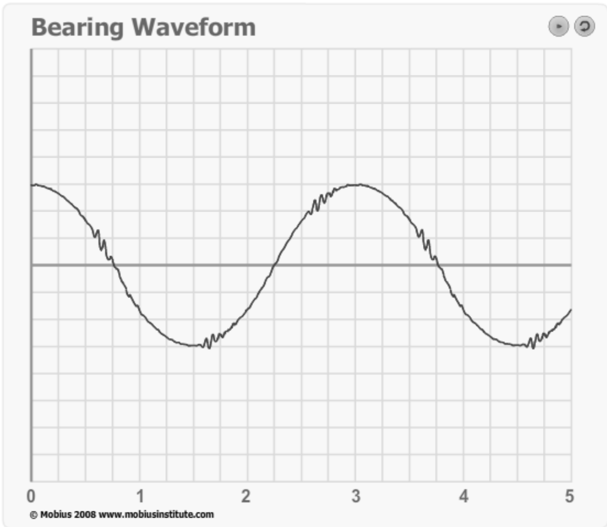


Figure 14-30

Now we will exaggerate the vibration amplitude still further.

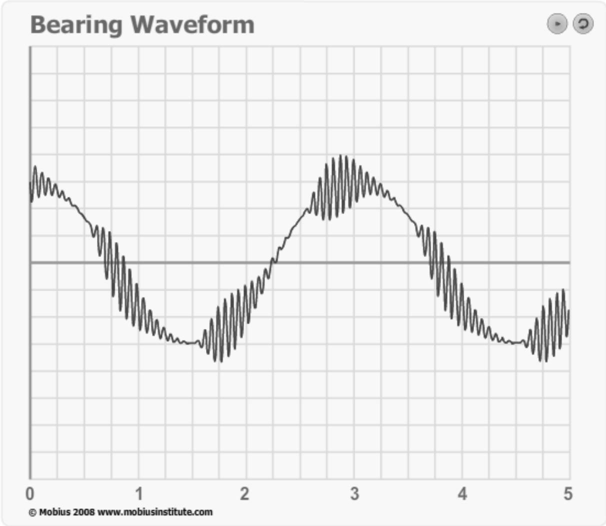


Figure 14-31

Once we filter the vibration signal we will have only the vibration that existed within the filter characteristics. The intent is for the bearing ringing to be captured. We must be careful to set the filter values so that we do not also capture vibration components from the motor or gearbox, etc.

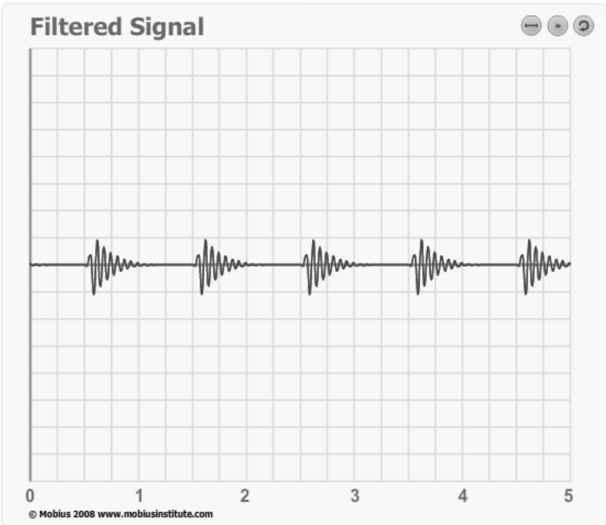


Figure 14-32

What we have now is a signal that shows only the periodic ringing of the bearing (or accelerometer, or of some other metallic component). We do not care about the frequency of the resonance (i.e. the high frequency component. All we want to know is whether we have ringing at all, what the amplitude is, and how often the ringing occurs – that should correspond to the bearing defect frequency. In this example we are simulating the vibration from a defect on the outer race.

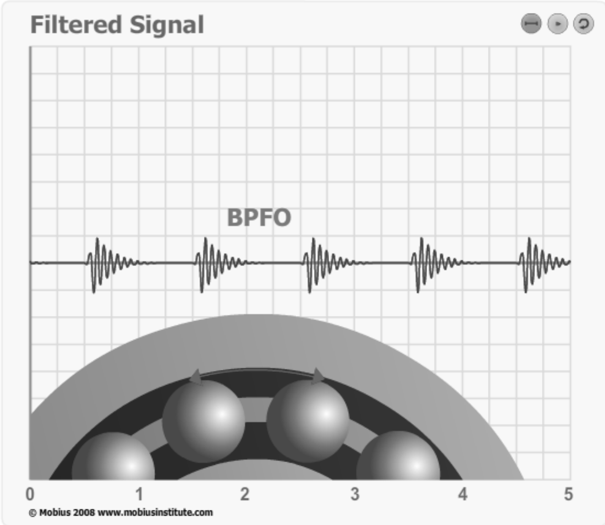


Figure 14-33

Step two: Rectify (or Envelope)

The task at hand is to capture the ringing transients and create a low frequency signal that we can view in a time waveform or spectrum. We can do this by "rectifying" the signal. (Later we will discuss the Spike Energy and PeakVue techniques that do this a little differently).

If we pass the signal through a "full-wave rectifying" in the analog domain, or perform the same task digitally, the negative-going vibration is folded over and made positive.

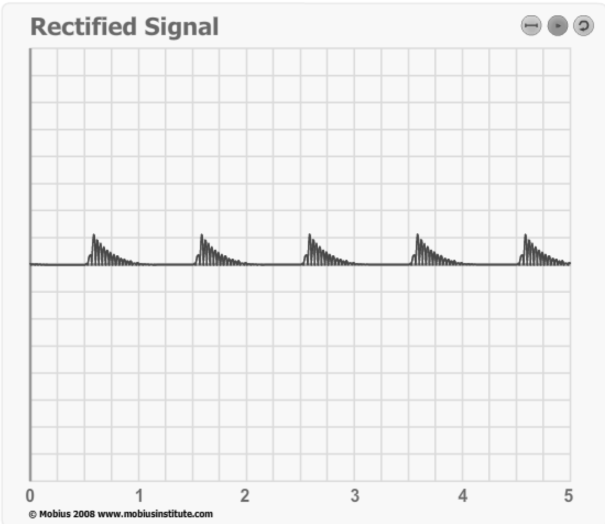


Figure 14-34

Step Three: Low pass filter

Now that we have the rectified signal, we can pass it through a low-pass filter. This has the effect of removing the high-frequency ringing. However, because we rectified the signal, we will be left with a waveform that is the same basic shape as the pulses (it looks like an envelope of the data).

The low pass filter may just be the anti-aliasing filters used to generate the spectrum. You must select an Fmax for the envelope/demod spectrum.

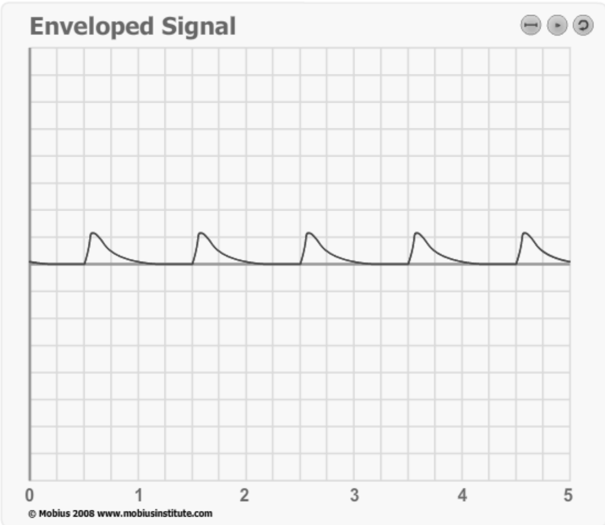


Figure 14-35

Setting up the measurement

The enveloped signal can be routinely collected and should be included in normal route measurements. Both the demodulated time waveform and spectrum can be analyzed. Remember that although internally, the demodulation technique is looking at the high frequency sounds made by the impacting in the bearing, what it returns to you is the repetition rate of the impacting, which is the bearing defect frequency and is relatively low. Therefore, when setting up the measurement, the demodulated spectrum should be configured with an Fmax setting equal to about 3.5x the calculated inner race defect frequency for the bearing. If you do not know the inner race defect frequency and the bearing has between 8 and 12 balls or rollers then you can make the Fmax 15x to 20x.

Step four: Analyze it

We can now analyze the waveform or spectrum. If we see peaks separated by BPFO, BPFI or BSF, then we can be confident that there is bearing wear – a defect exists.

We can use multiple band-pass or high-pass filter settings to focus on different frequency spans in order to assess the severity of the fault; the vibration appears in the higher frequencies first.

We can also study the demodulated (enveloped) spectrum and compare the height of the defect frequency peaks to the height of the noise floor.

If the bearing is in good condition, then the demod spectrum will be all noise (assuming that nothing else is generating impacts: gear wear, looseness, etc.).

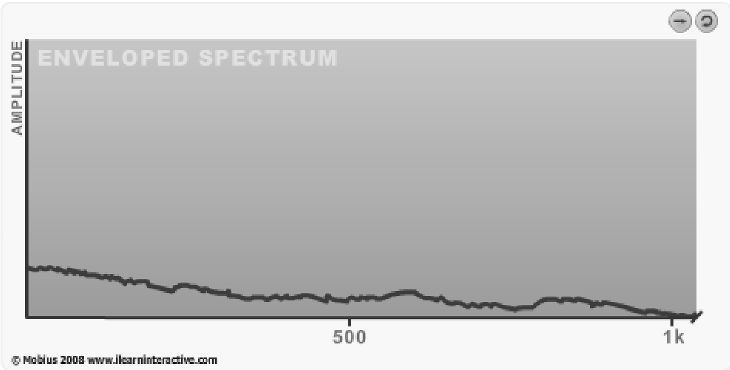


Figure 14-36

Once the fault begins to develop, the peaks will begin to grow. With most products on the market, it is not possible to state what the levels should be in order to assess the condition of the bearing. Instead it is a case of watching what frequencies are present and to compare the height of the peaks to the height of the noise floor.

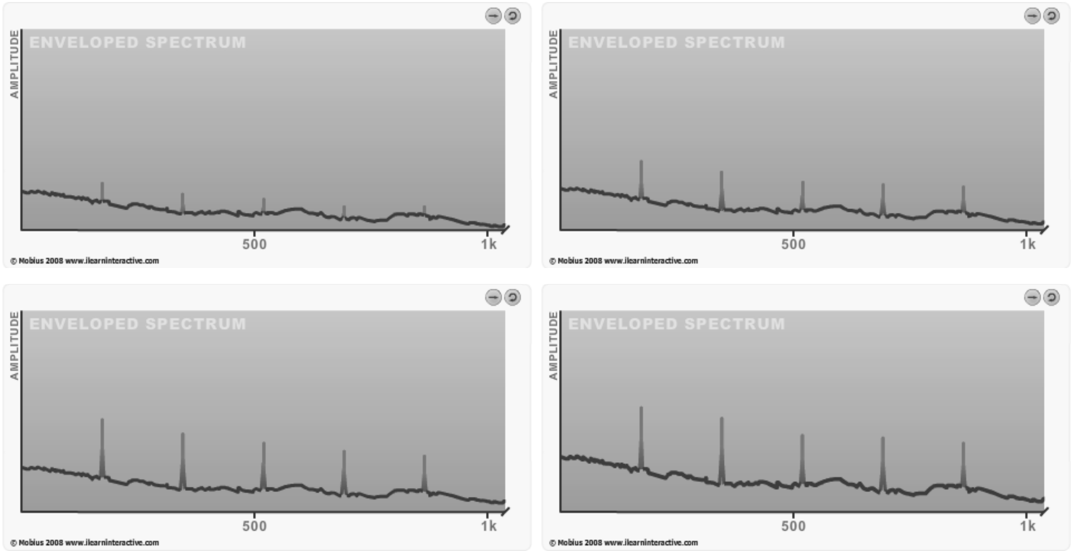


Figure 14-37

Once the noise floor begins to lift and swallow the peaks, so know that the bearing is in stage four and will fail soon.

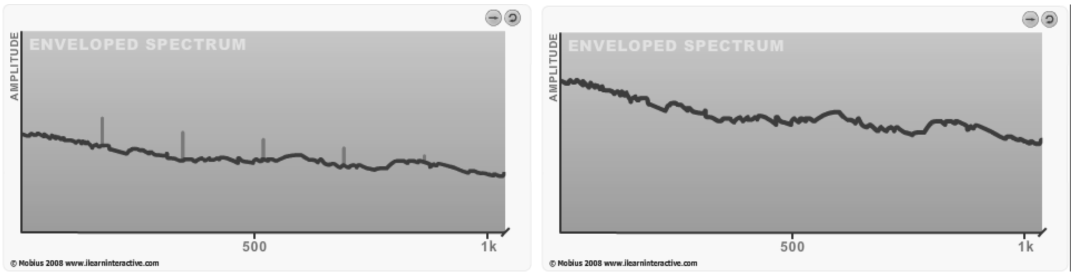


Figure 14-38

Four uses for high frequency tests

1. Early warning: Low frequency high amplitude vibration is filtered out leaving us with the relatively low amplitude high frequency “sound” of the impacts in the bearing. This allows us to “hear” this low level sound long before it appears in the normal velocity spectrum.

2. Confirmation: The high frequency tests are taking a different route to determine that repetitive impacting is happening in the bearing. If the (bearing) frequencies contained in the high frequency test also appear in the velocity spectrum or vice versa it gives us more confidence that this is in fact a bearing defect and not, say, vibration coming from another machine.

3. Location: High frequencies do not travel as far as low frequencies and therefore the high frequency tests are much more sensitive to the location of the measurement. It is

possible to see the same bearing tones in velocity spectra collected from 4 bearings on the machine because the vibration travels from one to another. If the same frequencies exist in the high frequency test at only one bearing then that is the bearing with the defect.

4. Low speed machines: It is difficult to measure low frequency vibration with an accelerometer. If the bearing defect frequency is a low frequency it may be difficult to measure directly however, the "sound" of the impacts and the stress waves created by the impacts are still high frequency even if the repetition rate of the impacting is not, therefore the high frequency tests are still valid.

Cautions

1. VFD's and SCR's have high frequency switching circuits that can confuse the demodulation routines in our data collectors. This high frequency switching looks like a bearing defect even though it is electrical.

2. Reciprocating machines: Demodulation is looking for repetitive impacting – that's what a reciprocating machine does! So the demodulation routine may simply pick up the piston rate rather than a bearing defect.

3. Don't overhaul too early! Remember that these high frequency tests are very sensitive. It is important to wait until you see indications of bearing wear in the velocity spectrum before doing an overhaul (except in the case of low speed machines). It is also important to gain historical information about your machines and how their failure patterns relate to the vibration data. It is possible to see clear signs of bearing wear in a high frequency test but upon opening up the bearing only see very tiny indications of wear.

Shock Pulse Method

In this Category II course we will not go into great detail about the shock pulse method, however given that it is used by the SPM Instrument company and the PRÜFTECHNIK company it is important to understand the difference between this technique and the enveloping/demodulation technique.

In the previous section we discussed the need to set up a high pass or band pass filter in order to remove the high amplitude, low frequency vibration and focus on the high frequency vibration generated by the bearing. The shock pulse method also uses a band pass filter, but it does so in combination with the resonant frequency of the transducer.

The enveloping/demod method relies on natural frequencies in the machine/bearing structure to amplify the vibration (or stress wave or shock pulse) that results from the bearing defect (or from a lubrication problem). It is not possible to predict where the natural frequencies are in a bearing or structure, so mounting a sensor with a known natural frequency removes the need to determine the optimal filter settings. In the case of the shock pulse method, the transducers used have a natural frequency of 32 kHz (SPM) or 36 kHz (PRÜFTECHNIK). When poor lubrication or a bearing defect exists, the transducer is 'excited' and the vibration is amplified.

It should be noted that most of the commercially available accelerometers do not have a controlled natural frequency; the frequency and response can be very different from one accelerometer to the next, even accelerometers with the same model number.

As with enveloped data, the shock pulse method can provide a time waveform or spectrum that can be analyzed. In addition, two values are available (with a third derived from the first two) may be trended or compared to alarm limits. One value is proportional to the "carpet" or noise level of the vibration (dBc), and a "peak" or maximum value that is proportional to the peaks that stand out above the carpet level (dBm).

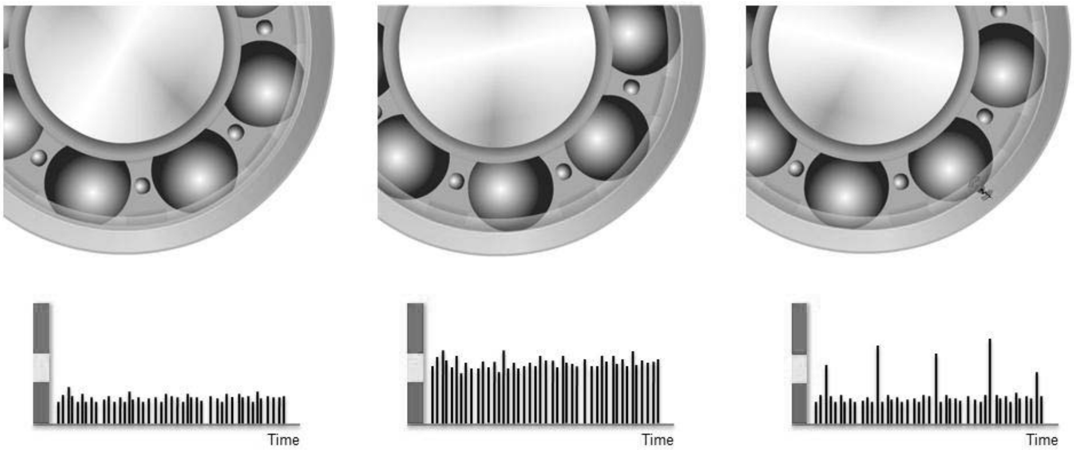


Figure 14-39 Artwork from the SPM Instrument company showing different carpet and peak values.

By analyzing changes in the carpet level (which can indicate lubrication problems) and the peak levels, which can indicate the presence of bearing damage, it is possible to determine what action should be taken.

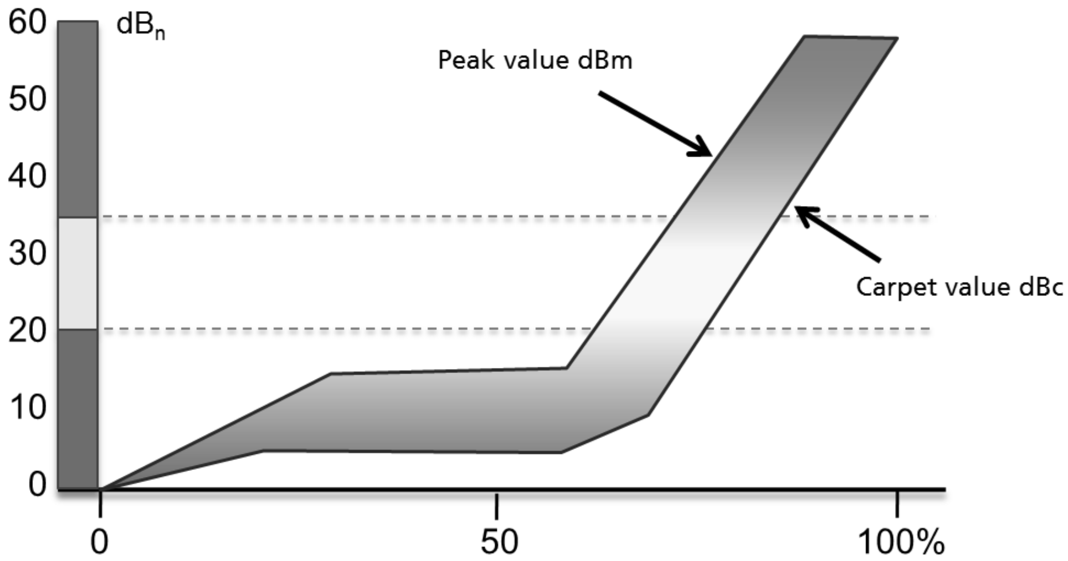


Figure 14-40 Graphic from SPM Instrument. Trending the carpet and peak values over the life of the bearing

Spike Energy Method

The spike energy method was once a very popular and powerful method. It is designed to work in a very similar way to the shock pulse method. Charts were developed that enabled users to compare spike energy readings to pre-defined limits in order to assess the severity of a bearing fault.

However, when the spike energy method was originally developed by the IRD company, it was used in association with an accelerometer manufactured by the company. As discussed in the shock pulse section, the natural frequency and response of this sensor was known which therefore meant that the readings could be correlated to the severity of the bearing condition.

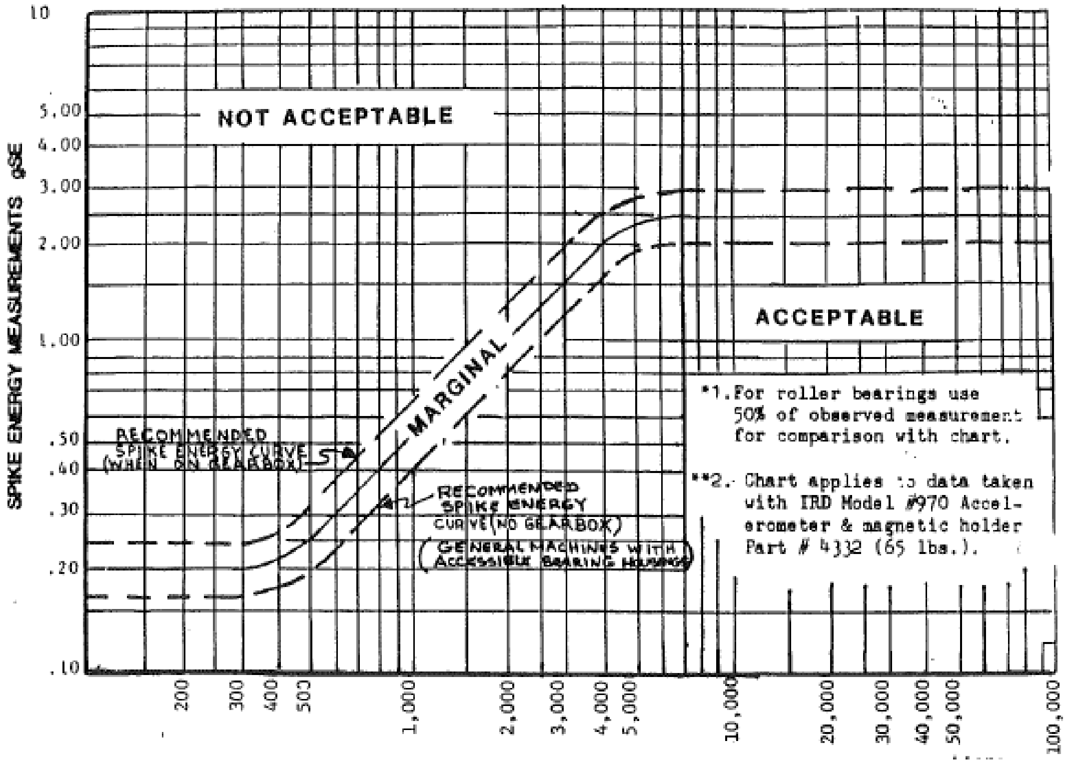


Figure 14-41 Spike Energy fault severity chart

Unfortunately, the production of that sensor ended many years ago, and now the accelerometers provided with the systems that utilize the spike energy method do not have a controlled natural frequency or response. That means that the charts are no longer usable.

Spike energy can still be used as a diagnostic tool in the same way as acceleration enveloping; to look for signs of bearing wear via the waveform or spectrum. However, while the spike energy amplitude (in gSE) can be trended, it should not be compared to the severity chart. Note that if the accelerometer is changed for any reason, the amplitude of the readings will change.

PeakVue Method

Unlike the shock pulse and spike energy methods, the PeakVue method does not rely on the natural frequency of the transducer. The PeakVue method uses a signal processing technique to detect signs of bearing wear using a similar technique as acceleration enveloping.

Like the enveloping technique, the PeakVue method requires you to select a filter setting that is appropriate for your application (based on machine speed and type). The original enveloping implementations utilized analog filters and rectifier (enveloping) circuits and thus could not detect faults from the earliest stage. More modern implementations utilize digital techniques to perform the enveloping process. The PeakVue technique goes one step further.

The PeakVue technique samples (digitizes) the vibration at a very high rate in order to detect any short duration "stress waves" that are generated in the earliest stage of the bearing fault. If the signal from the sensor is not sampled quickly enough these "stress waves" can be missed:

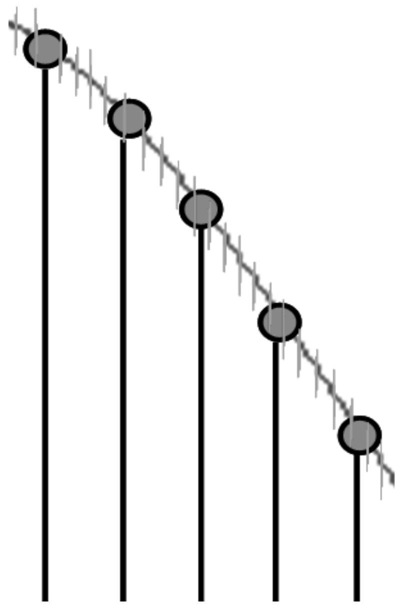


Figure 14-42 Stress waves going undetected due to low sample rate.

The manufacturer, the CSi division of Emerson Process, claims that sampling the signal at a very high rate (102.4 kHz) ensures that the stress waves will be detected. The system uses a "peak detection" process to capture the amplitude of these stress waves.

The resultant waveform can be sampled as a spectrum or as a waveform; both of which provide an early warning of the bearing defect.

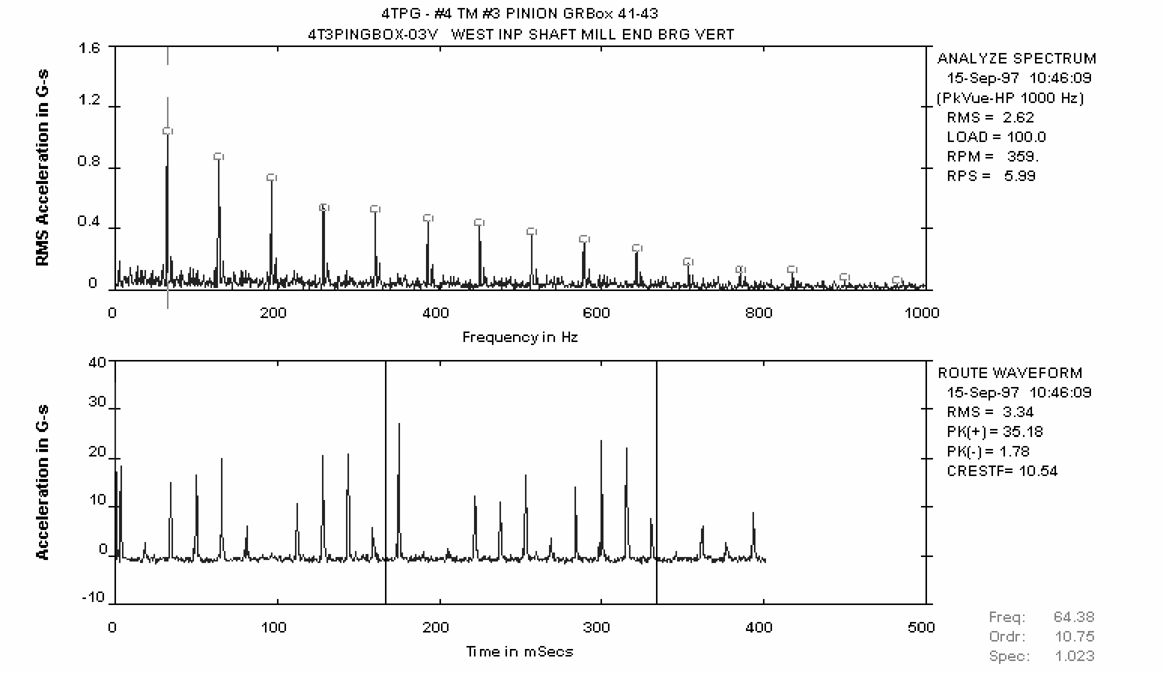


Figure 14-43 Sample PeakVue data from a gearbox

Beware of other fault conditions

The techniques discussed thus far are able to detect bearing faults because of the high frequencies generated. It should be noted, however, that there are other fault conditions that generate high frequencies.

For example, if there is rotating looseness, and thus impacting is occurring at a rate of once per revolution, the impacts will generate high frequency vibration which can be detected with all of the techniques above.

Certain types of defects on gear teeth will also generate high frequency vibration, and can thus be detected using the high frequency techniques just described.

It is important to understand, however, how looseness and gear defects will affect the shock pulse readings dBm and dBc. Cavitation, for example, will generate high frequency "noise" which will cause the dBm and dBc values to increase.

Stage Three Bearing Faults

The following points summarize the characteristics observed when a bearing is in Stage Two:

- More significant damage:
    - Minor damage through to more significant damage

- Bearing can fail in many ways for many reasons

- Very high frequency vibration continues to increase in amplitude.
- Envelope (demodulation) spectrum will be effective
    - Filters must be set up correctly
- Classic patterns appear in spectrum:
    - Harmonics due to impacts
    - Modulation (sidebands) due to cyclical change in load

The spectrum is now very important. The acceleration spectrum and log velocity spectrum may have exhibited peaks at the bearing defect frequencies, but there is no doubt that the velocity spectrum will have vibration patterns that indicate the fault.

Figure 14-44

Outer race fault (inner race rotating)

If there is a spall on the outer race (probably right in the load zone), there will harmonics of BPFO present in the data. To begin with it is possible that the harmonics are weak, but as the fault develops the harmonics will get stronger, and will likely increase in amplitude as the frequency gets higher.

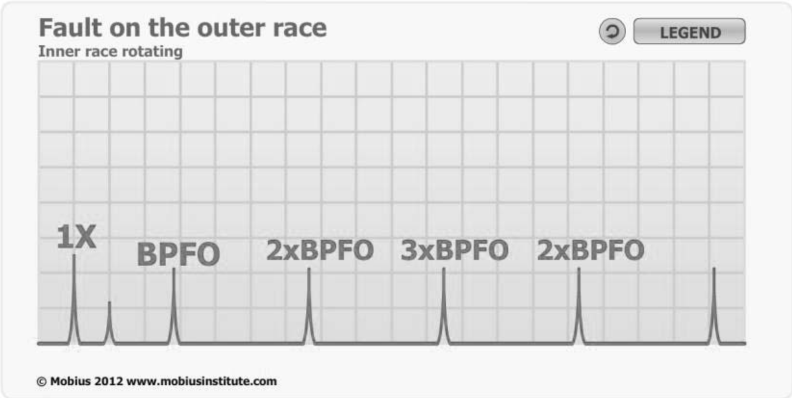


Figure 14-45

Outer race fault (outer race rotating)

If the outer race is rotating, then the pattern will change. There will still be harmonics; there always are harmonics when impacts occur. The forces in the impact will no longer be consistent. When the damaged area travels through the load zone, the impacts will be stronger compared with the impacts (and vibration level) when the damaged area is at the top of the bearing where it may be lightly loaded.

This rise and fall in the vibration amplitude generates sidebands in the spectrum. Because the outer race would be rotating at the turning speed, the sidebands would be spaced at 1X.

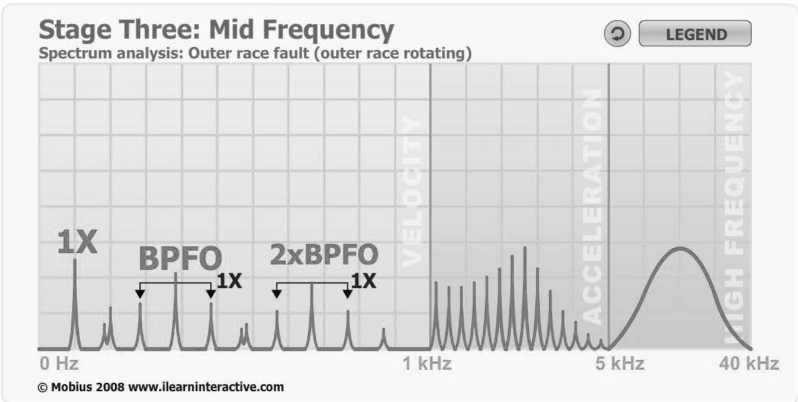


Figure 14-46

Inner race fault

If there is damage on the inner race on the bearing, we will observe three important characteristics:

1. Because there will be impacts, we will see harmonics of the BPFI frequency.

2. Because the damage is on the inner race and the vibration must travel through the balls to the outer race and on to the measurement location. Therefore the amplitude may be lower.

3. Because a spall travels around the bearing once per revolution, the impacts will not be equal in amplitude. So we will witness 1X sidebands. If the outer race is rotating then there will be no sidebands because the force behind each impact will be equal.

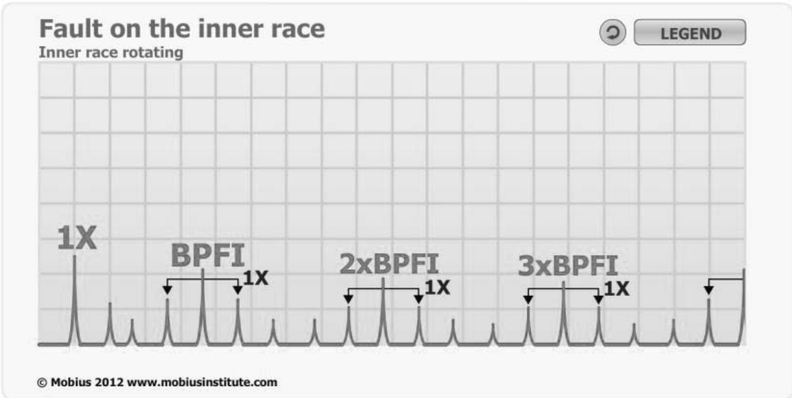


Figure 14-47

Ball or roller damage

If there is ball or roller damage, then we expect to see a peak at BSF with harmonics. It is likely that 2xBSF and harmonics will be stronger as the damaged area on the rolling element may make contact with the inner and outer race each rotation.

There will also be FT (cage rate) sidebands. As the balls move around the bearing they will go into the load zone of the bearing at the rate defined by the fundamental cage frequency, which will be less than half turning speed if the inner race is rotating, a little over 0.5X if the outer race is rotating.

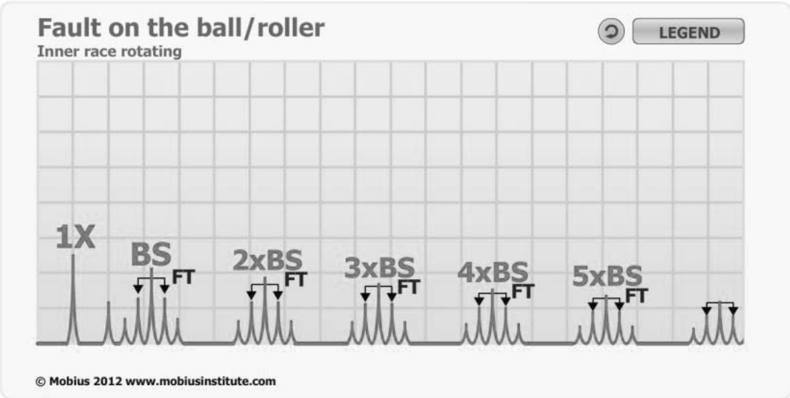


Figure 14-48

Knowing the forcing frequencies can be very helpful; especially if there is more than one bearing close to your measurement location. However knowing that the defect frequencies are non-synchronous, and that they generate harmonics, and that 1X sidebands indicate an inner race fault, and FT sidebands indicates a ball or roller fault, then we can determine with a lot of confidence that the bearing fault exists.

Overview of techniques

The effectiveness of each bearing detection technique is summarized below:

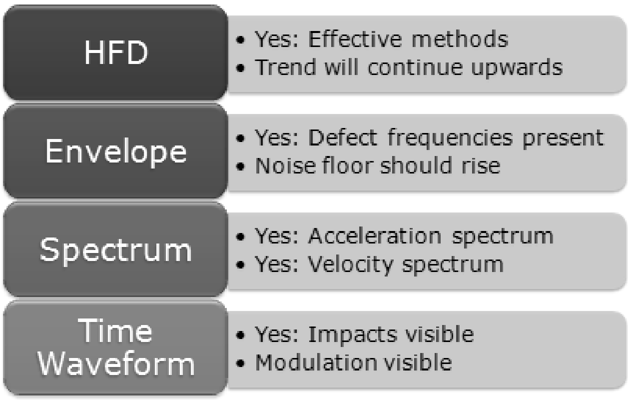


Figure 14-49

The following table summarizes the possible actions you could take at this stage:

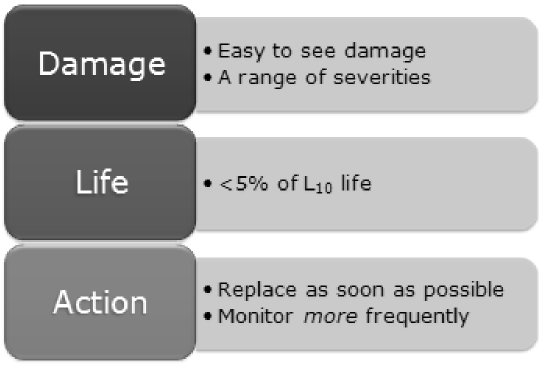


Figure 14-50

Data preparation

Remember that bearing defect frequencies are non-synchronous. In order to quickly identify which peaks are synchronous and non-synchronous it is helpful to order normalize the data. It is also important to have enough resolution in the spectrum to separate bearing defect frequencies from shaft rate harmonics or other forcing frequencies, especially in multi shaft machines. If you routinely take one spectrum with a high Fmax (say 60x) then it is a good idea to zoom in and view the data from 0-10x or 0-20x so that you can clearly see the bearing tones.

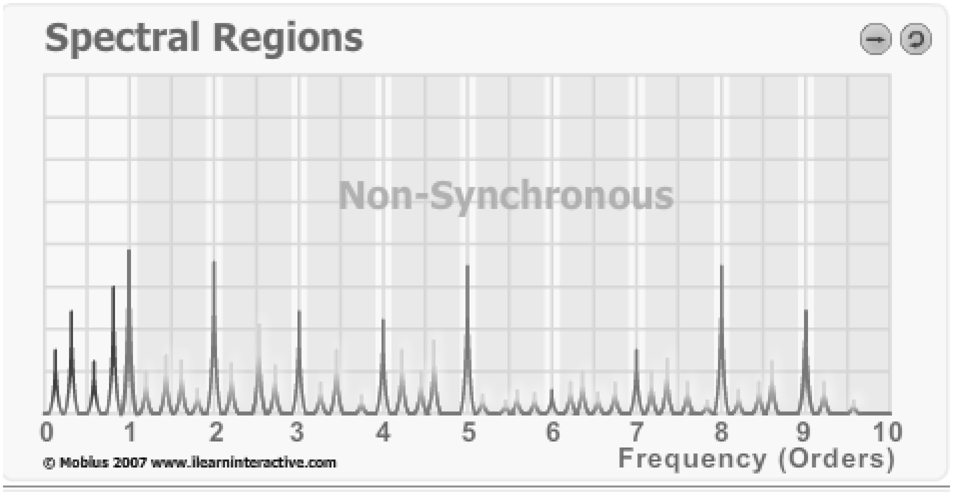


Figure 14-51

Logarithmic graph scales

One of the tricks for quickly identifying bearing problems is to format the data in a way that accentuates them. Log and dB graph scales on the "Y" or vertical axis serve exactly this purpose. In brief, when a graph is presented in a Log or dB format, one is able to see very small peaks in the presence of very large peaks. In a machine, bearing tones may be much smaller in amplitude than 1X or a vane pass frequency. When looking at the graph in a linear scale, one might only see these dominant peaks and the bearing tones might be down in the noise.

With a Log or dB amplitude scale one can see two peaks in the same graph even if one peak is 1,000x higher in amplitude than the other! If you want to try to visualize this, consider trying to take a photograph of a person standing next to a very high tree. The problem is, you always have to have the top of the tree in the photo. If you stand far enough back to capture the top of the tree, the person will look like a little speck of dust at its base. This is your bearing tone compared to 1x. As mentioned, if you simply change the scale to Log or dB you will be able to clearly see the person and the entire tree in the image even if the tree is 1000x taller that the person.

Even if a bearing tone has a low absolute amplitude, if its amplitude doubles or triples then it indicates there is a serious problem – so it is very important to be able to see these tones even if they are quite small.

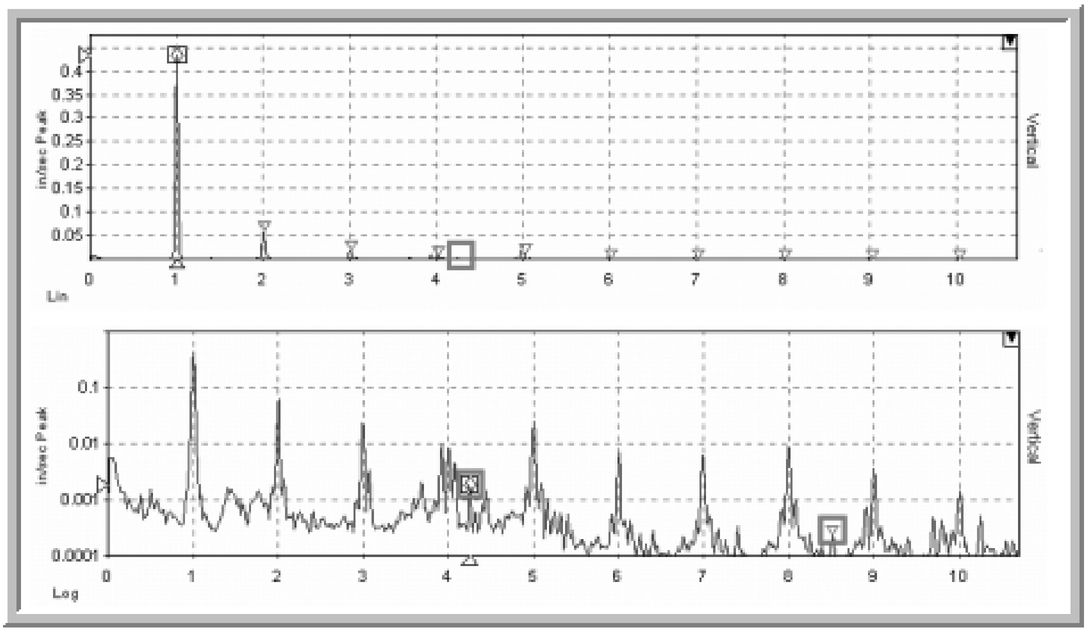


Figure 14-52

The two plots above are the same data, the bottom plot is in a Log format; the top one is linear. A bearing tone and harmonic are indicated by the red boxes – note that the tone is not visible at all in the top graph. The bottom graph also shows signs of looseness (1x harmonics) and a raised noise floor which is not visible in the top graph. Finally, also note that the data in both graphs are order normalized and there is plenty of resolution (and space) between the shaft rate harmonics so it is easy to find non synchronous tones.

The same can be said about the plots below, which are also identical. The red trace represents a baseline taken from the same machine; the blue trace is the latest data. Without knowing anything at all about vibration, would you say that top set of graphs indicates a problem? The answer would be “yes” and simply by presenting the data this way, our analysis can actually be as easy as saying “The data has changed – there is a problem.”

Look more closely and try to see specifically what is different between the red and blue traces in the upper set of graphs. You will notice that 1x harmonics out to 8x are all higher (looseness) you will notice the noise floor is higher which is related to impacting and in this case also a sign of rotating looseness. Finally, can you see the bearing tones? They are at approximately 3.8x, 4.8x and 5.8x. Which fault is this? Inner race fault – these are 1x sidebands. Did we need to know the bearing make and manufacturer? No. Is any of this visible in the linear graphs of the same data at the bottom? No!

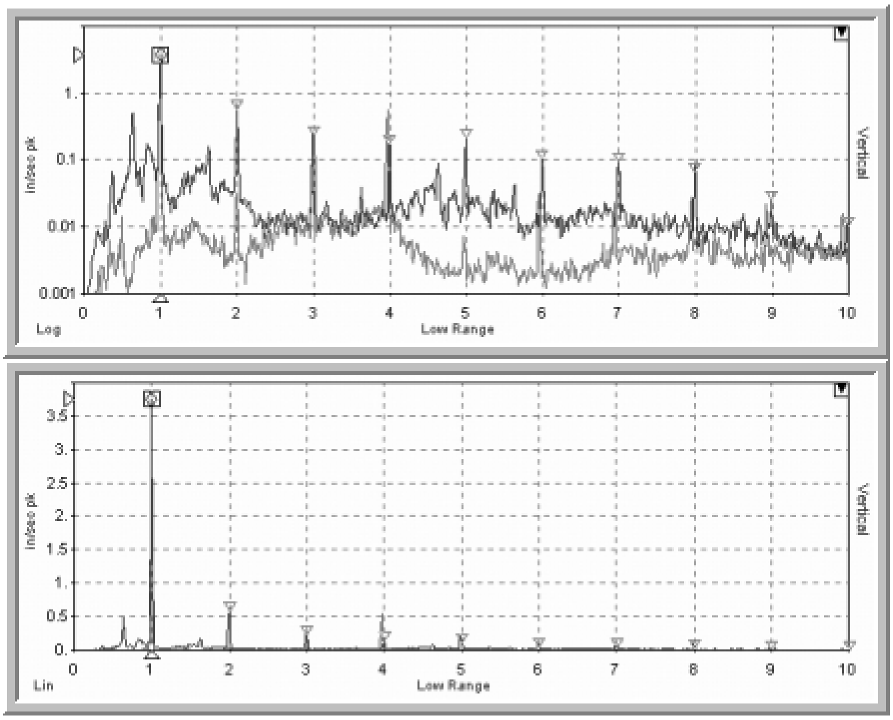


Figure 14-53

Practice examples

Try to identify the bearing tones in the following order normalized plots.

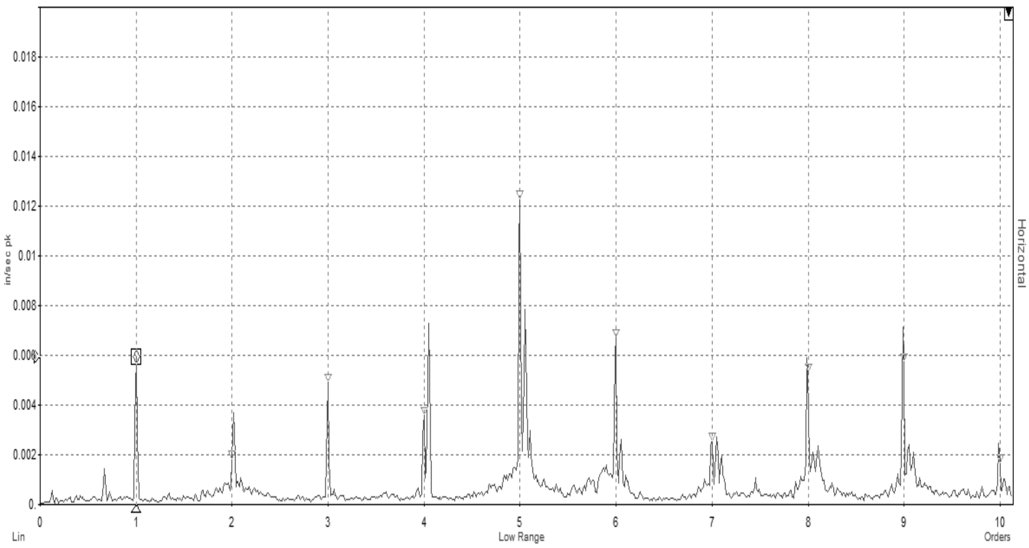


Figure 14-54

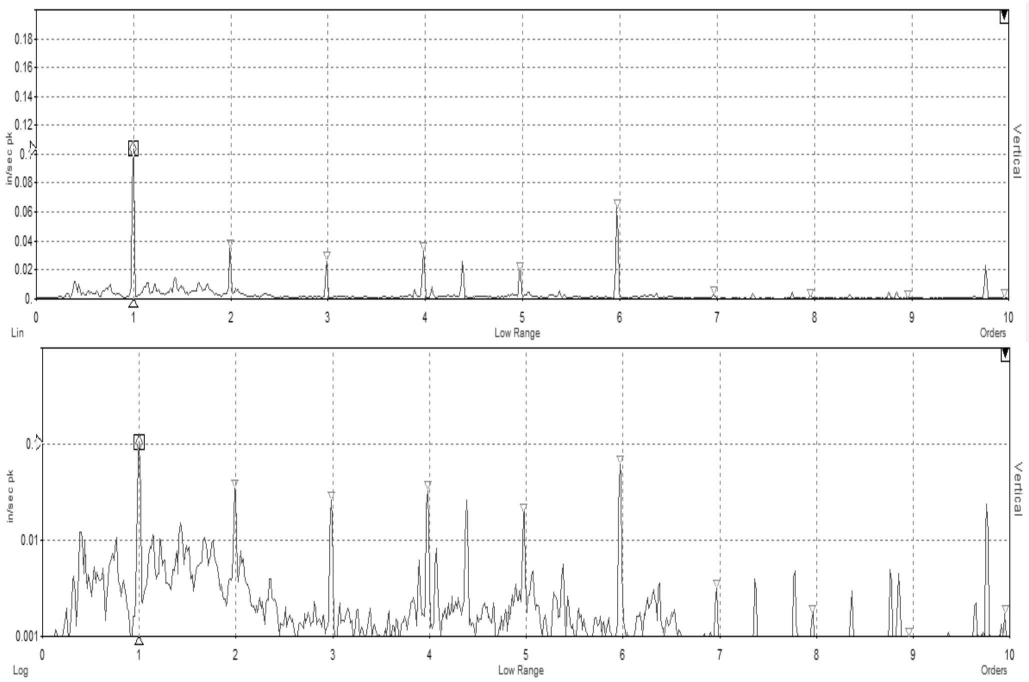


Figure 14-55

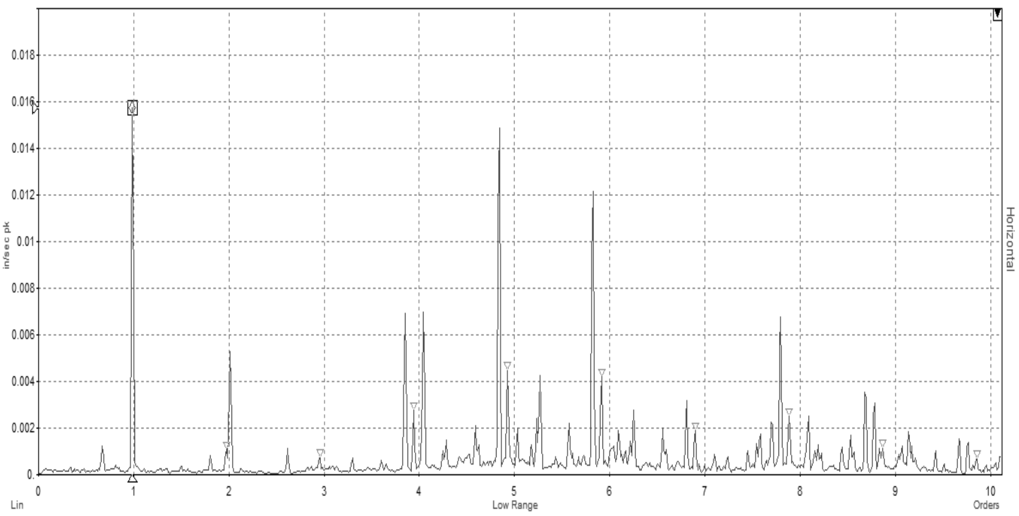


Figure 14-56

Time waveform analysis

Time waveform analysis is also useful for detecting bearing defects, but we will not go into it in great detail in this course.

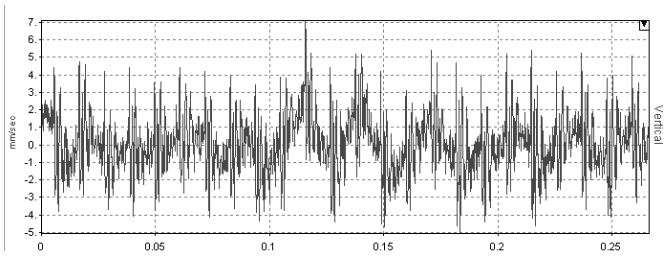


Figure 14-57

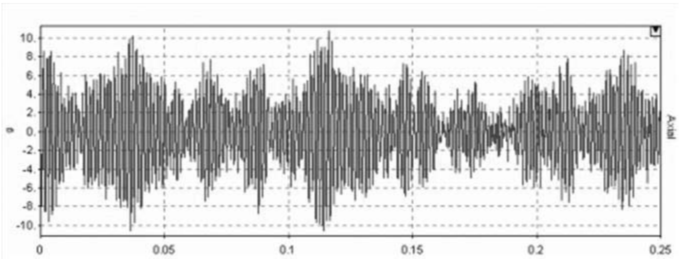


Figure 14-58

Time waveform analysis is particularly effective with low speed machines.

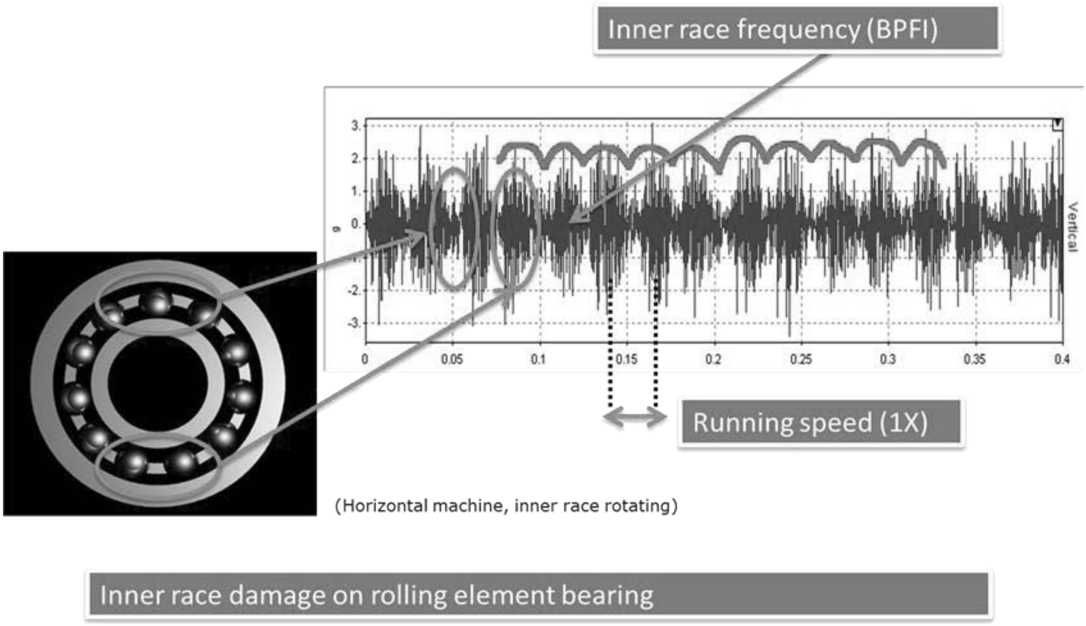


Figure 14-59

The time waveform makes it easy to detect amplitude modulation as you can see in the figure above. This is a time waveform from a bearing with an inner race defect. The amplitude modulation is happening at the shaft rate.

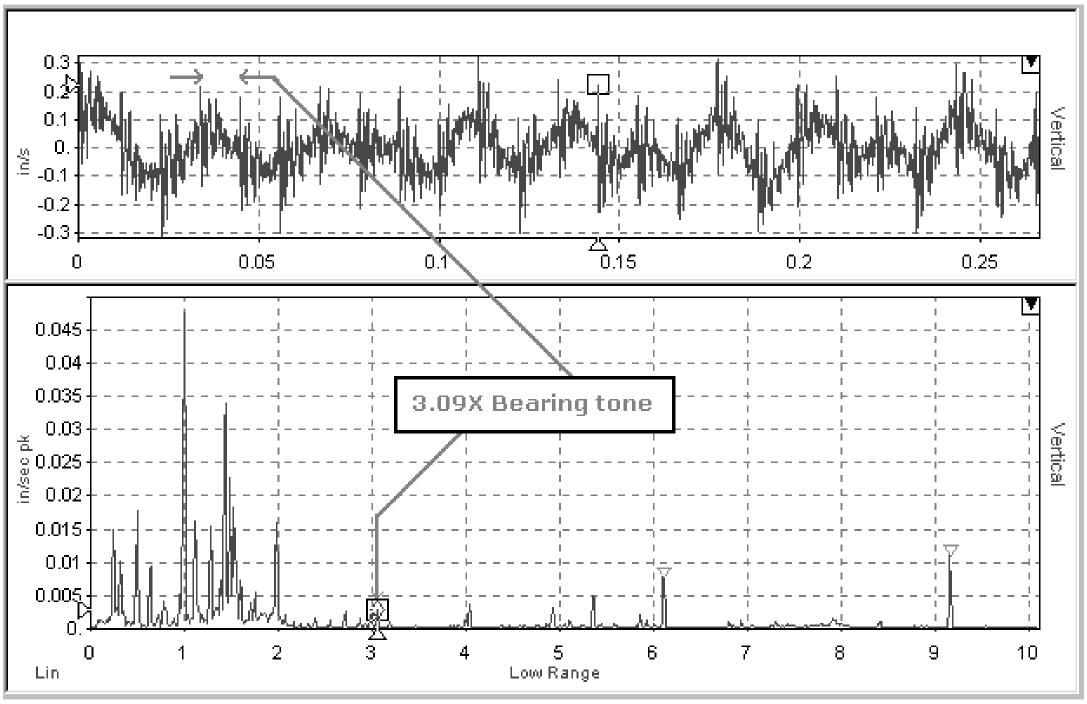


Figure 14-60

The figure above shows repetitive impacting in the time waveform. This corresponds to the defect in the bearing being struck by a ball or roller. By measuring the time (period = T) between two impacts and taking its reciprocal (1/T), we can calculate the frequency (Hz) of the impacting. We can then see the corresponding peak in the spectrum.

Stage Four Bearing Faults

The following points summarize the characteristics observed when a bearing is in Stage Two:

- Significant damage
    - Damage far more extensive
    - Damage in one component causes damage in other components
    - *Failure is imminent*
- Very high frequency vibration may trend *downwards*.
    - Smoothing of metal reduces sharp impacts
- Spectrum, time waveform and envelope spectrum analysis still effective – at first…

Figure 14-61

As we move into the stage four faults the vibration patterns will start to change dramatically; due in most part to the excessive wear. The vibration becomes less and less periodic – there are too many areas of damage. The vibration starts to become noisy which lifts up the noise floor – we call them "haystacks", but they are really just resonances.

As the fault develops further the defect frequency peaks will disappear altogether, and the spectrum will change to become more like the pattern expected with rotating looseness. This is because there has been so much metal loss that the clearances begin to grow.

Once you observe these changes you must take action; the bearing has less than 1% of its life remaining.

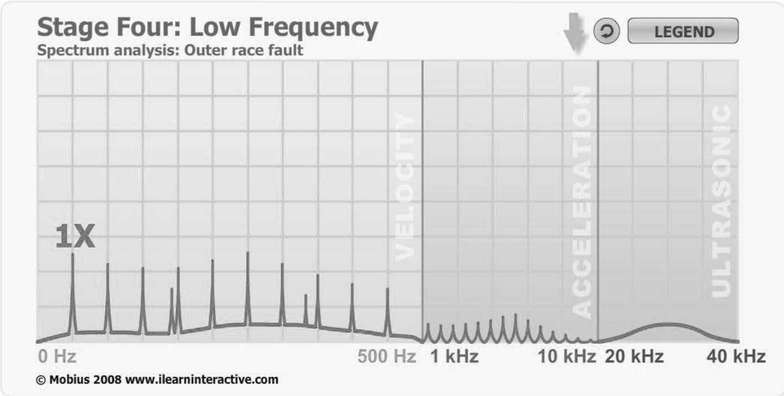


Figure 14-62

At this stage the high frequency techniques become far less effective. The shock pulse and gSE trends may dip. The spectrum will provide enough evidence of the fault condition. A person standing next to the machine could tell you that the bearing was damaged.

The effectiveness of each bearing detection technique is summarized below:

HFD	• No: Less high freq. vibration • Trend will drop
Envelope	• Yes: Defect frequencies present • Noise floor should rise
Spectrum	• Yes: Acceleration spectrum • Yes: Velocity spectrum
Time Waveform	• Yes: Impacts visible • Modulation visible

Figure 14-63

The following table summarizes the possible actions you could take at this stage:

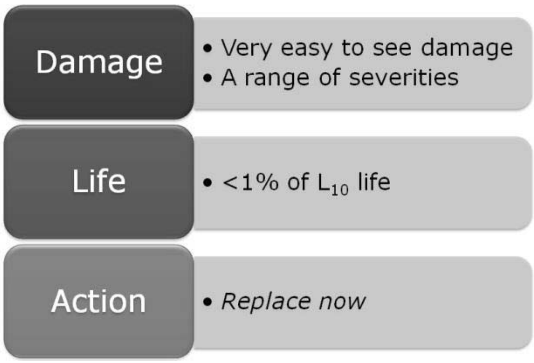


Figure 14-64

Optimizing your results

Now that we have nearly completed this section on detecting bearing defects, please remember that detecting defects is only part of the story. It is just as important to try to remove the root cause of bearing failure and extend the life of the bearings. This begins in the design phase, making sure one is using the correct machine and bearings for the job, and it continues into purchasing, shipping, handling, installation, acceptance testing, alignment, balancing, lubricating etc. In short, there is bearing wear and there is bearing care and we should be interested in both of them.

Bearing wear

Use all of the techniques at your disposal in order to detect wear at the earliest stages and track it up towards failure, providing information to planners along the way so that they can decide when it is the optimum time to change the bearing. Your tools include high frequency tests, beginning with ultrasound for monitoring lubrication and the earliest stages of wear. Most vibration data collectors will also offer some high frequency test for bearing wear. Vibration spectra, time waveforms and RMS overalls may all be monitored and trended.

Data presentation is key! This point cannot be over stated. If you present your data well in terms of proper resolution, good amplitude scaling, order normalization, a log or dB scale and some sort of baseline or graphical alarm then it will be very easy to detect bearing defects at the earliest stages and monitor them as they deteriorate.

Case Study: Air Washer #1

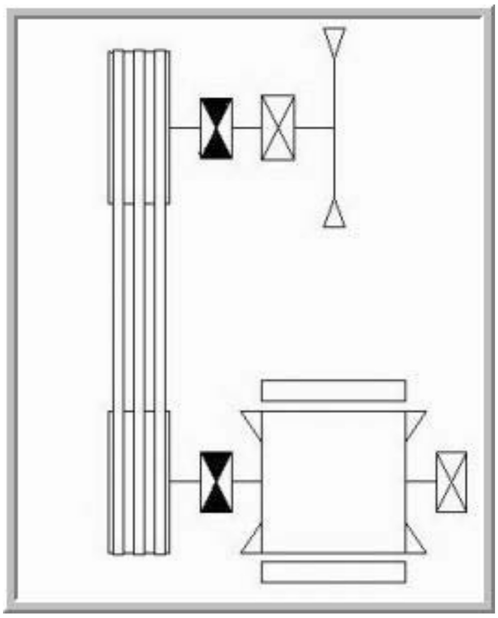


Figure 14-65 -Motor driving fan; 2:1 belt ratio

- Speed = 1770 CPM; 1XM
- 40 HP motor
- Overhung fan
- 16 blades on the fan; 16XF or 8XM
- 11 cooling fan blades on the motor
- Motor sheave 6.5" (165 mm)
- Fan shaft = 0.5XM; 885 CPM; 1XF
- Fan sheave diameter 13" (330 mm)
- Distance between centers 47" (1194 mm)
- Belt rate 0.1636XM

When monitoring commenced on 4/5/99, the vertical and horizontal data did not show very much at all. The amplitude levels are quite low. The time waveform does show some high frequency content, suggesting the earliest signs of bearing wear. This is a belt driven machine, so there are peaks associated with the belt rate and the fan shaft. The data is from the motor.

The axial direction does show a small non-synchronous peak at 3.06 orders. The high 1x indicates the sheaves may be misaligned.

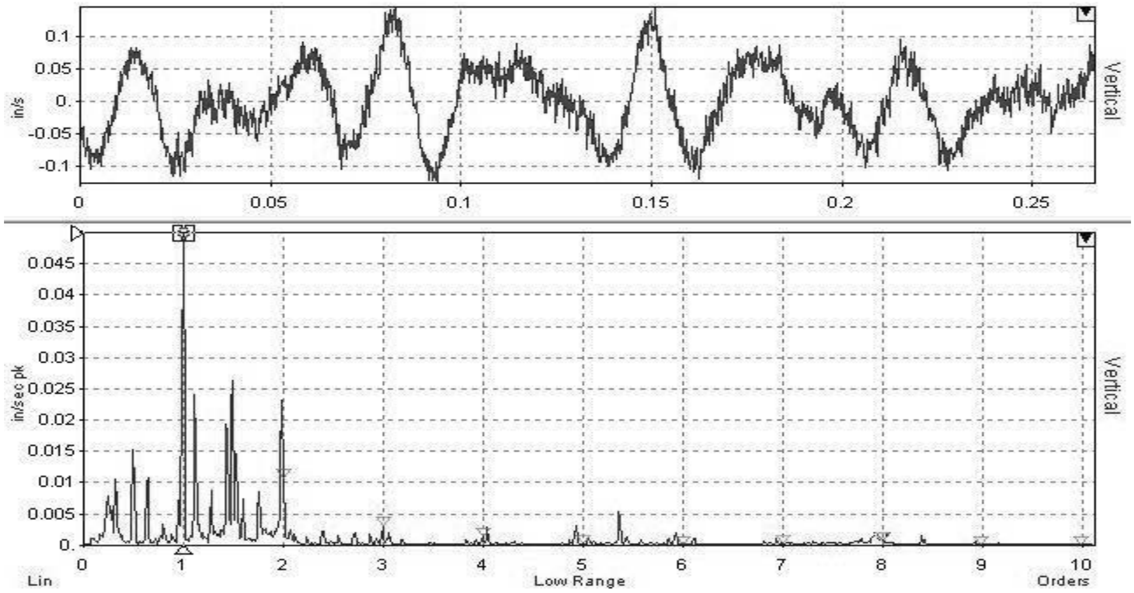


Figure 14-66 - Vertical and Horizontal show very little signs of bearing frequencies.

May 3 shows early signs of bearing wear, the levels were increasing, and the time waveform showed even more signs of bearing wear. It is difficult to see in a linear plot; however, in log view a peak at 3.1X with harmonics is visible. The bearing is at Stage Four now.

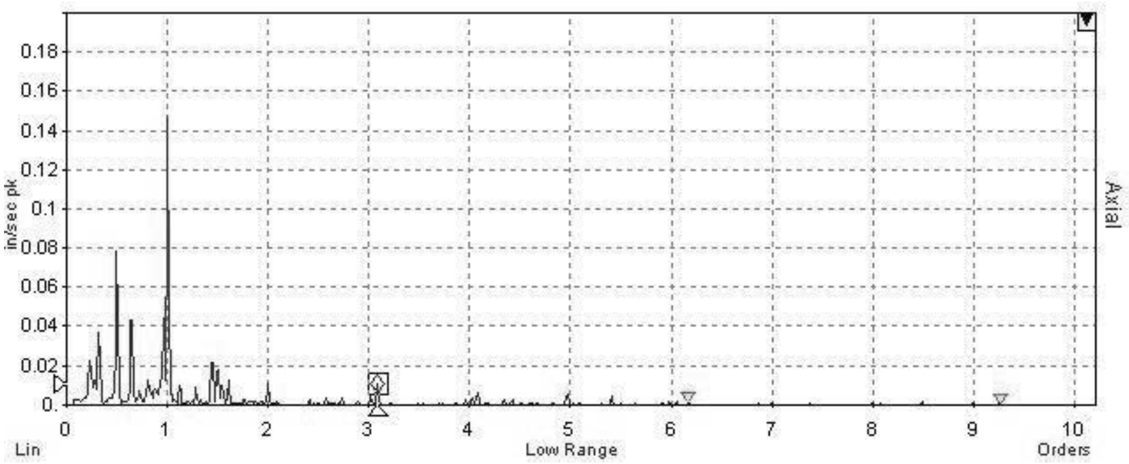


Figure 14-67 - Axial shows peak at 3.06 orders.

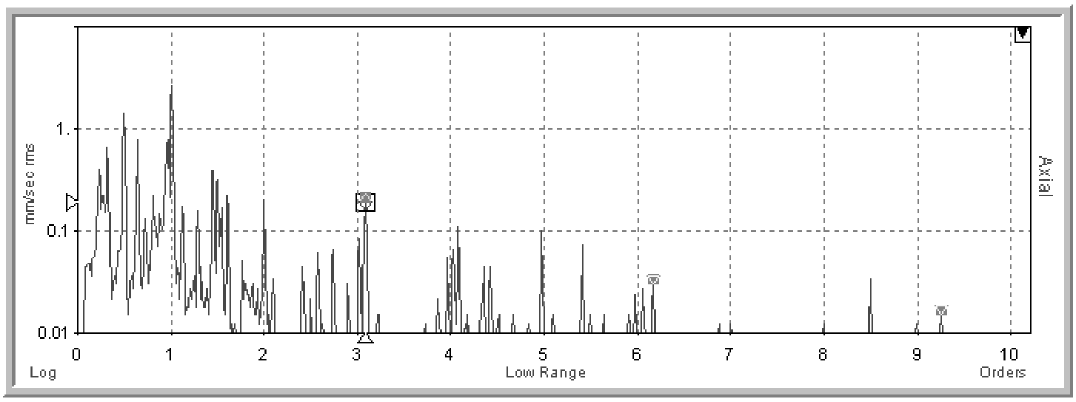


Figure 14-68 - The peaks at 3.06 and its harmonics are more visible in Log scale.

On 6/1/99, the signs of bearing wear are becoming quite strong. The time waveform is a classic. The impacts are occurring every 3.1X of running speed.

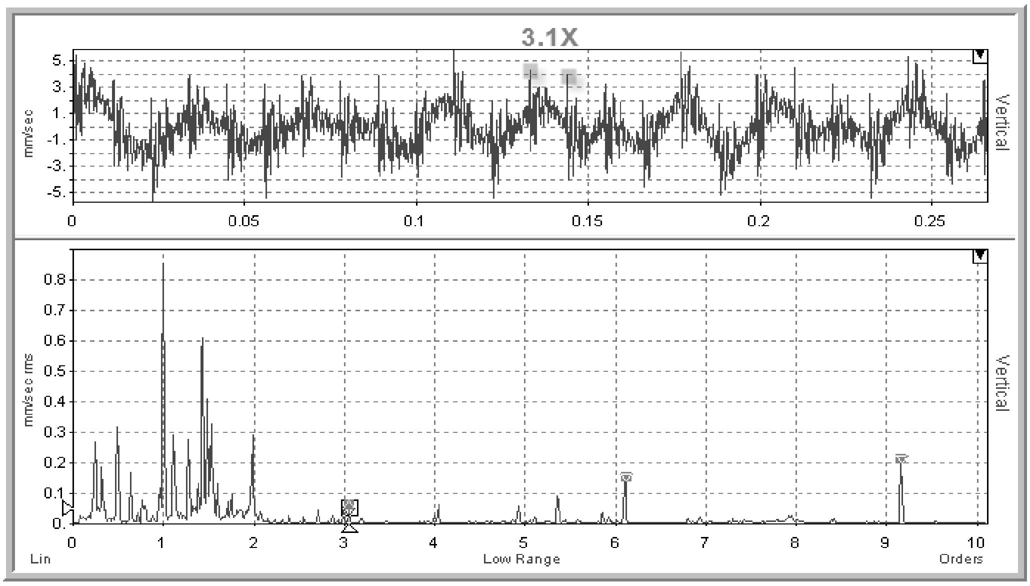


Figure 14-69 - The classic waveform shows the impacting at every 3.1 orders.

The high range of the spectrum shows there are harmonics of the bearing frequency to greater than 50 orders.

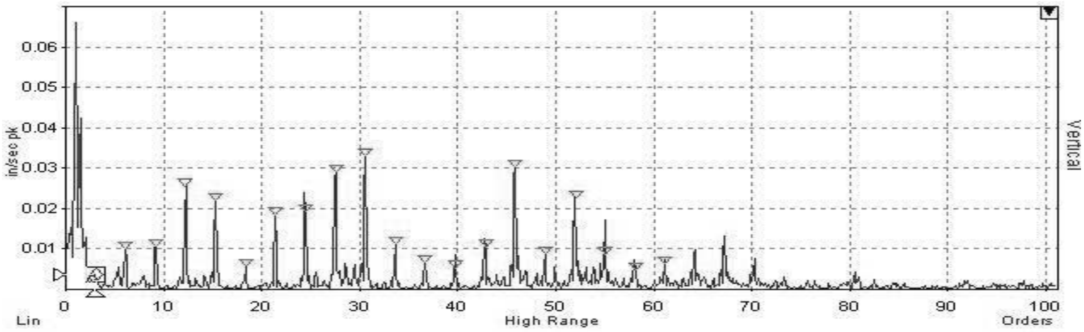


Figure 14-70 - High range shows bearing harmonics to greater than 50x.

Where are the sidebands?

Looking closely at the data, even in log, there are no strong sidebands. How can this be? Recall that sidebands occur if there is a ball fault (sidebands of FT) or an inner race fault (sidebands of 1X). Given that the bearing tone is low in frequency (BPI = 0.6 x # balls, BPO = 0.4 x # balls) it would be suggested that the fault is on the outer race. So there are harmonics, just no sidebands.

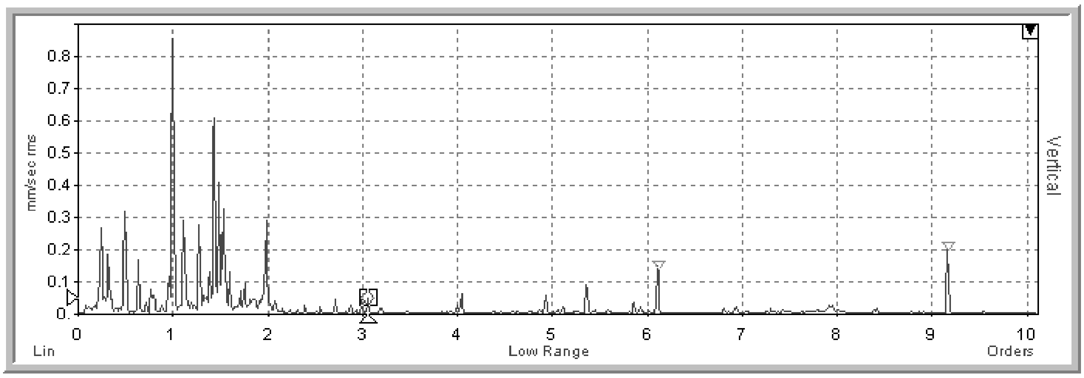


Figure 14-71 - There are no sidebands. This is an outer race fault which typically does not vary in amplitude like inner race frequencies.

Notice that we have been able to determine, with some confidence, the nature of the fault without even knowing what bearing is installed in the machine.

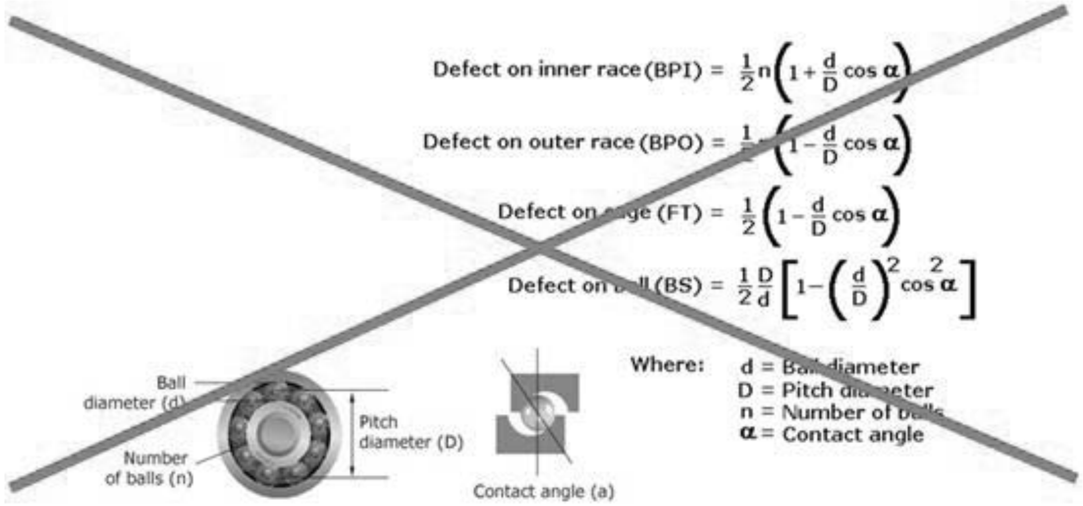


Figure 14-72 - Even though the bearing was not known, we were still able to determine the nature of the fault

On 5th October, 1999, the harmonic levels increase, haystacks appear, and the noise floor rises. This is the data from the high range, vertical direction. Fig. 15-32 shows the high range spectrum. Notice the bearing harmonics to more than 60 times turning speed.

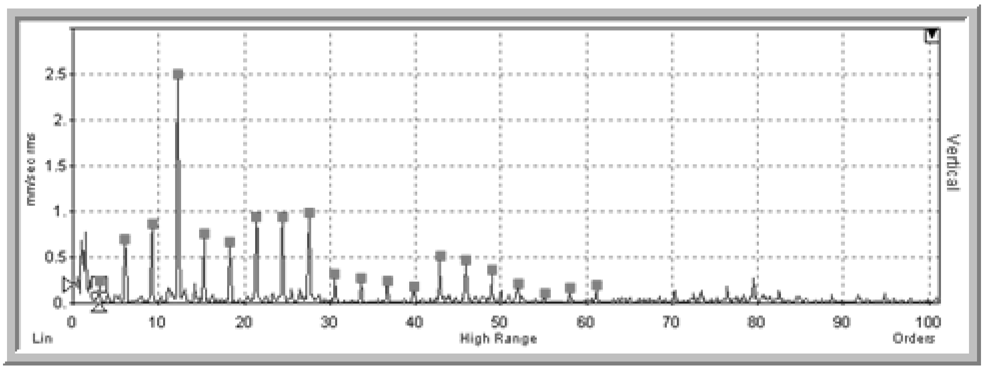


Figure 14-73 - High range showing bearing frequency harmonics to greater than 60x.

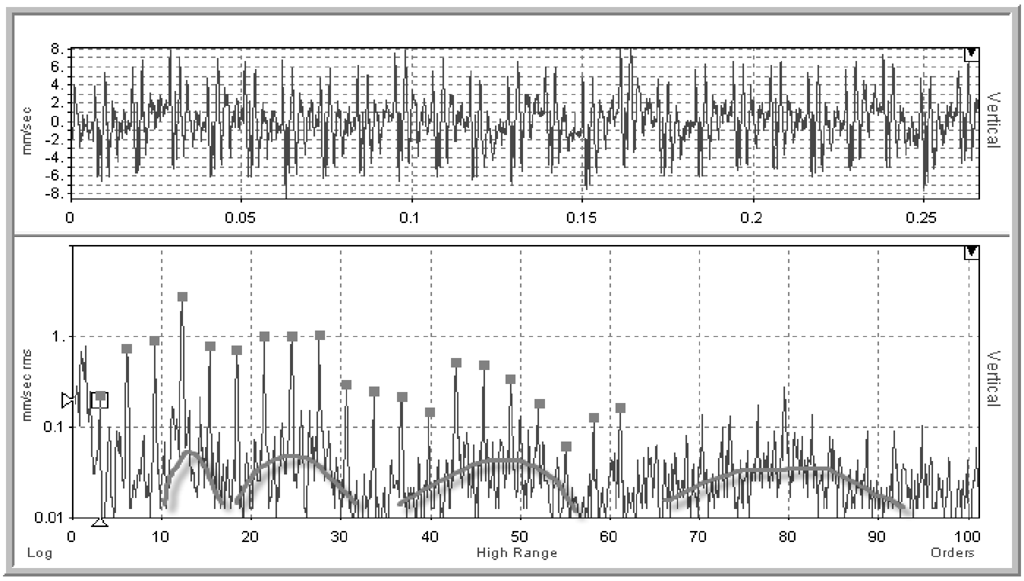


Figure 14-74 - by October, haystacks are evident

The 1/12/00 data looks very different! The time waveform does not have the same periodic spikes, the spectrum does not have the harmonics, but the noise floor is considerably higher.

The bearing is at Stage 8. The bearing is almost dead - in fact, the machine failed catastrophically just two days later!

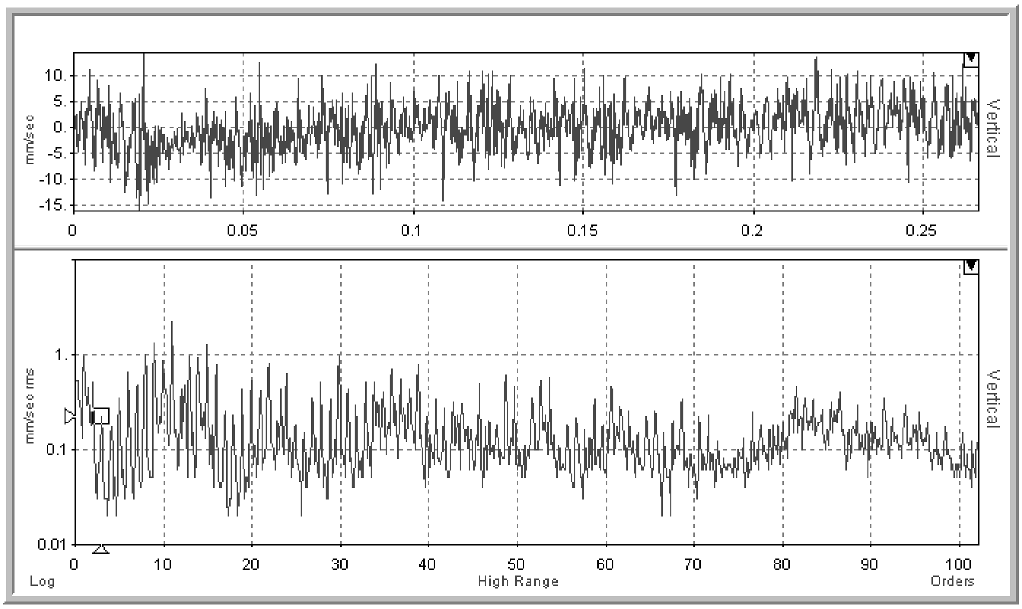
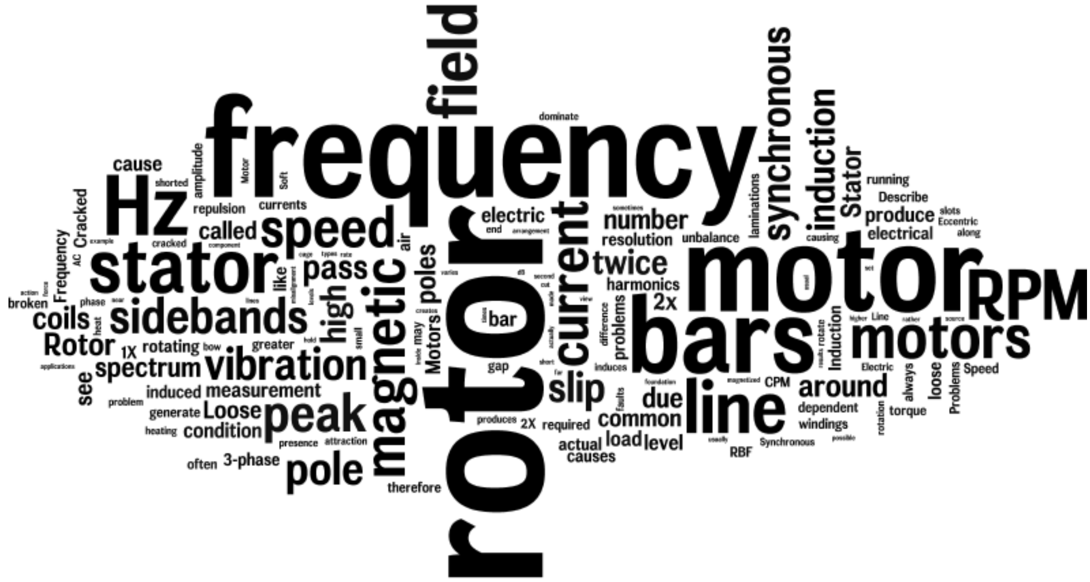


Figure 14-75 - Stage 8 - the waveform is different, the spectrum floor is raised. The machine failed 2 days later.



Chapter 15

Electric Motor Analysis

Objectives:

- Describe the spectral characteristics of electric motor problems
- Describe how an electrical problem can look like a 2x or 4x peak in spectra
- Describe how misalignment can invoke an electrical problem in a motor
- Describe the causes of 3 types of sidebands from electric motors

Electric Motors

Electric motors are the workhorse of industry and have unique faults along with the standard mechanical faults. Faults can develop due to misalignment, unbalance, resonance, and foundation problems. Motors with rolling element bearings therefore suffer from bearing failures.

Figure 15-1

This section examines their electrical properties, and the faults related to those properties.

There are two types of AC electric motors; the synchronous motor and the induction motor, and each may be powered by single phase or 3-phase current. In industrial applications, 3-phase motors are by far the most common, owing to their higher efficiency. In addition, induction motors are more common than synchronous motors.

Induction Motors

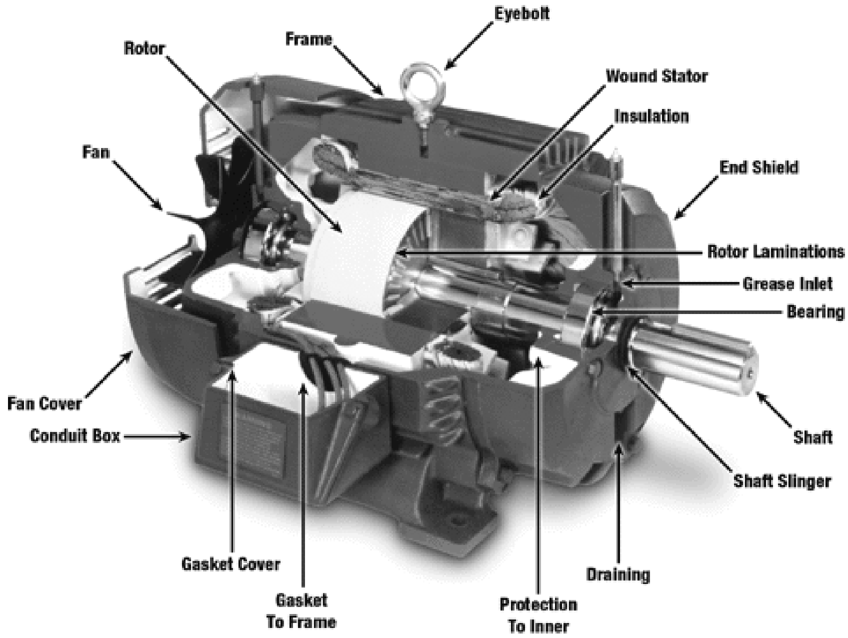


Figure 15-2 - Cutaway view of an Induction motor

The cutaway view shows the components of a standard induction motor. The fan on the non-drive end cools the motor. Bearings and end shields hold the rotor centered in the motor. Stator laminations are designed to expel heat from the motor and reduce losses

Between the stator and rotor windings is an air gap that must be maintained evenly about the stator. Uneven air gaps cause low level vibration and cause bearings to fail early.

Induction Motors typically have no electrical connection between the rotor and the outside world. They operate because a moving magnetic field induces a current to flow in the rotor. This current in the rotor creates the second magnetic field required (along with the field from the stator windings) to produce a torque. Induction motors are simple and therefore relatively cheap to construct. They do not rely on brushes like the DC motor, and usually have a longer life. They are by far the most common type of motor for applications above 1kW.

The stator of the AC motor contains a number of coils of wire wound around and through the stator "slots". There are always many more slots than there are coils, so the coils are intertwined in a fairly complex manner.

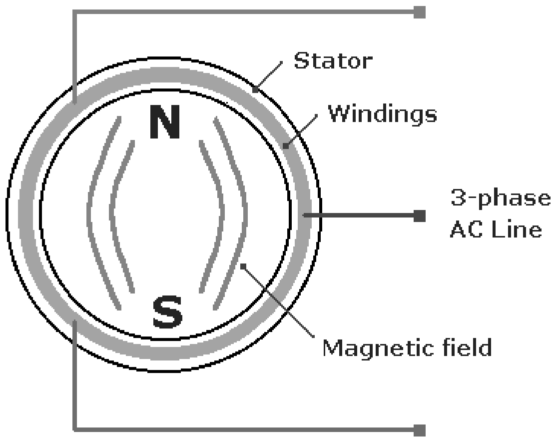


Figure 15-3 - Induction Motor

When the coils are energized, a rotating magnetic field is set up inside the stator. The speed of the rotation is dependent on the number of coils, or the number of "poles".

3-phase, 3 coils = 2 poles

- Speed = 60 Hz (50 Hz)
- Speed = 3600 (3000) RPM

3-phase, 6 coils = 4 poles

- Speed = 30 Hz (25 Hz)
- Speed = 1800 (1500) RPM

In a 3-phase motor, three coils will form 2 magnetic poles due to the action of the currents, which have a 120-degree phase difference between them. With a 60 Hz line frequency, and two poles in the stator, the field rotation rate will be 60 per second, or 3600 RPM. If there are 4 poles (6 coils), the field will rotate at 1800 RPM, and so forth.

In countries with 50 Hz line frequency, two-pole motors rotate at 3000 RPM, and four-pole motors rotate at 1500 RPM.

The induction motor differs from the synchronous motor in that the rotor is not a permanent magnet, but is rather an electromagnet. It has conducting bars running along its length imbedded in slots spaced uniformly around its periphery. The bars are all connected together by rings at each end of the rotor that are welded to the ends of the bars.

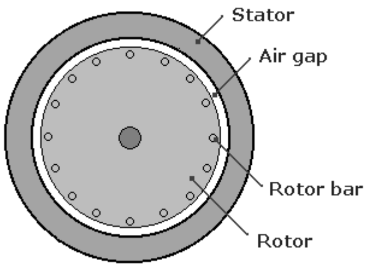


Figure 15-4 - Rotor bars in an induction motor

This arrangement resembles the design of small rotary cages used to exercise pet hamsters, and is therefore sometimes called a "squirrel cage", and induction motors are called squirrel cage motors.

Each pair of bars is actually a "shorted turn", magnetically speaking. The rotor becomes magnetized by induced currents in the rotor bars due to the action of the rotating magnetic field in the stator. As the stator field sweeps past the rotor bars, the changing magnetic field induces large currents in them, generating its own magnetic field. The polarity of the induced rotor magnetic field is such that it repels the stator field that creates it, and this repulsion results in a torque on the rotor, causing it to spin.

Because the induction motor works by magnetic repulsion rather than attraction like the synchronous motor, it has been called a "repulsion induction" motor.

If there were no friction in the system, the rotor would turn at synchronous speed, but the motor would produce no useful torque. Under this condition, there would be no relative motion between the rotor bars and the rotating stator field, and no current would be induced in them.

As soon as any load is applied to the motor the speed is reduced, causing the rotor bars to cut the magnetic lines of force of the stator field, creating the repulsion force in the rotor. The induced magnetic field in the rotor migrates around in the opposite direction of the rotation, and the speed of this migration is dependent on the applied load. This means the RPM will always be less than synchronous speed.

The difference between the actual speed and synchronous speed is called the "slip". The greater the slip, the greater the induced current in the rotor bars, and the greater the output torque. The current in the stator windings also increases in order to create the larger currents in the bars. For these reasons, the actual speed of an induction motor is always dependent on the load.

There are very strong forces between the rotor and stator. The distance between the rotor and stator must be equal and balanced and the magnetic field should be uniform. For a variety of reasons this may not be the case:

- Rotor off center
- Rotor movement: bent shaft

- Stator eccentric (not round)
- Stator or rotor lamination or other problems
- Broken rotor bars (inconsistent magnetic field)
- Broken/damaged stator slots (inconsistent magnetic field
- Loss of a phase, or unbalance phase voltage (3-phase)

Synchronous Motors

If a round rotor that is permanently magnetized is placed inside the stator, it will be "dragged" around by magnetic attraction at the speed of the rotating field. This is called the "synchronous" speed, and this arrangement is called a synchronous motor.

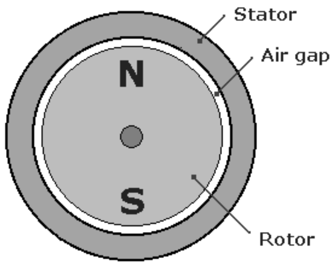


Figure 15-5 - Synchronous Motor

There is no slip so the motor operates at 3600 RPM (3000) or 1800 RPM (1500)

Sources of Vibration in Electric Motors

Twice the line frequency (100 Hz or 120 Hz) is always a measurable vibration component in an electric motor. The magnetic attraction between the stator and rotor varies at this rate, and the iron itself changes dimension a little in the presence of the varying magnetic field due to "magnetostriction".

Magnetostriction is the deformation of a magnetic material in the presence of a magnetic field, and it causes vibration at 100 or 120 Hz in all electric devices such as motors, generators, transformers, etc.

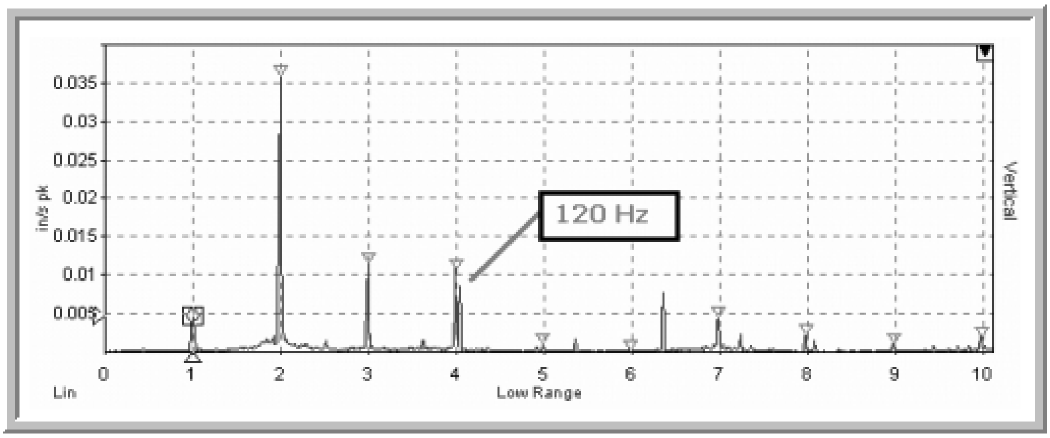


Figure 15-6 - A peak at 2x the Line Frequency is near the 4x peak. 2x line frequency is common in electric motors.

In two-pole motors (synchronous speed of 3000 CPM or 3600 CPM), it is sometimes difficult to distinguish the 100 or 120 Hz (twice line frequency) peak from the 2X peak in the vibration spectrum. This is another good reason for collecting spectra with high resolution.

One test to verify the presence of a 2X peak instead of 100 or 120 Hz, is to take a measurement while the motor is running, and then cut power to the motor. The 100 or 120 Hz peak will disappear, whereas 2X will remain (as the motor drops in speed).

The 2x line frequency peak can be very strong and dominate the spectrum.

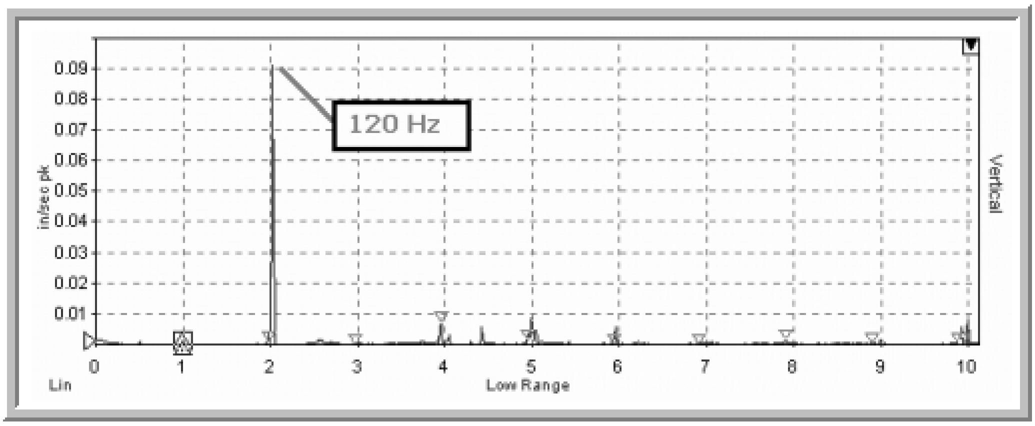


Figure 15-7 - The 2x line frequency can dominate the spectrum. It is near the 2x peak.

Variable frequency drives work in a similar way, however instead of 50 Hz or 60 Hz, the voltage is supplied at a different frequency, thus controlling the motor speed. The 2xLF peak will still be present however the motor speed and the frequency of 2xLF will be different.

Stator related fault conditions

Stator Problems

Stator problems will generate high vibration at twice the line frequency (100 or 120 Hz).

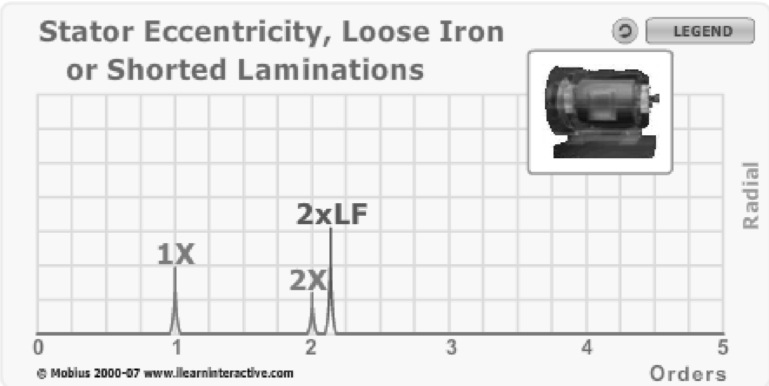


Figure 15-8 - Stator Problems

Stator eccentricity produces an uneven stationary air gap between the rotor and stator that produces a very directional source of vibration.

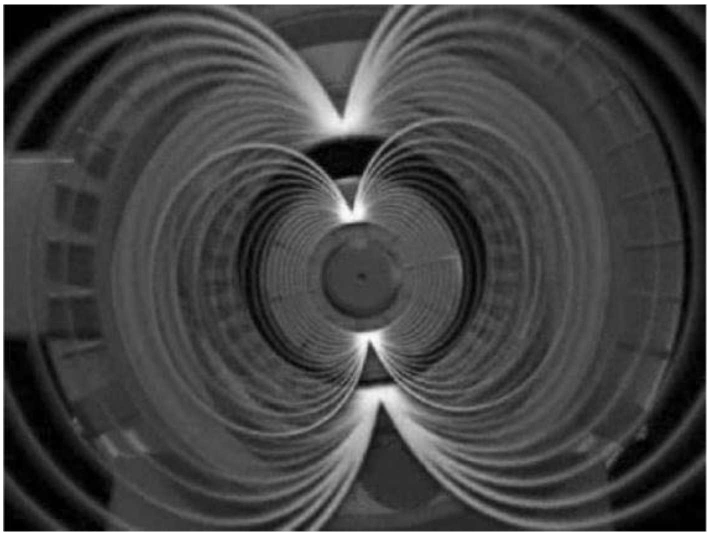


Figure 15-9 - Stationary air gap

Soft Foot

Soft foot and warped bases can produce an eccentric stator. Altering the tension on the hold down bolts, correctly shimming the feet, reducing stress from conduit or piping, or repairing the foundation can resolve this problem.

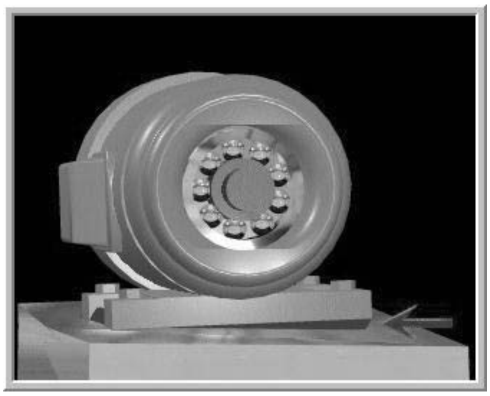


Figure 15-10 - Soft foot causes Stator Eccentricity

Rotor related fault conditions

Eccentric Rotors

Eccentric rotors produce a rotating variable air gap between the rotor and the stator, which induces a pulsating source of vibration.

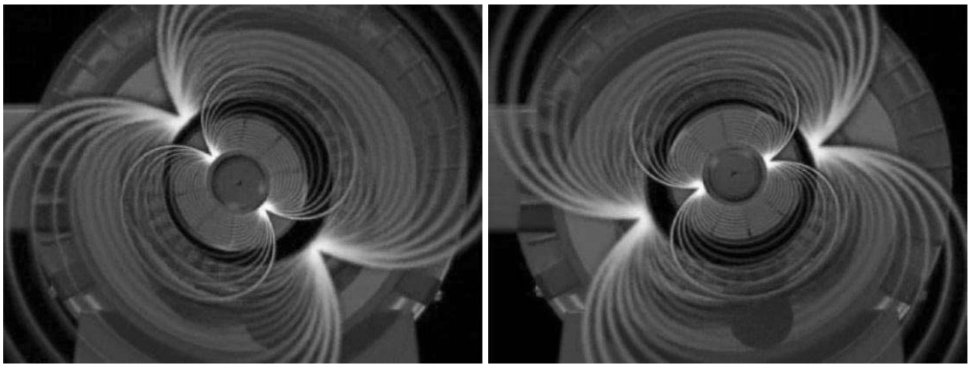


Figure 15-11 - Rotating air gap

Again you will see the twice line frequency component, however this time there will be pole pass sidebands around this frequency and the 1X peak.

The **pole pass frequency is the slip frequency times the number of poles**. The slip frequency is the difference between the actual RPM and the synchronous speed.

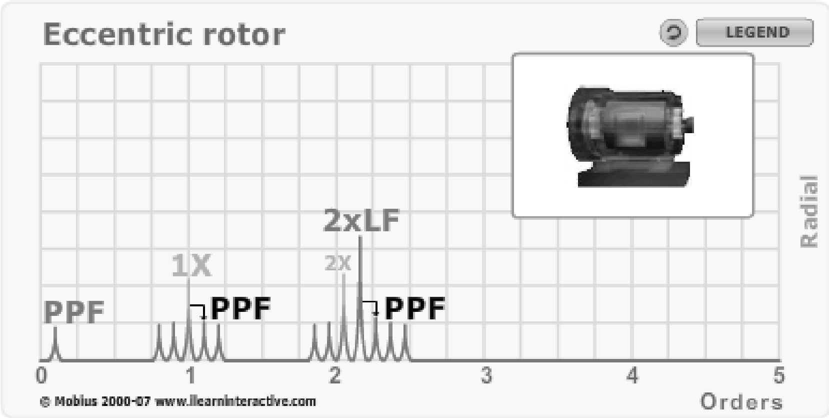


Figure 15-12 - Eccentric Rotors

Because the slip frequency is quite small, the pole pass frequency is therefore small, and thus a high resolution measurement may be needed to identify these sidebands.

Synchronous Speeds for 60 Hz & 50 Hz

- 2 pole: 3600 RPM or 3000 RPM
- 4 pole: 1800 RPM or 1500 RPM
- 6 pole: 1200 RPM or 1000 RPM

Thermal Rotor Bow

Uneven heating of the rotor due to unbalanced rotor bar current distribution causes the rotor to warp, or "bow", and rotor bow results in an unbalance condition with all its usual symptoms. It can be detected by the fact that it goes away when the motor is cold. Local heating can be so severe in motors that the offending bar can actually melt and lodge in the air gap.

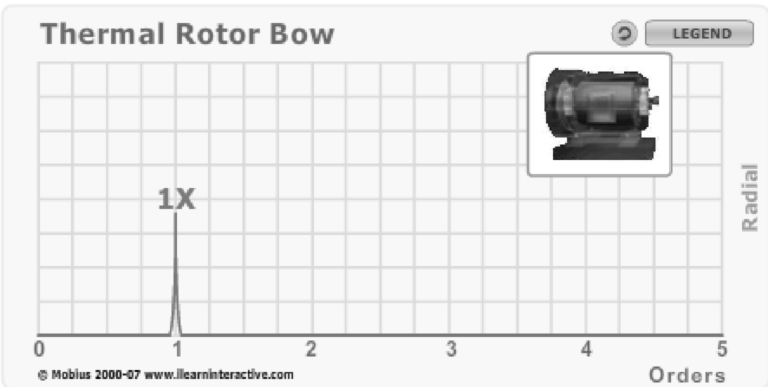


Figure 15-13 - Rotor bow looks like unbalance

Rotor Problems

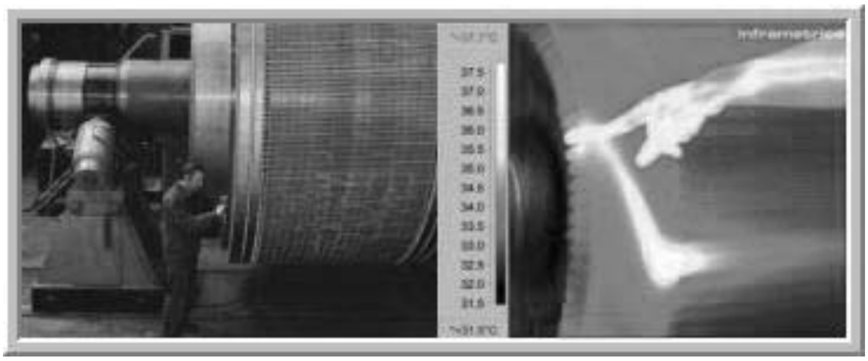


Figure 15-14 - Cracked or broken rotor bars short and produce heat

Broken, cracked, or corroded rotor bars are a fairly common failure mode of induction motors, especially in motors that are frequently started and stopped under load. The starting current is much greater than the running current, and puts a strain on the rotor bars, causing them to heat.

Cracked Rotor Bars

Cracked rotor bars will generate pole pass frequency sidebands around 1X and its harmonics (2X, 3X, and so on). You will often see a very busy spectrum with harmonics of 1X, each with "skirts" of pole pass sidebands.

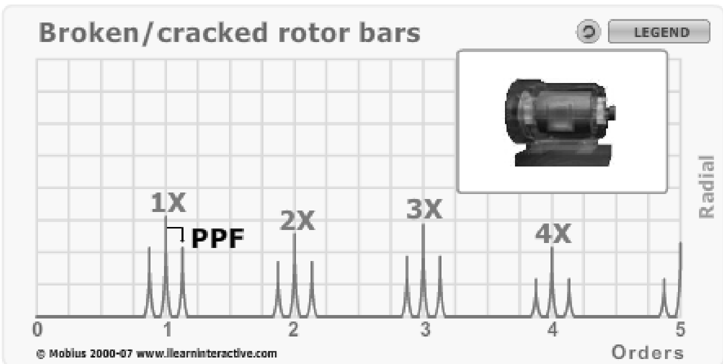


Figure 15-15 - Cracked rotor bars generate pole pass frequency sidebands around 1x and harmonics.

An induction motor with defective rotor bars will produce a vibration signature that slowly varies up and down in amplitude at twice the slip frequency of the motor. **This phenomenon is called beating,** and can often be heard as well as measured. The amplitude and frequency of the beats are dependent on the load on the motor (because that affects the slip frequency).

Rotor Bar Passing Frequency

It is common to see Rotor Bar Frequency (RBF - the number of rotor bars times the RPM) due to the fact that each rotor bar passes slight disruptions in the magnetic field due to the current's path through the rotor and stator. A high RBF indicates cracked or broken rotor bars or defective or loose rotor bar joints.

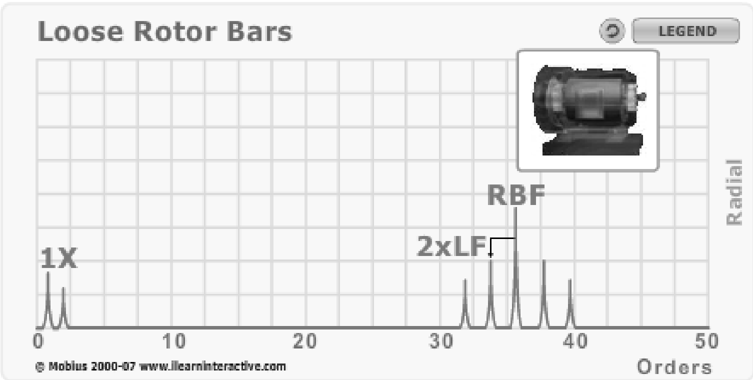


Figure 15-16

Measuring motor current

A test to check the condition of rotor bars that has become very common is to measure the motor current (just one phase is required) and generate a high-resolution spectrum. A portable current clamp or current transformer is used. The output is connected to the data collector (or spectrum analyzer) as usual (see your data collector manual for set up requirements).

Figure 15-17 - Current clamp

If levels are high a step-down current transformer (CT) may be required. A high resolution measurement is required centering around line frequency if possible. Measurements are often made at the motor control center.

It is better than vibration because the Line Frequency amplitude can accurately be compared to the sidebands for a more accurate indication of broken rotor bars. Vibration is influenced by unbalance and other sources but a current measurement is current only.

Some challenges are that a high resolution measurement is needed and it can be difficult to determine slip frequency if not at the machine.

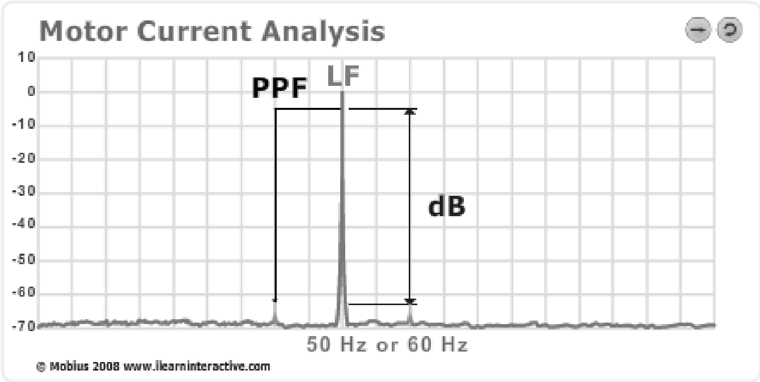


Figure 15-18

The high resolution measurement is to see pole pass frequency sidebands around line frequency (50 or 60 Hz). Use 1600 lines and a frequency range of 0-70 Hz.

The goal is to compare the amplitude level of peaks at sidebands of the pole pass frequency (# poles * slip frequency), and compare those to the level of the line frequency (not twice line frequency).

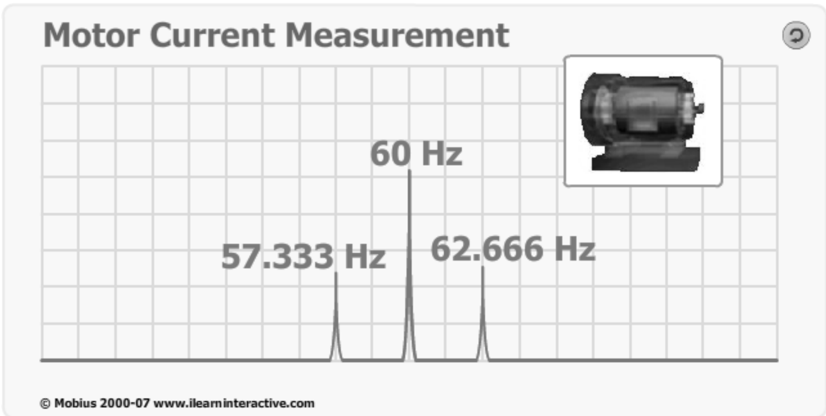


Figure 15-19 - Motor current spectrum

For example, if the synchronous speed was 1800 RPM, and the actual running speed was 1760 CPM, the slip frequency will be 40 CPM, or 0.667 Hz. In this example (in the US), we would examine the peak at 60 Hz, and at 58.666 Hz and 61.334 Hz.

If the sidebands are 55 to 60 dB down from the line frequency peak, the rotor bars are considered good, but if they rise to 40 dB below the peak, damaged rotor bars are indicated. It is possible to calibrate a system like this to relate the actual number of open bars to the sideband level if the number of bars in the rotor is known.

Loose Rotor

Sometimes the rotor can slip on the shaft, usually intermittently depending on temperature, and this causes severe vibration at 1X and harmonics. Abrupt changes in load or line voltage can instigate this condition.

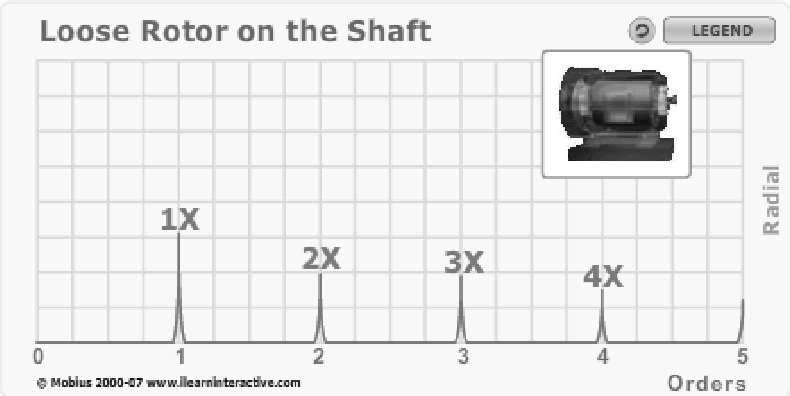


Figure 15-20 - Loose rotor produces harmonics of turning speed.

Loose Stator Windings

If the electrical windings of the motor stator are even a little loose, the vibration level at twice line frequency will be increased. This condition is very destructive because it abrades the insulation on the wire, leading to shorted turns and eventual short circuits to ground and stator failure.

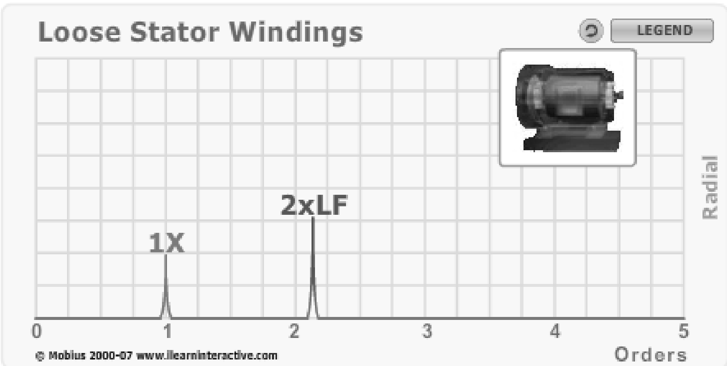


Figure 15-21 - Loose Stator Windings increase the 2x Line Frequency peak.

Lamination Problems

The rotor and stator of AC motors are made of thin laminations that are isolated from each other. If the laminations are shorted together, local heating and resultant thermal warping will occur.

Shorted laminations also cause higher twice line frequency vibration levels. The warping can cause the 1X level to increase, and often pole pass sidebands are observed.

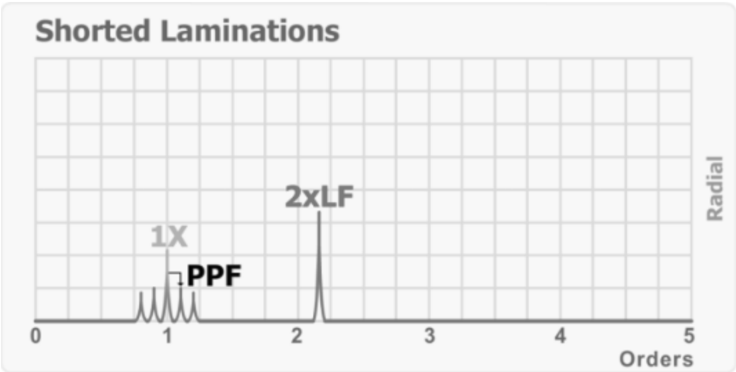


Figure 15-22 - Shorted Laminations

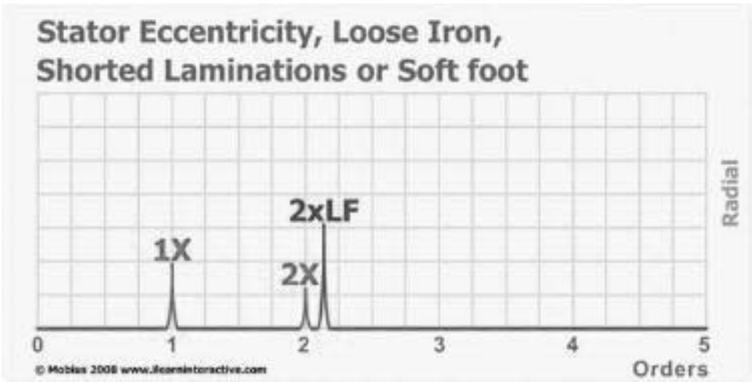


Figure 15-23

Loose Connections

Phasing problems due to loose connectors can cause excessive vibration at twice line frequency (100 or 120 Hz), with sidebands of one-third line frequency (33 or 40 Hz).

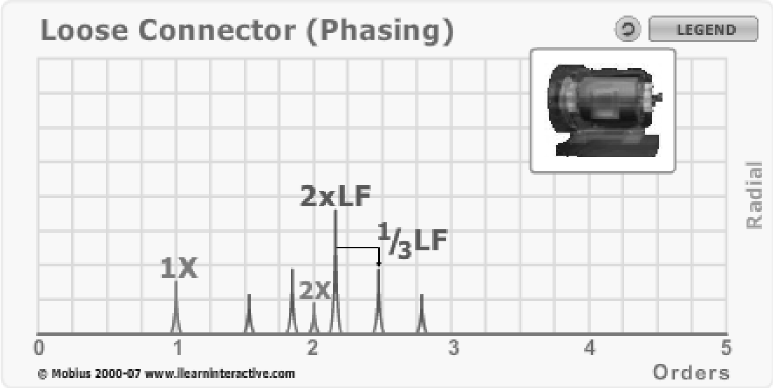


Figure 15-24 - Loose connections cause phasing problems

Chapter 16
Gearbox Analysis

Objectives:

- Calculate Gearmesh frequency
- Diagnose a broken tooth
- Describe the causes of sidebanding in gears

Gearboxes

Gearboxes have unique frequencies and depending on the configuration may be difficult to analyze.

There are a number of reasons a gearbox may fail:

- Tooth wear
- Tooth load
- Gear eccentricity
- Backlash
- Gear misalignment
- Broken or cracked teeth
- …. And others

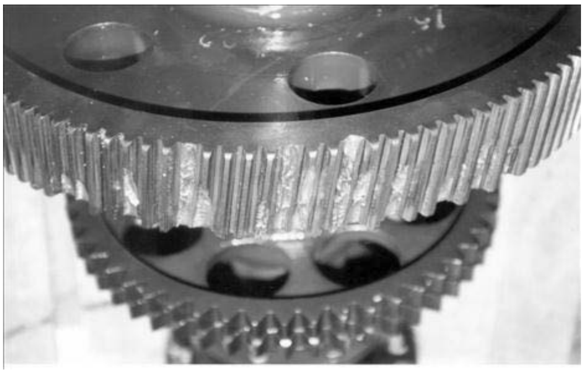


Figure 16-1

A recent study showed that 60% of failures could be attributed to lubrication skills and practical issues.

In the section on forcing frequencies gearboxes were examined in some detail. There are three key frequencies involved, the input speed, the frequency of the gearmesh, which is the number of teeth multiplied by the speed of the shaft, and the output speed.

Gear mesh = Number of teeth x Shaft speed

Output speed = Input speed x Input teeth/Output teeth

Other forcing frequencies include the **Hunting Tooth Frequency** and the **Gear Assembly Phase Frequency.**

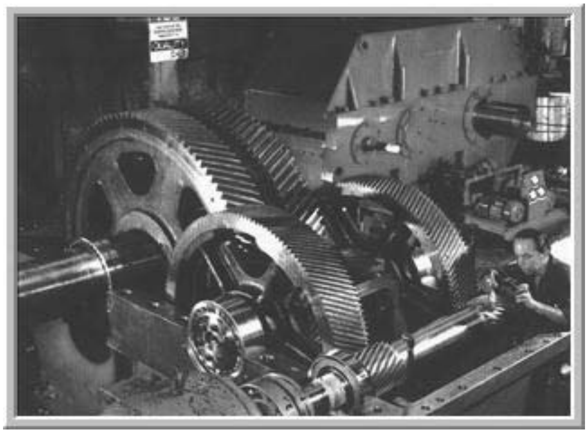


Figure 16-2

Complex Gearbox - When a gearbox has two or more stages (or teeth meshes to be precise), the calculation of the forcing frequencies becomes more challenging (as there are multiple shaft speeds and multiple gear mesh frequencies), and thus it becomes more difficult to relate increases in level with specific faults.

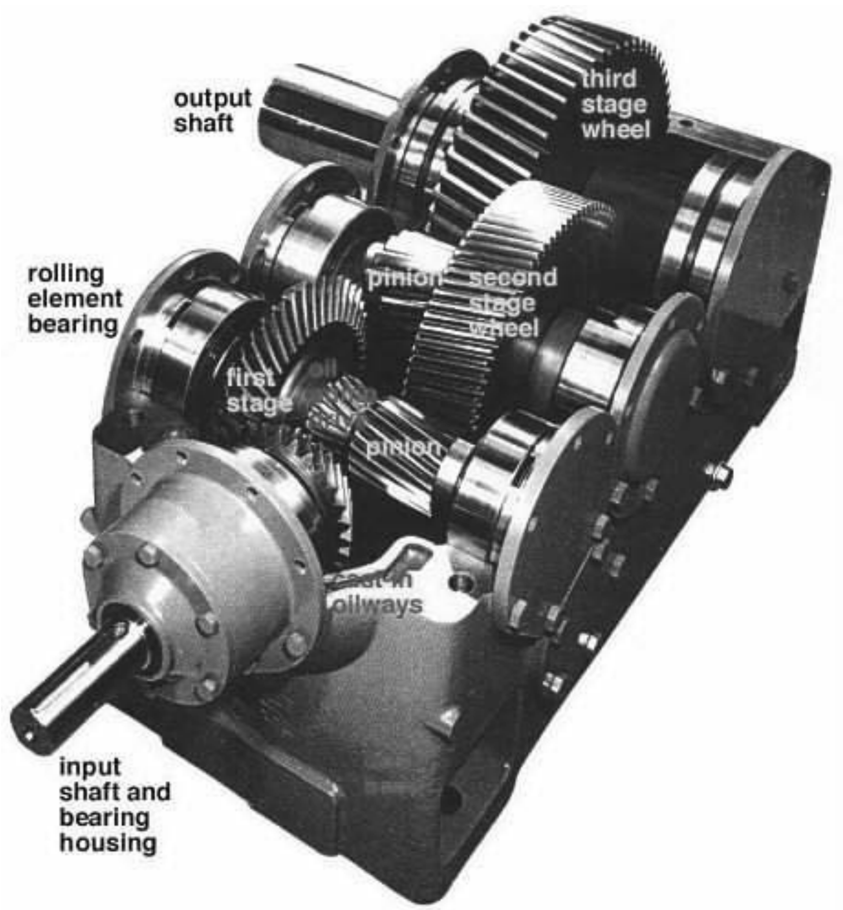


Figure 16-3 - Multi stage gearbox

There will normally be peaks at the shaft speeds and gearmesh frequency; however they will be low level. There may be a 2X gearmesh peak, and there may be sidebands of shaft speed around the gear mesh frequency.

Some faults also show up at 3x gearmesh frequency. It is therefore necessary to set the Fmax high enough to see 3x gearmesh and sidebands.

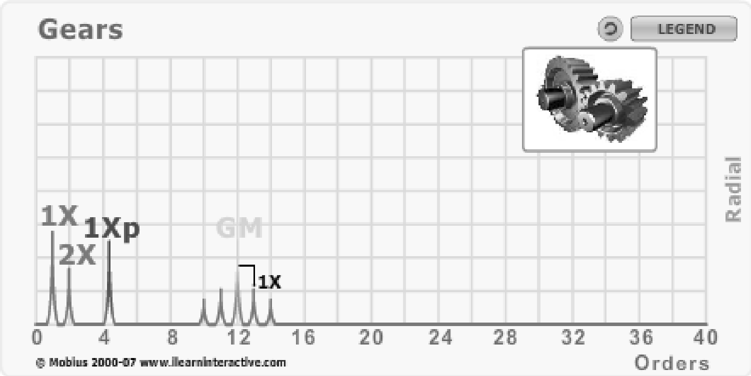


Figure 16-4 - Gears generate peaks of shaft speeds and gearmesh frequency.

These frequencies will be most prominent in the radial direction for spur gears, and in the axial direction for helical gears.

Most faults are detected by studying the gearmesh frequency and 2x and 3x gearmesh frequencies along with their sidebands. The following is a partial list of faults that can be detected through this study.

- Tooth wear
- Tooth load
- Gear eccentricity
- Backlash
- Gear misalignment
- Broken or cracked teeth

It is beyond the scope of the course to cover these in greater detail.

Waveforms and Gear Analysis

The waveform is a VERY powerful tool when attempting to diagnose gear faults.

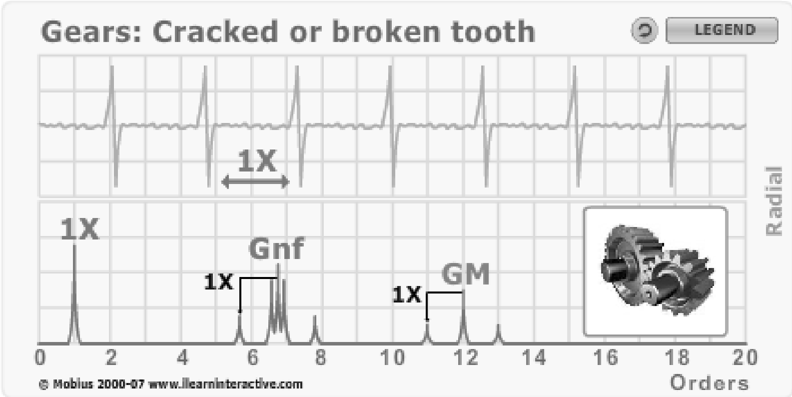


Figure 16-5 - A cracked or broken tooth will show up in the waveform as a pulse at gear rpm.

As each tooth meshes there is a pulse in the waveform. The teeth can typically be counted in the time waveform. Depending upon the nature of the fault, one of the impacts/pulses per cycle may be at a higher amplitude; or lower amplitude (if it were missing). See Figure 16-6

The waveform data in Figure 16-6 shows impacts as each gear meshes. The variance in amplitude is due to the gears being out of alignment, causing varying loads and therefore varying amplitudes. The pulses relate to the contact between two teeth. This waveform also shows the cycle that relates to a complete revolution of the shaft.

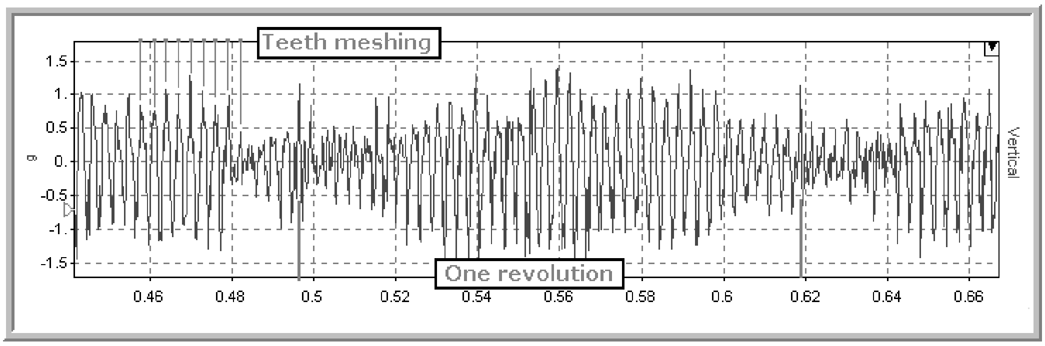


Figure 16-6 - Each impact relates to two teeth meshing.

The time waveform will be largely sinusoidal with limited modulation. In healthy gearboxes there should be no distinct transients.

A rule of thumb is to collect 6-10 rotations of each shaft with adequate resolution to see sidebands and detail.

Time synchronous averaging (TSA) is commonly used when attempting to diagnose gear faults. TSA will average away all of the vibration sources that are not synchronous with the tach pulse, which is taken from the shaft of the gear of interest.

This means that other sources of vibration, from bearings, the motor, resonances, and so on, are removed, leaving a clean time waveform. The TSA is time consuming (a large number of averages is required), and it is time consuming to set up in the first place. However the results are worth all the effort.

Tooth Wear

When teeth begin to wear, two things will happen. The first is that the sidebands of gear mesh will increase in level. The sidebands will correspond to the speed of the gear with the wear. The sidebands will develop as a result of amplitude or frequency modulation.

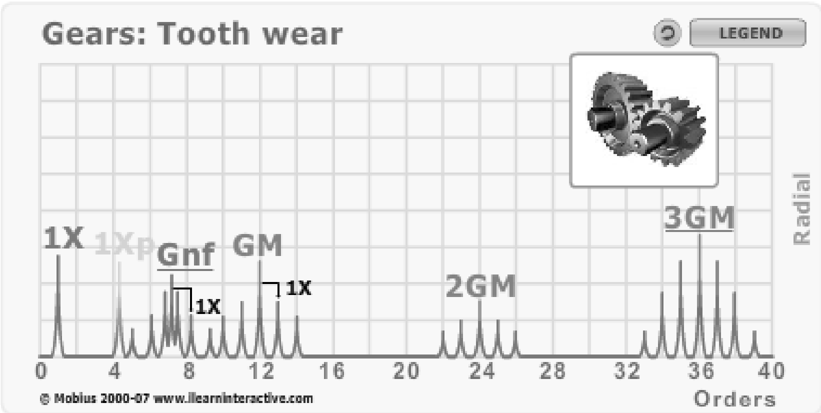


Figure 16-7 - Tooth Wear

The second thing that occurs is that the natural frequency of the gear is excited due to the impacting of the gears meshing. This peak will also exhibit sidebands, and as a natural frequency, it is likely to have a broader base.

The 3x gearmesh frequency will increase in amplitude and multiple sidebands will appear.

Tooth Load

The amplitude of the tooth mesh frequency is dependent on the alignment of the shafts carrying the gears, and the load on the gear. The higher the load, the higher the amplitude. A high peak at the gear mesh frequency does not necessarily indicate a problem. Excessive load may result in wear and damage in the future.

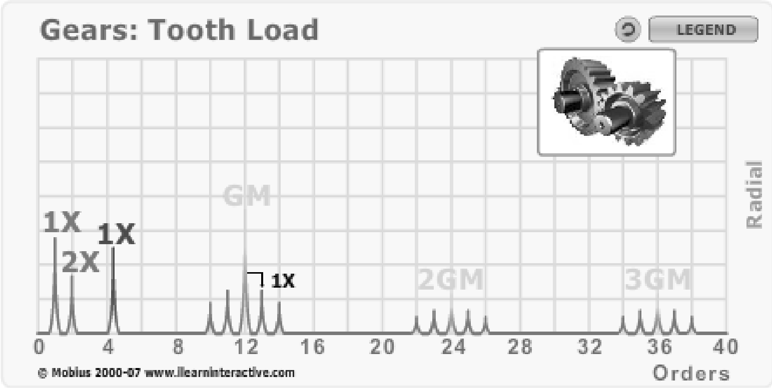


Figure 16-8 - Tooth load increases the gearmesh levels.

Eccentric Gears

Eccentric gears and gears with bent shafts cause the load to vary at once per rev (amplitude modulation) producing sidebands spaced at shaft speed (of the offending gear) around gearmesh frequency. However, often you will only see a single sideband, rather than an entire family.

The 1x Gearmesh and the 3x Gearmesh frequencies will be dominant due to “non-linear” vibration. This is caused when the teeth slide in and out of contact.

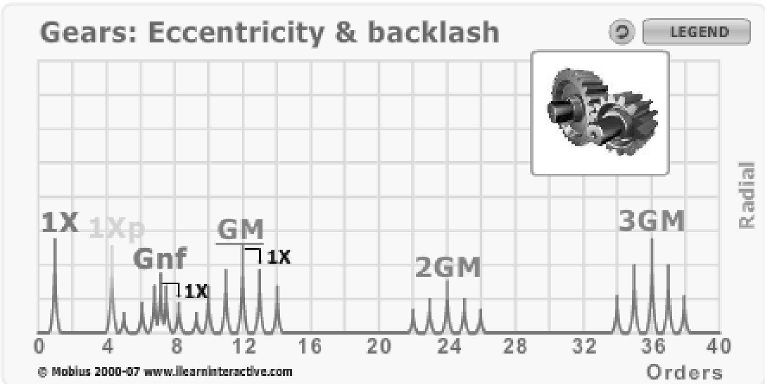


Figure 16-9 - Gear Eccentricity and Backlash

Gear Backlash

Gear backlash also generates shaft speed sidebands around the gearmesh frequency. The gear mesh peak and the gear natural frequency peak will often decrease with increased load when this problem exists. See Figure 16-9

Misaligned Gears

Misaligned gears also generate high gearmesh frequencies with sidebands, however it is common to have harmonics of gearmesh frequency, with higher levels at twice and three times gear mesh frequency.

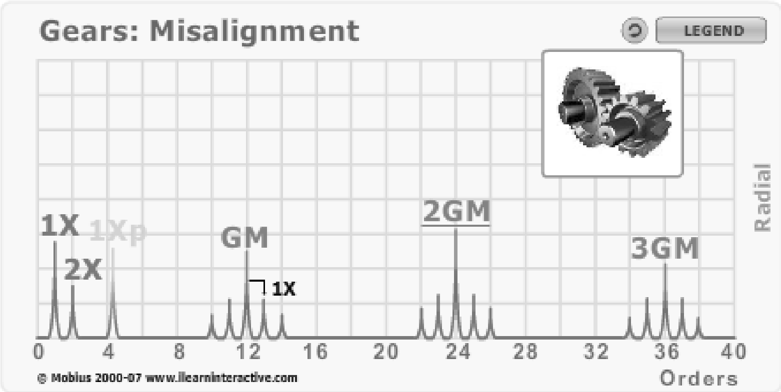


Figure 16-10 - Gear Misalignment

Wear on the gear is uneven or skewed due to the angular contact between meshing teeth. This produces sidebands around 1x, 2x, and 3x gearmesh frequency may be at 2x RPM instead of 1x RPM. The sidebands may be lower below 2x gearmesh compared to the sidebands above 2x gearmesh.

The spectral data shows the sidebands when zoomed in around gearmesh frequency.

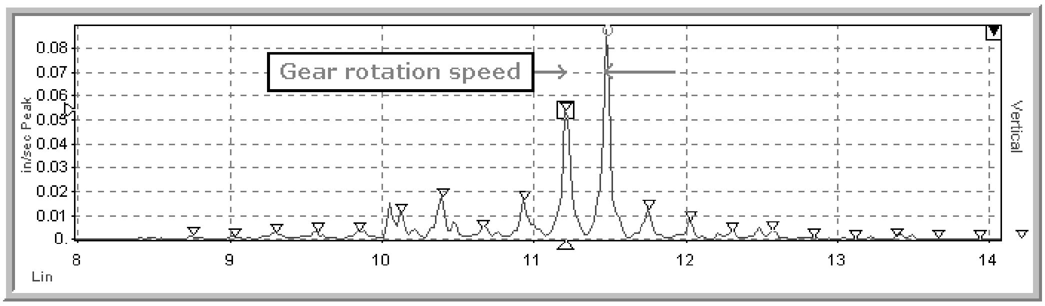


Figure 16-11 - Sidebands of shaft speed are evident when zoomed in around gearmesh frequency.

Harmonics of the gearmesh frequency indicate the presence of sidebands especially around 2x gearmesh frequency. 2x gearmesh frequency is also the highest in amplitude.

The sidebands would be clearer if the data were not integrated and would be even better in a logarithmic scale. This data actually came from the axial direction.

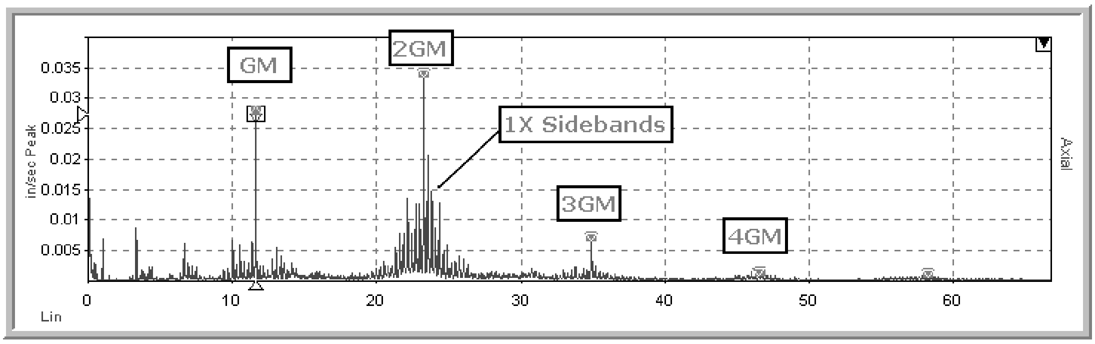


Figure 16-12 – Example - 2x Gearmesh Frequency is highest and has many sidebands.

Cracked or Broken Tooth

A cracked or broken tooth generates a high amplitude peak at the turning speed of that gear. And it will cause the gear natural frequency to be excited. There will be sidebands of turning speed of that gear around gearmesh frequency.

However, the best way to see a cracked or broken tooth is in the waveform. If there were 12 teeth, one of 12 pulses in the waveform will be very different from the others. The time difference between these pulses will be equal to the period of the turning speed of the gear because the tooth comes into contact once per revolution.

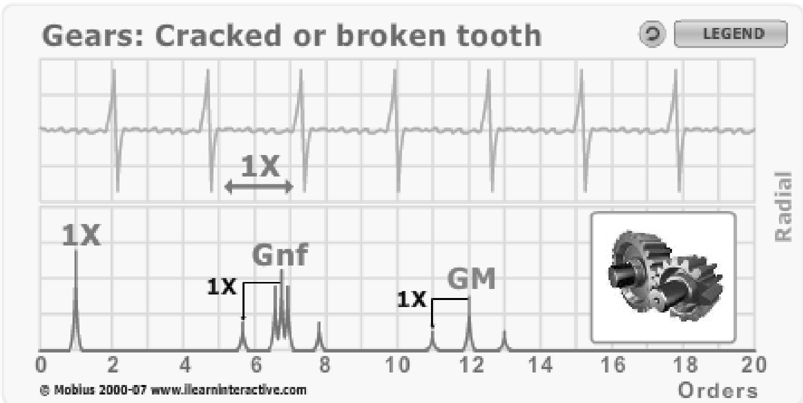


Figure 16-13 - Broken tooth shows up best in the time waveform as a pulse spaced at shaft speed.

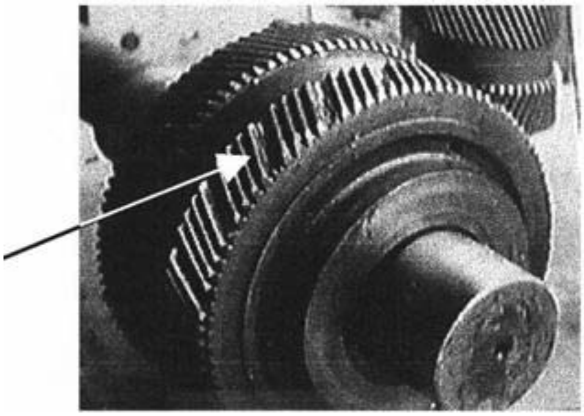


Figure 16-14

Hunting Tooth Frequency

The "hunting tooth frequency" is the rate at which a tooth in one gear mates with a particular tooth in the other gear. If the gear ratio is an integer such a 1, 2 or 3, the hunting tooth frequency will be the RPM of the larger gear, and the same teeth will be in contact once per revolution. This will cause uneven wear on the gears - a small defect in one tooth will repeatedly contact the same teeth in the other gear causing localized wear on those teeth.

For this reason, gearboxes are not made with these simple ratios unless absolutely necessary. Ideally, the hunting tooth frequency should be as low as possible to evenly distribute the wear around both gears. This means the number of teeth on each gear should be a prime number.

In practice, the hunting tooth frequency is used to detect faults on both the gear and pinion that may have occurred during manufacturing or as a result of mishandling. It is typically a low frequency, and a "growling" sound comes from the gearbox.

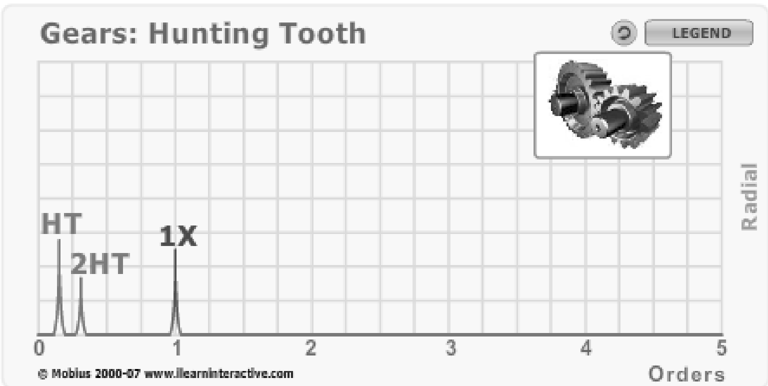


Figure 16-15

Gear fault frequencies can be calculated with the iLearnInteractive gear calculation program. Simply enter the input shaft speed, the number of teeth on the gear. The calculations for the

Prime and Common Factors are provided along with gear related frequencies such as Gear Assembly Phase and Hunting Tooth Frequency. The calculations and animations are displayed are made for the gearmesh frequencies and various faults.

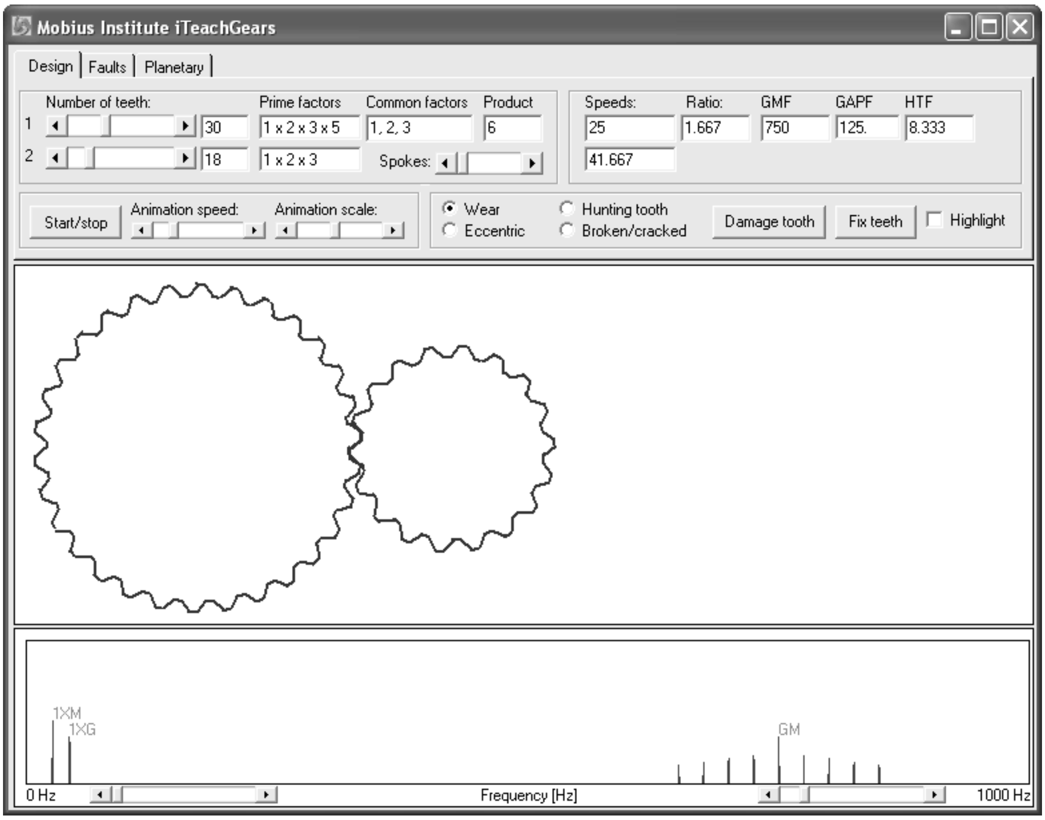


Figure 16-16 - Gear Frequency calculator from iLearnInteractive. The bottom of the screen shows how the spectrum will look if there are faults.

Time Waveforms and Gearbox analysis

Figure 16-17

Here is an example of gears with spalling on the teeth. The spectrum does not make it very clear what the problem could be, but the waveform clearly reveals the problems. There is not a peak over 0.006 in/sec.

The spacing of the impacts in the waveform are at the RPM of the gear with spalling. In this case notice the low g levels that would not trigger an alarm.

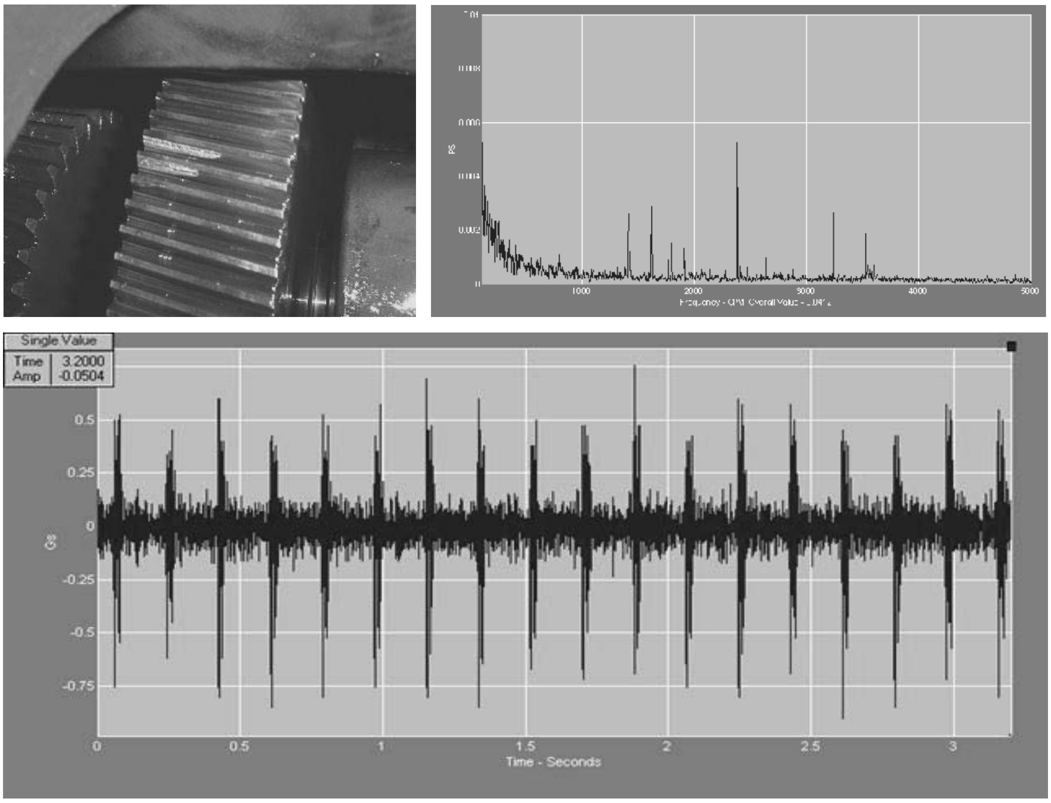


Figure 16-18 - The impacts are spaced at the RPM of gear with spalling. No alarms were triggered.

Wear Particle Analysis

Wear Particle Analysis often provides a warning of gear damage well before vibration analysis does.

Note that wear particle analysis is typically separate from the routine oil analysis which misses abnormal wear particles. Cutting wear, abrasive wear, and sliding wear produce particles that are larger than the capabilities of spectroscopy.

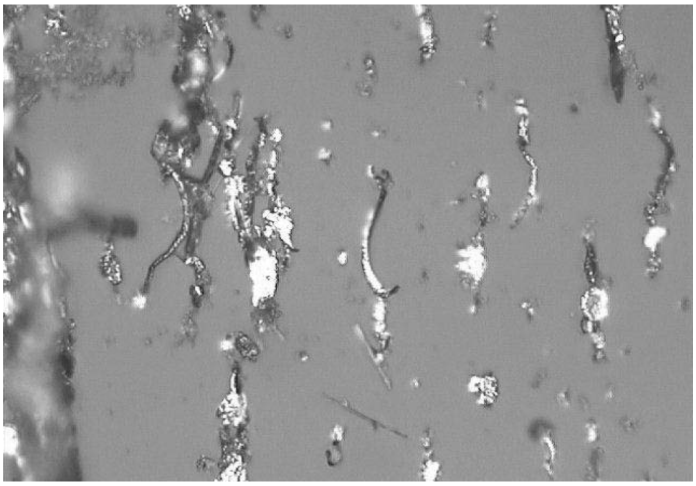


Figure 16-19 - Wear particle analysis often reveals gear damage before vibration does.

Chapter 17

Pumps, Fans and Compressors

Objectives:

- Calculate Forcing Frequencies associated with pumps, fans, compressors
- Diagnose Vane Pass in spectral data
- Identify Cavitation in spectral data
- Describe the characteristics of Cavitation in Waveform Data
- Identify air flow problems in spectral data

Pumps, Fans, and Compressors

Pumps, fans and compressors can all suffer from the classic fault conditions such as unbalance, misalignment, bearing faults, and looseness. However, some faults are unique to these components.

There are many types of pumps in common use, and their vibration signatures vary over a wide range. When monitoring pump vibration, it is important that the operating conditions are consistent from one measurement to the next to assure consistent vibration patterns. Suction pressure, discharge pressure, and especially air induction and cavitation will affect the vibration pattern greatly.

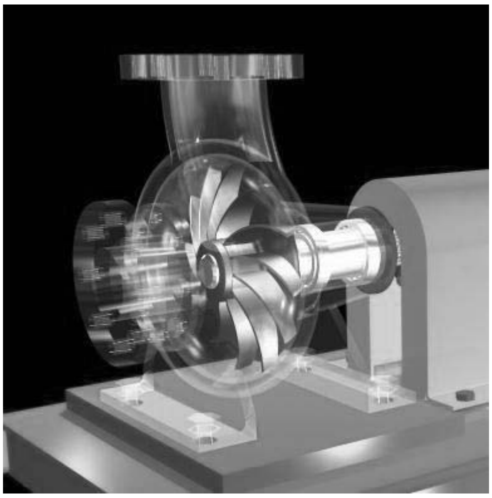


Figure 17-1 - Pumps, fans, and compressors have some unique vibration characteristics.

Centrifugal pumps always have a prominent vibration component at the vane pass frequency, which is the number of impeller vanes times the RPM. If the amplitude increases significantly, it usually means an internal problem such as erosion of impellers or a flow related problem, or possibly misalignment. Harmonics of vane pass are also common in such pumps.

Forces are generated by the pressure variations as a rotating blade, vane, or lobe passes a stationary housing or component.

The stationary component creates a non-uniform flow disturbance in the fluid or gas.

Blade Passing Frequency

In its broadest sense, Blade Passing Frequency includes similar components such as vanes, lobes, and pistons. The forcing frequencies are:

- Number of blades x turning speed
- Number of vanes x turning speed
- Number of lobes x turning speed
- Number of pistons x turning speed

Causes of high blade passing frequency are:

- Rotor or housing eccentricity
- Non-uniform variable pitch blades
- Loose, bent, or misaligned housing diffuser vanes
- Blade or vane wear (abrasion or cavitation)
- Operation (improper performance parameters)
- Improper damping settings
- Dirty, damaged or missing filters
- Inlet or discharge line restrictions

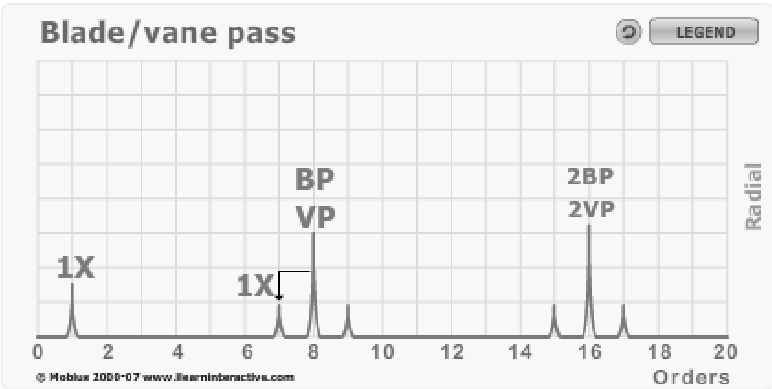


Figure 17-2 - Blade/vane pass and harmonics

Other faults include the impeller loose on the shaft and pump starvation.

When the impeller is loose on the shaft, there will be vane pass frequency with sidebands of turning speed.

Pump starvation may look like unbalance as the flow into the volute is uneven. The time waveform would show distortion due to the flow being uneven. The distorted time waveform produces harmonics of turning speed.

Cavitation

Cavitation normally creates random, higher frequency vibration or "noise". It is often observed as a "hump" in the vibration spectrum raising the floor from about 15X to 35X.

Cavitation normally indicates insufficient suction pressure or starvation - i.e. low inlet pressure. The liquid tends to vaporize while coming off the impeller creating vacuum bubbles that implode. The waveform can often sound like gravel in the pump.

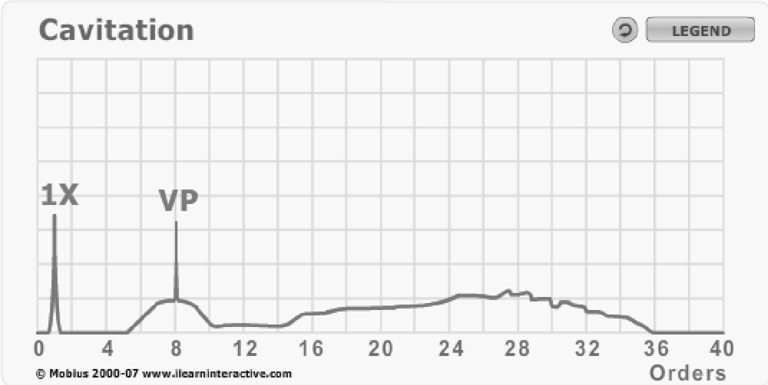


Figure 17-3 - Cavitation often has a hump of energy in the floor

The data in Figure 17-4 is from a 20 hp centrifugal pump that has a cavitation problem. It sounds like it is pumping gravel. The time waveform is a very useful analysis tool, as the high frequency bursts of energy are often clearly visible - however, more time data (i.e. a longer record) than normal may be needed.

The spectrum reflects the impacting in the raised noise floor and mounds of energy under the peaks. The highest peak is at vane pass.

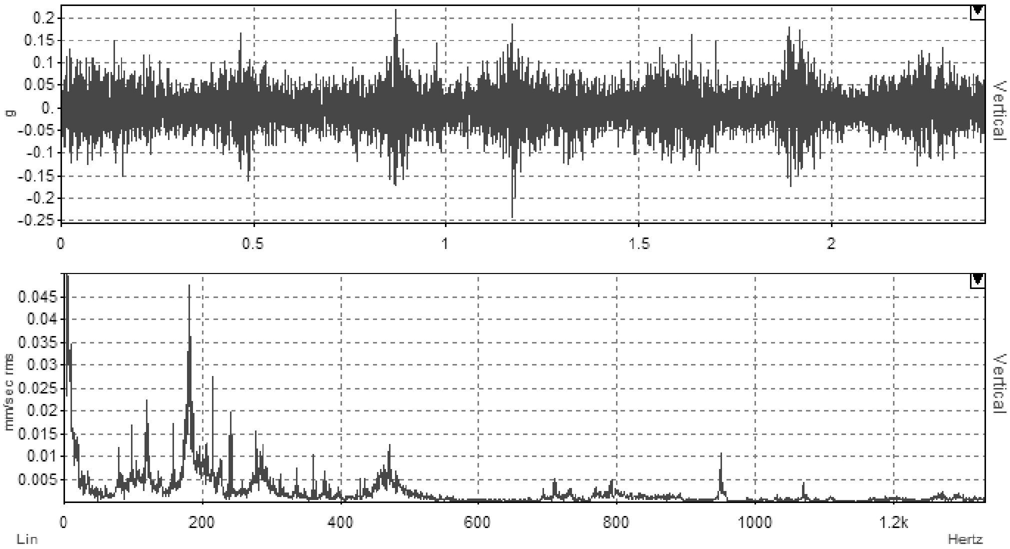


Figure 17-4 - Waveform has random bursts of energy. Spectrum has raised noise floor throughout

Turbulence

Turbulence in fans, pumps, and compressors is induced when the normal flow is restricted or impeded during operation.

It shows up in spectral data as low frequency, broad peaks between 0.5 and 0.8 orders.

Figure 17-5 - Flow Turbulence in fans and pumps

Causes of turbulence are:

- Obstruction in air ducts or plumbing lines
- Sharp radius turns in piping
- Abrupt diameter changes in lines
- Over capacity operation.

Harmonics

Lobed blowers and screw compressors generate harmonics under normal conditions.

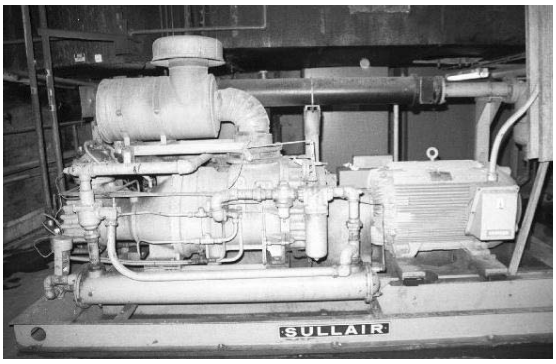


Figure 17-6

Sullair Rotary Screw Compressor, 1650 CFM

Speed is 1780 CPM

300 HP motor

Female rotor has 6 vanes and the male has 4 vanes.

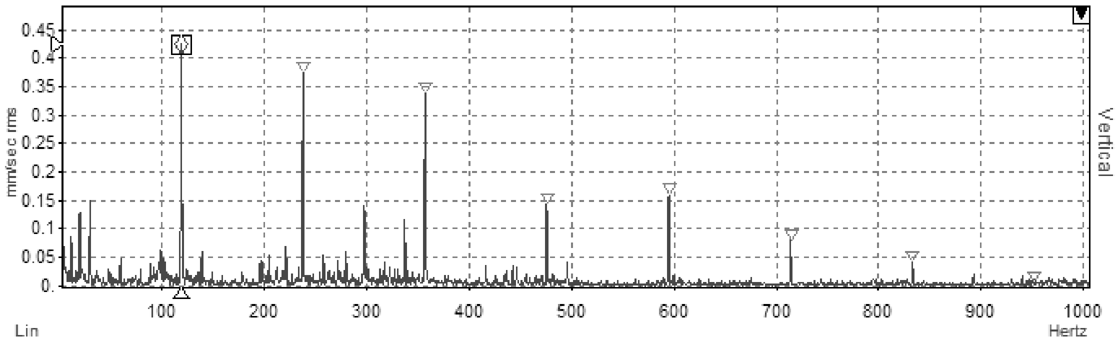


Figure 17-7

Chapter 18

Vibration Analysis Process

Objective:

- Learn the analysis process
- Learn to analyze spectra

The analysis process

Before you analyze data, remember that you did not begin this process with data, you began by:

- Understanding the machine, its components, its failure modes
- Calculating forcing frequencies
- Defining standard test conditions
- Testing the machine in the correct positions with the correct data collector setups etc.

Vibration data does not mean much on its own. We must begin the process by understanding the machine we are testing, understanding what faults the machine can have, determining how to test the machine in order to detect those faults, testing the machine properly, ensuring that the data is valid etc. These are all topics we have covered thus far in the course.

What this means is that if we just have a vibration spectrum and no other information about where it came from, we would not be able to interpret it. Analyzing vibration data is actually a tiny part of a condition monitoring program, most of the work is in understanding the machines and their sources of vibration, defining standard test procedures, ensuring that data is collected in a repeatable way and setting up good baselines. If this is all done correctly, one can simply compare new data to a baseline and very quickly determine if the machine has a problem or not.

Here is a general procedure that describes the analysis process:

- Validate the data
- Find the running speed / normalize data
- Look for harmonics
- Identify forcing frequencies
- Compare data to a baseline
- Look for and identify faults the machine can have

We will now review these items step by step.

Validating the Data

Every time a measurement is collected, there is a chance that something will go wrong. The transducer may be loose, the machine may surge in mid-test, the operating conditions may not be correct, and more.

We have already reviewed the causes of bad data in the data acquisition section when we discussed how to collect good data. Now pretend that the data has already been collected and you are sitting in your office, looking at the computer screen and about to review the data and look for problems in the machines. At this stage you also need to verify that the data is valid. It also may be a case of one person collecting data and another person doing the analysis. We are now considering the person doing the analysis and the steps he needs to take.

Here is a suggested process:

1. Ensure that the spectrum has peaks - and not just electrically related peaks (at line frequency and multiples). Make sure that there is some information about the mechanical state of the machine.

2. Look for the classic "ski-slope" in the graph. If the spectrum begins at a high level at the low frequency end of the graph, and slowly decays across the graph, then it is likely that there has been a problem with the transducer.

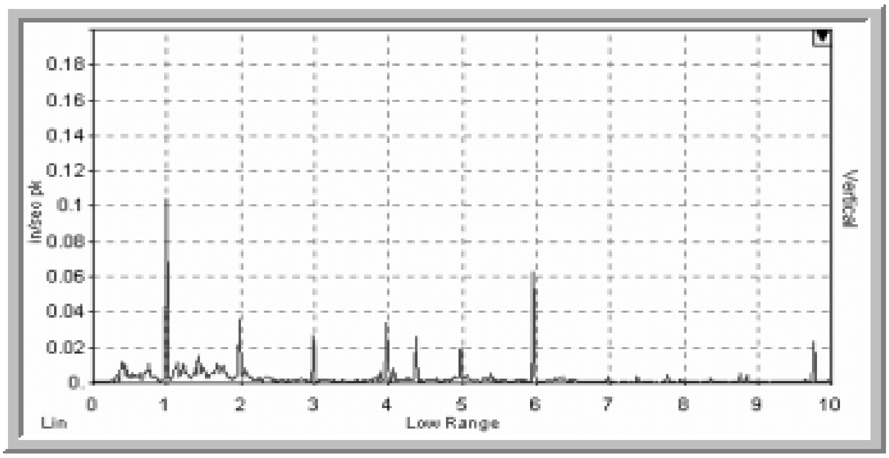


Figure 18-1 Ensure the data has peaks related to mechanical occurrences.

The most common problem is transducer related. The most common cause for the ski-slope is a transient of some kind. The transient can be mechanical, thermal, or electrical.

As discussed in the transducer section, there are a number of telltale signs that there has been a fault in the transducer or in the way it has been mounted:

- Poor setup can cause a chunky time waveform
- Loose or broken cable
- Loose mounting
- Insufficient settling time
- Electrical
- Temperature
- Transducer bumped during collection time

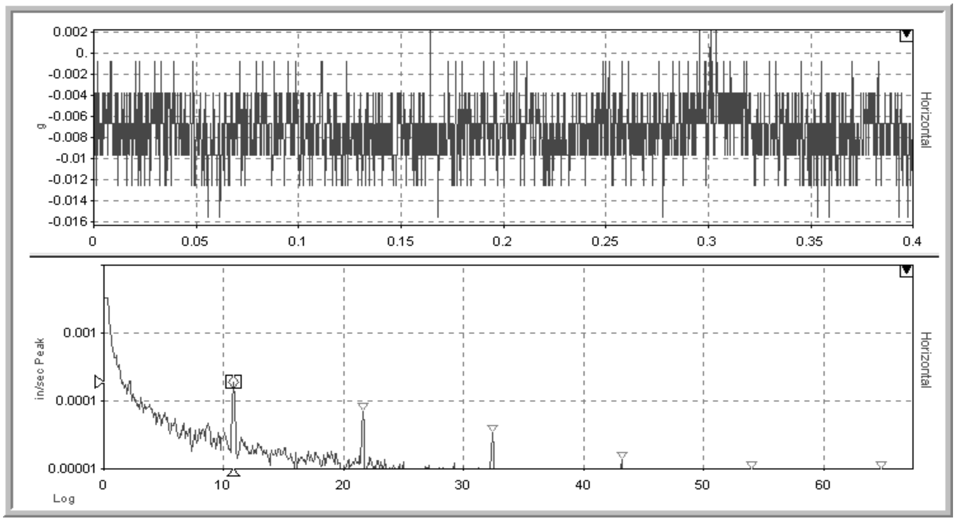


Figure 18-2 Bad Data. Chunky time waveform and ski slope in spectrum.

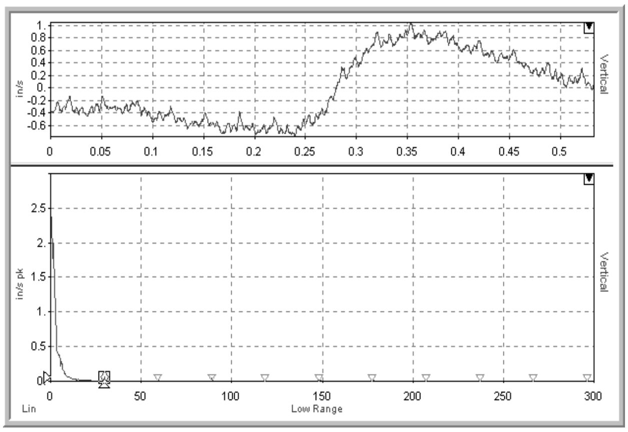


Figure 18-3 Inadequate Settling Time causes shifts in the time waveform and a ski slope in the spectrum.

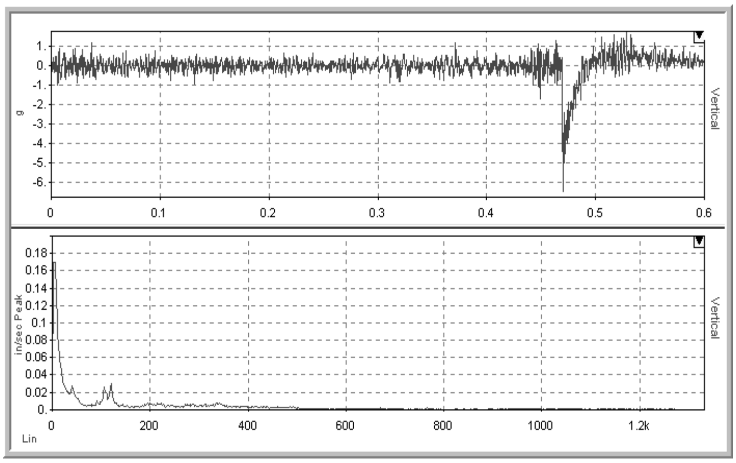


Figure 18-4 This type of transient can be caused by bumping the transducer. It causes a ski slope in the spectrum.

3. Verify the running speed peak. Determine whether the machine was running under the correct condition and speed?

Assuming that the data looks "clean", perform the next check: was the machine running under the correct condition? Identify the running speed peak? Was it running at the correct speed?

It can be difficult to tell if the load or speed were not correct, unless of course they were considerably different to the norm. Does the data look "different?" Or is it necessary to perform a comparison against other data from the same point.

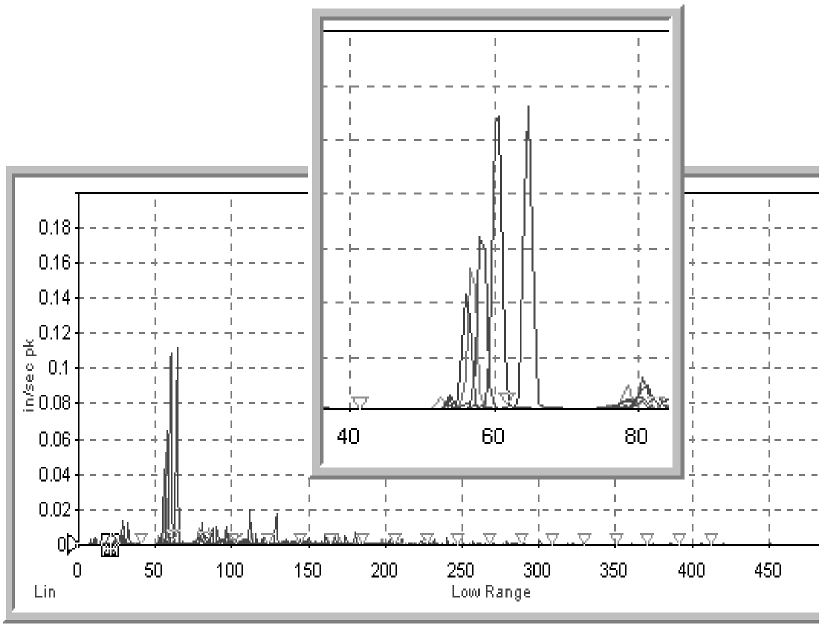


Figure 18-5 This plot is two measurements overlaid on each other to compare running speed peaks.

4. Next, if possible verify that the data is from the correct machine, measurement point, and direction.

Many times either Unit A **or** Unit B will be running. Is the data from the correct Unit?

Is the data from the correct point and direction on the machine? I.e. is it really from the drive end of the component, and in the vertical direction?

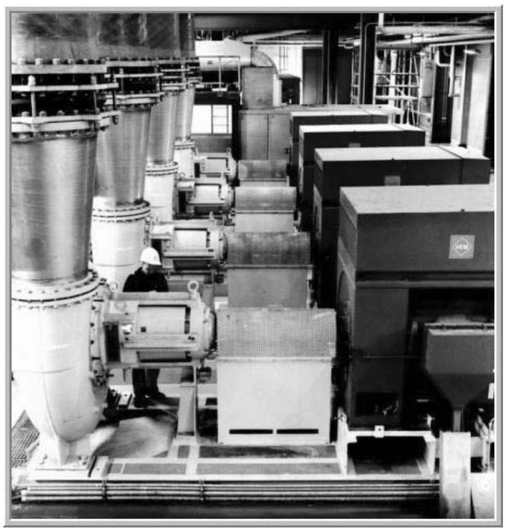


Figure 18-6 Verify that the data is from the correct unit.

These checks may seem unreasonable, but mistakes do happen. It is good to quickly verify these things whenever possible.

Data presentation

What can be learned about the machine from the spectrum in Figure 18-7? Is there a problem? Are the vibration levels normal? Is the pattern changing?

Unfortunately there is no universal database of standard spectra to compare this with.

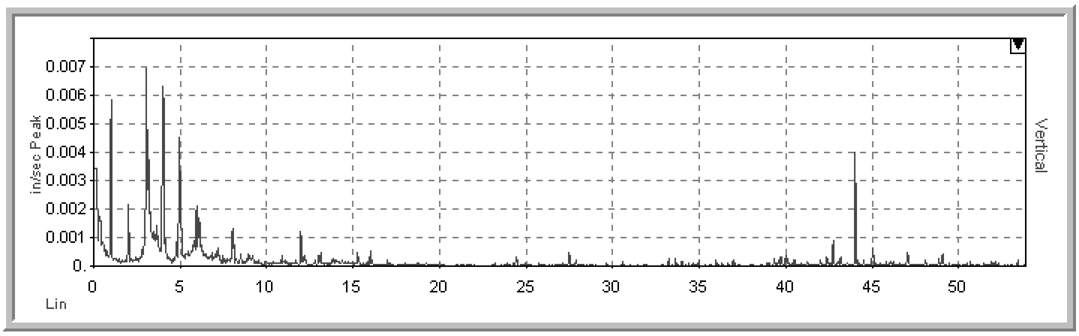


Figure 18-7 What can be learned from a single spectrum?

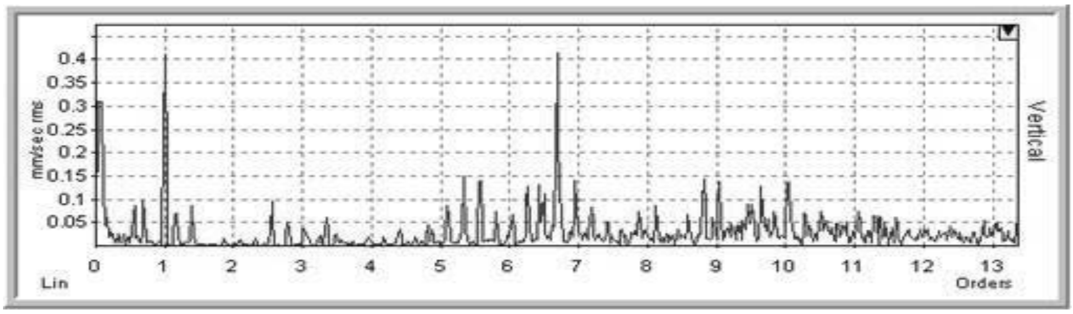


Figure 18-8

In order to do a proper job of analyzing the data, it helps to present the data in certain ways that make it easier for us to quickly identify the frequencies of interest and also to quickly see if there have been any changes in levels or patterns. Remember, the goal of a condition monitoring program is to detect changes in machine condition, so it is helpful to be able to quickly compare the new data to some reference based on prior data.

These may include: prior measurements from the same point, reference spectra, alarm limits, spectra from other axes, spectra from other locations on the machine, and data from the same location and axis but from another identical machine.

Comparisons are performed for two basic reasons: to see how the data has changed (or how it differs from the "norm"), and to understand the motion of the machine.

Condition monitoring is all about change. We may think we can recognize problems by just looking at a single spectrum, however if we do not know how the vibration data normally looks, or whether the levels or patterns are changing, then do we really know if there is a problem?

Figure 18-9 How is the machine moving in all three axes?

It would be great to have a universal database of how every machine should vibrate; a reference we could use to test our machines. But that does not exist (and it can't exist).

Instead we have to use previous test data, data collected on other identical machines within the plant, and reference data. The reference data may be spectra taken immediately after an overhaul (or soon after the machine was new), or statistically derived baseline data

The spectra and waveforms are then watched for change and we use our growing knowledge of vibration analysis and the machine to interpret the change.

The second reason to compare vibration data is to understand how the machine is vibrating as a structure. Because the whole machine vibrates, data can be compared from one axis to another, and between locations to better understand the motion of the machine.

It would good to be able to visually animate the entire machine in slow motion, exaggerating the movement so that we can clearly see how it is vibrating. Faults such as unbalance, misalignment and resonances, and to a lesser extent looseness can be best diagnose by understanding the whole motion.

But this is not possible (not without special software and more time than is probably available during routine collection), but there is a lot that can be done with the data you routinely collect.

Spectral comparisons to reference data

Condition monitoring relies heavily on comparisons between current data and older data. If it is possible to know how much the data has changed, and where it has changed (what frequencies) then that is a real help.

All analysis software allows previous measurements to be recalled, giving a quick view of how much the vibration levels have changed since the last test.

Overlay Graphs

The easiest way to view the data is to overlay the graphs one atop the other. Then it is plain to see exactly how the data has changed at each spectral peak. Three or four graphs can be displayed this way and still keep track of which spectrum came from which date.

Given that the goal is to see change in the level, if all of the spectra overlay closely, then there is really nothing to worry about.

When there is more significant change, then it is time to analyze the data more closely.

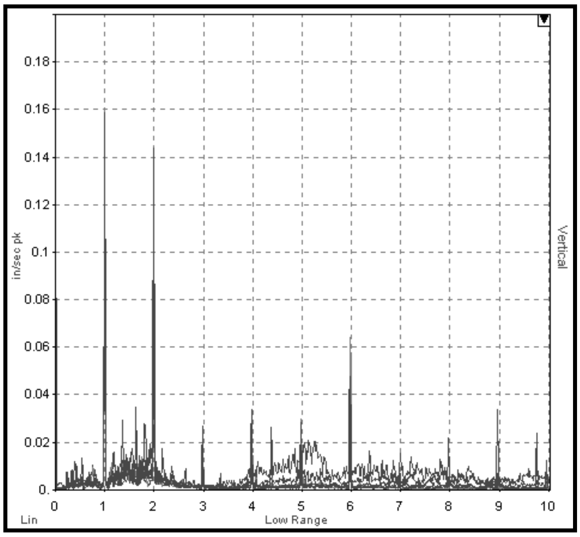


Figure 18-10 Plots overlaid on each other to show change

One of the graphs can be singled-out, or viewed in a way that better highlights the change. In some software programs a trend of the level over time at a specific frequency is available. With other software, trends will be defined by the alarm bands that are set. The intent is the same, even if there is not quite the same level of control.

Act on larger changes

- Look for changes in amplitude – especially at known forcing frequencies

- Look for changes in pattern such as new peaks, harmonics, sidebands and raised noise floor (Figure 18-11)

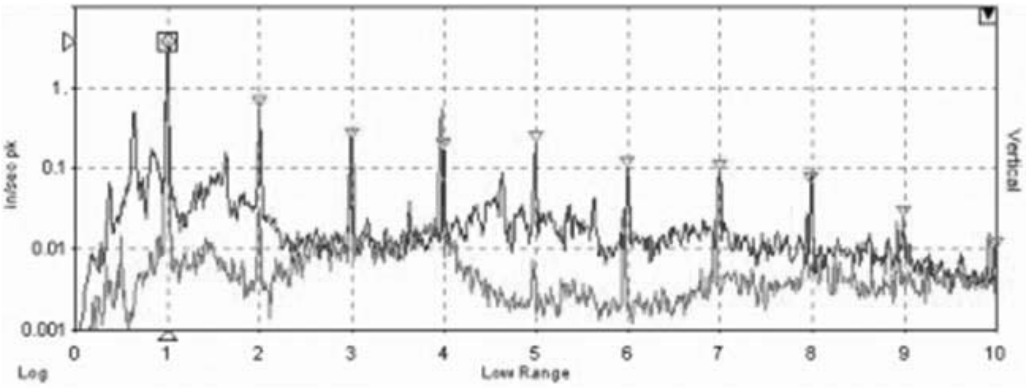


Figure 18-11 Look for changes in pattern and amplitude

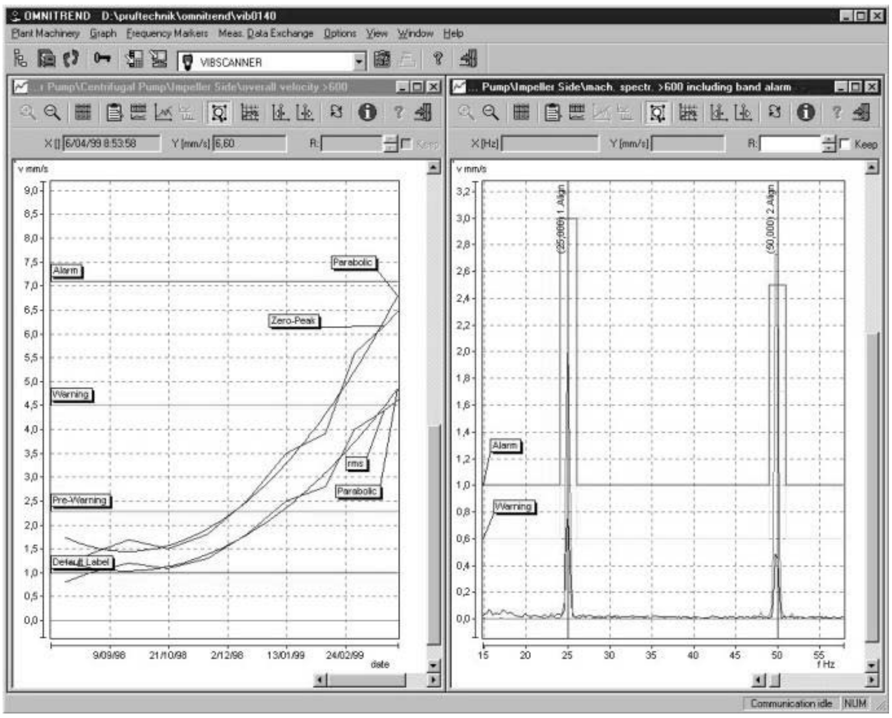


Figure 18-12 Trends of a particular area of a spectrum can be trended for a quick indicator of changes occurring.

Trending

Trend graphs provide a quick visual view to the changes that are occurring. Figure 18-13 shows a trend of vibration levels at particular frequencies and in frequency bands. There are many different trend plots available in most software packages.

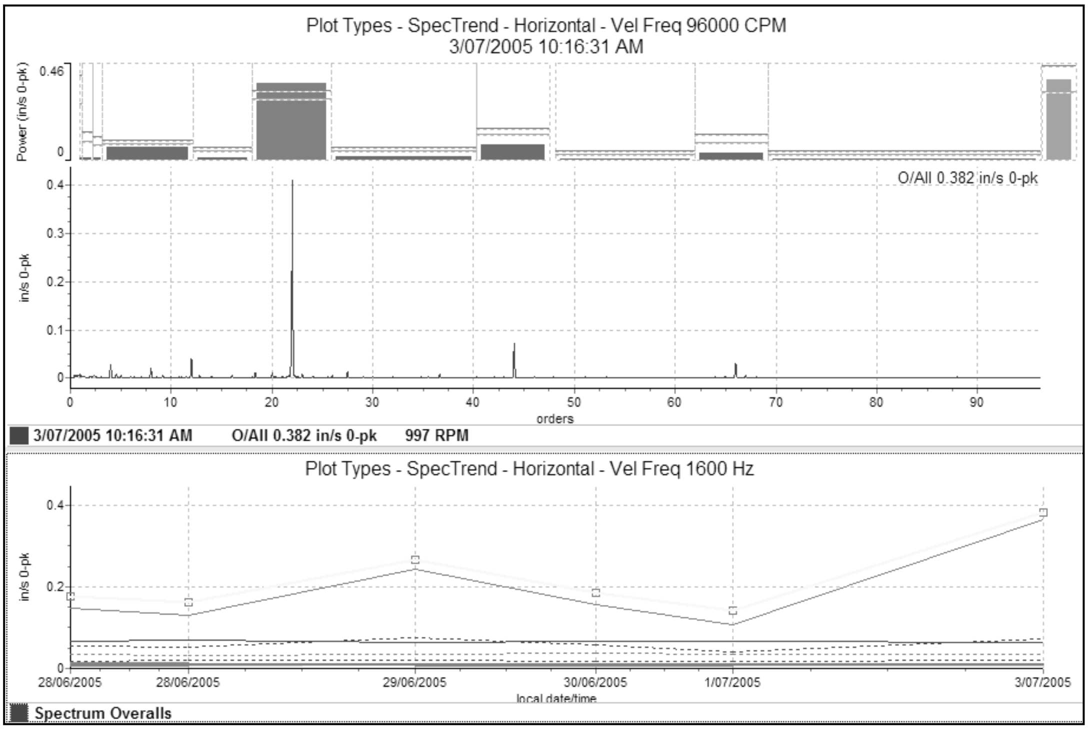


Figure 18-13 Trend plots

Trending is probably the most important part of analysis, whether it is temperature, pressure, vibration, oil analysis, or any other process. Simply looking at a single value, single time waveform or spectrum and trying to diagnose machine condition is often impossible to do. The more important question to ask is: "What has changed?"

This is where trend plots are useful. They allow us to select items of interest, such as forcing frequencies, overall levels or entire spectra and ask the question "What has changed?" and if we display the data well, we should be able to answer that question very quickly. In other words, if we plot things correctly it should be very obvious to us very quickly, what has changed. Then we can confirm the problem and take action.

Stacked Plots

Another graphical comparison method available in some of the software programs allows the spectral graphs to be stacked vertically. The difference in level of each major peak can still be viewed quite clearly while still seeing the complete spectrum.

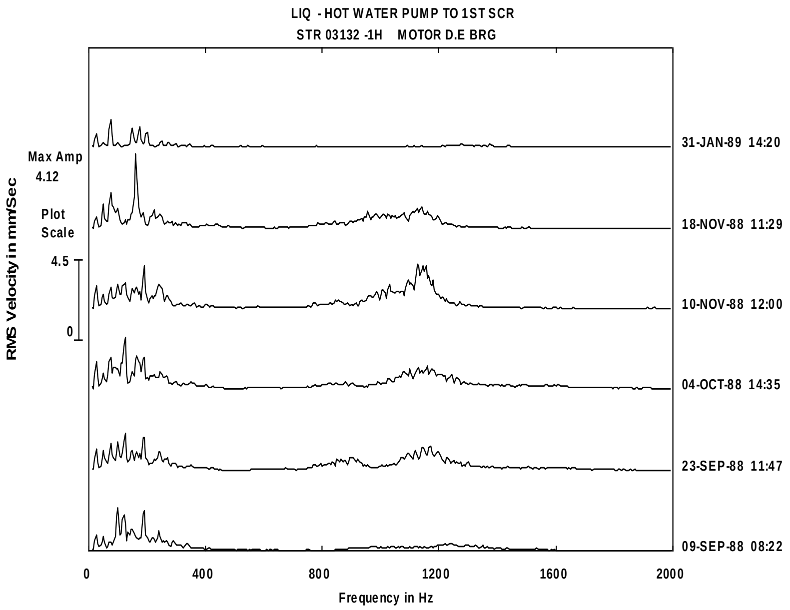


Figure 18-14 Changes from one data set to another are apparent when stacked vertically

One shortfall of the stacked plot is that with larger data sets there is insufficient room available for each graph; they become cramped, and it is difficult to see sufficient detail. The waterfall plot can solve this problem.

Waterfall Plots

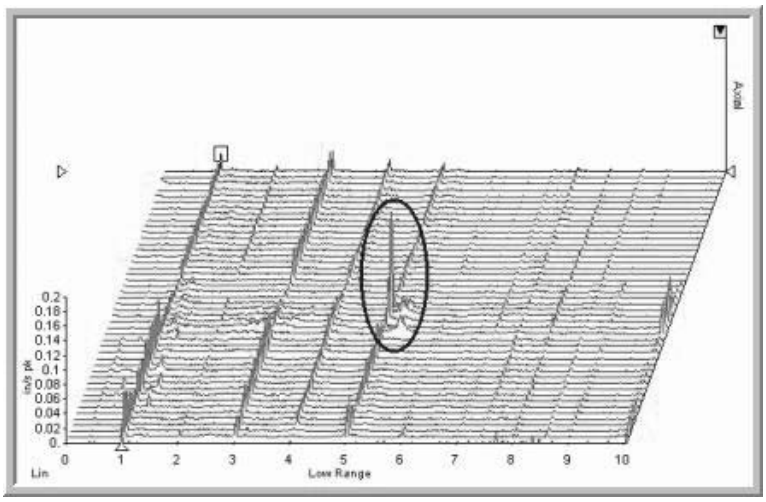


Figure 18-15 Waterfall plot of dozens of spectra showing the changes over time. Note that each spectrum comes from a different date.

A very popular way to study a larger sample of spectra is the waterfall plot. This plot is used to show how the peaks and patterns have changed over a period of time.

Waterfall plots reveal a great deal of information. They can reveal all manner of trends and interesting patterns.

After a few years of data it is easy to see where faults have begun to develop and then disappear after the fault was dealt with (after the repair for example). They therefore serve as an excellent historical review of how faults develop, and what levels are reached before repair is necessary.

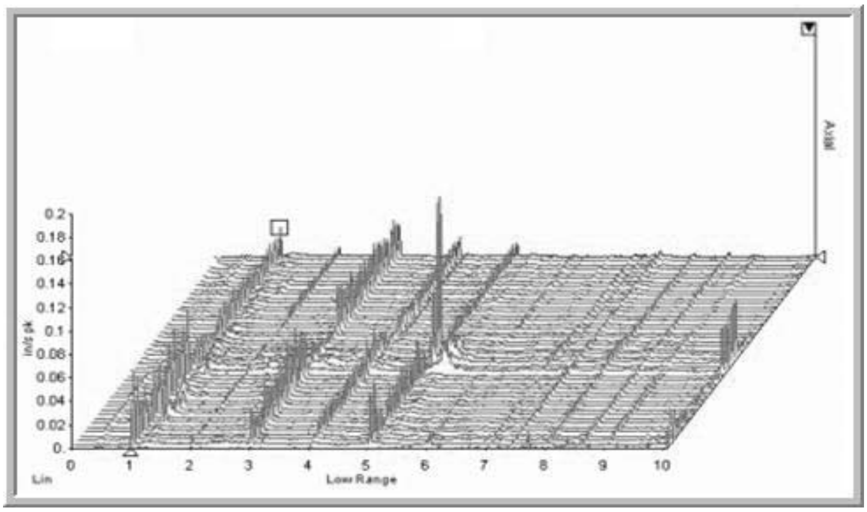


Figure 18-16 Historical spectral data provides information about the stable and changing frequencies.

Logarithmic Displays

The logarithmic graph scale is often used to highlight harmonics, sidebands, and other patterns in the data. This is because logarithmic scale displays small amplitudes in the presence of large amplitudes.

It goes without saying, therefore, that logarithmic scaling is also useful when performing graphical comparisons between two sets of data (or between spectral data and reference data). Just look at the difference between these two graphs - they both contain the same data.

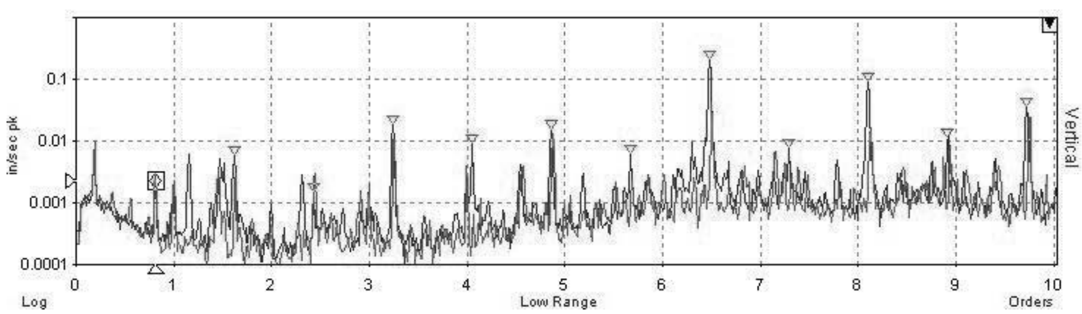


Figure 18-17 Two overlaid spectra in logarithmic scaling. Note how clear the harmonics appear now.

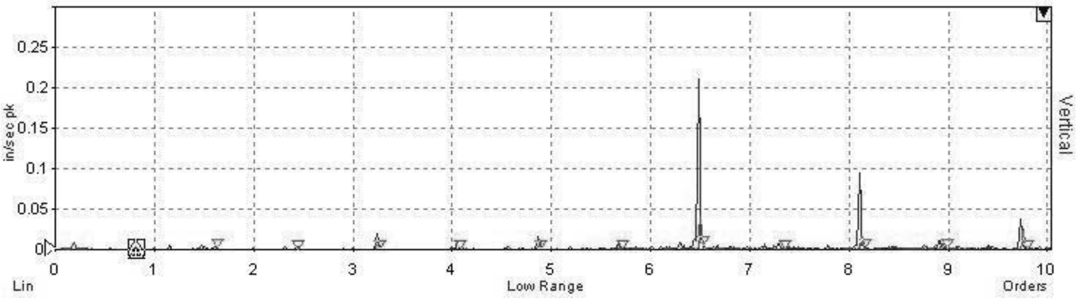


Figure 18-18 The same data as Figure 18-17, but linear scaling. Where are the harmonics?

Whole Machine Approach

Machines move in three axes, horizontally, vertically, and end to end.

Simply put, knowing how the whole machine is vibrating, conditions like unbalance, misalignment and structural resonances would be much easier to solve. Bearing faults would be easier to pinpoint as well - identifying which bearing had the fault. And it would be easier to pinpoint which bolts were loose.

We will study how and why the vibration from one location should be compared to the next, and between the axes.

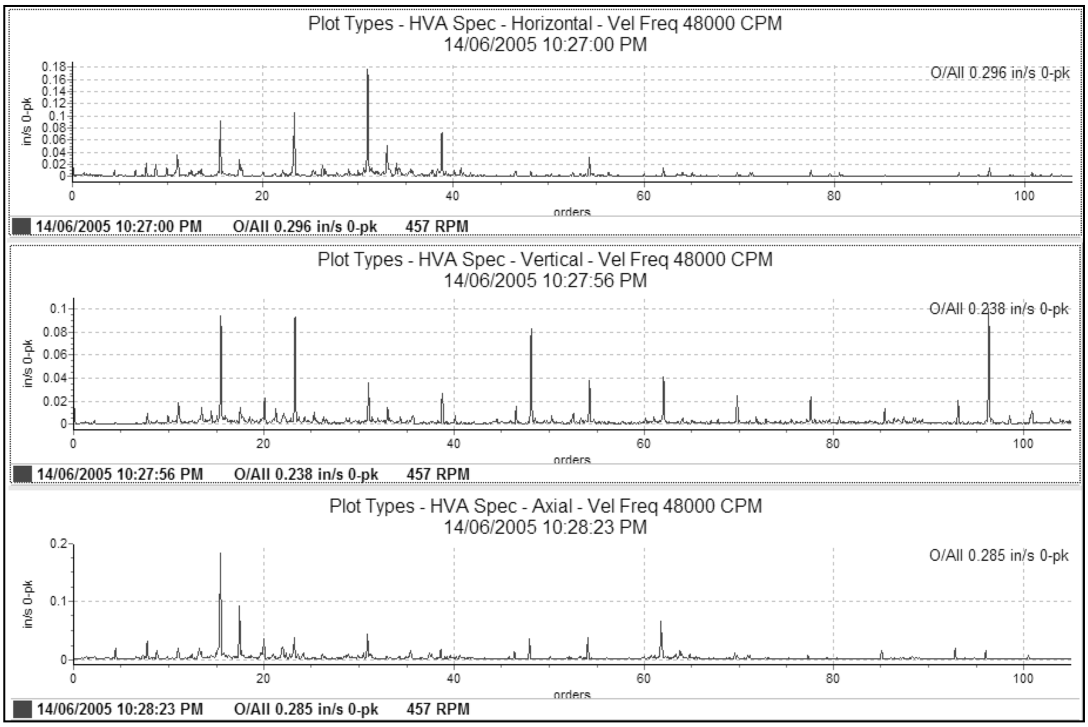


Figure 18-19 It is best to collect and analyze all three axes

Example: Pump flexibility

For example, if you had a simple horizontal motor-pump and you suspect unbalance (because you saw a high 1X peak), then you should look in the other axes as confirmation. Why; because the rotational forces should produce vibration in the vertical and horizontal directions but very little in the axial direction.

But what if the vibration was much higher in the horizontal axis than the vertical axis (Figure 18-20)? This could mean that the real problem is a flexible foundation, not unbalance. In other words, different faults show up in different axes and some are diagnosed by comparing the relative levels and patterns in different axes.

This is just one example. When you do study the various fault conditions in greater detail, you will be doing yourself a favor if you try to understand the forces involved. This will help you to understand what to look for in the three axes of vibration.

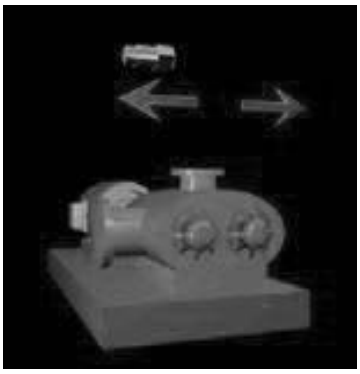


Figure 18-20

Machine Orientation

Take the machine orientation into account, too. A vertical machine will vibrate differently than a horizontal machine. Given that a horizontal pump is restricted in its movement in the vertical direction as compared to horizontal, the vertical and horizontal vibration are not expected to be equal.

A **vertical pump**, on the other hand is far more likely to have equal vibration in the radial and tangential (the two horizontal directions), unless the movement is restricted in some way.

This is just one reason why we recommend collecting as many axes of vibration as possible, either with a triaxial sensor, or with as many individual single-axis measurements as possible. Of course, additional tests can be made as follow up just to confirm a diagnosis.

Compare to other locations

Now that there is understanding of how the knowledge of the whole-machine vibration can help with a diagnosis, it follows that comparisons from one point on the machine to others should be made.

For example, if a misalignment problem is suspected, compare measurements from both sides of the coupling - on the pump and motor for example.

If unbalance or a resonance/flexibility problem is suspected, check the spectrum at different points on the same machine component to see if they have higher levels in the same direction. If a machine is "bouncing" up and down, then expect all vertical measurements to be high at the resonant frequency.

The bottom line is that when the different fault conditions become familiar, think about how the whole machine will vibrate, and then adapt the measurements and analysis accordingly.

Compare to another machine

It can often be useful to compare data from one machine to another. These comparisons help in understanding what is normal for a machine.

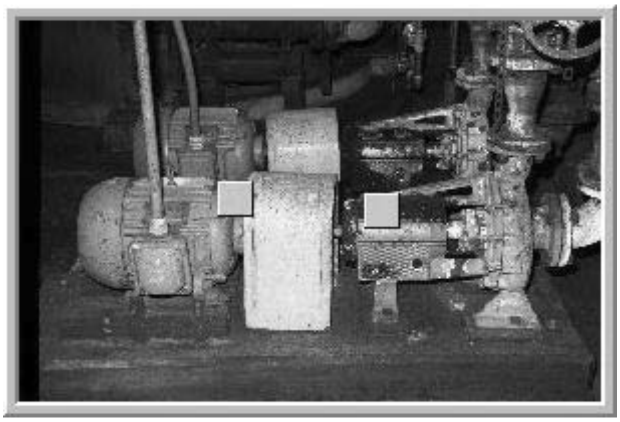


Figure 18-21 Compare Units 1 and 2

In this example, a comparison between two identical machines reveals quite different patterns - indicating that one has a higher than normal pump vane rate peak (at 6X), and the other has a high running speed peak. Without this reference it is difficult to know what is normal for this pump.

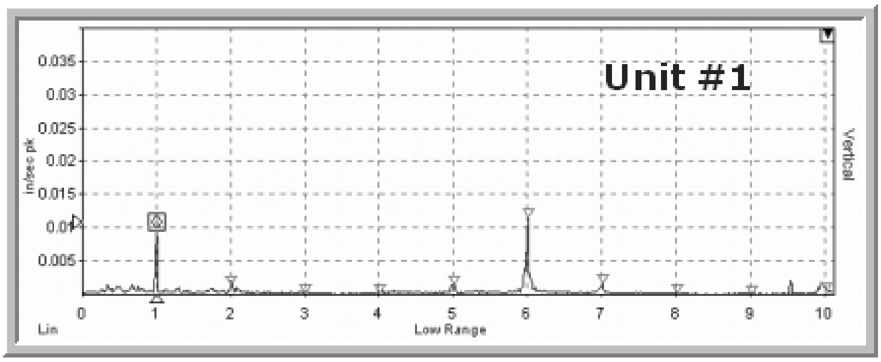


Figure 18-22 Compare Unit 1 and Unit 2. Which is normal? 6x peak is higher than unit 2.

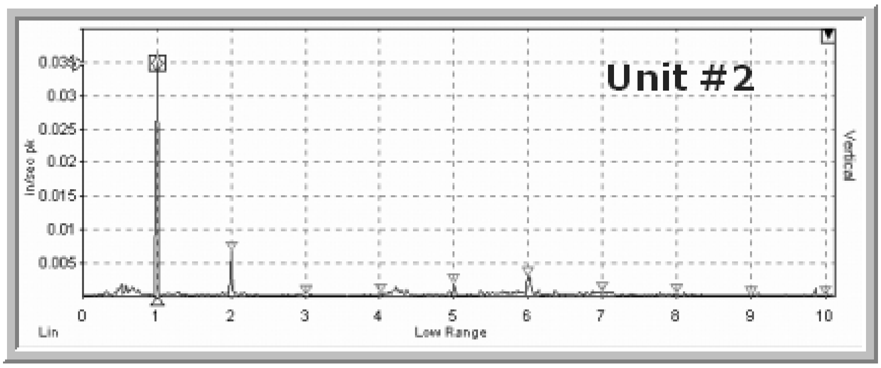


Figure 18-23 Is the 1x peak in Unit 1 or Unit 2 normal for this pump?

Summary:

- In a condition monitoring program, the best reference is data taken from the same machine, tested in the same exact way month after month, year after year.
- Develop a baseline and see how the vibration changes rather than focusing on vibration troubleshooting where you try to diagnose faults from one spectrum.

Step-by-step approach to analyzing vibration

Analysis phase: Verify and normalize

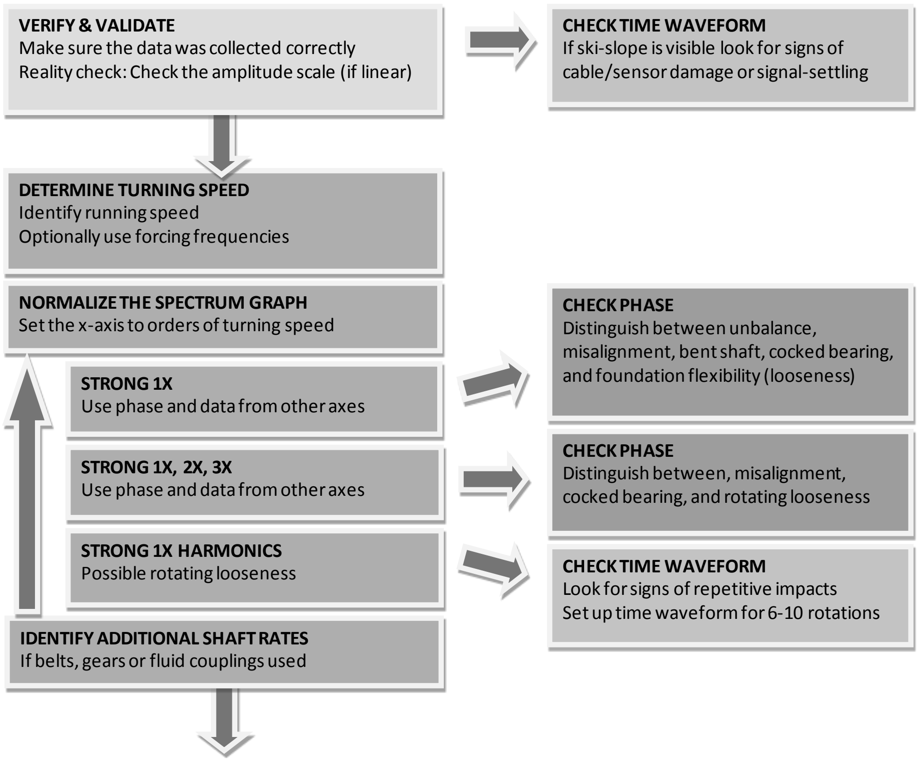


Figure 18-24

The first main task is to verify that the data we are analyzing is "good data", and then to "normalize" the data and based on what we see, launch our investigation in three different directions.

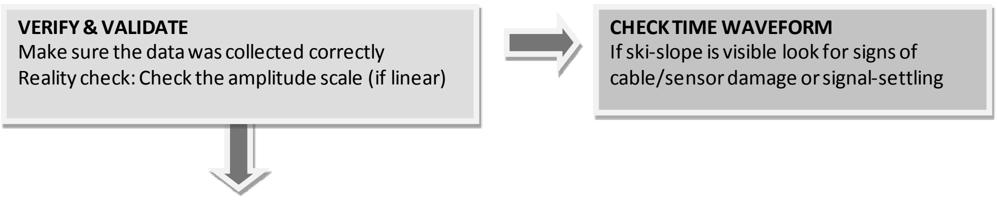


Figure 18-25

Step one is to look at the data and get an overall impression as to the state of the data (how high is the amplitude, and if it is very low you may be able to move on); does it appear to have come from the correct machine/point (compare it to a baseline reading), does it appear to have changed much (compare it to the baseline or previous reading or statistically generated alarms – if it has not changed you may be able to move on); and does it appeared to have been measured correctly (is there a “ski-slope” – and if so, check the time waveform to see if you can learn why).

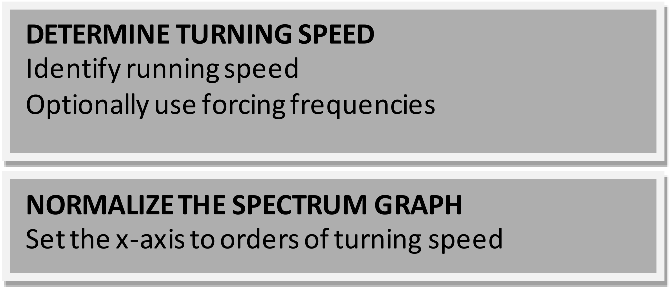


Figure 18-26

Now you must identify the turning speed for the machine, and find the 1X peak. So many fault conditions can be identified in relation to the 1X peak (or frequency). It is the author’s recommendation to normalize the graph (view the graph in terms of orders or multiples of 1X).

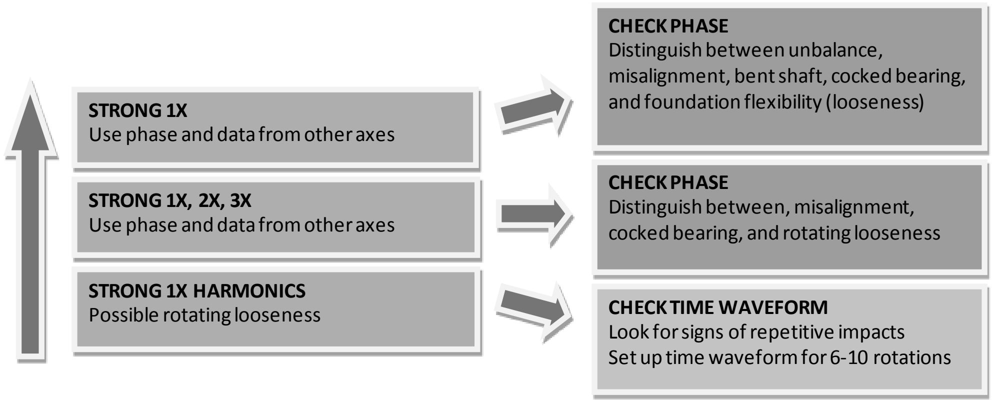


Figure 18-27

Now you can investigate the 1X peak. If it is higher than normal you need to compare the amplitude in other axes, and you need to look at the 2X ,3X, and 4X peaks, and you need to see if there are strong harmonics. There are a whole range of fault conditions that could be indicated, including: unbalance, misalignment, bent shaft, rotor bow, eccentricity, coked bearing, looseness (all kinds), and resonance – and possibly other faults in the event that a forcing frequency (e.g. vane pass rate) happens to be a low order (e.g. 2X, 3X, etc.). Phase analysis and time waveform analysis can help to distinguish between all of these fault conditions.

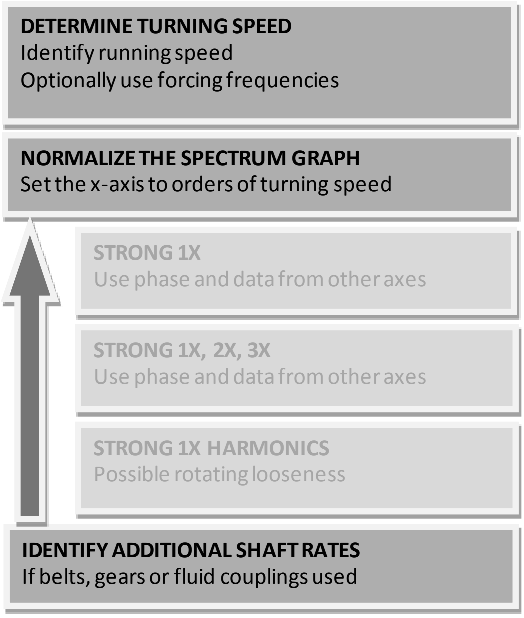


Figure 18-28

If the machine has multiple shafts (e.g. it is belt or chain driven or it has a gearbox), then you need to repeat this process for the turning speed of each shaft. For example, this is what helps you to distinguish faults related to the motor versus those on a belt-drive fan.

Analysis phase: Forcing frequencies

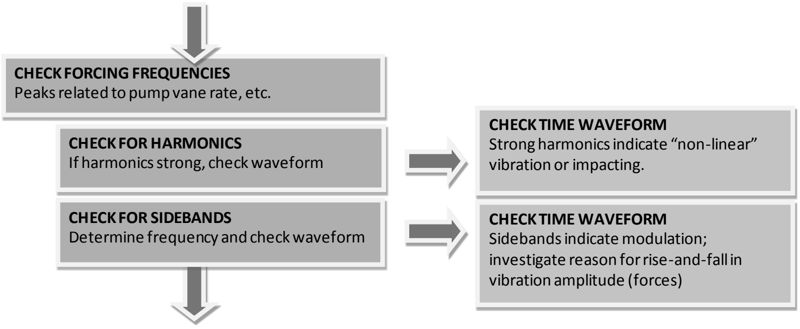


Figure 18-29

The next step is to consider all of the *defined* forcing frequencies such as vane pass, blade pass, lobe pass, gearmesh, and bearing frequencies. You must identify if there is a peak at one or more of these frequencies; whether the amplitude of the peak has changed (and if it is significant), and then see if there are 1X sidebands centered around that peak, or if there are harmonics of that peak. For each different fault condition there are different patterns that you will look for, but the presence of harmonics and/or sidebands tells you a great deal. As a rule, if you see harmonics or sidebands, the time waveform could reveal additional information.

Analysis phase: Unexplained peaks

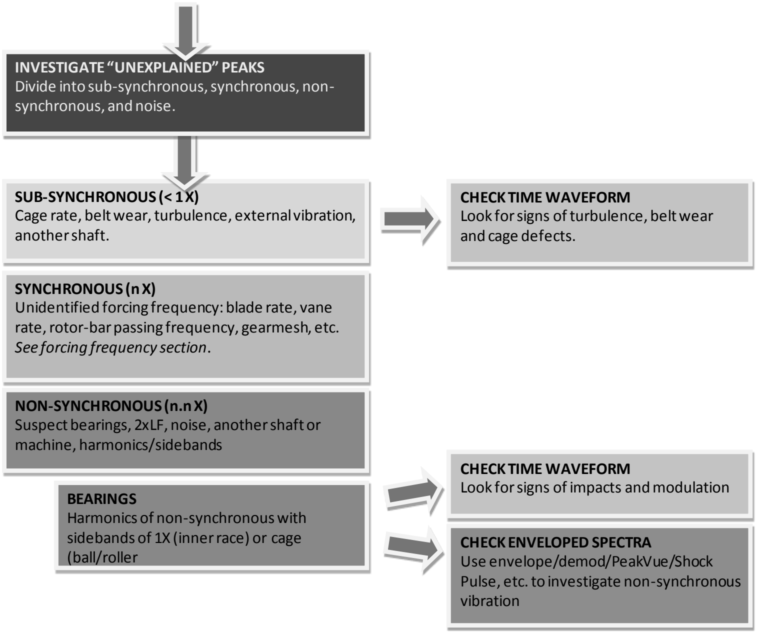


Figure 18-30

Once you have accounted for 1X (and 2X, 3X, 4X, and harmonics) and the defined forcing frequencies, you may be left with a number of peaks that are "unexplained". They may be unexplained because you have not fully defined your forcing frequencies (including the bearing defect frequencies), or they could be peaks from external sources (nearby machines), or they could be generated by the machine.

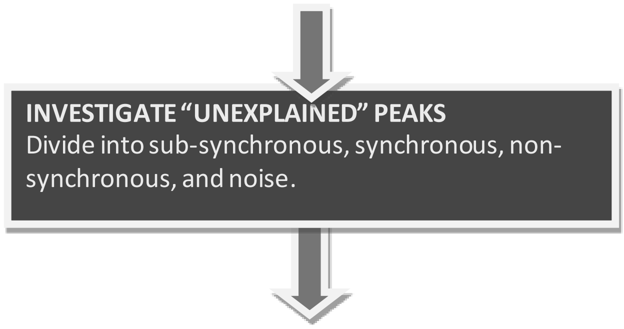


Figure 18-31

It is very helpful to "mark-off" the peaks that you can identify so that the "unexplained peaks" are easier to identify.

Now you must categorize the peaks according to their frequency relative to the turning speed of the shaft.

ANALYSIS PHASE: UNEXPLAINED PEAKS: SUB-SYNCHRONOUS

SUB-SYNCHRONOUS (< 1X)
Cage rate, belt wear, turbulence, external vibration, another shaft.

→

CHECK TIME WAVEFORM
Look for signs of turbulence, belt wear and cage defects.

Figure 18-32

Sub-synchronous peaks are lower (in frequency) than the turning speed. They could relate to the bearing cage frequency, belt frequencies, hydraulic instability such as oil whirl or oil whip, turbulence, rotor rub, shaft rub, compressor wheel rub, or fractional order peaks from severe looseness. It is always important to look for harmonics, and if found, look at the time waveform.

ANALYSIS PHASE: UNEXPLAINED PEAKS: SYNCHRONOUS

SYNCHRONOUS (n X)
Unidentified forcing frequency: blade rate, vane rate, rotor-bar passing frequency, gearmesh, etc.
See forcing frequency section.

Figure 18-33

If the peaks are synchronous, that is, they are an exact multiple of the turning speed of one of the shafts, then they could relate to blade, vane, lobe pass frequencies; reciprocating motion; gears; slot frequency or rotor bar pass frequency in motors. (Of course, we have already assessed unbalance, misalignment and other faults that would generate 1X, 2X, and other low-order synchronous frequencies.) As we discussed with the identified forcing frequencies; we would look at the amplitude, the change from previous and the presence or harmonics or sidebands.

ANALYSIS PHASE: UNEXPLAINED PEAKS: NON-SYNCHRONOUS

NON-SYNCHRONOUS (n.n X)
Suspect bearings, 2xLF, noise, another shaft or machine, harmonics/sidebands

Figure 18-34

That leaves the "non-synchronous" peaks; the peaks at non-integer multiples of the running speed. These peaks generally fall into one of four categories: vibration from an external machine or component; peaks at twice-line frequency; peaks generated through "intermodulation", also known as "sum-and-difference" frequencies; noise (discussed a little later); and "bearing tones" – discussed next. For now we will just say that if the peak corresponds with twice line frequency, i.e. 100 Hz or 120 Hz (6000 CPM or 7200 CPM) – or twice the frequency generated by the VFD, then you would consider a number of electric motor faults. Otherwise you should consider the possibility of a bearing defect – even if you believe you know the defect frequencies of the bearing at that point.

ANALYSIS PHASE: UNEXPLAINED PEAKS: BEARING DEFECTS

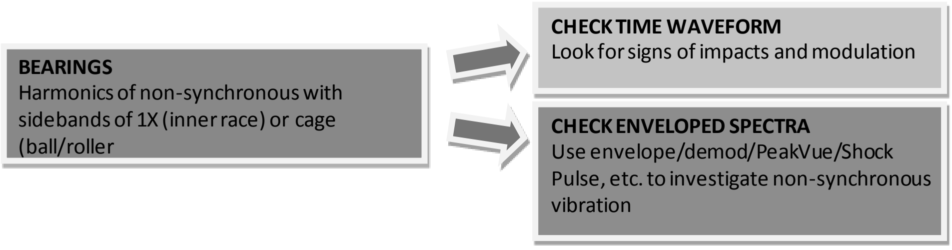


Figure 18-35

If there are unidentified peaks at non-synchronous frequencies, and you can rule out external sources of vibration then you must consider the possibility that the peaks could relate to bearing defects. This is quite a detailed analysis process, and there is much we could discuss, however if you see harmonics of these frequencies then you suspicion that a defect exists would increase significantly. If you can see 1X sidebands, then your suspicion of an inner race defect would increase. If there were sidebands of approximately 0.47X then your suspicion of a rolling element defect would increase. And if there were no sidebands, you would mostly suspect an outer race defect. Measurements such as demodulation, enveloping, PeakVue, Shock Pulse and acceleration time waveform should be consulted to confirm your suspicions.

Analysis phase: Noise

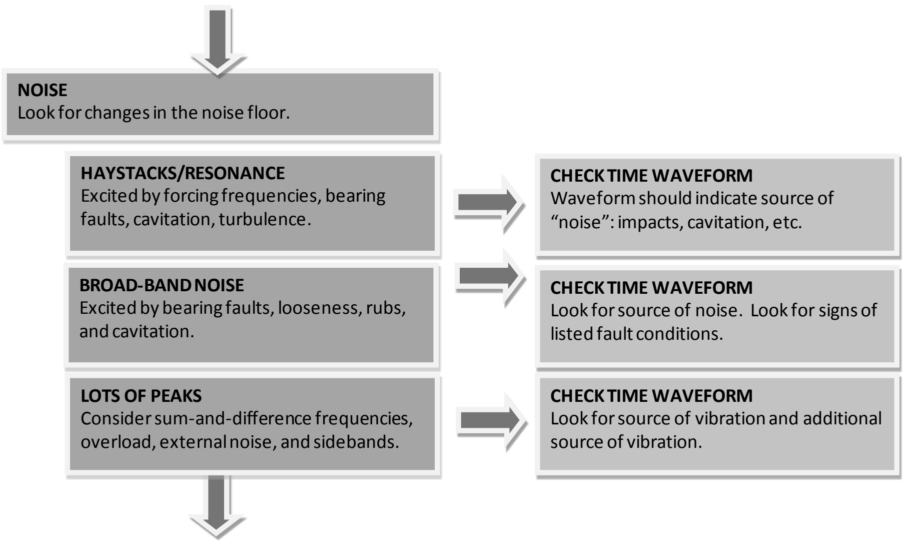


Figure 18-36

Noise in a spectrum is also an indicator of certain fault conditions. Noise could indicate a range of fault conditions, from late-stage bearing defects, to rotating looseness, to cavitation, to process noise or external machine vibration. The time waveform can help identify the nature of the vibration that is causing noise to appear in the spectrum.

Having a systematic approach will help you to detect fault conditions that you may have missed, and accurately diagnose any defects in the machine. Understanding how the spectrum (and waveform) will change as the different fault conditions develop will help immeasurably. Seeing harmonics and sidebands should trigger a series of thoughts in your mind.

The analysis phase can often identify the fault condition with sufficient accuracy and confidence that no further diagnostic work is required. However, to reduce the chance of costly error, additional diagnostic tests are recommended.

Figure 18-37

Example: An example of the Analysis Process

After validating that the data is good, the most important concern is to correctly identify running speed. Your vibration software may be designed to locate running speed either from information entered in the field or by automatically analyzing the data. Either way, make sure it has performed this important task correctly.

Recall in the "Principles of Vibration" chapter, how it was shown that many peaks which can show up in a spectrum are directly related to the running speed of the machine (or turning speed of the shaft.) The analyst is always looking at peaks as multiples of the various shaft turning speeds. Locating the running speed is vital to the successful analysis of vibration spectra.

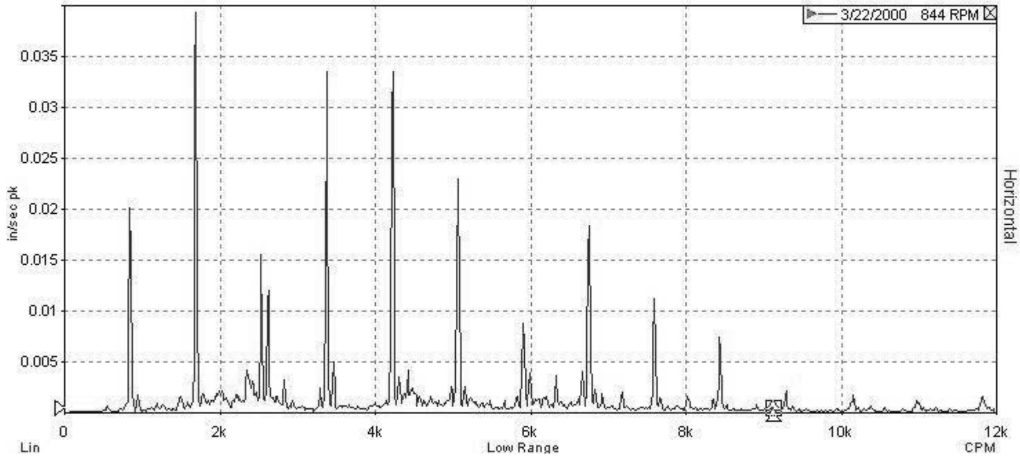


Figure 18-38 Data from a variable speed fan. Frequency is in CPM.

In many cases the 1X peak will be obvious. When it is not, it may be necessary to either look for a peak at a known multiple of running speed and work backwards (i.e. find the 6X peak and divide the frequency by 6).

One good way to relate the peaks to running speed is to view the spectra in "order normalized" format. In a normal spectrum graph the x-axis is marked off in intervals of Hertz (cycles per second), or more typically CPM (cycles per minute). See Figure 18-39.

When the spectrum is order normalized, the x-axis is marked in intervals of the running speed. In Figure 18-39, which is the same data as Figure 18-38, the peaks related to the running speed clearly stand out

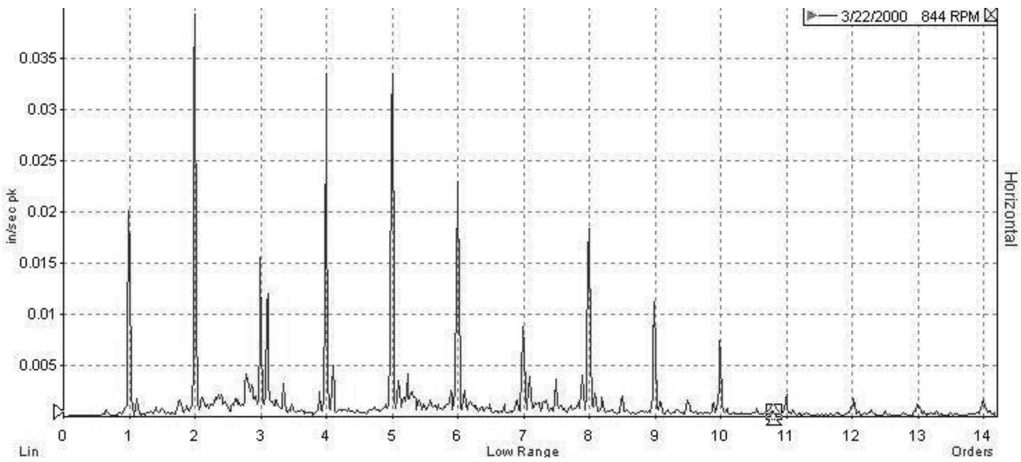


Figure 18-39 The same data, but order normalized to running speed.

Now it is easy to very quickly determined if there are any synchronous peaks (related to the running speed), or any non-synchronous peaks (non-integer multiples of the running speed). It is also easy to see if there are any peaks below the running speed (the sub-synchronous peaks).

Some machines have two or more shaft speeds to identify due to the drive method. See Figure 18-40. Belt drives and gearboxes change the speed of the next shaft and affect the spectral data. The multiple shaft running speeds may be in the spectrum.

Figure 18-40 Belt drives and gear boxes produce multiple shaft running speeds

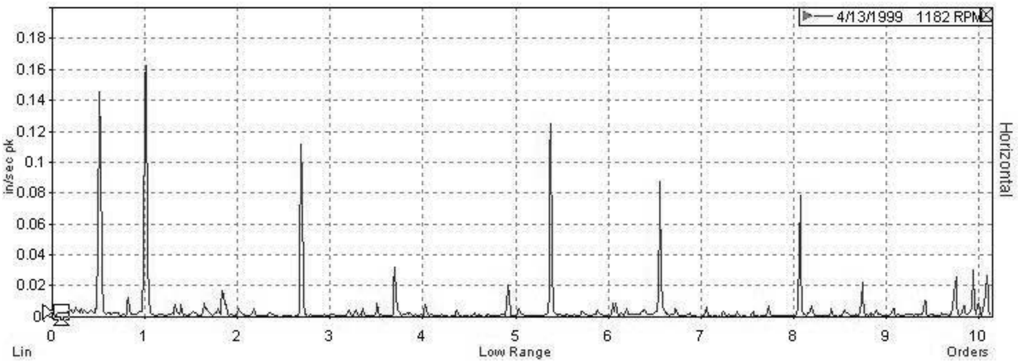


Figure 18-41 Belt drives can show running speeds from both shafts.

Figure 18-41 is a spectrum from a belt driven machine. It has two shafts that must be identified. Identify all turning speeds and order normalized to each one so the multiples of the shaft speeds are clearly evident. This helps determine whether the non-synchronous peaks are actually multiples of the other shaft turning speed.

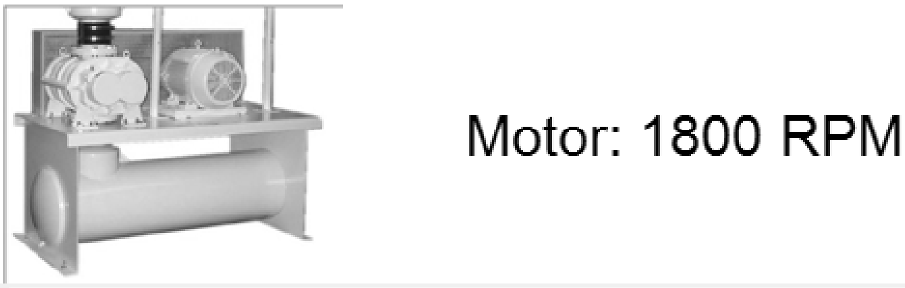
One of the analyst's goals should be to set up the analysis software to minimize the work that needs to be performed. Order normalizing the graph is certainly a step in this direction. Quickly scanning the data and immediately assessing what kind of pattern there is saves a lot of work and effort, and fault conditions are less likely to be missed.

Now we will take you through the entire process step by step.

Start with the machine

Again, vibration analysis is not about looking at data alone and trying to figure out what is wrong. It is not like reading tea leaves – rather it is about relating the data to the machine and attempting to detect problems that the machine is likely to have. In order to do this we always start with the machine. You should have a schematic of the machine in front of you that clearly shows the test points and test operating conditions. You should have a chart that shows all of the machine components and their forcing frequencies (Figure 18-42). These are the things you are going to look for in the spectra.

The first step is to identify the running speed of the machine on the graph. You should have a good idea of where to look because you should have a standard test procedure that states that the machine should only be tested at a certain speed!



	Code	Name	On Secondary	Elements	Final Ratio
▶	BR	BELT ROTATION	No	0.335	0.335
	1XM	1 X MOTOR SHAFT	No	1	1
	1XW	1 X BLOWER SHAFT	Yes	1	1.2169
	2XM	2 X MOTOR SHAFT	No	2	2
	BL	BLOWER LOBES	Yes	2	2.4338
	3XM	3 X MOTOR SHAFT	No	3	3
	2BL	2 X BLOWER LOBES	Yes	4	4.8676
	3BL	3 X BLOWER LOBES	Yes	6	7.3014
	4BL	4 X BLOWER LOBES	Yes	8	9.7352
	MB	MOTOR ROTOR BARS	No	64	64

Figure 18-42 A chart showing the machine speed and forcing frequencies

Identify the running speed

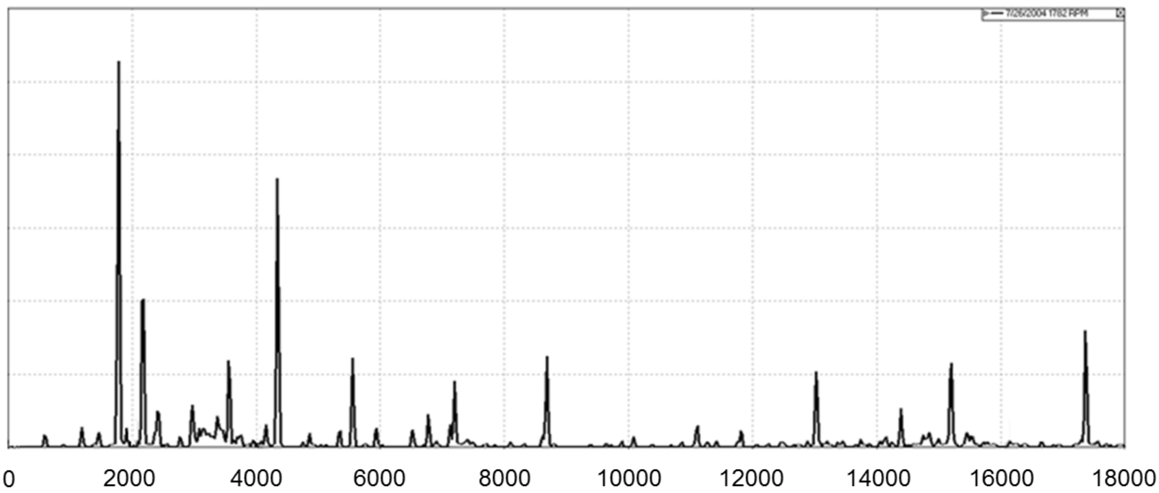


Figure 18-43 Can you find 1800 RPM on this graph?

The X-axis of Figure 18-43 is in CPM – can you find the motor running speed? It should be just less than 1800 RPM.

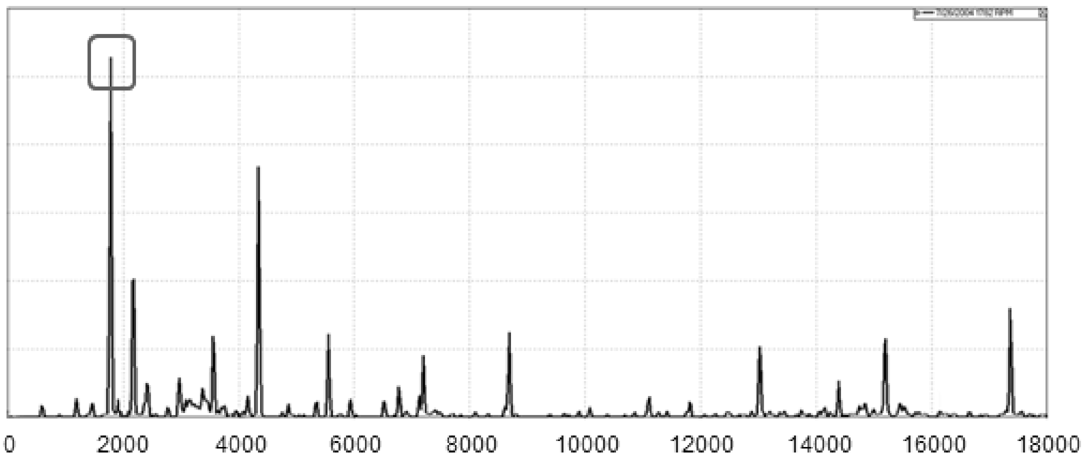


Figure 18-44 The motor running speed peak

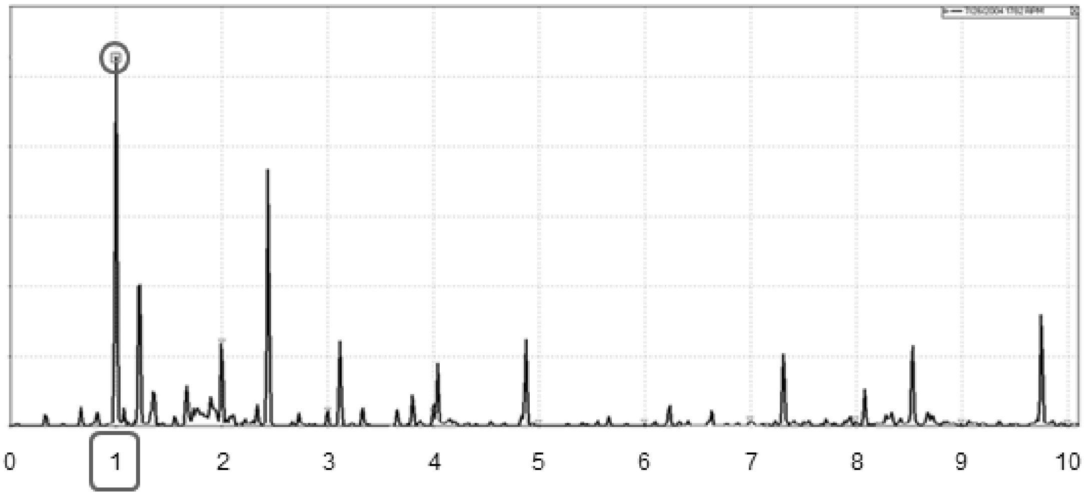


Figure 18-45 Change the X axis to "orders"

With the motor shaft rate identified, it should now be labeled “1X” or “normalized” and the X-axis should be changed to “orders.” (Figure 18-45) Now we can immediately see which peaks are and are not related to the motor shaft. The next step is to identify the peaks that are related to the motor shaft.

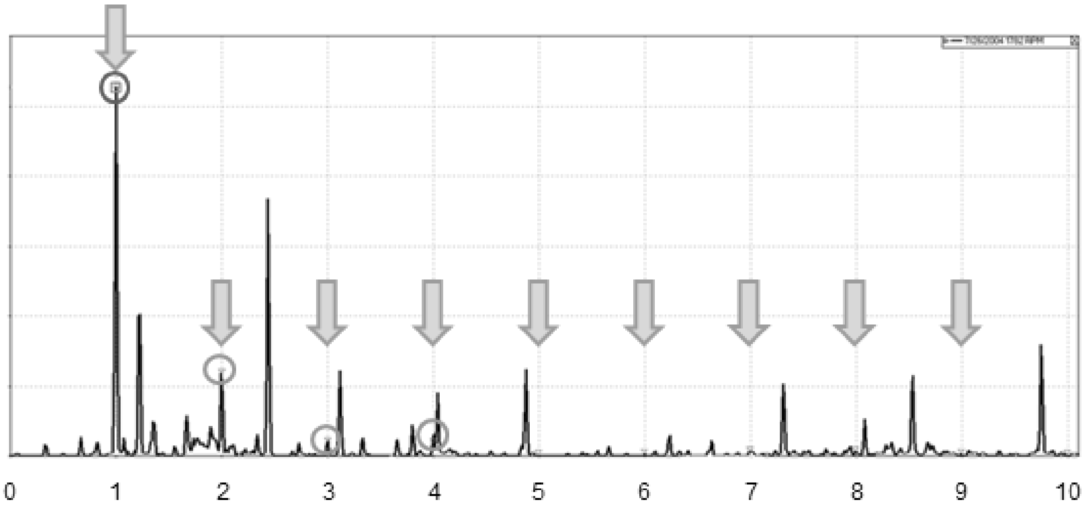


Figure 18-46 Motor shaft rate harmonics

These peaks (Figure 18-46) should now be marked in your software. The peaks beneath the green arrows are harmonics or multiples of the motor shaft rate.

Looking back at the forcing frequency table (Figure 18-42) we see that the blower is turning at 1.22X. The next step is to identify this peak in the spectrum and its harmonics. We will now use red arrows to point to the blower shaft (Figure 18-48).

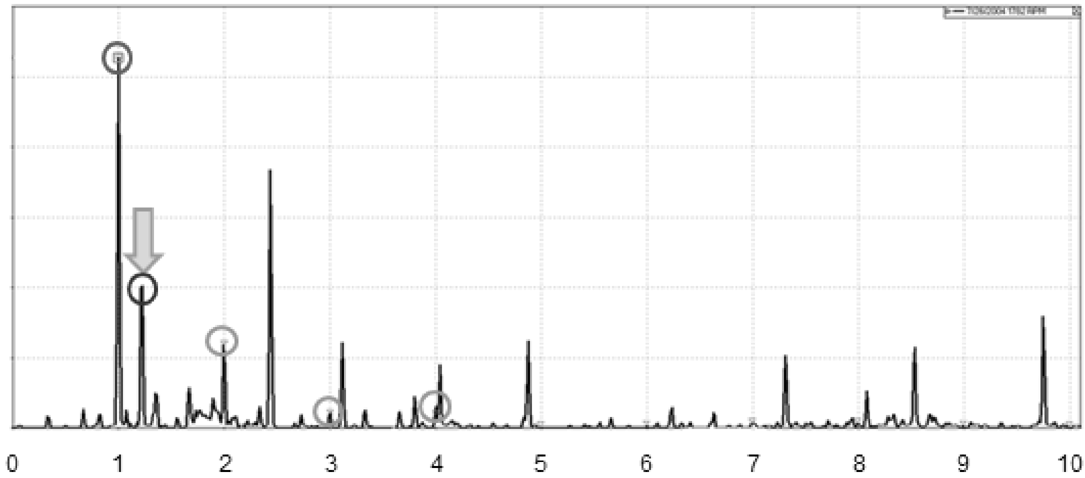


Figure 18-47 Blower shaft rate at 1.22x

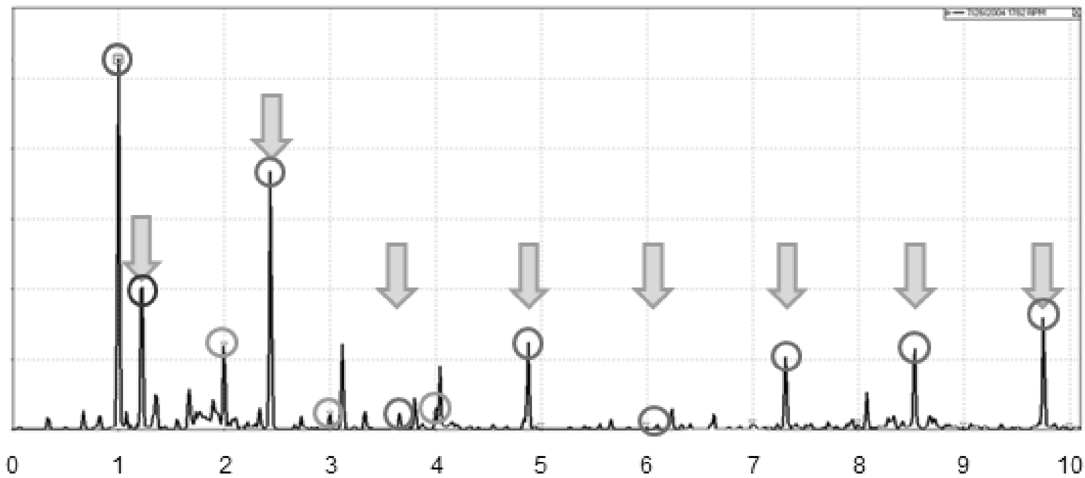


Figure 18-48 Blower shaft rate harmonics

Now we have identified the blower shaft and its harmonics, we need to return to the forcing frequency table again and see what other components exist in the machine and what their forcing frequencies are and then we need to identify them on the graph. We can see that the blower has two lobes (Figure 18-49). What is the lobe pass frequency?

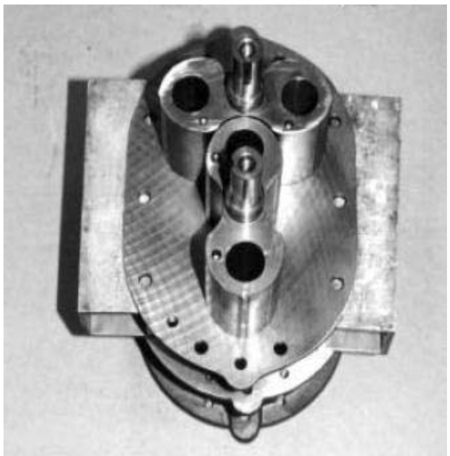


Figure 18-49 The blower has 2 lobes and a timing gear

Two lobes = 2 x the blower shaft rate or 2.44x. The blower lobes will also produce harmonics. These are indicated by the red arrows in Figure 18-50. Note again that we are marking families of peaks in different colors. If you were doing this analysis on a piece of paper, you would be doing the same exact thing with colored pencils!

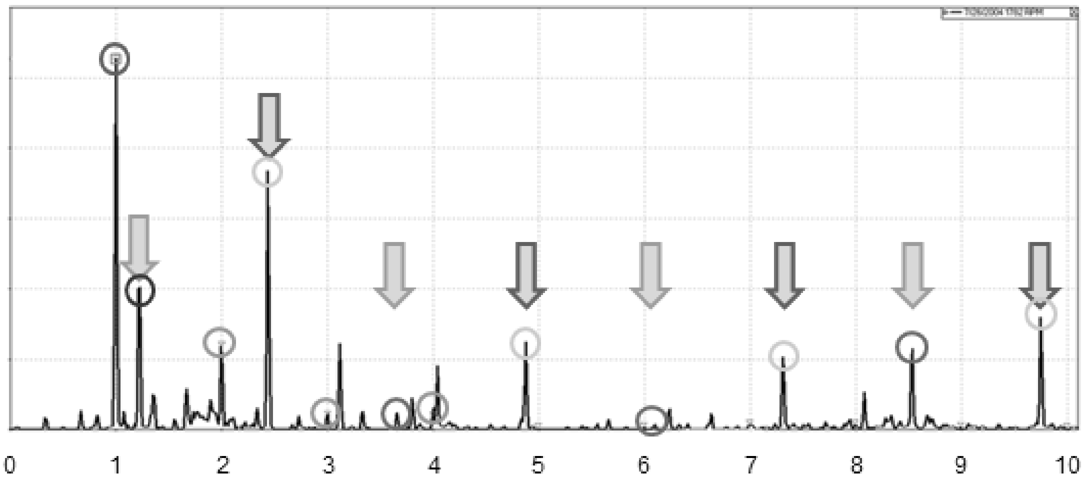


Figure 18-50 Blower lobes at 2.44x plus harmonic indicated by red arrows.

This is a belt driven machine, so the next step is to find the belt rate on the graph and see if it has any harmonics. The belt is running at 0.36x. We have marked it and its harmonics with blue arrows (Figure 18-51)

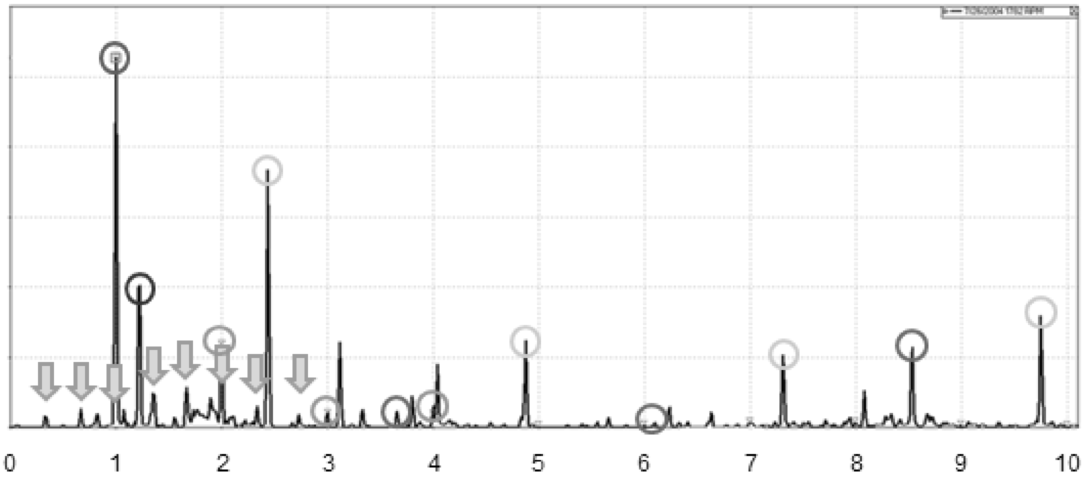


Figure 18-51 The belt rate and harmonics indicated by blue arrows

It appears that we have marked or identified most of the peaks in the spectrum in Figure 18-52. Can you find any peaks that are not marked? Are these related to each other in any way? Are they sidebands or harmonics of each other?

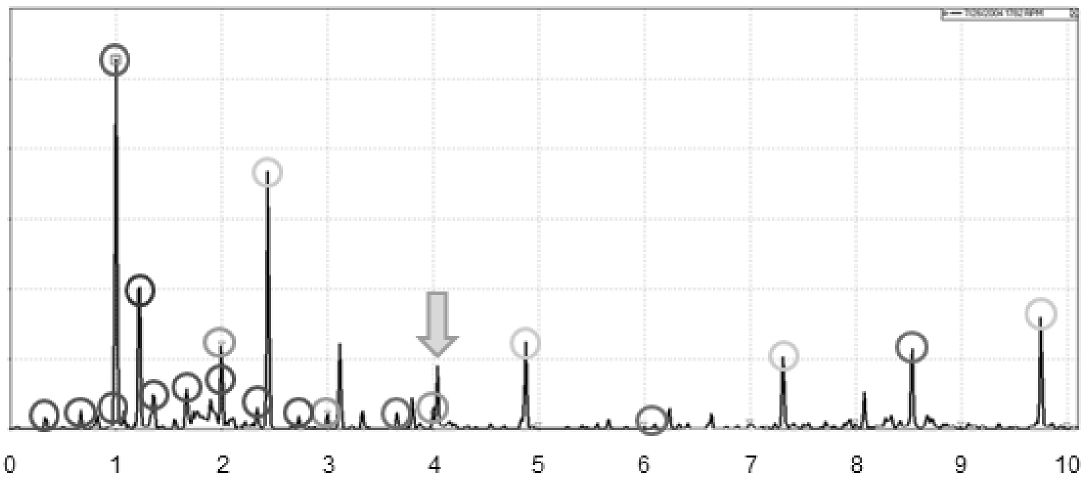


Figure 18-52 The peak just above 4x is 120 Hz - electrical.

In Figure 18-52 we have indicated a peak just above 4x that has not yet been identified. If the motor is turning just below 1800 RPM, then what is 4x in Hz? First, 1x in Hz is 1800 RPM / 60 = 30 Hz. If we multiply this by 4 we get 120 Hz. Remember that the motor is actually turning a little slower, so 120 Hz will be a bit higher than 4x. This is 2x the electrical line frequency (60 Hz) in the United States. This vibration is related to the A.C. motor and is further discussed in Category II.

Anything else?

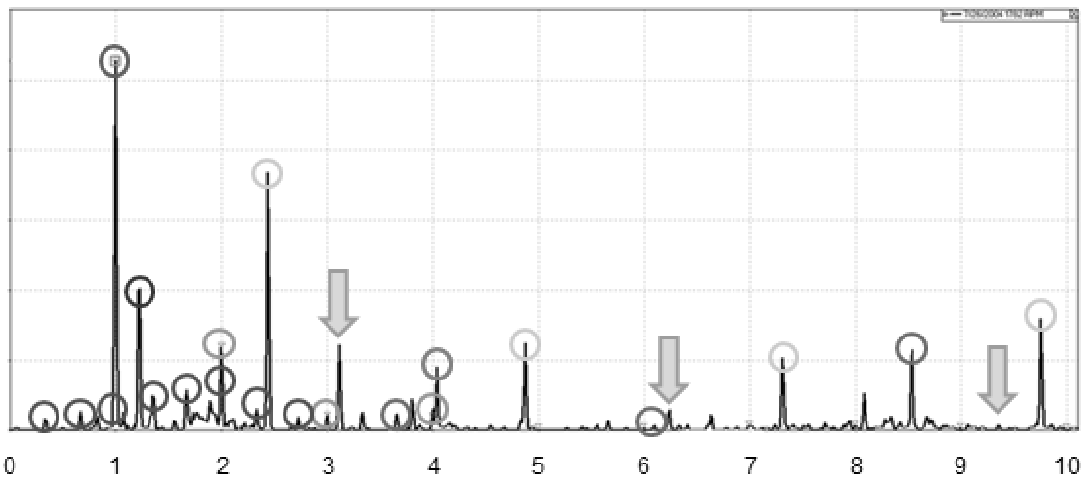


Figure 18-53 Peaks at 3.1x and 6.2x

We aren't going to get into it now, but I'll give you a hint: This vibration comes from the machine component that allows the shaft to spin. Yes, these peaks come from a rolling element bearing and they are called "bearing tones." We'll talk more about this later.

A little bit more…

- We have related peaks in the spectrum to the machine using a step by step procedure.
- We have also identified some peaks not related to either of the shafts or the belt – these may be bearing tones.
- Notice our graph only went from 0 to 10X – are there other forcing frequencies in this machine we need to find?

Remember we talked about data presentation and the idea of taking more than one measurement on a point; one with a broad view that gave us the big picture (high Fmax) and another more zoomed in focused image that allowed us to see what was going on in a smaller frequency range? Do you see how much information we found in the 0 – 10x range? If we had any less resolution than this then many of these peaks would have been mashed together and we would not have been able to identify them. The 120 Hz peak might have been joined to the 4x peak. Just look at the range from 0 – 10 in the graph in Figure 18-54 – the peaks are all on top of each other.

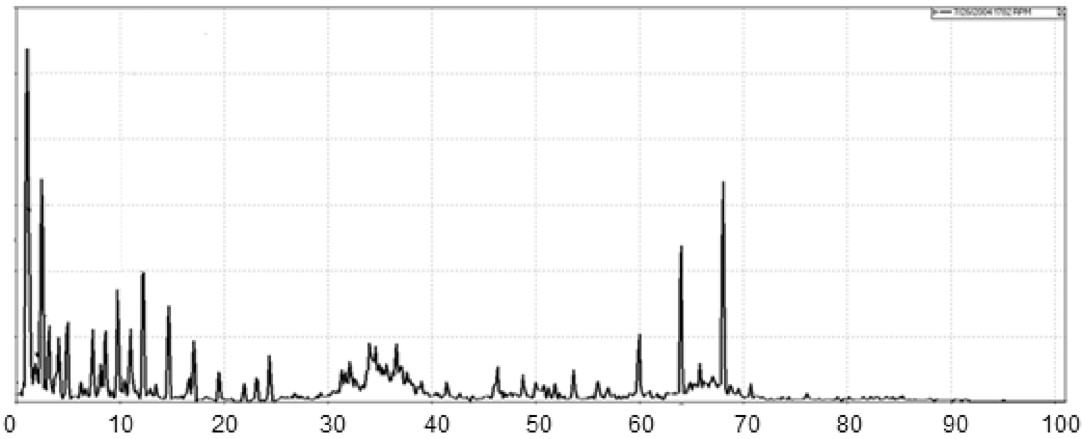


Figure 18-54 The same data but 0 - 100x

Identify unknown forcing frequencies

We know that the A.C. motor has between 30 and 80 motor bars but in real life we rarely know the exact number. It is not something that is typically written on the name plate of the machine. That said, this machine component produces vibration (i.e. it is a forcing frequency) and not only that, it creates a specific pattern that is easy to identify. Therefore we should look in the vibration spectrum taken from one of the motor bearings, and look for this pattern. The pattern is the number of motor bars x the shaft rate (between 30 and 80 bars) and on either side of it will be sidebands separated by two times the electrical line frequency or 120 Hz in this case (Figure 18-55).

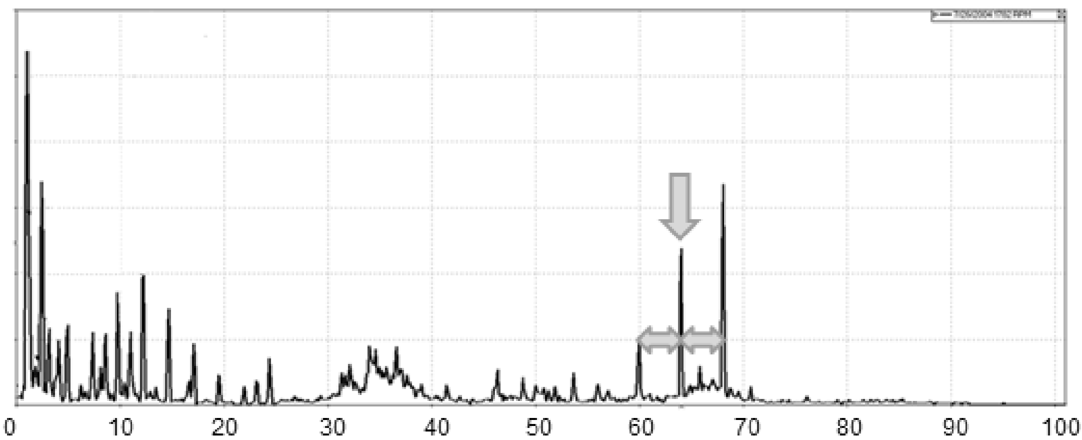


Figure 18-55 Motor bar peak with 2x LF sidebands (120 Hz)

This pattern is easier to identify when using a software program where you can place cursors on the peaks and measure the distance between them.

Once having identified the number of motor bars, this information should be added to the forcing frequency table and to your analysis software so that we do not have to look for it again next time we collect data.

Note one more item. The blower had a timing gear. We do not know how many teeth it has and at the moment, it is not creating an obvious peak in the spectrum. The thing to do would be to look in the spare parts closet and see if we can find a timing gear and count the teeth. Another option would be to contact the manufacturer or look in the spectral data from other identical blowers if there are any. In any case, we will need to have this information in order to detect problems with this component, so eventually we must remember to look for this information one way or another.

In Figure 18-56 you will find one more example of how to find unknown forcing frequencies. This is data from a directly driven centrifugal pump. We do not know how many vanes the pump has. Can you make an educated guess?

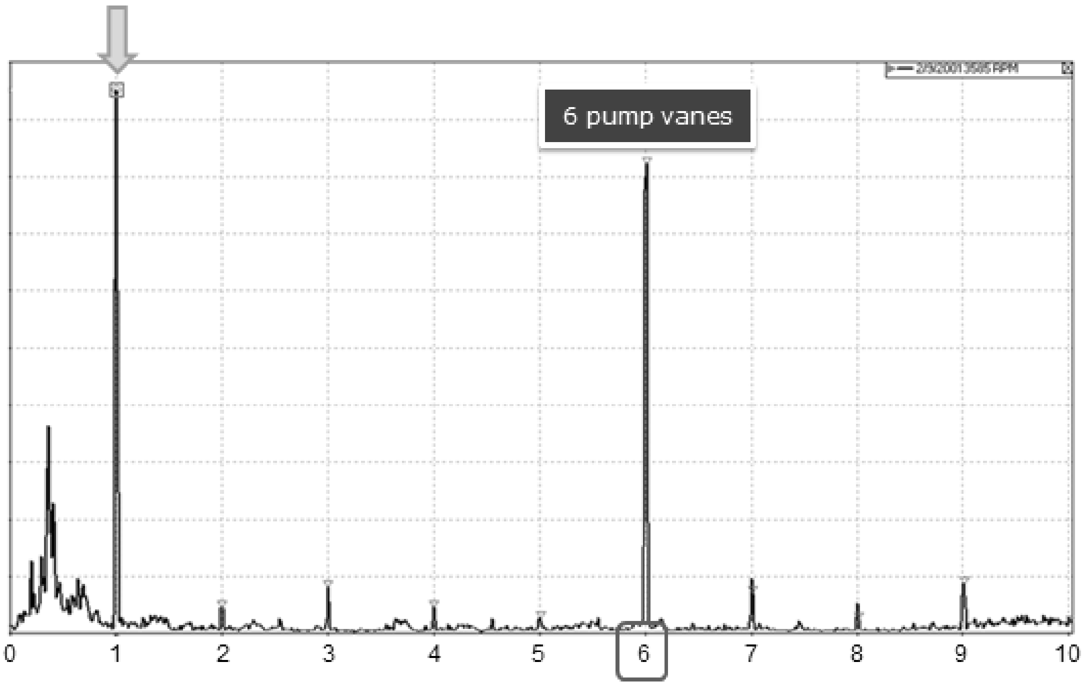


Figure 18-56 How many vanes does this pump have?

It has 6 vanes, this is why 6x (6 vanes x 1x) is so much higher than the other 1x harmonics. It is higher because it is not just a harmonic; it is also a forcing frequency.

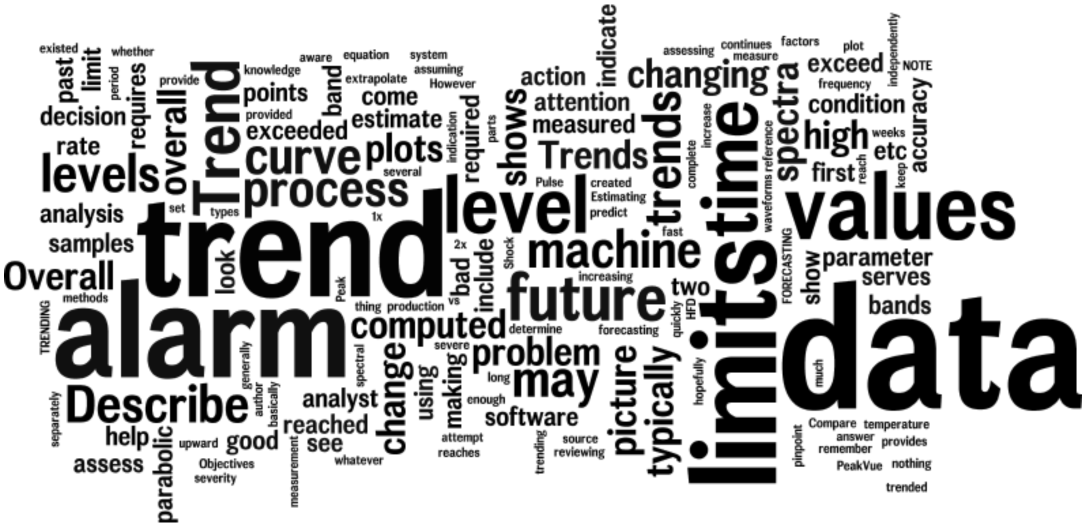
Looking for machine faults

Going back to the blower example, we have completed the process of preparing the data for analysis. If we are running a condition monitoring program then all of the things we identified on the graphs should be entered into your analysis software. The next time you collect data all you should have to do is identify 1x – tell the software which peak is 1x – and all of the other peaks should automatically be defined by the software.

Your software should then also allow you to overlay the data with either a set of alarm criteria or with other data from the same machine. In other words, we identified all of the forcing frequencies but we still have no idea whether the levels are acceptable or not. As we mentioned earlier, the best way to know this is to compare the data to historical data from the same machine tested under exactly the same test conditions (speed, load, data collector setup, sensor, test point etc) Then we can create meaningful trends and easily identify changes in condition.

Here is a summary of what we did, and the steps that should follow:

- We now have the data properly validated, normalized and formatted with the forcing frequencies identified.
    - We may have also noticed peaks that are not identified
- We will hopefully have a baseline, alarm or other criteria to compare the data to.
- Now, we go back to the machine and ask "what are the most common faults this machine can have?
- What do these faults look like (in the data)?
    - In which measurement points / axes do they appear?
- Does this machine have this fault? (yes or no)
- Does this machine have the next fault? (yes / no)



Chapter 19

Setting Alarm Limits

Objectives:

- Describe the ISO rms alarms
- List 3 benefits of band alarms
- Compare and contrast band alarms and mask or envelope alarms
- Describe a benefit of using statistical alarms

Large amounts of data are collected during the day to day process of running a predictive maintenance program. In order for the program to be successful, this data must not be ignored. It must be analyzed so that decisions can be made regarding repairs. Parts must be ordered, downtime scheduled, and priorities shifted.

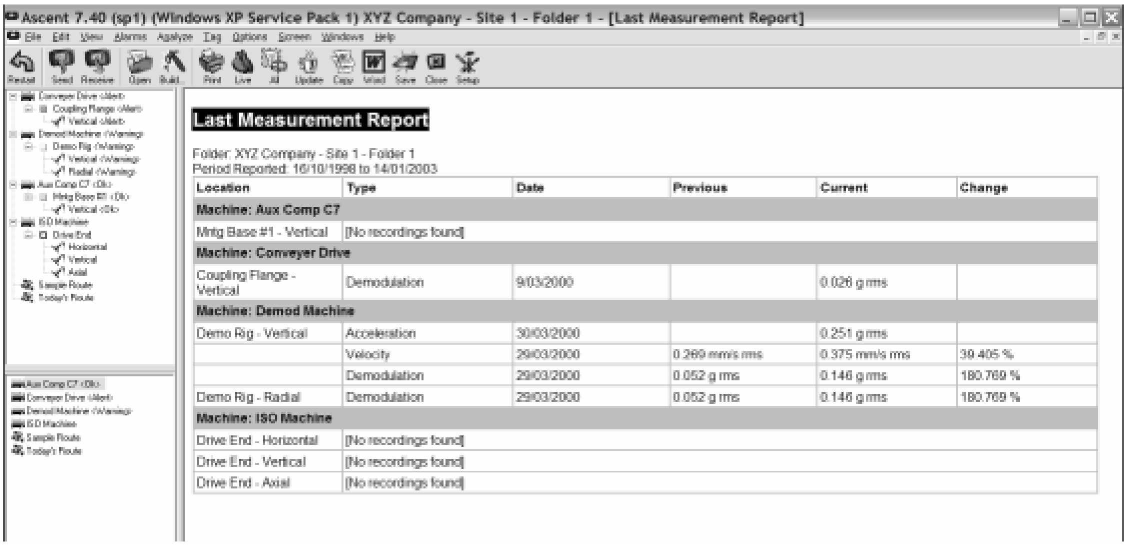
But should we do with all the data? How will we know if there is a problem?

Some analysts choose to look at every piece of data, both the spectra and waveform. This is very costly in time and energy. Often the sheer volume of data prevents a good analysis in a timely manner.

One approach is to view the trend data, either trends of overall levels, specific bands, or of specific bearing measurements. This process combines the detection and analysis phase on a machine by machine basis.

Smart analysts invest the time to set up good alarm levels for the machines so that the data can be scanned in the computer for measurement points that are not within acceptable limits. Alarm limits can be applied to RMS readings, Shock Pulse/HFD/Spike Energy readings, spectra, and even time waveforms. This way the analyst only studies the data for machines listed in the exception report.

Good alarm limits can save a huge amount of time.


Ascent 7.40 (sp1) (Windows XP Service Pack 1) XYZ Company - Site 1 - Folder 1 - [Last Measurement Report]

Last Measurement Report

Folder: XYZ Company - Site 1 - Folder 1
Period Reported: 16/10/1998 to 14/01/2003

Location	Type	Date	Previous	Current	Change
Machine: Aux Comp C7					
Mntg Base #1 - Vertical	[No recordings found]				
Machine: Conveyer Drive					
Coupling Flange - Vertical	Demodulation	9/03/2000		0.026 g rms	
Machine: Demod Machine					
Demo Rig - Vertical	Acceleration	30/03/2000		0.251 g rms	
	Velocity	29/03/2000	0.269 mm/s rms	0.375 mm/s rms	39.405 %
	Demodulation	29/03/2000	0.052 g rms	0.146 g rms	180.769 %
Demo Rig - Radial	Demodulation	29/03/2000	0.052 g rms	0.146 g rms	180.769 %
Machine: ISO Machine					
Drive End - Horizontal	[No recordings found]				
Drive End - Vertical	[No recordings found]				
Drive End - Axial	[No recordings found]				

Figure 19-1 - Exception Report list the measurements that are outside the preset limits.

At the start of most programs, there are two options that typically occur. Either a large percentage of the machines are on the Exception Report or very few, if any, are on the report. As programs mature, machine problems corrected, and alarm levels tweaked, the report is much more accurate and reflective of the true condition of the machines. Getting to this point can sometimes be difficult.

For many years the major software packages have included an "**exception report**". The software scans through all of the new vibration measurements, compare them to alarm limits, optionally compare them to previous readings, and then generate a list of machines with the results.

The results of the report typically indicate which machines "failed", and give an indication of the severity. Machines with data that is significantly above alarm are treated differently to machines with only slight exceptions.

The results indicate the current level of the data, as well as the level of exceedance. The exceedance may be reported as a percentage (100% indicates the new level is twice the alarm limit) or less frequently in decibel (6 dB is the same as 100%).

The results typically also indicate how the latest readings have changed compared to the previous readings, and/or to a reference or baseline reading. Again, the results are presented as a percentage increase or a dB ratio.

Let's explore this issue of the "reference" reading a little more carefully.

The reference level is dictated by so many factors. The size of the machine, the load the machine is under, the importance of the machine, and the history of the machine are all important factors.

For example, a precision machine tool should not vibrate to the same level as a ball mill.

A machine that has been running at 0.1 ips for months is less of a concern than a machine that has changed from 0.005 ips to 0.1 ips.

When we have a new reading, whether it is a simple overall level reading or a complete spectrum, one of the biggest challenges is to know what the levels should be. How high is too high? At what point should monitoring be increased? At what point should a repair be performed?

There really needs to be a way to say that up to a certain vibration level the machine is OK, and above that level the machine needs to be repaired. Unfortunately, it is not that easy.

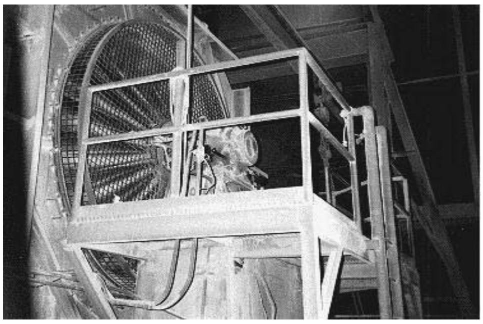


Figure 19-2

So, where does a person start? And what levels are too high? What is acceptable? How much time should be spent on critical machines versus smaller, less critical machines? Fortunately there are guidelines. Unfortunately they are just that, guidelines. However, the guidelines provide a very good starting place. Most people want a "formula" or "magic number" to compare their vibration to. It's just not that simple.

There are many published charts for acceptable vibration levels. These may work as a good starting place. However, be aware that each plant and process is different and the same machine in different plants may not produce similar characteristics. There are many variables.

Still, a good starting place is with the published alarm limits.

ISO 10816 RMS Alarm Limits

The ISO 10816 Standard has 5 parts. Part 1 is for General Machines. Part 3 is for Industrial Machines. These 2 Parts will be the dominant range for most plants. The 5 parts are shown below.

Part 1: General guidelines

Part 2: Large land-based steam turbine generator sets in excess of 50 MW

Part 3: Industrial machines with nominal power above 75 kW and nominal speeds between 120 r/min and 15000 r/min when measured in place.

Part 4: Gas turbine driven sets excluding aircraft derivatives

Part 5: Machine sets in hydraulic power generating and pumping plants

ISO 10816.1 gives the limits for General Machines. ISO 10816 Severity Chart is shown in Figure 19-3.

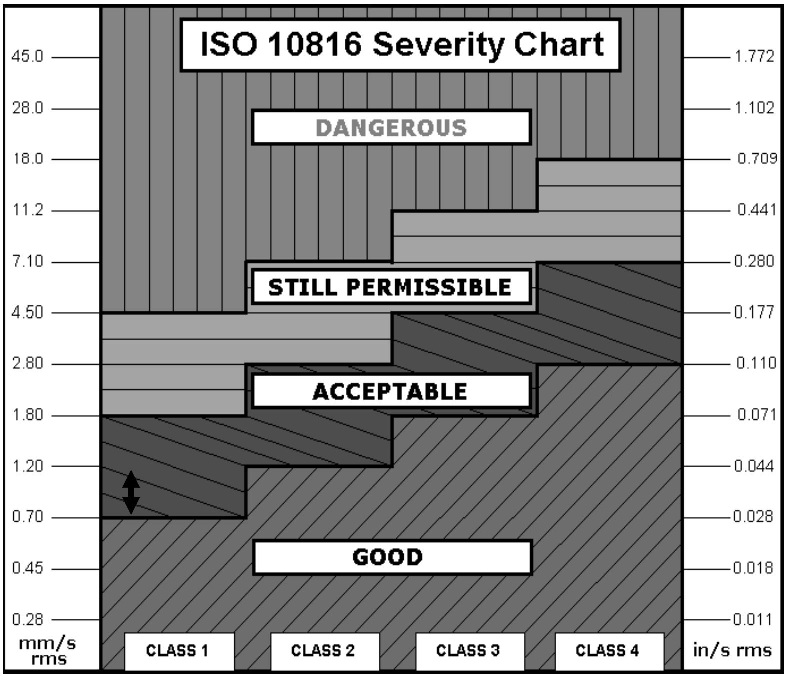


Figure 19-3 - 10816 Severity Chart for General Machines

This chart has the limit values for 4 classes of machinery.

From one limit value to the next above vibration severity increases 1.6 times (1 step). From one condition zone to the next above vibration severity increases by a factor of 2.5 (2 steps). 3 steps up is a fourfold increase.

Notice the values are in both metric and imperial velocity units.

Class I: Individual parts of engines and machines, integrally connected to the complete machine in its normal operating condition. (Production electrical motors of up to 15 kW are typical examples of machines in this category.)

Class II: Medium-sized machines (typically electrical motors with 15 kW to 75 kW output) without special foundations, rigidly mounted engines or machines (up to 300 kW) on special foundations.

Class III: Large prime-movers and other large machines with rotating masses mounted on rigid and heavy foundations which are relatively stiff in the direction of vibration measurements.

Class IV: Large prime-movers and other large machines with rotating masses mounted on foundations which are relatively soft in the direction of vibration measurements (for example, turbo-generator sets and gas turbines with outputs greater than 10 MW).

Figure 19-4 is the Vibration severity chart for ISO 10816.3, the standard for industrial machines. Notice they are categorized by size and whether the driver is integral to the pump. The limits are in both metric and imperial velocity units.

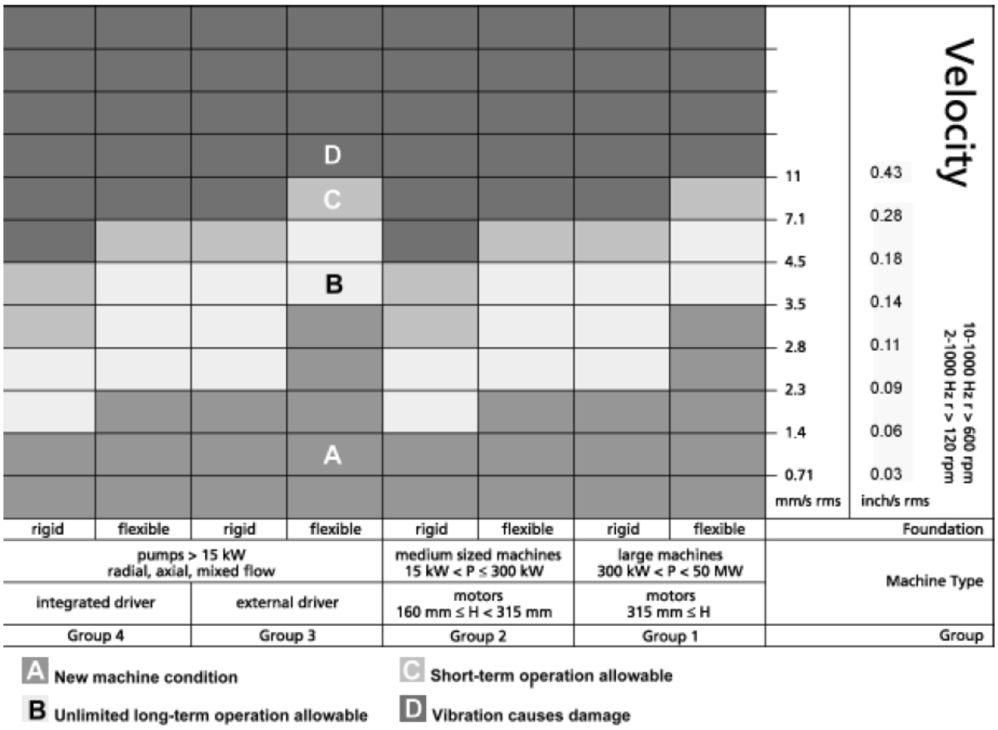


Figure 19-4 - The Velocity Severity Chart for ISO 10816.3 Industrial Machines

Figure 19-5 is the same chart in Displacement units.

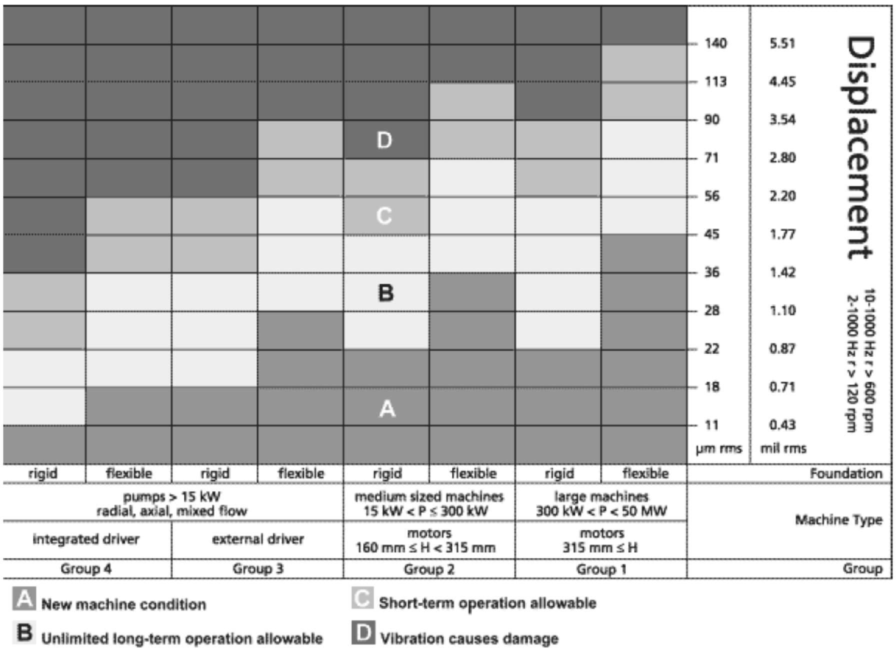


Figure 19-5 - The 10816.3 Severity Chart in Displacement units.

Note that these are single number, overall values in the frequency range of 3 Hz to 1000 Hz. They are a good number to get started with and to compare to.

Recall from the "Principles of Vibration" chapter that for a pure sine wave, the rms value is 0.707 times the Peak value.

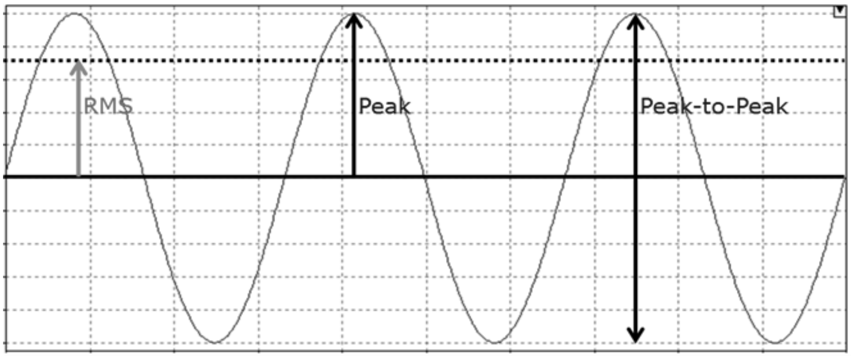


Figure 19-6

There are two ways the RMS is derived based on whether it is computed digitally or is from an analog signal.

Remember that this is the simple equation for a pure sine wave. Real data is not sinusoidal.

TRUE RMS FROM AN ANALOG SIGNAL.

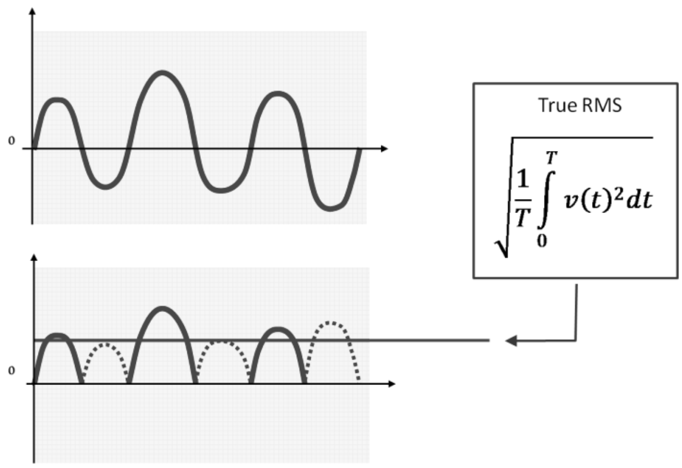


Figure 19-7 - The True RMS calculated from an Analog signal. It has both positive and negative values.

Recall that RMS is calculated as the reverse of its name. First the values under the curve are squared so that they are all positive. Then the average value is calculated of the values under the curve. And finally the square root is computed of that averaged value. Figure 19-7 shows the process for the Analog signal.

The process is not quite the same for a digital signal.

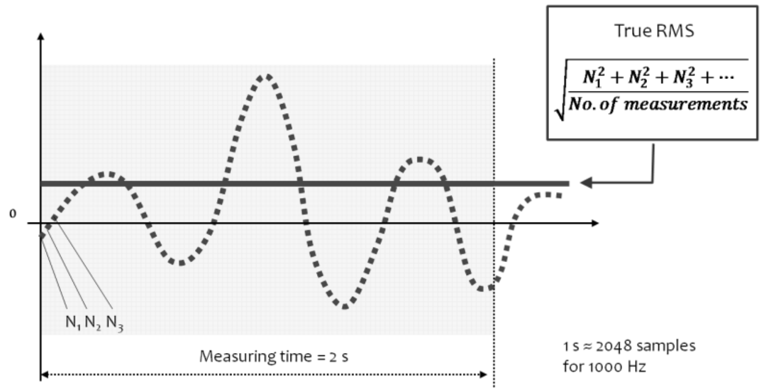


Figure 19-8 - True RMS computed from a digital signal

The signal must have a large number of samples over a length of time. Each sample "n" is squared. The sum of the squares is divided by the number of samples. The square root of the quotient is the true RMS value.

VIBRATION SEVERITY AND ISO 10816.

The ISO 10816 defines vibration severity as the RMS level of vibration velocity measured over a frequency range of 3 Hz to 1000 Hz.

Instead of measuring the amplitude of a transient at a single high frequency, the vibration severity reading represents an average of all vibration components within a wide and comparatively low frequency range.

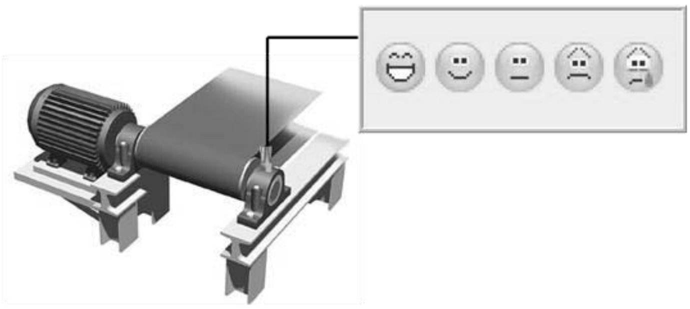


Figure 19-9

Vibration severity is directly related to the energy level of machine vibration, and thus a good indicator of the destructive forces acting on the machine.

According to ISO 10816 vibration severity is defined as the RMS level of vibration velocity, measured over a frequency range of 3 to 1000 Hz

Is 1000 Hz enough?

Figure 19-10 is data from a pump with a bad bearing. The frequency range is the ISO specified range of 3 Hz to 1000 Hz. Does this frequency range include all the data needed to accurately diagnose the problems with the pump?

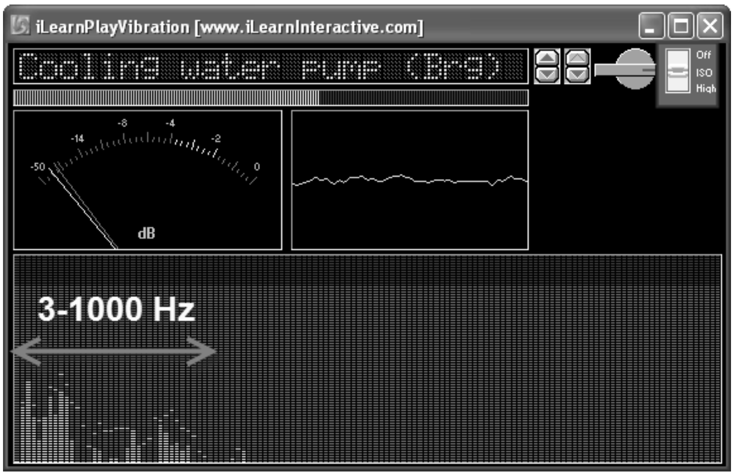


Figure 19-10 - Data from a pump with a bad bearing. Using ISO specified range.

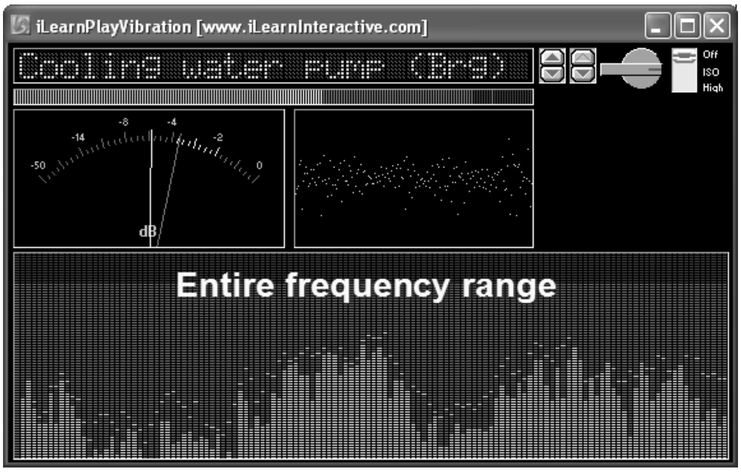


Figure 19-11 - Data from the same pump but out to a much higher frequency range.

Figure 19-11 shows data from the same pump but out to a much higher frequency range. It has all the information in it to make good diagnosis of the pump condition. Notice how much information is ignored if the Fmax is set to ISO specifications.

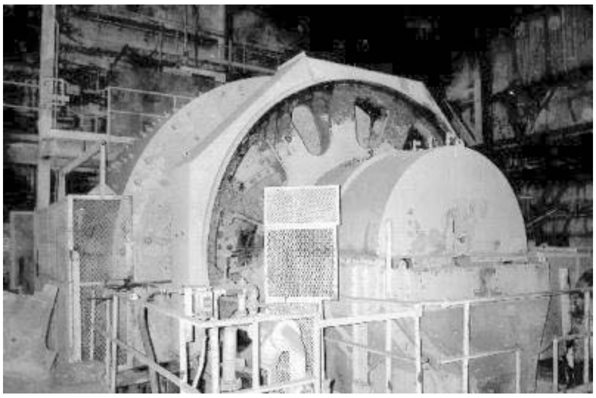


Figure 19-12

The question that everyone is asking is, "How high is too high?"

The correct answer is…"it depends." That is not an answer anyone likes to hear, but the fact of the matter is it really does depend on several factors.

The normal vibration level for a machine is dependent upon:

- The function of the machine and the forces involved
- The rigidity of the machine structure

A large diesel engine vibrates more than a small electric motor, because different forces are involved. A machine on a stiff concrete foundation vibrates less than the same machine bolted to a flexible metal frame, because its overall structure is more rigid.

Excessive machine vibration on new machines is a sign of inherent structural weakness or bad resonance characteristics. An increase in the vibration level from good condition has basically three causes: something is loose, misaligned, or out-of-balance.

ISO 7919

The ISO standard 7919 provides guidelines for measurements and evaluation criteria for a variety of machine types:

Part 1: General guidelines

Part 2: Land-based steam turbines and generators in excess of 50 MV with normal operating speeds of 1500 r/min, 1800 r/min, 3000 r/min and 3600 r/min

Part 3: Coupled industrial machines

Part 4: Gas turbine driven sets

Part 5: Machine sets in hydraulic power generating and pumping plants

The evaluation zones allow for assessment and possible action on machines based on the measured vibration.

Zone A: The vibration of newly commissioned machines normally falls within this zone.

Zone B: Machines with vibration within this zone are normally considered acceptable for unrestricted long-term operation.

Zone C: Machines with vibration within this zone are normally considered unsatisfactory for long-term continuous operation. Generally, the machine may be operated for a limited period in this condition until a suitable opportunity arises for remedial action.

Zone D: Vibration values within this zone are normally considered to be of sufficient severity to cause damage to the machine.

ISO 14694: 2003

This International Standard gives specifications for vibration and balance limits of all fans for all applications except those designed solely for air circulation. ISO 14694 describes the measurement locations, vibration amplitudes, and satisfactory/ alarm/shutdown limits.

Table 1 — Fan-application categories

Application	Examples	Limits of driver power kW	Fan-application category, BV
Residential	Ceiling fans, attic fans, window AC	⩽ 0,15	BV-1
		> 0,15	BV-2
HVAC and agricultural	Building ventilation and air conditioning; commercial systems	⩽ 3,7	BV-2
		> 3,7	BV-3
Industrial process and power generation, etc.	Baghouse, scrubber, mine, conveying, boilers, combustion air, pollution control, wind tunnels	⩽ 300	BV-3
		> 300	See ISO 10816-3
Transportation and marine	Locomotive, trucks, automobiles	⩽ 15	BV-3
		> 15	BV-4
Transit/tunnel	Subway emergency ventilation, tunnel fans, garage ventilation, Tunnel Jet Fans	⩽ 75	BV-3
		> 75	BV-4
		none	BV-4
Petrochemical process	Hazardous gases, process fans	⩽ 37	BV-3
		> 37	BV-4
Computer-chip manufacture	Clean rooms	none	BV-5

NOTE 1 This standard is limited to fans below approximately 300 kW. For fans above this power refer to ISO 10816-3. However, a commercially available standard electric motor may be rated at up to 355 kW (following an R20 series as specified in ISO 10816-1). Such fans will be accepted in accordance with this International Standard.

NOTE 2 This Table does not apply to the large diameter (typically 2 800 mm to 12 500 mm diameter) lightweight low-speed axial flow fans used in air-cooled heat exchangers, cooling towers, etc. The balance quality requirements for these fans shall be G 16 and the fan-application category shall be BV-3.

Figure 19-13

Table 4 — Vibration-levels limit for test in manufacturer's work-shop

Fan application category	Rigidly mounted mm/s		Flexibly mounted mm/s	
	Peak	r.m.s.	Peak	r.m.s.
BV-1	12,7	9,0	15,2	11,2
BV-2	5,1	3,5	7,6	5,6
BV-3	3,8	2,8	5,1	3,5
BV-4	2,5	1,8	3,8	2,8
BV-5	2,0	1,4	2,5	1,8

NOTE 1 Refer to Annex A for conversion of velocity units to displacement or acceleration units for filter-in readings.

NOTE 2 The r.m.s. values given in this Table are preferred. They are rounded to a R20 series as specified in ISO 10816-1. Peak values are widely used in North America. Being made up of a number of sinusoidal wave forms, these do not necessarily have an exact mathematical relationship with the r.m.s. values. They may also depend to some extent on the instrument used.

NOTE 3 The values in this Table refer to the design duty of the fan and its design rotational speed and with any inlet guide vanes "full-open". Values at partial load conditions should be agreed between the manufacturer and user, but should not exceed 1,6 times the values given.

Figure 19-14

Table 5 — Seismic vibration limits for tests conducted *in situ*

Condition	Fan-application category	Rigidly mounted mm/s		Flexibly mounted mm/s	
		Peak	r.m.s.	Peak	r.m.s.
Start-up	BV-1	14,0	10	15,2	11,2
	BV-2	7,6	5,6	12,7	9,0
	BV-3	6,4	4,5	8,8	6,3
	BV-4	4,1	2,8	6,4	4,5
	BV-5	2,5	1,8	4,1	2,8
Alarm	BV-1	15,2	10,6	19,1	14,0
	BV-2	12,7	9,0	19,1	14,0
	BV-3	10,2	7,1	16,5	11,8
	BV-4	6,4	4,5	10,2	7,1
	BV-5	5,7	4,0	7,6	5,6
Shutdown	BV-1	Note 1	Note 1	Note 1	Note 1
	BV-2	Note 1	Note 1	Note 1	Note 1
	BV-3	12,7	9,0	17,8	12,5
	BV-4	10,2	7,1	15,2	11,2
	BV-5	7,6	5,6	10,2	7,1

NOTE 1 Shutdown levels for fans in fan-application grades BV-1 and BV-2 should be established based on historical data.

NOTE 2 The r.m.s. values given in this Table are preferred. They are rounded to a R20 series as specified in ISO 10816-1. Peak values are widely used in North America. Being made up of a number of sinusoidal wave forms, these do not necessarily have an exact mathematical relationship with the r.m.s. values. They may also depend to some extent on the instrument used.

Figure 19-15

Spectrum Alarm Limits

The Overall value is a good number to trend. But it does not catch everything. It does not let us know about small amplitude values that may indicate severe or critical conditions. A better method is to have the system scan particular regions of the spectrum, and compare against a level for that region.

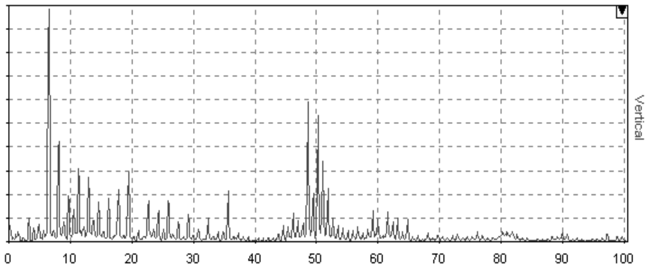


Figure 19-16

Several questions need to be answered.

- Which method should be used… Band, envelope, or statistical?
- What should the limit be… an absolute value or a relative value?

That may depend upon the capabilities of your software, or what you are comfortable with.

But a reference is still needed to compare against - particularly when starting out. There are basically two ways to set a reference alarm level. The first is to utilize published alarm limits and set fixed alarm limits, and the second is to start with existing vibration readings and perform a calculation to derive the alarm limit. There are pros and cons to both approaches.

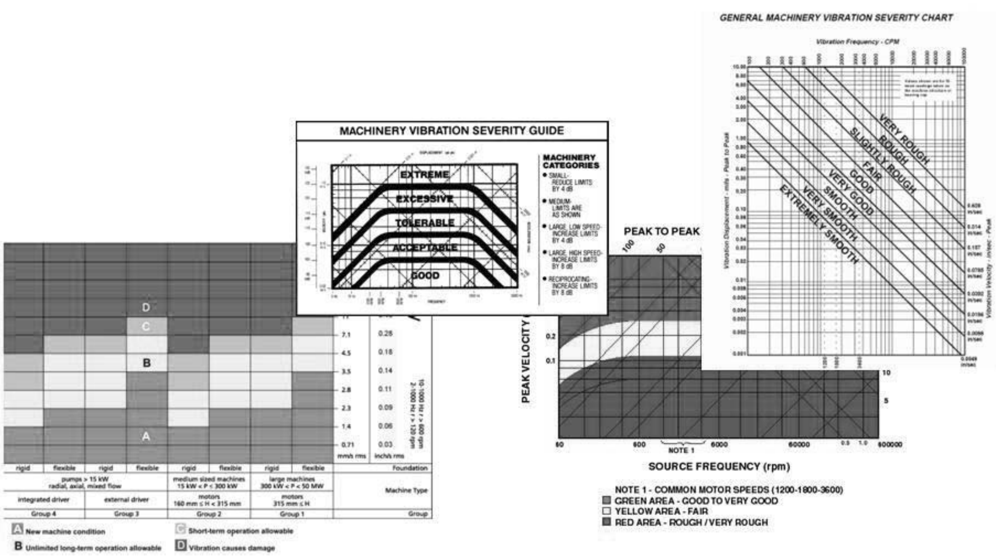


Figure 19-17 - Various charts for setting Alarm Limits.

Many attempts have been made by various groups to characterize all machines and come up with a set of vibration levels that define how a machine should vibrate.

Attempts have been made by software vendors, the military, training companies and standards bodies alike.

The result is a table of limits based on machine type (defined by load, speed or function). A few of those tables are presented here for your reference.

Rathbone Chart	
Machine Running Condition	**Overall Vibration Velocity**
Very Rough	0.628 in/sec peak
Rough	0.314 in/sec peak
Slightly Rough	0.157 in/sec peak
Fair	0.0785 in/sec peak
Good	0.0392 in/sec peak
Very Good	0.0196 in/sec peak
Smooth	0.0098 in/sec peak
Very smooth	0.0049 in/sec peak

Table 19-1 - Rathbone Chart

The Rathbone Chart, introduced in 1939 by a UL agent named T. C. Rathbone, was the first official attempt to characterize the behavior of mechanical devices under vibrations. It compares overall vibration velocity (measured in Inch Per Second - IPS) to varying degrees of machine smoothness. It is now considered obsolete because it fails to relate the vibration level with the frequency spectrum or to the machine size.

The ISO standard number 2372/10816 provides vibration amplitude acceptance guidelines for machinery with operating speeds from 10 to 200 revolutions per second (600 to 12000 RPM). The ISO standard classifies a medium size machine (15-75kW) as being "good" if the overall rms vibration level is between 0.18 to 1.12 mm/second (.007 to 0.042 Inch/Sec or 85 to 100 VdB). Larger machines may run rougher and smaller ones smoother.

in/sec pk	mm/sec rms	Level in VdB	Less than 15kW (<20HP)	15 – 75kW 20 to 100HP	>75kW (100HP)
1.0	18.0	125	Not permissible	Not permissible	Not permissible
0.63	11.2	121	Not permissible	Not permissible	Just tolerable
0.4	7.10	117	Not permissible	Just tolerable	Just tolerable
0.25	4.50	113	Just tolerable	Just tolerable	Allowable
0.16	2.80	109	Just tolerable	Allowable	Allowable
0.1	1.80	105	Allowable	Allowable	Allowable
0.06	1.20	101	Allowable	Good	Good
0.04	0.70	97	Good	Good	Good

Table 19-2 - ISO 2372/10816.1 General Machinery Standard

As stated earlier, ISO limits are defined for overall limits. Because 1X vibration normally dominates, it can be used for 1X limits. But limits are needed for the bearing frequencies, etc. Refer to Figure 19-4 for a full size view of this chart.

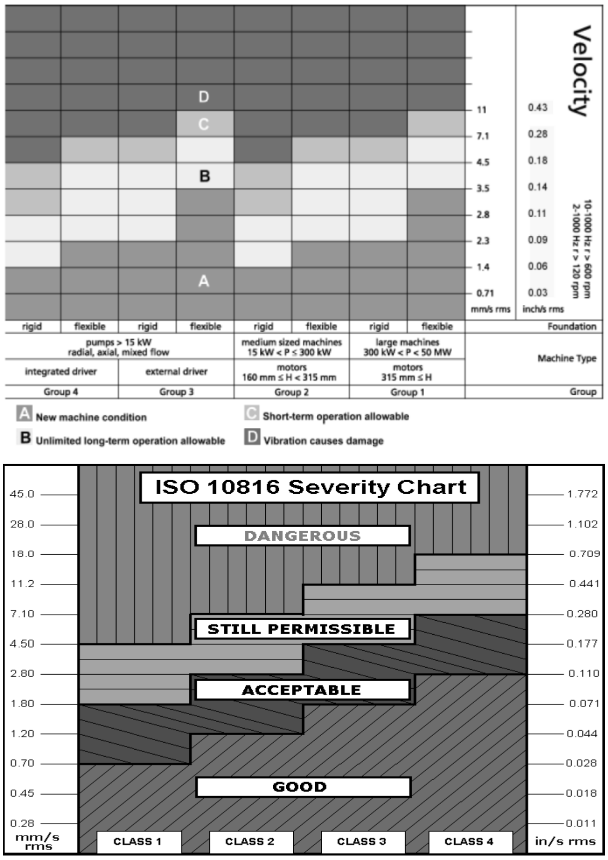


Table 19-3 - ISO 10816.1 and 10816.3 Alarm Limits. See Figure 19-3 and Figure 19-4.

This chart from Entek/IRD has been in use for more than 30 years. It is an improvement on many others because it relates the amplitude (either in Velocity or Displacement) to particular frequencies.

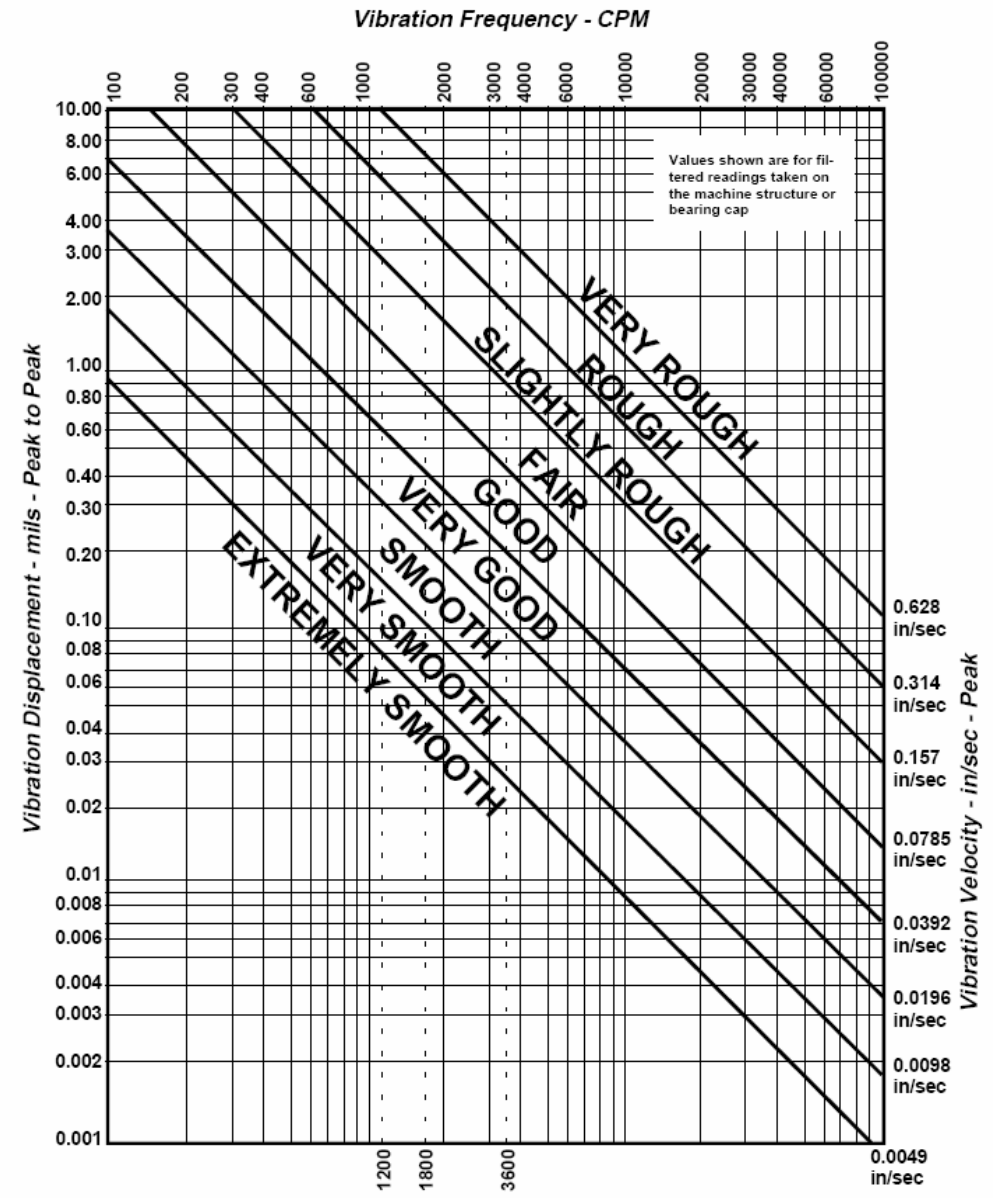


Figure 19-18

DLI Engineering publishes this guide to machine severity. It is quite useful because it caters for different frequencies, speeds, and units (acceleration, velocity and displacement).

A better copy of this and copies of the other charts are in the Quick Reference guide.

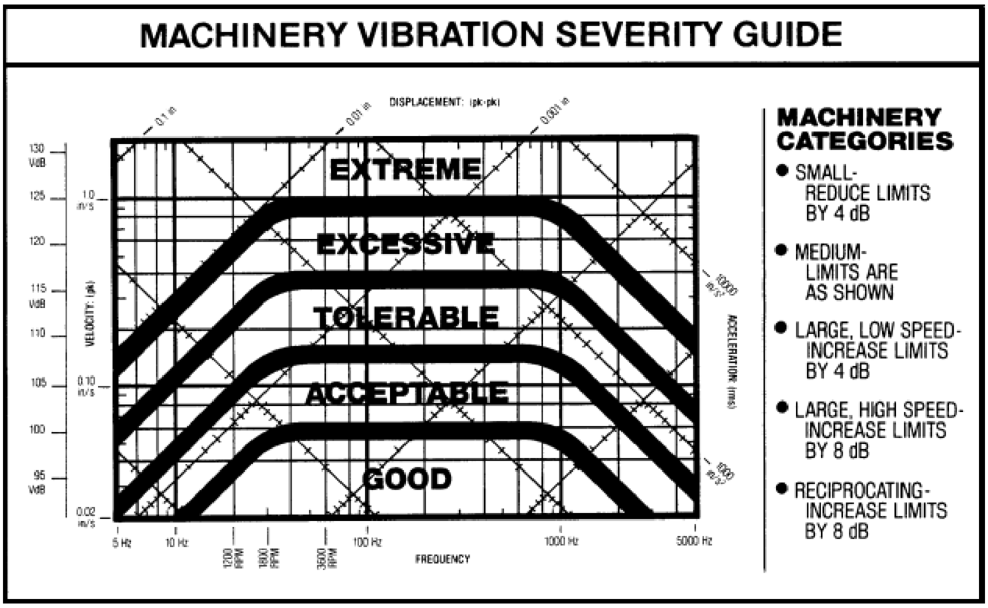


Figure 19-19 - DLI Engineering guide to Alarm Limits

The charts can be used to set a fixed amplitude alarm for various machinery and specific frequencies. It can be tedious. Most software packages have "canned" alarm limits for various types of machines and these can be a good resource, too.

However, it is difficult to use these fixed alarms and know that they are right for your machines. There needs to be a better way to tailor the alarm limits to your specific machines in your specific processes. Alarms can be calculated from actual vibration levels on your machinery.

Band Alarms

The previous discussions have shown how fixed alarm limits and statistical calculations can be used to compute values that can be applied to vibration limits. But how are they applied?

This is an area where the approach taken is largely dictated by the software package being used. In brief, there are "band alarms", "envelope (or mask) alarms", "expert systems" and "artificial intelligence systems". This course covers the band alarms and the envelope alarms.

Simply put, band alarms work on the principle that it is possible to consider different portions of the spectrum, and different scalar measurements (overall readings, bearing measurements, etc.) and apply different alarms to each band.

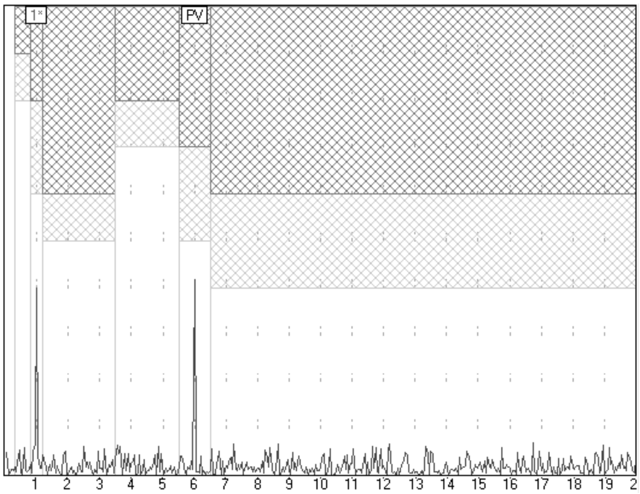


Figure 19-20 - Six to twelve bands may be available for setting alarms. Bands are often defined by Analysis Parameter Sets.

For example, in a spectrum, one band could be created around the running speed peak. The software will focus on the vibration level between 0.9X to 1.1X, for example. The software will look at the vibration in that frequency band and compute the maximum level, an average level, the rms level, or some other parameter and see if it exceeds a limit.

The same technique is also applied to other "scalar" data such as overall readings, waveform pk-pk readings, crest factor calculations, bearings measurements and other useful parameters.

The band can be given a special name, for example "1X", "Running speed peak", "Unbalance", or whatever the user prefers. The limits applied are also typically user selectable; either fixed limits, or limits that are computed statistically. It is also common for the user to be able to set more than one limit per band, for example an "alert" limit and a "fault" limit.

Six or twelve bands may be available for use in this way (the exact number available is set by the software package), and the bands may cover narrow frequency ranges (1X, 2X, 3X, etc.) or broader ranges 1-10X, 0.2X - 0.8X (sub-synchronous), 10X-50X, and so on.

Naturally the options vary from software package to software package. And the name given to these bands also varies.

HOW IT WORKS:

If the amplitude of the peak increases to the point that it crosses over the alarm level, the band limit will be exceeded and the alarm triggered.

The alarm can be set up in a way that also measures the total Power in the band. This means that if the total energy of the peaks is at a certain level, a “power” alarm will be triggered although no peaks have crossed the actual alarm level. See Figure 19-21.

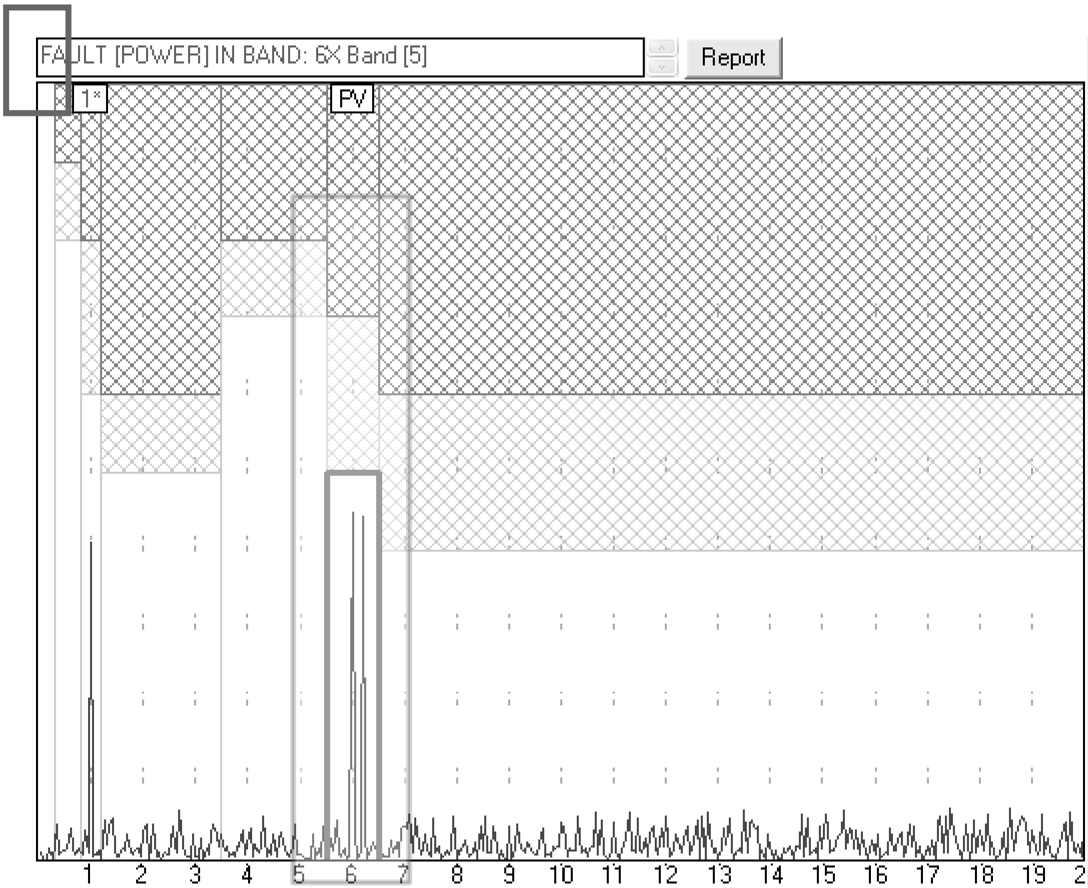


Figure 19-21 - A Power alarm is triggered based on the total power in the band rather than a single peak penetrating the alarm level.

When the exception report is generated, the software performs whatever calculation is necessary to first extract the value from the band, and then to compare that value to the alarm limit(s). A report is then generated to inform the user of any exceptions. This is where the fun can begin…

In a system that has been set up correctly, the report should list a small number of machines, and the information presented should give a clear indication of what may be happening with the machine. Over the years the reports and the methods used to derive the alarm limits have been improved, so the reports have been improved.

However, unfortunately the effort, skill, experience and patience required to correctly set up the bands (or analysis parameter sets) has been too great, and the report has therefore included too many machines. If over one-third of the machines typically appear in the report, then the system has not been set up correctly.

This fact has so often caused vibration analysis engineers to give up in frustration and turn to manual analysis instead. Fortunately the bands also act as useful analysis tools, and the data extracted from the bands can be trended to see how values are changing over time.

There is great value in having a working exception report. Working correctly, exception reports should save a great deal of time. The data collector should be uploaded to the computer, the report run, a list of the machines "in alarm" viewed, and then time and attention focused only on the machines that appear to have a problem.

BAND FREQUENCIES

Most software packages offer at least 6 frequency bands for the spectral data. Each band has a minimum (Fmin) and maximum frequency (Fmax). Companies who use the products have tweaked the bands to work well with their machinery. One of these companies is General Motors. The chart graph below (Figure 19-36) shows the frequency bands typically used although there is quite a bit of variation after the third band.

Common settings include:

- Band 1: (sub-synchronous): 0.3 – 0.8x
- Band 2: (1x): 0.8x – 1.2x
- Band 3(synchronous) 1.2x – 3.5 x
- Band 4: 4x to N depending on the machine.

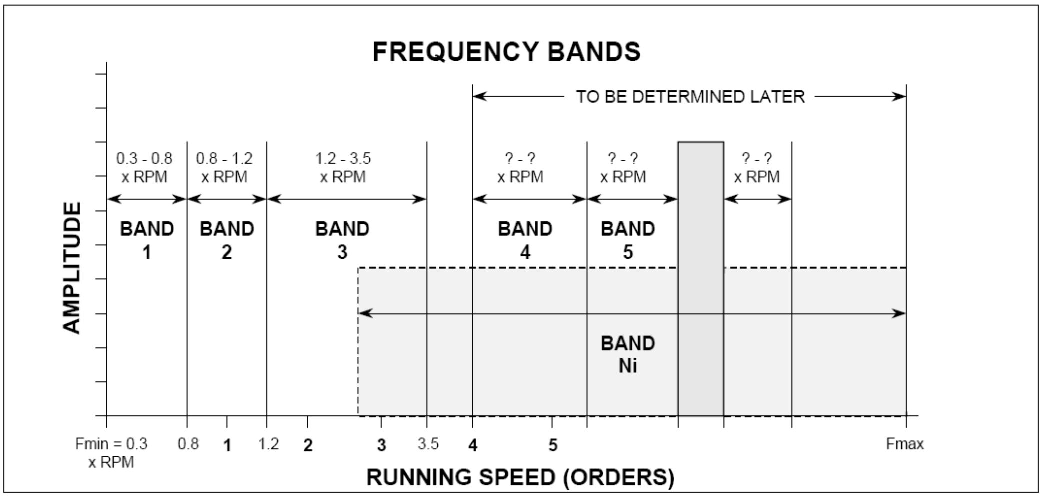


Figure 19-22 - Sample band alarms from the GM specification No. V1.0a-1999

In Figure 19-23 and Figure 19-24 notice the guidelines for the vane pass frequency.

TABLE 9.4 MAXIMUM ALLOWABLE VIBRATION LEVELS FOR POSITIVE DISPLACEMENT AND CENTRIFUGAL PUMPS				
LINE-AMPLITUDE BAND LIMITS				
BAND	FREQUENCY RANGE Hz (CPM)	VELOCITY	MM/SEC RMS	(INCH/SEC - PEAK)
1	[0.3 → 0.8] x RPM		0.718	(0.04)
2	[0.8 → 1.2] x RPM		1.35	(0.075)
3	[1.2 → 3.5] x RPM		0.718	(0.04)
4	3.5 x RPM → 2,000 Hz (120,000 CPM)		0.54	(0.03)
BAND-LIMITED OVERALL AMPLITUDE LIMITS				
BAND	FREQUENCY RANGE Hz (CPM)	ACCELERATION	g's RMS	(g's PEAK)
1	0.3 x RPM → 5K Hz (300K CPM)	POSITIVE DISPLACEMENT	1.06	(1.5)
		NON-POSITIVE DISPLACEMENT	0.707	(1.0)
PUMPING FREQ. BAND (PF)	FREQUENCY RANGE Hz (CPM)	VELOCITY	MM/SEC RMS	(INCH/SEC - PEAK)
BAND 5	5 Lines of resolution centered on PF.	PISTON	1.35	(0.075)
		VANE	0.89	(0.05)

Figure 19-23 - GM standard limits for positive displacement and centrifugal pumps.

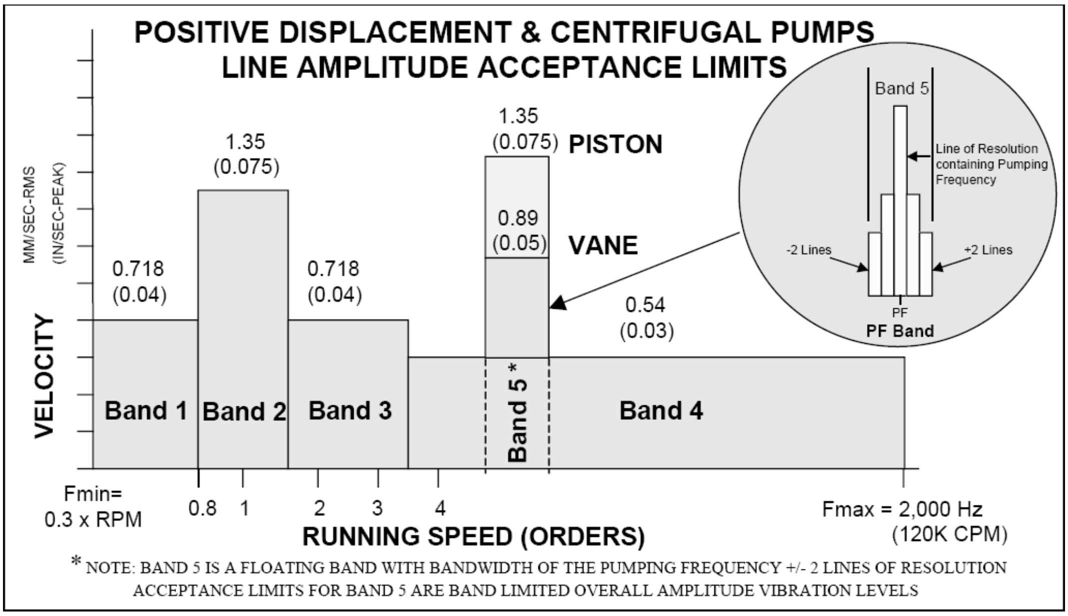


Figure 19-24 - Frequency band descriptions for pumps

Setting the limit for each band is the challenge. This is where most programs fail. Use printed guides such as the GM specification or the ISO standards. Another solution is use an offset from the reference or baseline data. The final option is to compute statistics.

Continue tweaking the alarms so that the Exception Report is meaningful. When the Exception Report continually shows alarms that are not real, people tire of looking at the report and ignore them. It is much like the fairy tale of the boy crying wolf, one day when there really is a problem, it is ignored and there is a machine failure that could have been avoided. Recall from the chapter on Condition Monitoring how important it is to do something with the information or Predictive Maintenance is not really being practiced, just the expense of condition monitoring without the benefit. Adjust the alarm levels so they provide accurate warning.

One **limitation of Band Alarms** is that the bands are not necessarily sensitive enough. The RMS calculations are dominated by the highest peaks and harmonics and sidebands may be ignored. Small peaks can grow next to big ones and the alarm will not be tripped. This is especially true of bearing frequencies. See Figure 19-25

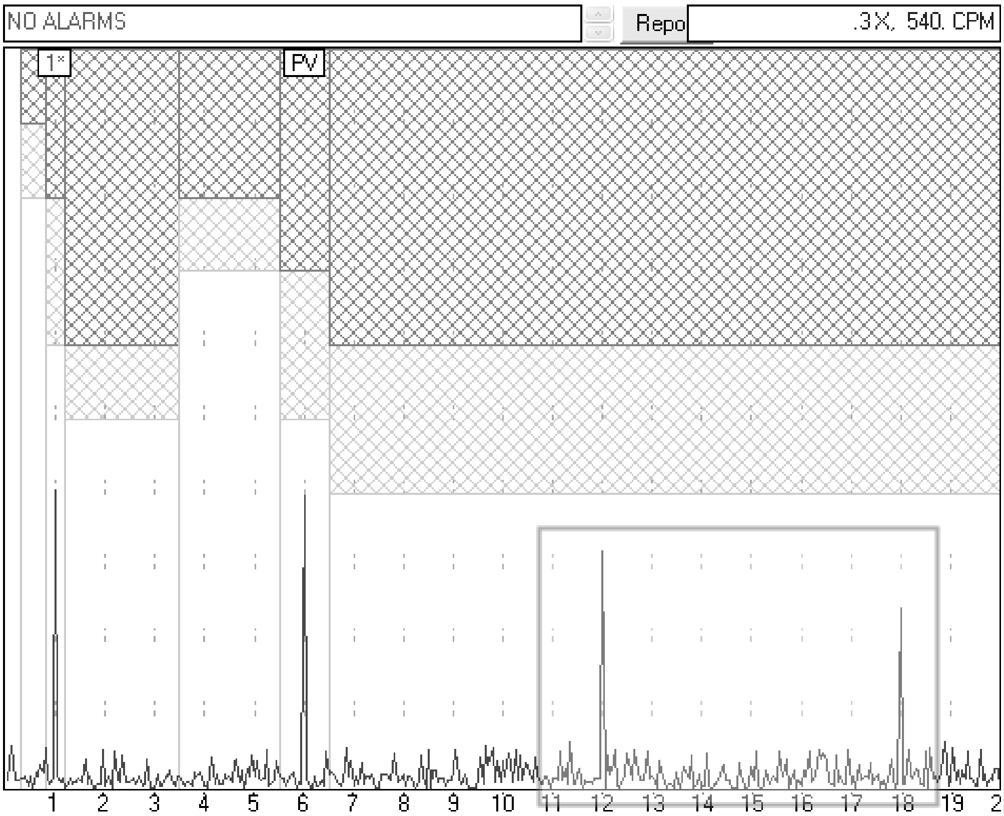


Figure 19-25 - Band alarms may not be sensitive enough to catch small peaks such as bearing frequencies.

Mask/Envelope Alarms

Envelope alarms (also known as mask alarms, and not to be confused with envelope detection used in bearing analysis), take a different approach. Rather than breaking the spectrum into individual bands, an alarm limit is applied to the entire spectrum.

As seen in this example, it is not a single line across the spectrum; it is an envelope that hugs the spectrum at all frequencies.

The benefit to this approach is that every frequency is covered, and it is potentially more sensitive to peaks that can appear at unexpected frequencies. Whereas a single "band" may be used to cover a wide frequency range, from 1X to 10X, for example, an envelope/mask may be computed to have up to 50 individual limits that follow the shape of the spectrum.

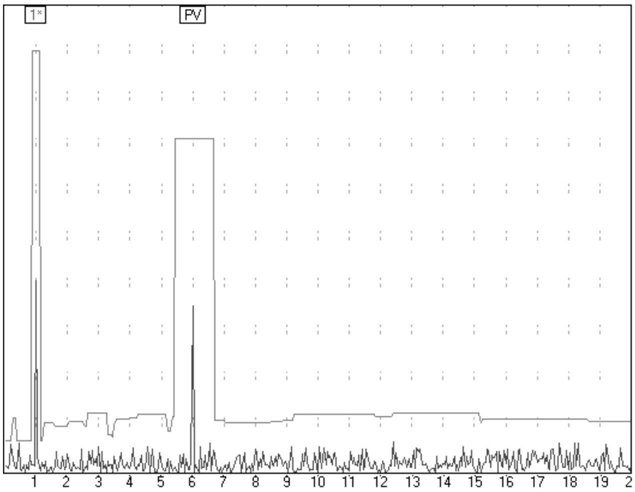


Figure 19-26 - Envelope or mask alarms

The downside is that, depending upon the specific implementation; it is possible that the exception report is not very specific as to the cause of an exception. For example, if a peak in the spectrum were found to exceed the at 3560 CPM, you would like to know what that means. A band-based exception report may label that as "2X", and allow you to view a trend of previous values from that band - a feature that may not be available in a system based on envelopes.

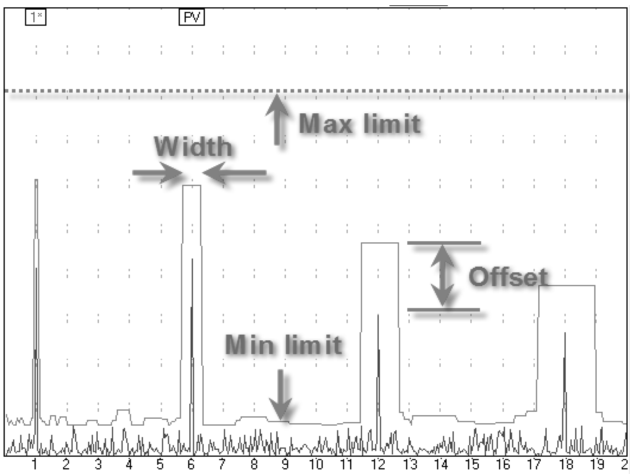


Figure 19-27 - Several limits can be specified when setting the alarm limits.

However, some envelope-based systems also provide for entering forcing frequency information (the running speed, the number of pump vanes, etc.) and the exception report automatically relates exceptions to these known frequencies. In this example, the software would see that 3560 CPM is close to twice the running speed and would label it appropriately.

The software typically computes the envelope alarms, although there is one package that allows the limits to manually be drawn on the spectrum.

When a program is begun, there will not be a history of data to use in a statistical calculation. Instead the envelope may be calculated by simply taking the baseline spectrum (the first measurement collected) and doubling all of the levels across the spectrum (perhaps applying a minimum and maximum threshold). Other calculations may be performed in order to enable the envelope to be less sensitive to slight speed variations.

Again, it is necessary to consult the software manual, or question the sales person. Each approach certainly has its merits and the real issue is how much time a person is willing to spend in order to understand the technique and to set up the alarms.

SPEED COMPENSATION

For variable speed machines there must be a compensation for the speed or the band will trip an alarm when the speed increases. This compensation is made when the band widths are defined. They can be defined with constant bandwidths or with bandwidths that are proportional to speed. Figure 19-28 shows the proportional bandwidth envelope.

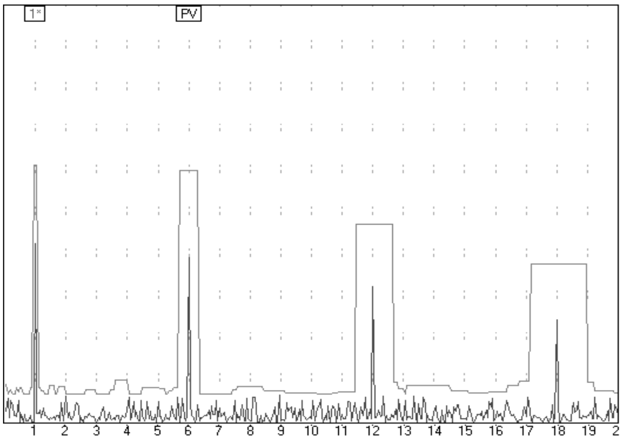


Figure 19-28 - Variable speed is compensated for with proportional bandwidths

AMPLITUDE OFFSET

The amplitude can be defined in one of two ways. It can be defined as a fixed offset which sets a value across the entire spectrum. For example, an offset of 0.1 ips means that every bin will have to increase by 0.1 ips before an alarm is triggered. This can be too low for some peaks such as vane pass but too high to catch bearing problems. See Figure 19-29.

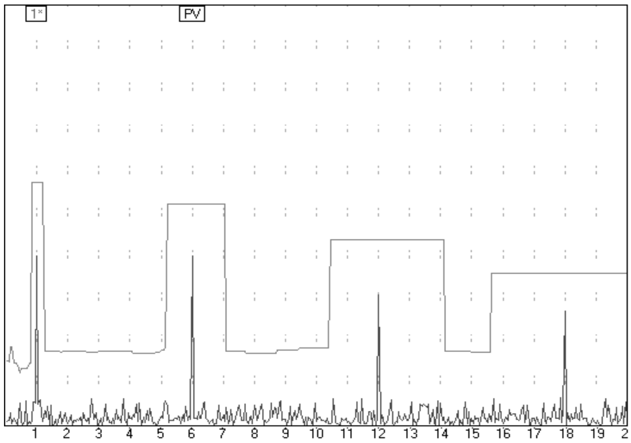


Figure 19-29 - Fixed amplitude offset

A preferred method is to use a ratio offset such as 150% above the baseline data. This allows all peaks to change, but have it scaled for each frequency. See Figure 19-30.

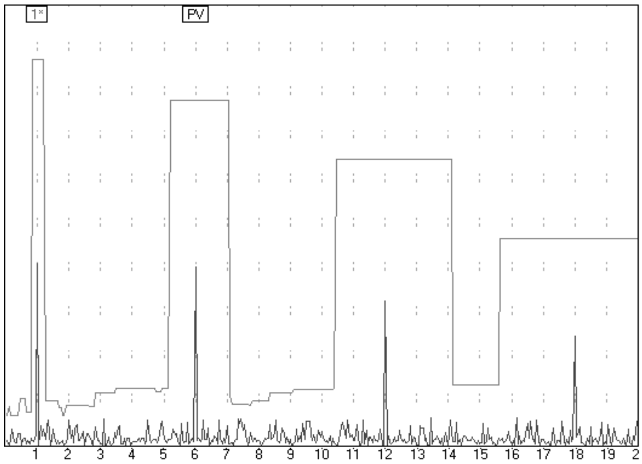


Figure 19-30 - The amplitude offset based on a ratio of the baseline value

WEAKNESS

A weakness of the envelope alarms is demonstrated in Figure 19-31. The alarm level envelope around the 6x vane pass was set based on the 6x peak. However, now a small peak at 6.2 orders has grown considerably. It will have a long way to grow before it reaches the alarm limit. If this peak is a bearing frequency, it may reach the final stages or even fail before this peak reaches the alarm limit.

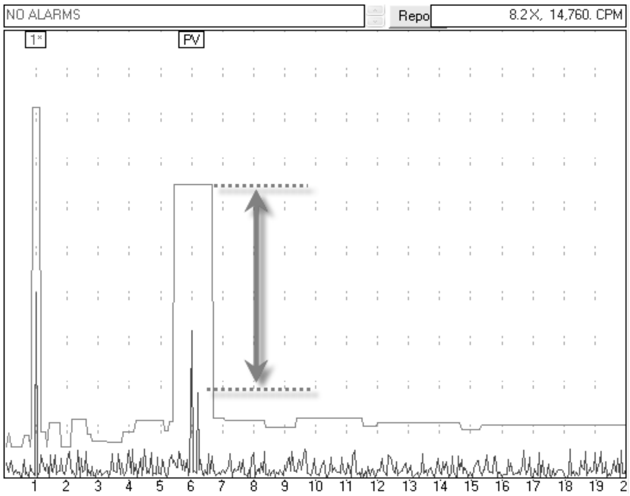


Figure 19-31 - A weakness is that a nearby peak may have to grow by 10 times to reach the alarm.

Calculated Alarms relative and computed alarms

Rather than using these fixed limits, we can attempt to calculate what the limits should be based on previous readings. Since all machines are different, and even similar machines (AC motors for example) can operate under different loads, as well as being mounted differently, and manufactured differently, and so on, it is difficult to use fixed alarm limits.

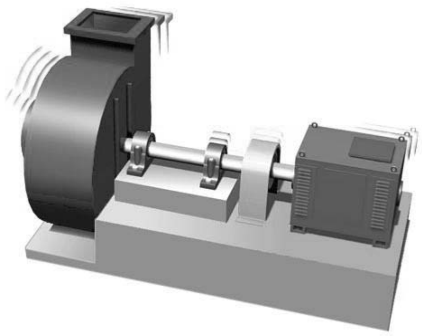


Figure 19-32

If a set of readings is taken on a machine, and then an analysis performed to assess the quality of the data and the approximate condition of the machine, then alarm limits can be applied based on the current levels.

A common rule of thumb is that a problem exists if the vibration levels double from their original "healthy" levels. But what are the healthy levels?

It is important to get the program started, so use fixed alarms at first. After readings have been collected, switch to calculated alarm limits. Enough time has to pass so that that a good idea of the condition of the machine is realized. In short, if the vibration levels do not change significantly, more than about 15% over six months, then the machine's condition is probably stable.

Setting the Baseline

Based on the earlier statement that a doubling of vibration indicates that a problem exists, baseline measurements can be taken, the observed vibration levels doubled, and set those as the limits. But what is the "baseline" measurement?

The baseline measurement is ideally a measurement that we believe best represents how the machine should vibrate. It is a measurement with which we can compare against to determine if a change has occurred.

>150% increase is “substantial” – investigate with priority. The simplest approach is to take the first ever set of data collected and set it as the "baseline". Many software programs easily compare the current vibration data with the baseline measurement.

Another approach is to take a reading after a machine has been repaired/overhauled (and after it has had a chance to "run-in") and set it as the baseline. In this case the machine is possibly running as well as it ever will run (that is, the vibration levels will be as low as they will ever be). In this case however, it may be good to set the alarm limits at greater than twice the baseline; otherwise the system may be too sensitive (giving false alarms).

Yet another approach, the approach preferred by the author, is to use statistical calculations.

Let's start at the basics so we understand what this is all about. In other parts of this training system we have said that we are interested in absolute vibration levels. But we are probably more interested in how the levels change.

If a vibration level does not change over a period of time, then it is believed a machine is stable and does not warrant any repair. The author has seen and heard of examples where the vibration level and pattern has appeared quite serious, however it has not changed for 18 months. That does not mean that a problem does not exist, it only means that urgent/unscheduled repair action is not required.

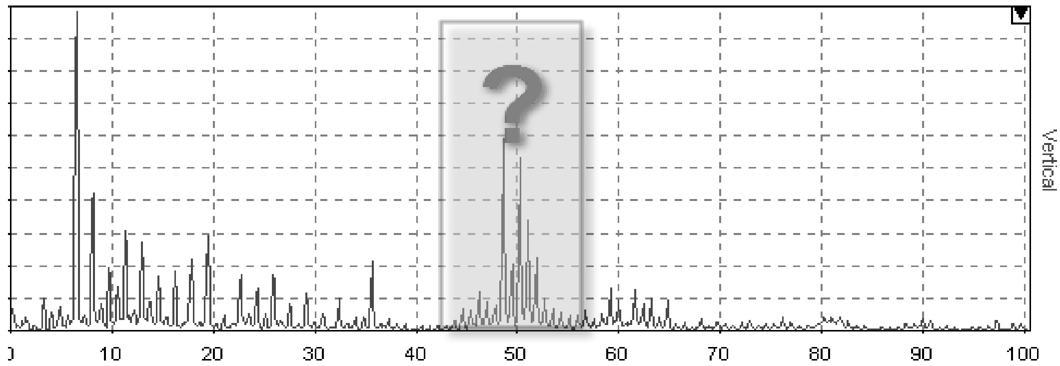


Figure 19-33 - it is difficult to know what limits to place on individual regions or lines in a spectrum.

If after four months a region of the spectrum does not change, should it be expected to change? On the other hand, if it normally changes considerably, should it be expected to change?

For example, 5 sets of data are collected on a machine, 30 days apart, and the running speed (1X) peak barely changed, whereas the peak at 6X, which corresponds with the vane pass rate, varied by over 30%.

What would be expected in the next reading? A 1X level that was similar to the other readings would be expected, and a 6X peak that was within the range (or maybe a little outside) of the previous 5 readings. Notice there has been no discussion of its absolute levels, just relative levels. In many cases, particularly after accumulating a history of readings, absolute levels (i.e. the actual level in ips or mm/sec) are less interesting to us.

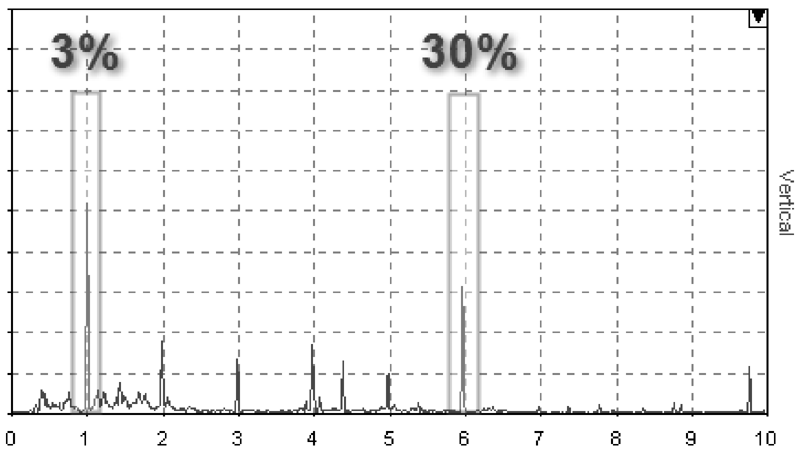


Figure 19-34 - Historically, how much has the frequency band changed over 5 months of data.

If the next month's data shows that the 1X amplitude level was 10% higher than the previous level, would it indicate a problem? It would be a little surprising because this is outside the norm for this machine. A 10% increase at 6X would not be as surprising, would it? It seems normal for this level to be changing from test to test.

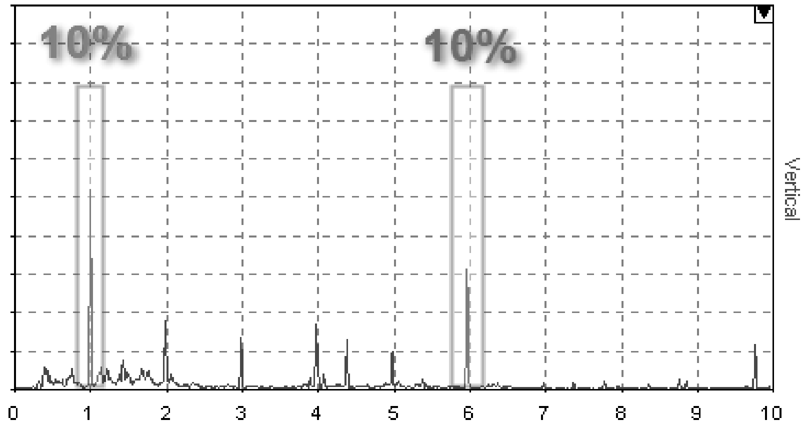


Figure 19-35 - If the next month's readings changed by this amount, is it a problem?

These observations reflect standard human understanding. Statistics are just the application of these observations into mathematical formulas.

Standard Deviation

Before entering into the discussion on Standard Deviations, it should be pointed out that most software programs can do this at the push of a few buttons. A short description of the process is given here. A more detailed description is in the shaded box.

In general terms, the Standard Deviation is a measure of how much a set of data varies. The number is described in terms of σ (sigma) variation. For example, using alarm limits set to 2 σ then 95% are expected to fall within the limits.

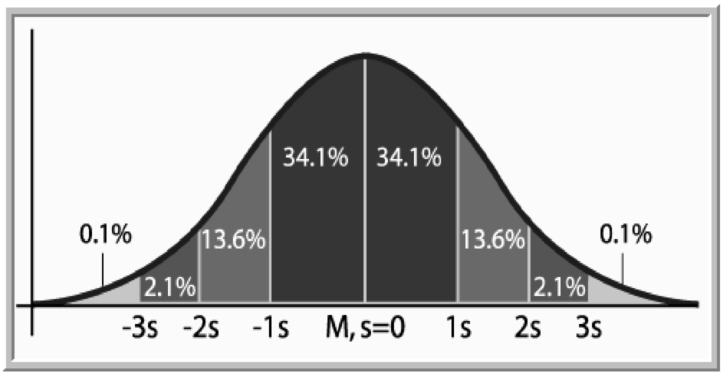


Figure 19-36 - Bell curve of standard deviation. Approx 95% fall within the limits

The variation can be calculated quite easily, and from a parameter called the "standard deviation" can be calculated. “Standard deviation" - is a term that describes the amount of variation that is "normal" for the quantity being measured.

It is not worthwhile to explore the actual math involved, however, the standard deviation is typically represented by the Greek symbol "sigma". It can be shown that there is a very high probability that a new reading with fall within 3-sigma, and a lower probability that it will fall inside 2-sigma.

The great thing about computers is that they can do all this for us. If your software supports it, your job is simply to review the data to determine which data are "worthy" of being used in the statistical process, and then asking the software to perform the calculations.

The result is that the alarm limits, however they are used, are based on readings taken from your machine, and take into account how the vibration levels normally vary.

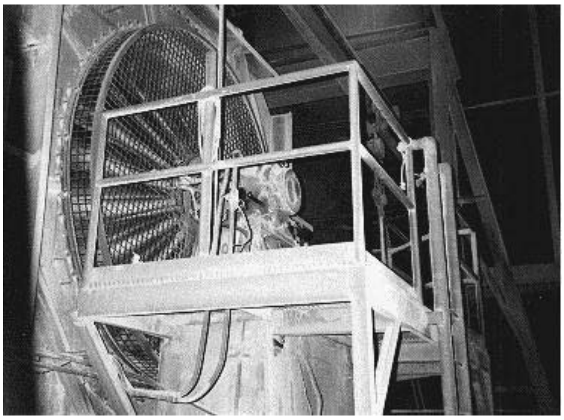


Figure 19-37 - Statistics builds alarms based on the machine's history

The variation can arise from process noise (flow, external noise, etc.), normal fluctuations in load, natural variation in vibration (two measurements taken 2 minutes apart will not be the same!), and other operational variations.

The application of statistics varies from package to package, so consult your software vendor (or their manuals) to see how you can utilize this powerful tool.

It should be said that a vibration analysis program is normally begun with fixed alarms (because there is not a history of data to use in the calculations), and then when between five and ten readings are collected (five is OK, ten is better) switch over to a statistical approach.

Statistics with a Twist – Identical Machines

Thus far the discussion of generating alarm limits using statistical calculations has been based on readings taken from the machine. It is normally assumed that the readings used in the statistical calculations would come from a single source. That is, the statistical alarms for the free end vertical bearing point would be generated with data from that point only. But does it have to be that way?

Let's think "outside the box" for a moment. If there are six motor-pump units in a row, all doing the same thing, how would the vibration readings compare from the bearing on unit "A" to the same bearing on unit "B"?

If they were manufactured by the same company, and were pumping the same fluid, wouldn't the levels the levels be expected to be similar? And if they were not, to what would that be attributed? If there was a difference in level at one frequency or another, perhaps it could be assumed that they were in different condition.

Knowing that, would it not be possible to learn something about pump "A" from pump "B"? For example, if these machines were monitored for six months and the bearing on pump "B" failed, and the vibration levels reached before it did fail are known (or you witnessed the extent of wear when it was repaired before failure), wouldn't that help set the alarm limits on pump "A"?

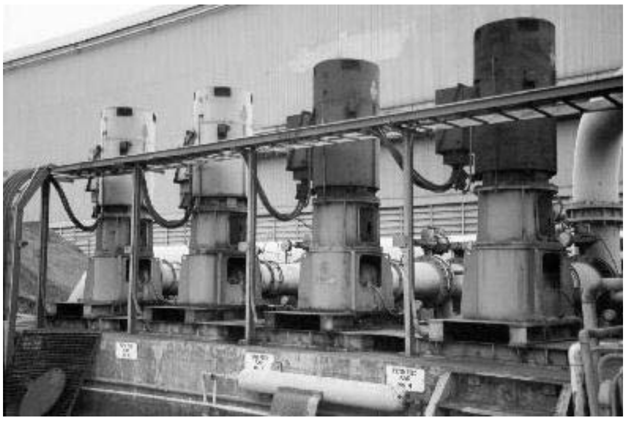


Figure 19-38

If that makes sense, then perhaps the data can be utilized from multiple identical machines to generate the statistical alarms for all of those machines.

This concept can be taken one step further. What if while setting up the alarms it is noticed that units "A" through "E" were OK, whereas unit "F" had some kind of problem. Perhaps the alarms could be generated based on the data available on the first five pumps (and if they were all tested twice then there would be ten sets of data for the calculations), and apply those alarms on all six pumps.

Doing this may even highlight just how severe pump "F" really is, as now there is a good reference to compare against.

Not all vibration analysis packages support these calculations (utilizing vibration data from multiple machines), so verify to see if it is possible with the package you are using. The benefits are that "mature" alarm limits can quickly be generated, and the severity of the condition of one of the family can be highlighted based on the data from the others.

This technique also serves as a reminder that when performing vibration analysis, data should also be accessed from other identical machines in order to determine the nature and severity of a problem - because it is always tough to know what is "normal".

Standard Deviation:

In a vibrating quantity, the instantaneous deviation from the equilibrium position, if considered over a long time interval, will have an average, or mean value. If these deviation values are squared and then averaged, the result is called the variance of the vibration. The square root of the variance is defined as the standard deviation of the vibration. It can be thought of as the RMS value of the deviation. A vibration with a small standard deviation never strays very far from its equilibrium position, while one with a large standard deviation does make larger excursions.

The standard deviation is defined as the square root of the variance. This means it is the root mean square (RMS) deviation from the average. It is defined this way in order to give us a measure of dispersion that is (1) a non-negative number, and (2) has the same units as the data.

In practice, one often assumes that the data are from an approximately normally distributed population. If that assumption is justified, then about 68% of the values are at within 1 standard deviation away from the mean, about 95% of the values are within two standard deviations and about 99.7% lie within 3 standard deviations.

Background:

The first characteristic, of raw data, is the **average** or **mean**. This number is generated by adding the numbers contained in the raw data and dividing by the total number of numbers. This concept should be familiar to you since most vibration data is averaged.

A second characteristic of raw data is **variance**. This is a measurement of the data's dispersion or spread. Variance describes the degree to which a group of numbers is scattered away from their average or mean. Variance is a good measure of dispersion, but the numerical value is not intuitive and therefore difficult to interpret.

A better measure of dispersion is derived by taking the square root of the variance. This third data characteristic is called a **standard deviation**. The units associated with a standard deviation are the same as the measurement units contained in the data. This makes the standard deviation easier to relate to the raw data and average.

A standard deviation possesses a couple of interesting properties. First, the percentage of numbers from any raw data within x standard deviations of the average is $100*(1-1/x^2)$. Therefore, at least 88.88% of the numbers in any raw database will be within 3 standard deviations of the mean.

The second property applies to raw data with a **normal distribution**. A normal distribution is produced when random numbers occur between limits. The dispersion of data is reduced by a normal distribution of data. The result is 99.74% of the numbers in a database with normal distribution are within 3 standard deviations of the average.

Variable Speed Machines

There is a very real situation that can cause us a great deal of grief - varying machine speed and load.

As pointed out in various ways, rule one in testing rotating machinery is to establish repeatable test conditions. Do everything possible to ensure that the machine is running at the same speed and load each and every time. However, if that cannot be done, then action must be taken to deal with the consequences.

In an earlier example we discussed creating a band around the running speed peak, from 0.9X to 1.1X, for example. But what if the speed changes? What does the band really mean?

When the speed of a machine varies (or when the load varies), the vibration levels also change. So, what if we have set some alarms, statistically or by referencing a standards chart, and the speed changes enough to affect the amplitude level? Are the alarm levels still valid? Is the 1X peak in the spectrum still going to lie within the band? This question must be addressed no matter which method is used (bands, envelope, expert system or AI).

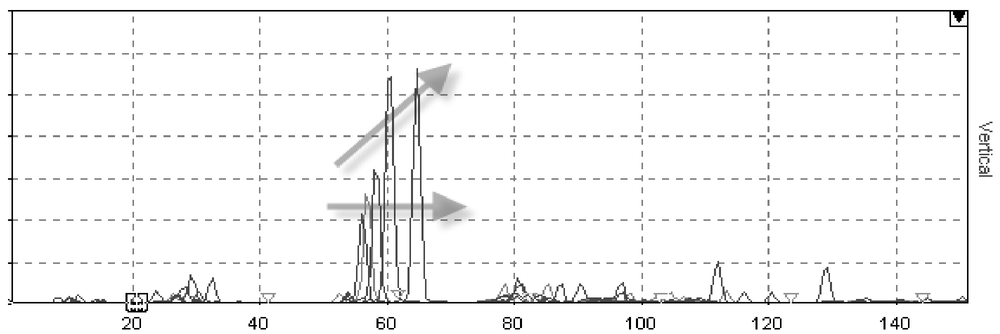


Figure 19-39

When speed changes it affects more than just the frequency range, it affects the amplitude as well. An increase in speed or load does cause an increase in the amplitude of the vibration related to turning speed.

First we need to introduce the concept of **normalization**. We are often talking about frequency in terms of the multiple of running speed. 1X is the running speed peak. 3X is the peak at three times that frequency, and so on. The process of normalization acts to "line up" the peaks in the spectrum.

If we have spectra from a machine running at different speeds, we would expect to see the peaks in different places.

Instead, when we graph the normalized data, the peaks at 1X are overlaid, the peaks at 2X are overlaid, and so on. We can perform this normalization in our calculations as well, so that the 1X peaks will always fall within the 1X band, and the 1X peak is always compared to the correct level in the envelope.

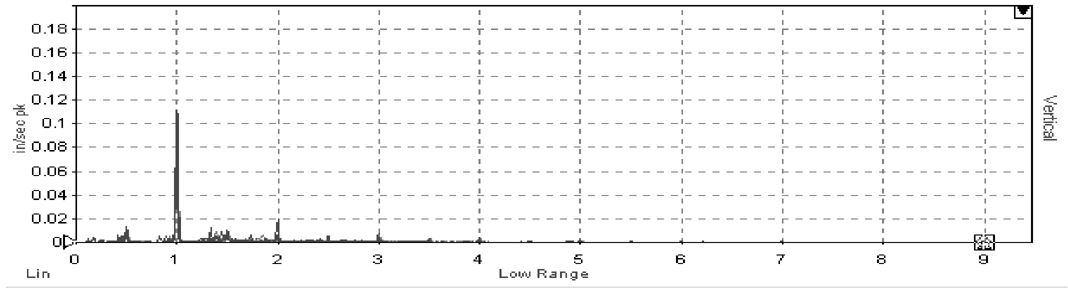


Figure 19-40 - Order normalized spectra

But what about other frequencies, like the line frequency that is not affected by the running speed? Unfortunately, when you normalize the spectrum, the peaks related to running speed line up, and those that are not related to running speed (line frequencies, resonant frequencies, external sources of vibration, and so on) do not.

The answer is not simple, and again depends upon the system you are using. Some systems are able to cater for these situations, and others are not. When it comes time to perform the spectral analysis, you may have to be aware of these issues and adjust the analysis accordingly.

Some software packages provide the option of describing the machine as a variable speed machine. Then when beginning to collect data on the machine, the analyzer asks for the speed which can be typed in. The actual speed could be from a readout and can be in RPM or for roll processes, feet or meters/min.

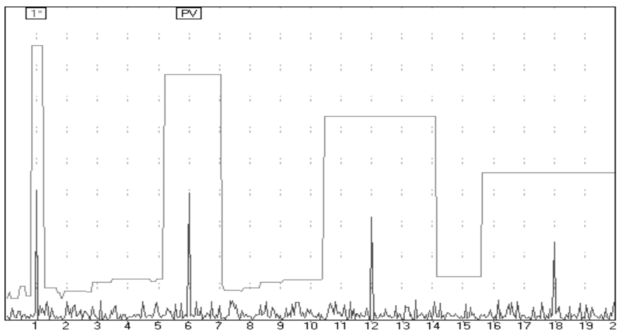


Figure 19-41 - Mask alarm set to allow for speed variation

A similar option is available for the load. Defining the machine as variable load forces the analyzer to ask for an input via the keypad. This data then shows up in the Measurement Exception Report to let the analyst know the load so that compensations can be applied if appropriate for that machine.

The other more important issue is how much variation can be tolerated. If a machine varies in speed by just a few percent then it is OK to normalize the spectra and more-or-less analyze the data as usual. Outside this range it is likely that the levels will be different (due to resonances, the force in impacts, load changes, and other factors) so normalization is not appropriate.

Figure 19-42

There are a few different approaches to take, and again it depends upon the system used. In some cases it may be necessary to set up different machines in the database to store the data when tested in different conditions.

For example, two machines in the database could be named "Chiller A summer" and "Chiller A winter". In the summer time when the chiller is under greater load collect the data as if the name of the machine was "Chiller A summer". And in winter, when the load is lower and the levels are down, collect data on the machine "Chiller A winter".

In the same way, a machine can be set up as "Pump A 1000 RPM", "Pump A 1200 RPM", and so on. If the speed cannot be controlled, then store the data in different "machines" in the database, and apply different alarm limits to each machine.

There have been attempts to characterize a machine so that the effect of the speed and load change can be "normalized" out of the machine, however the author is not aware of any successful implementations.

So, the first rule is to control the speed and load of the machines if possible. The second goal should be to minimize the variations so that data can still be compared to the data from all tests. And finally, if this is not possible, set up system so that the data and alarm limits can be kept separate.

Conclusion

Start with published alarms to get the program started. Use calculated alarms once you have a history of data. Set your alarms based on change above the reference levels. As a suggestion, a 50% increase is “significant” and should be investigated, a greater than 150% increase is “substantial” and should be investigated with priority.

Chapter 20

Acceptance Testing

Objective:

- Describe 4 considerations for developing criteria for New or Rebuilt equipment Acceptance procedures.

Vibration Analysis can be used for more than just diagnosing fault conditions. It can be used in an organization's "acceptance testing" program to ensure new and rebuilt machinery is free from faults before placing it in service.

When Acceptance Testing is part of the purchasing process, new or rebuilt equipment arrives in better condition and has less defects and startup failures. When the criteria is part of the purchase order, plants do not have to accept machinery that does not meet the standard.

The purchase order can specify where and how measurements are made on the machinery, if they are to be made at the supplier's location, and whom they are to be made by. It can also specify the procedures for measuring at the plant before and during the installation and after startup. The sign-off on the machine does not have to be made until the machine is in place, under normal working loads, and within preset limits.

Following these acceptance guidelines can result in better equipment with fewer problems, and longer equipment life.

Figure 20-1

Organizations can set Criteria for:

- Manufacturers to meet
- Repair shops to meet
- Checking new and overhauled machines against a standard.

New Machinery Standards

Develop standards for new machinery that the **supplier** must meet. They must make the measurements and they must do it where and how you specify.

The purchase order should include the following specifications:

- The measurement points where you want to collect vibration data
- Acceptable vibration amplitudes at each measurement point
- The type of probe and attachment (fixed, magnetic etc) you plan to use.
- The type of instrumentation you plan to use.

You can take the measurements yourself, and tell them where/how you will take the measurements and what levels they must not exceed.

Or you can provide the standards to them and have them take the measurements.

Specification should include:

- Measurement locations
- Type of measurements (spectra, shock pulse, etc.)
- Type of sensors to be used
- Load conditions during the test(s)
- Run-in period before the test is performed
- Qualifications of the person performing the tests
- Amplitude limits that must not be exceeded

Tests may be performed at the manufacturer's facility. However, if that is not possible, the tests or measurements may be performed once the equipment is installed, but it may still be rejected.

Figure 20-2

Machines can be tested by the purchaser's staff. The nature of the tests must be specified in the purchase order.

Specify the measurement locations and specify the nomenclature that everyone understands exactly where data was collected.

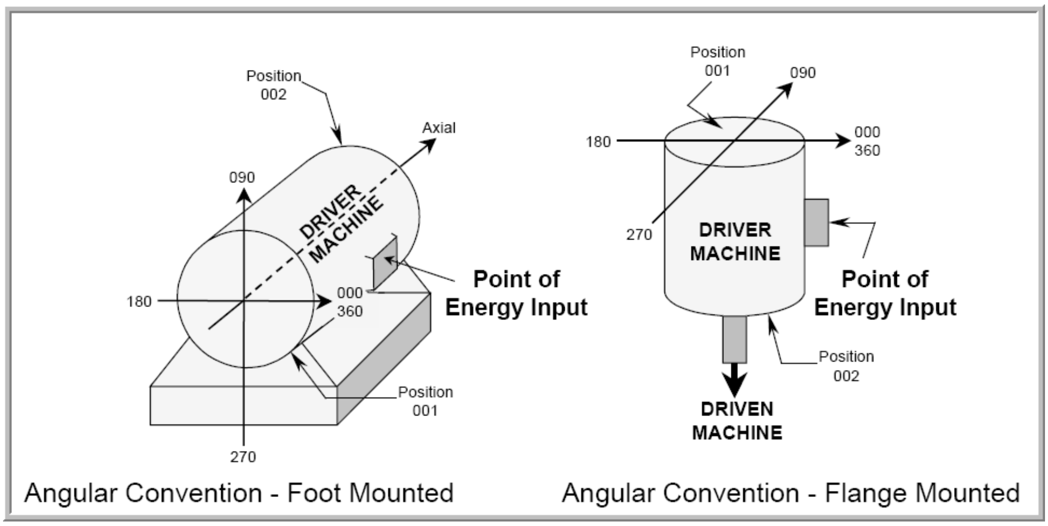


Figure 20-3 - From GM Specification No. V1.0a-1999

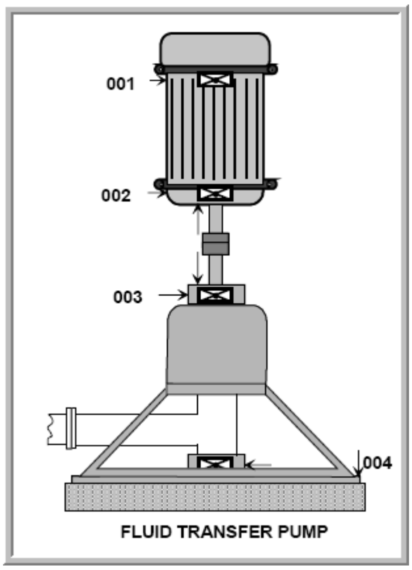


Figure 20-4 - From GM Specification No. V1.0a-1999

Specify Alarm Limits for different types of machines. These alarms are typically referred to as "Band Alarms." Alarm limits for a motor and a pump are shown below. The bands are Order based and the limits are shown in metric and imperial units.

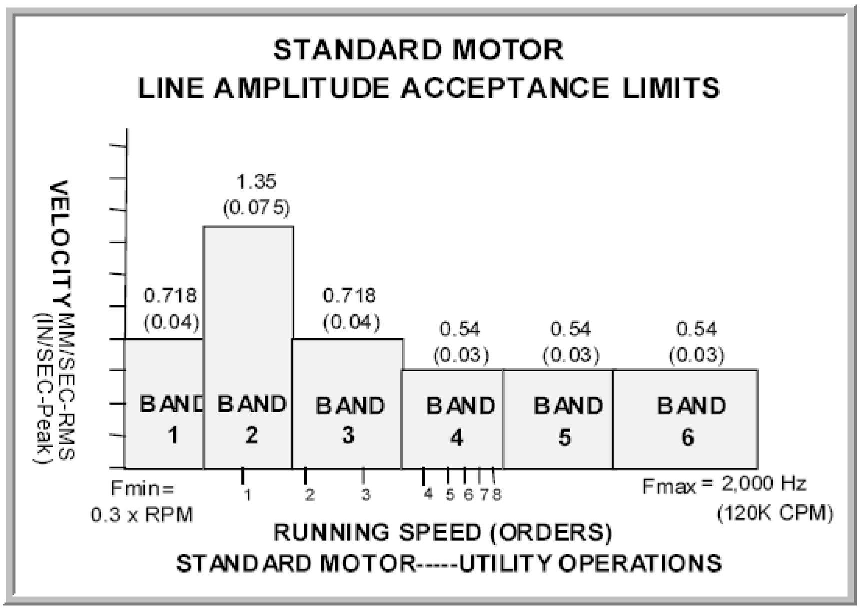


Figure 20-5 - Standard Motor alarm limits

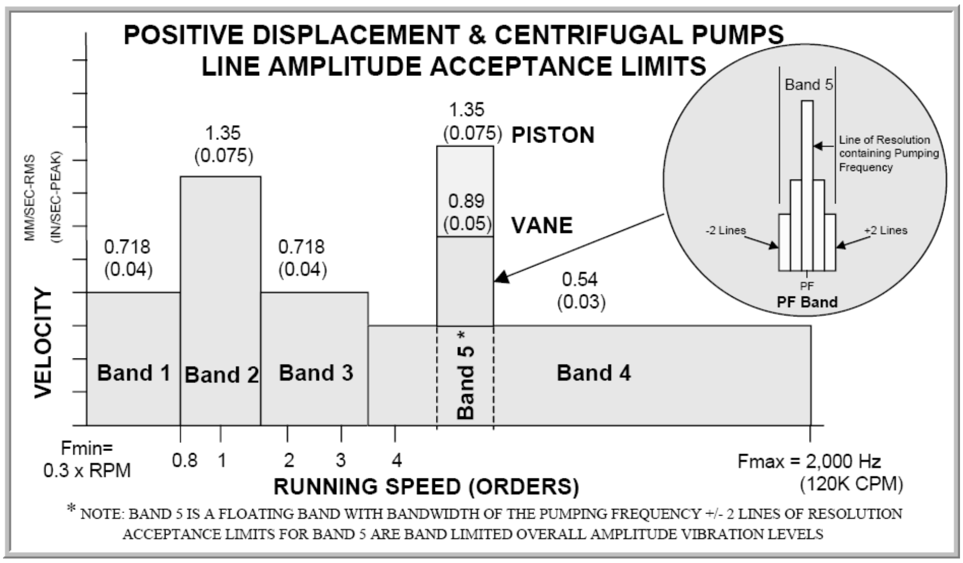


Figure 20-6 - Band alarms for Centrifugal and Piston pumps. Note limits for vane pass.

Collect a Baseline Measurement – A representative from the purchaser's company should perform a baseline analysis on the machine during the final test run at the vendor's facility. Record all the vibration and phase data for comparison at start-up.

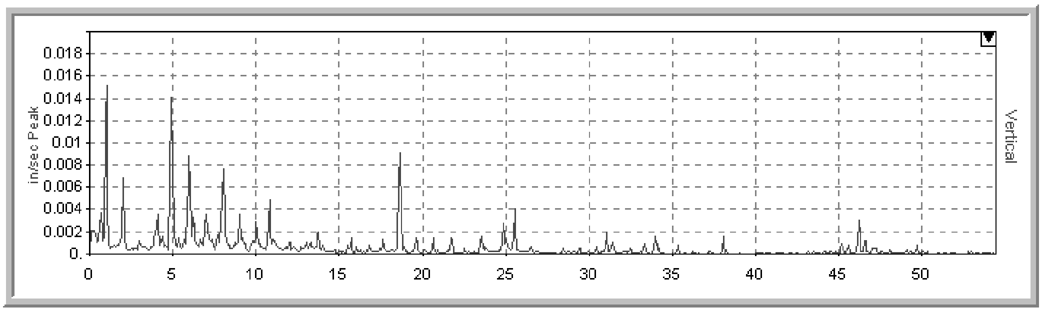


Figure 20-7 - Collect Baseline data at the vendor's factory. Collect both phase and vibration data for comparison at start-up.

Check for resonances

After installation, perform resonance bump tests on all piping, frames, and foundations. Record bump test data for future reference. If major problems are detected, you may require a full vibration test.

Ensure forcing frequencies will not excite resonances…they should be greater than 20% away from natural frequencies.

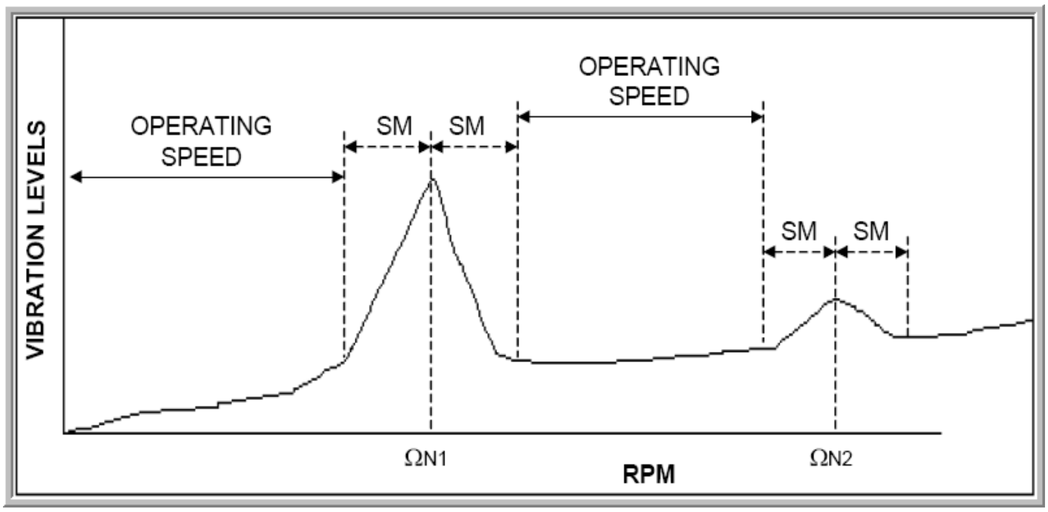


Figure 20-8 - Collect run-up and coast-down data as well as bump tests. The Separation margin should be at least 20% from the Natural Frequency. From GM.

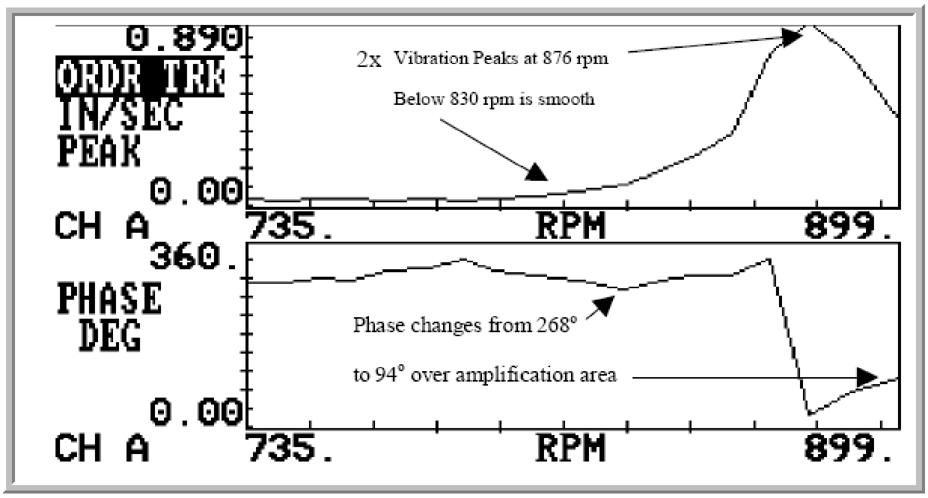


Figure 20-9 - Collect Sart-up and Coast-down data to determine best operating speeds

Collect Run-up and Coast-down data to determine critical speeds and resonance zones. Correct any problems detected during start-up before continuing the start-up.

RESONANCE: (from GM)

If the frequency of any harmonic component of a periodic forcing phenomenon is equal to or approximates the frequency of any mode of a machine's natural frequency of vibration, a condition of resonance might exist.

Operating speeds must have a separation margin (SM) of at least 25% of the resonance speed (0). Where multiple resonances exist, the operating speed shall also be above or below any given resonance and removed from the resonance by a separation margin of at least 25% of the resonance speed.

Analyze the new machine at Start-up

- Run the driver uncoupled and collect vibration and phase data
- Couple the driver and machine, collect vibration and phase data with the machine running unloaded and cold
- If possible, collect vibration and phase data as the machine warms up
- Collect data with machine unloaded after it reaches operating temperature
- Collect data with machine running at full load
- Collect additional data when foundation has heated up
- Cross compare all data collected with data on existing machinery
- Analyze all data to determine if the machine meets specifications, if not, then why?
- Was a problem detected on the test stand? Did the problem deteriorate when installed?
- Was the problem introduced in installation?
- Was the machine damaged in transit?

Overhauled Equipment Acceptance Standards

The acceptance standards for Overhauled Equipment are very similar to the standards for new equipment, but normally we are dealing with a company different from the manufacturer. The specifications must be communicated in writing before the start of any equipment overhaul.

When **Balancing** is involved, specify all the particulars required including:

- The speed or speeds at which to balance the rotating element
- Acceptable vibration limits on balance
- Whether a single-plane or two-plane balance is required
- A corrective balance of critical rotating elements
- A trim balance at the plant site when rotating elements are installed in their own bearings (record all vibration data).

ISO 1940

Mechanical Vibration – Balance quality requirements for rotors in a constant (rigid) state

Part 1: Specification and verification of balance tolerances
Part 2: Balance errors

Specifications for rotors in a constant (rigid) state:

- Balance tolerance
- Necessary number of correction planes
- Methods for verifying the residual imbalance
- Recommendations are also given concerning the balance quality requirements for rotors in a constant (rigid) state, according to their machinery type and maximum service speed

Table 1 — Guidance for balance quality grades for rotors in a constant (rigid) state

Machinery types: General examples	Balance quality grade G	Magnitude $e_{per} \cdot \Omega$ mm/s
Crankshaft drives for large slow marine diesel engines (piston speed below 9 m/s), inherently unbalanced	G 4000	4 000
Crankshaft drives for large slow marine diesel engines (piston speed below 9 m/s), inherently balanced	G 1600	1 600
Crankshaft drives, inherently unbalanced, elastically mounted	G 630	630
Crankshaft drives, inherently unbalanced, rigidly mounted	G 250	250
Complete reciprocating engines for cars, trucks and locomotives	G 100	100
Cars: wheels, wheel rims, wheel sets, drive shafts Crankshaft drives, inherently balanced, elastically mounted	G 40	40
Agricultural machinery Crankshaft drives, inherently balanced, rigidly mounted Crushing machines Drive shafts (cardan shafts, propeller shafts)	G 16	16
Aircraft gas turbines Centrifuges (separators, decanters) Electric motors and generators (of at least 80 mm shaft height), of maximum rated speeds up to 950 r/min Electric motors of shaft heights smaller than 80 mm Fans Gears Machinery, general Machine-tools Paper machines Process plant machines Pumps Turbo-chargers Water turbines	G 6,3	6,3
Compressors Computer drives Electric motors and generators (of at least 80 mm shaft height), of maximum rated speeds above 950 r/min Gas turbines and steam turbines Machine-tool drives Textile machines	G 2,5	2,5
Audio and video drives Grinding machine drives	G 1	1
Gyroscopes Spindles and drives of high-precision systems	G 0,4	0,4

NOTE 1 Typically completely assembled rotors are classified here. Depending on the particular application, the next higher or lower grade may be used instead. For components, see Clause 9.

NOTE 2 All items are rotating if not otherwise mentioned (reciprocating) or self-evident (e.g. crankshaft drives).

NOTE 3 For limitations due to set-up conditions (balancing machine, tooling), see Notes 4 and 5 in 5.2.

NOTE 4 For some additional information on the chosen balance quality grade, see Figure 2. It contains generally used areas (service speed and balance quality grade G), based on common experience.

NOTE 5 Crankshaft drives may include crankshaft, flywheel, clutch, vibration damper, rotating portion of connecting rod. Inherently unbalanced crankshaft drives theoretically cannot be balanced; inherently balanced crankshaft drives theoretically can be balanced.

NOTE 6 For some machines, specific International Standards stating balance tolerances may exist (see Bibliography).

Figure 20-10

The standard provides a chart for “easy” look-up (Figure 20-11). Search along the bottom of the chart for the RPM of the machine in question. Then search upwards for the G grade number you wish to apply, and then move across to the y-axis for the “Permissible residual specific unbalance” (g-mm/kg) per correction plane.

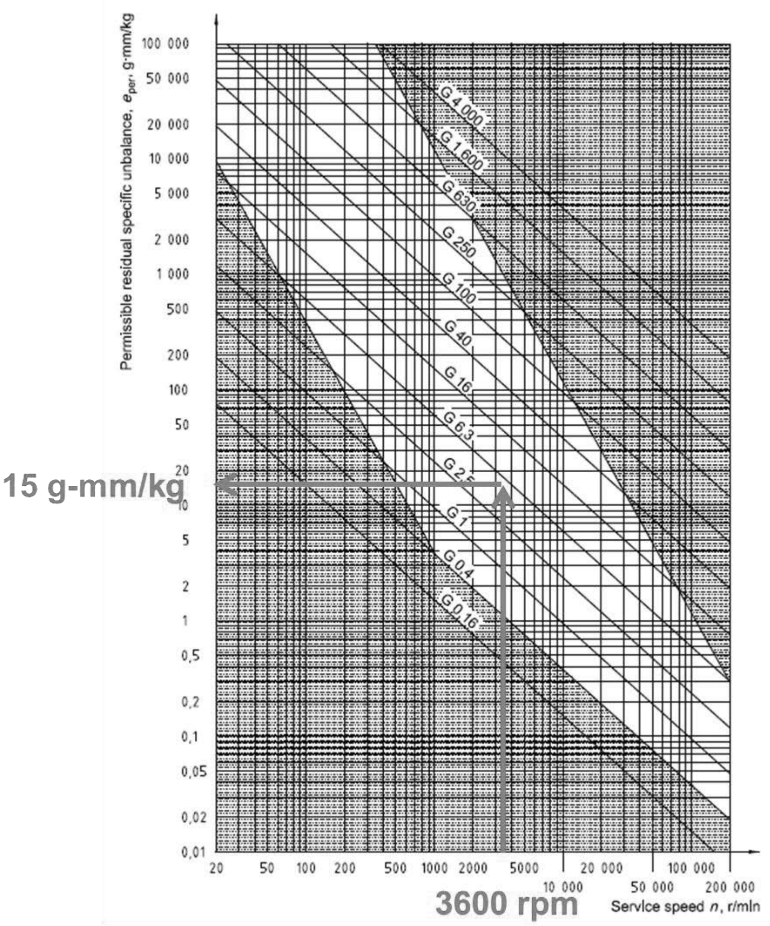


Figure 20-11

Specify Alarm Limits

Develop specifications for acceptable vibration limits after overhaul. Inform vendor's service personnel of limits before they start to overhaul. Base overhaul limits the same for new equipment if the foundations and mountings are problem free.

Below are two Acceptance Alarm Limit examples for fans and gearboxes. These are from the GM specification No. V1.0a-1999

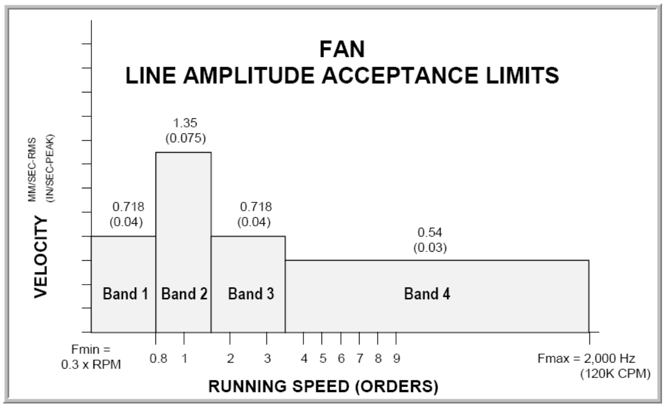


Figure 20-12 - Fan Acceptance Alarm Limits - from GM

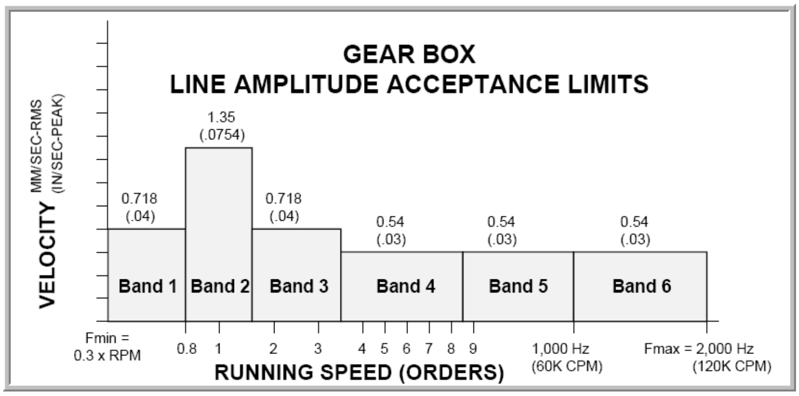


Figure 20-13 - Gearbox Acceptance Alarm Levels - GM

Analyze Overhauled Equipment at Start-up

The start-up procedure should be:

- Run driver uncoupled, record vibration and phase data and compare with previous data of existing machine
- Couple driver and machine, align the machine properly, and run the machine unloaded, while cold, collect, compare, and store vibration and phase data
- Did the overhaul correct the problem or was a new problem introduced?

Procedure for Rewound Electric Motors

- Check rewound electric motors for electrical faults by analyzing vibration at 1x and 2x electrical line frequency.
- Monitor electrical motors on the set stand and the motor repair shop.
- Compare the data to the uncoupled data at start-up.
- Vibration levels should not increase by more than 10%.

ISO 14694: 2003

This International Standard gives specifications for vibration and balance limits of all fans for all applications except those designed solely for air circulation. ISO 14694 describes the measurement locations, vibration amplitudes, and satisfactory/ alarm/shutdown limits.

Table 1 — Fan-application categories

Application	Examples	Limits of driver power kW	Fan-application category, BV
Residential	Ceiling fans, attic fans, window AC	⩽ 0,15	BV-1
		> 0,15	BV-2
HVAC and agricultural	Building ventilation and air conditioning; commercial systems	⩽ 3,7	BV-2
		> 3,7	BV-3
Industrial process and power generation, etc.	Baghouse, scrubber, mine, conveying, boilers, combustion air, pollution control, wind tunnels	⩽ 300	BV-3
		> 300	See ISO 10816-3
Transportation and marine	Locomotive, trucks, automobiles	⩽ 15	BV-3
		> 15	BV-4
Transit/tunnel	Subway emergency ventilation, tunnel fans, garage ventilation, Tunnel Jet Fans	⩽ 75	BV-3
		> 75	BV-4
		none	BV-4
Petrochemical process	Hazardous gases, process fans	⩽ 37	BV-3
		> 37	BV-4
Computer-chip manufacture	Clean rooms	none	BV-5

NOTE 1 This standard is limited to fans below approximately 300 kW. For fans above this power refer to ISO 10816-3. However, a commercially available standard electric motor may be rated at up to 355 kW (following an R20 series as specified in ISO 10816-1). Such fans will be accepted in accordance with this International Standard.

NOTE 2 This Table does not apply to the large diameter (typically 2 800 mm to 12 500 mm diameter) lightweight low-speed axial flow fans used in air-cooled heat exchangers, cooling towers, etc. The balance quality requirements for these fans shall be G 16 and the fan-application category shall be BV-3.

Figure 20-14

Table 4 — Vibration-levels limit for test in manufacturer's work-shop

Fan application category	Rigidly mounted mm/s		Flexibly mounted mm/s	
	Peak	r.m.s.	Peak	r.m.s.
BV-1	12,7	9,0	15,2	11,2
BV-2	5,1	3,5	7,6	5,6
BV-3	3,8	2,8	5,1	3,5
BV-4	2,5	1,8	3,8	2,8
BV-5	2,0	1,4	2,5	1,8

NOTE 1 Refer to Annex A for conversion of velocity units to displacement or acceleration units for filter-in readings.

NOTE 2 The r.m.s. values given in this Table are preferred. They are rounded to a R20 series as specified in ISO 10816-1. Peak values are widely used in North America. Being made up of a number of sinusoidal wave forms, these do not necessarily have an exact mathematical relationship with the r.m.s. values. They may also depend to some extent on the instrument used.

NOTE 3 The values in this Table refer to the design duty of the fan and its design rotational speed and with any inlet guide vanes "full-open". Values at partial load conditions should be agreed between the manufacturer and user, but should not exceed 1,6 times the values given.

Figure 20-15

Table 5 — Seismic vibration limits for tests conducted *in situ*

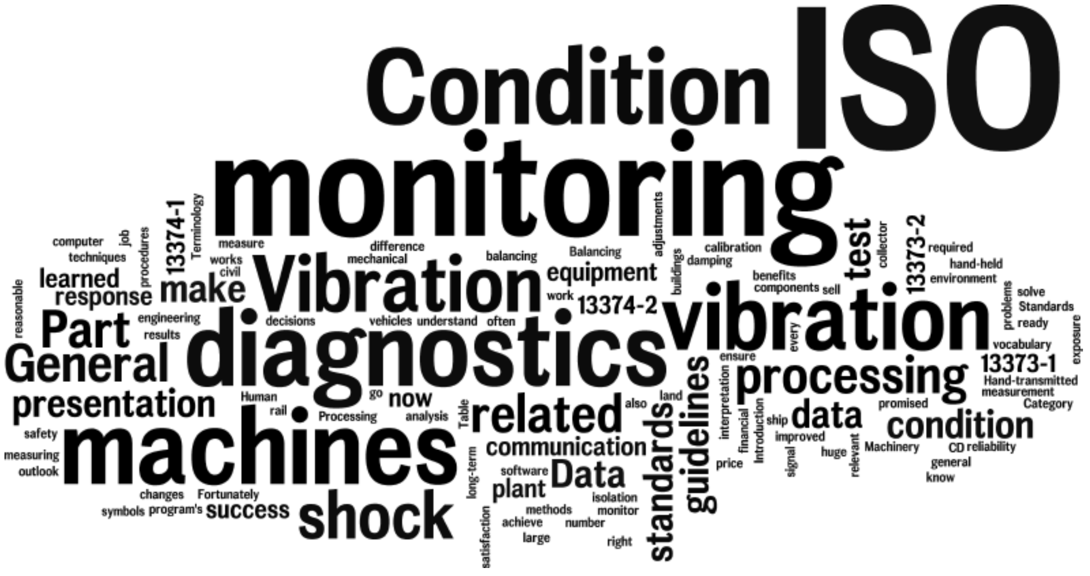
Condition	Fan-application category	Rigidly mounted mm/s		Flexibly mounted mm/s	
		Peak	r.m.s.	Peak	r.m.s.
Start-up	BV-1	14,0	10	15,2	11,2
	BV-2	7,6	5,6	12,7	9,0
	BV-3	6,4	4,5	8,8	6,3
	BV-4	4,1	2,8	6,4	4,5
	BV-5	2,5	1,8	4,1	2,8
Alarm	BV-1	15,2	10,6	19,1	14,0
	BV-2	12,7	9,0	19,1	14,0
	BV-3	10,2	7,1	16,5	11,8
	BV-4	6,4	4,5	10,2	7,1
	BV-5	5,7	4,0	7,6	5,6
Shutdown	BV-1	Note 1	Note 1	Note 1	Note 1
	BV-2	Note 1	Note 1	Note 1	Note 1
	BV-3	12,7	9,0	17,8	12,5
	BV-4	10,2	7,1	15,2	11,2
	BV-5	7,6	5,6	10,2	7,1

NOTE 1 Shutdown levels for fans in fan-application grades BV-1 and BV-2 should be established based on historical data.

NOTE 2 The r.m.s. values given in this Table are preferred. They are rounded to a R20 series as specified in ISO 10816-1. Peak values are widely used in North America. Being made up of a number of sinusoidal wave forms, these do not necessarily have an exact mathematical relationship with the r.m.s. values. They may also depend to some extent on the instrument used.

Figure 20-16

In order to ensure reliable operation you can insist that new machines and overhauled machines meet your requirements.



Chapter 21

ISO Standards

- Overview of ISO standards related to condition monitoring

Introduction

Condition monitoring and improved reliability can make a huge difference to the financial outlook of a plant, and on the work environment (safety, job satisfaction).

We know that, with the right changes, every plant can achieve the promised benefits.

We understand vibration, we have learned how to measure vibration, and we have learned how to solve problems. But what do you do now?

There is your data collector and computer software all ready to go. But which machines do you test? How often do you test them? What do you do with your results? And what can you do to ensure long-term success?

You have to make the decisions, and you will have to monitor your program's success and make adjustments as required.

ISO Standards and condition monitoring

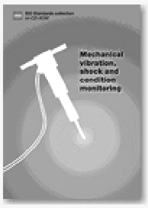
There are a large number of relevant ISO standards

# of standards	Category
7	Terminology, symbols and vocabulary
46	Vibration and shock measuring equipment, calibration, signal processing and general test methods
24	Condition monitoring and diagnostics
12	Balancing and balancing machines
13	Vibration and shock isolation and damping
29	Machinery vibration and vibration measurement of components

Table 21-1

There are also 34 related to “Hand-transmitted vibration and hand-held equipment”, 15 related to “Human exposure to mechanical vibration and shock”, 13 related to “Vibration and shock response of land, ship and rail vehicles”, and 8 related to “Vibration and shock response of buildings and civil engineering works”. Fortunately the ISO now sell a CD with all of these standards for a very reasonable price.

Mechanical vibration, shock and condition monitoring



A CD-ROM compilation of 202 standards and related documents addressing the field of mechanical vibration, shock and the condition monitoring of machines, including vehicles, and structures, such as bridges and buildings. It includes the entire portfolio of ISO/TC 108, Mechanical vibration, shock and condition monitoring, as well as a selection of other related ISO standards.

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More information »

Publication / e-product	Lang	Format	ISBN	Add to basket
Mechanical vibration, shock and condition monitoring *Year of publication: 2010*	English	PDF on CD		452,00 CHF

Figure 21-1

ISO 17359

ISO 17359: Condition monitoring and diagnostics of machines — General guidelines

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Figure 21-2

ISO 13373-1

ISO 13373-1: Condition monitoring and diagnostics of machines — Vibration condition monitoring — Part 1: General procedures

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Figure 21-3

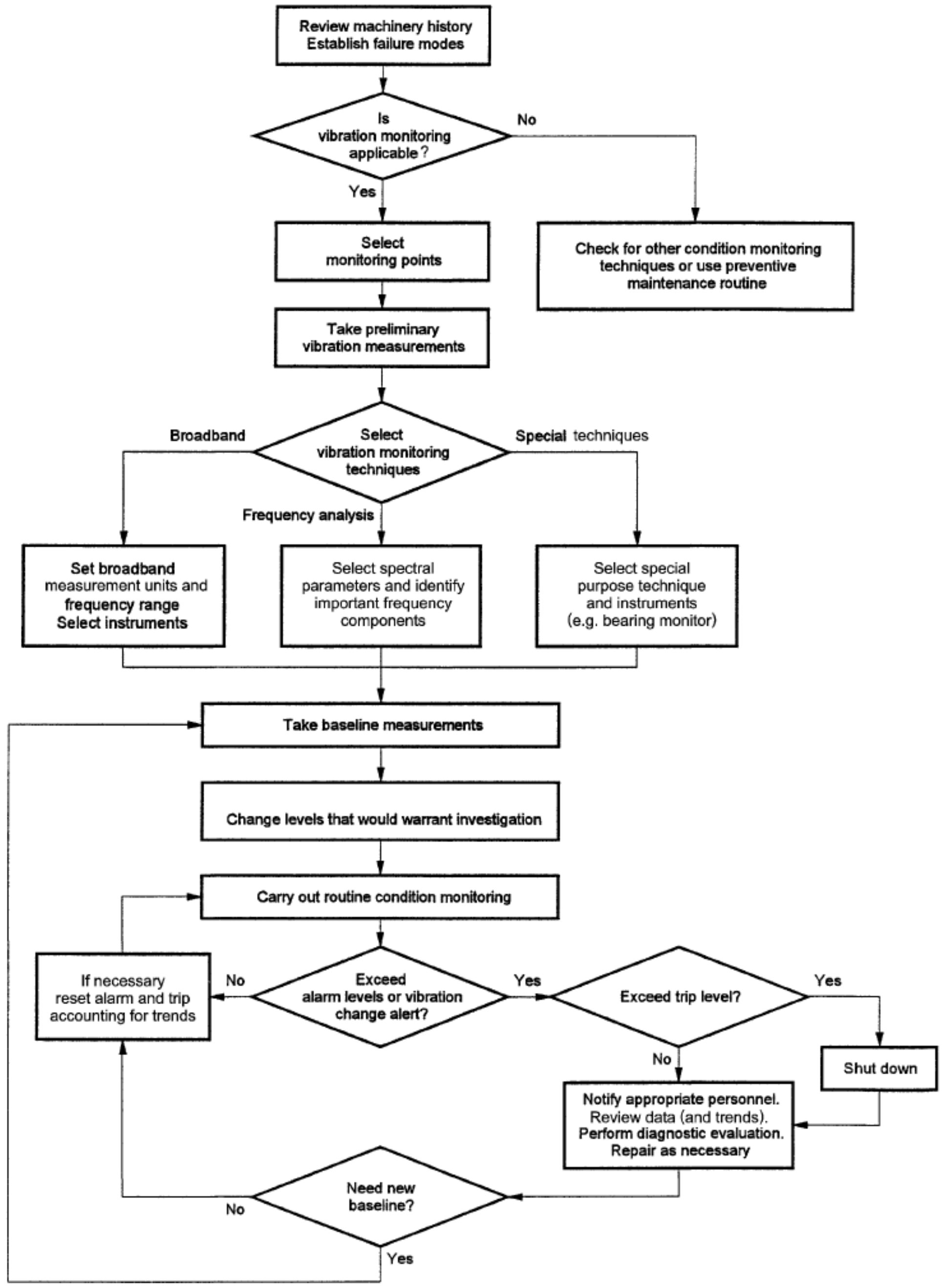

Figure 3 — Vibration condition monitoring flowchart

Figure 21-4

Table A.1 *(continued)*

Machine type	Evaluation parameters	Transducer type	Measurement locations	Direction	Standard reference	See note
Large electric motors and generators with fluid-film bearings	relative displacement velocity or acceleration shaft axial displacement phase reference and speed	non-contacting transducer velocity transducer or accelerometer non-contacting transducer or axial probe eddy current/inductive/optical transducer	shaft at each bearing housing thrust collar shaft	radial ± 45° radial X and Y axial Z radial	ISO 7919-3 ISO 10816-3, ISO 10816-5	
Medium and small motors with rolling element bearings	velocity or acceleration phase reference and speed	velocity transducer or accelerometer eddy current/inductive/optical transducer	each bearing and motor housing shaft	radial X and Y, axial Z radial	ISO 10816-3	5
Compressors – package centrifugal (four-poster), with fluid-film bearings and rigid housings	relative displacement velocity or acceleration phase reference and speed	non-contacting transducer velocity transducer or accelerometer eddy current/inductive/optical transducer	each bearing, pinion housing and ball gear shaft at each bearing or gear mesh each shaft	radial ± 45° radial X and Y radial	ISO 7919-3 ISO 10816-3	
Compressors – centrifugal process	relative displacement velocity or acceleration shaft axial displacement phase reference and speed (if driving through gear box)	non-contacting transducer velocity transducer or accelerometer non-contacting transducer or axial probe eddy current/inductive/optical transducer	shaft at each bearing bearing housings thrust collar or shaft end shaft	radial ± 45° radial X and Y axial Z radial	ISO 7919-3 ISO 10816-3	6
Compressors – screw-type with two interlocking shafts, fluid-film bearings	relative displacement velocity or acceleration shaft axial displacement phase angle of each shaft	non-contacting transducer velocity transducer or accelerometer non-contacting transducer or axial probe phase reference transducer	shaft at each bearing bearing caps and compressor housing thrust collar each shaft	radial ± 45° radial X and Y, axial Z axial Z radial	ISO 10816-1	

Figure 21-5

ISO 13373-2

ISO 13373-2: Condition monitoring and diagnostics of machines — Vibration condition monitoring — Part 2: Processing, analysis and presentation of vibration data

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Figure 21-6

ISO 13374-1

ISO 13374-1: Condition monitoring and diagnostics of machines — Data processing, communication and presentation — Part 1: General guidelines

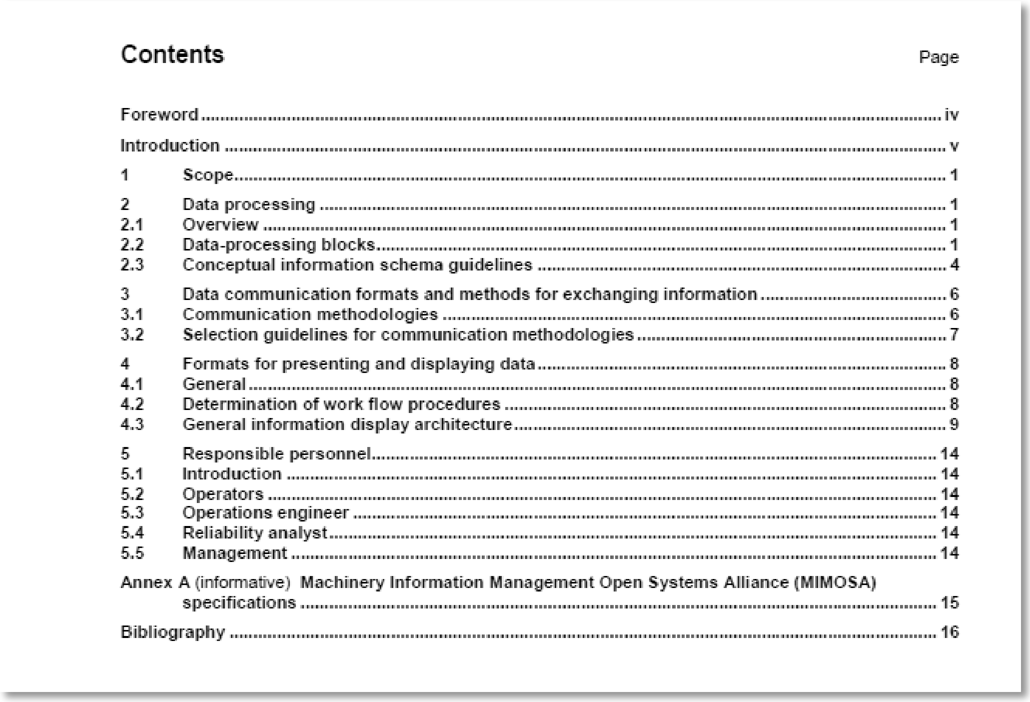

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Figure 21-7

ISO 13374-2

ISO 13374-2: Condition monitoring and diagnostics of machines — Data processing, communication and presentation — Part 2: Data processing

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Figure 21-8

ISO 13379

ISO 13379: Condition monitoring and diagnostics of machines — General guidelines on data interpretation and diagnostics techniques

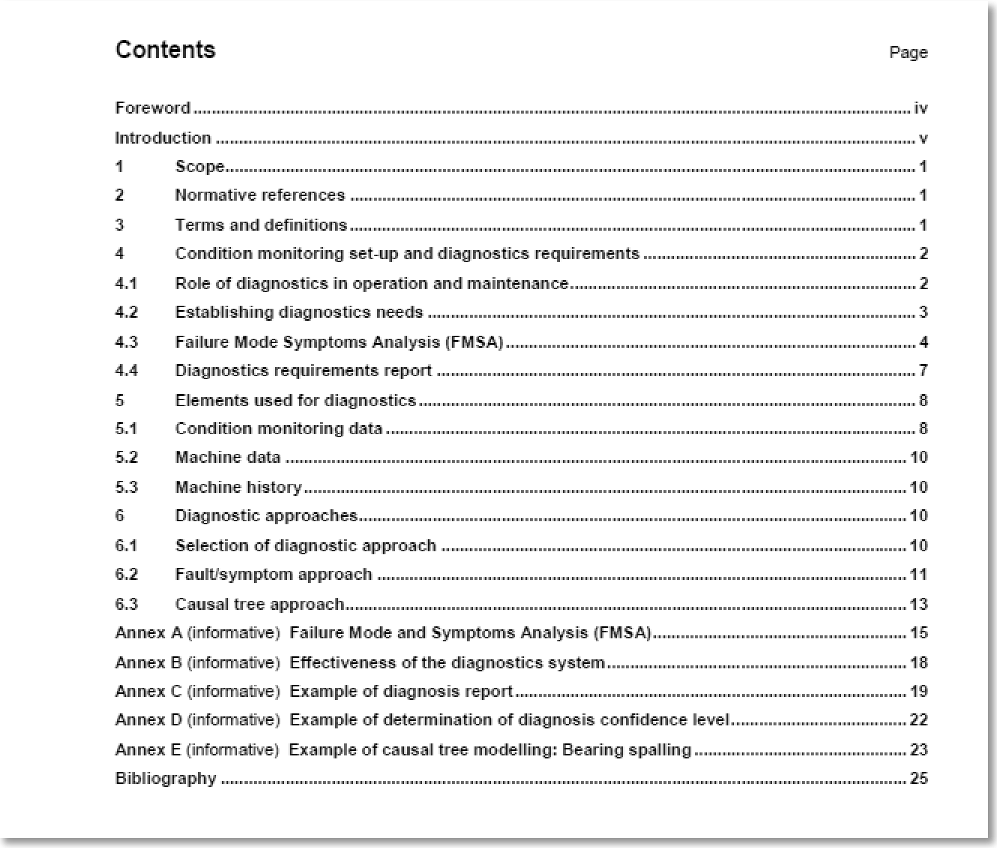

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Figure 21-9

Chapter 22

Maintenance Practices

Objective: *Describe two characteristics of the four maintenance practices and how each fits in a plant environment.*

Performing vibration measurements on a hot noisy machine can make one wonder what this is all about and why it is necessary. Understanding why and how-to can make the job so much more rewarding. This chapter provides background information regarding the value and place for vibration monitoring. An overview of the four maintenance practices compares the strengths and weaknesses of each.

The Four Maintenance Practices:

- Breakdown maintenance
- Preventive maintenance
- Predictive maintenance
- Precision maintenance

Why Do We Do Maintenance?

Before we learn about condition monitoring technologies, it is important to consider the goals of a maintenance program. Before acquiring tools to solve a problem it is important to understand what the problem is that you want to solve. So let's begin with the question: "Why do we do maintenance?"

Figure 22-1

Is the purpose to fix machines that are broken or damaged? Does this describe the real purpose of maintenance? Is there a more forward looking reason to do maintenance, perhaps as a way of preventing machines from failing and therefore avoid the consequences of failure? Is there a difference between these two views of maintenance?

Do we do maintenance simply to comply with regulations or for insurance reasons? Do we overhaul machines because they are "due for overhaul" whether they need it or not?

Do we do maintenance in order to meet production goals, increase uptime, increase plant efficiency and thus increase profitability? When you are replacing the bearings in a motor, are you thinking of plant profitability?

The main point here is that we need to understand our goals first; then we can work towards those goals and measure the degree to which we have been successful in meeting them or not.

Two views of maintenance strategies

There are two ways to look at maintenance strategies. We can try to improve our maintenance strategy, including process, procedures and technology in order to do maintenance more efficiently. Another view is that we can do maintenance better in order to produce our product more efficiently. These are not mutually exclusive goals, however, the way we implement our strategies and measure our success will depend on how we view the goal.

Machines are supposed to last a lot longer than they actually do. If we can do maintenance more efficiently then the following can be avoided:

- Catastrophic failure
- Secondary damage
- Additional spare parts costs
- Unnecessary overtime
- Injury to staff

When these are eliminated then downtime is reduced and costs are reduced.

What Costs Can Be Reduced?

Maintenance costs run deep in an organization. From the spares that need to be kept, to the overtime maintenance hours, costs mount up. Loss of production is also a huge opportunity cost to an organization, and secondary damage to equipment when failure occurs can also be substantial.

In addition, reduced energy consumption can help a company's bottom line, and improved product quality and manufacturing times go straight to profitability.

Figure 22-2

Every industry is different, and different organizations have their own priorities and ways to measure their level of success. You must understand your organization's priorities so that you can do your part to help meet the goals.

In some batch production processes, keeping equipment running during that process is paramount. A failure can cause huge production costs and the expensive waste of production material.

In the fishing industry, fishing fleets may have a very short season. If their factory ships are inoperable for any reason during that season, their lost revenue can grow into the millions.

In the power industry, loss of generation can not only affect revenue, but also incur penalties.

Other entities such as the US Navy have special criteria. In addition to regular maintenance costs, their concern is "ship readiness" - their ability to get under way when duty calls.

It is important to understand the special situations within your industry in order to make more informed maintenance calls.

Since it is important to detect a problem before a failure occurs, we must understand why machines fail in the first place.

Why Do Machines Fail?

The source of machine failure can start on the designers drafting board and end with poor maintenance practices and operating conditions. The way the machine is manufactured, the way it is installed, and the way it is overhauled all contribute to the ultimate life of the machine.

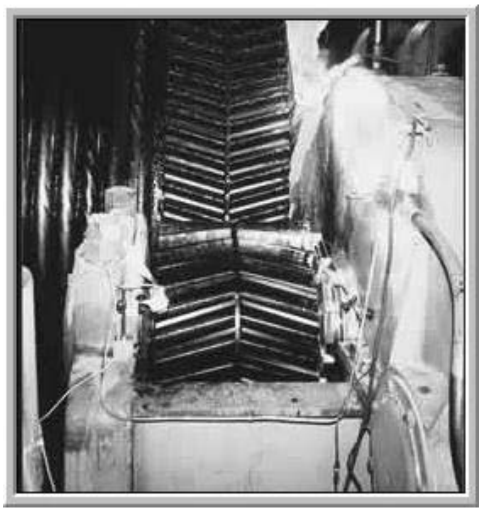


Figure 22-3

Failure starts with the initial specification and purchase, balancing and alignment, routine maintenance, lubrication, overhaul procedures and acceptance testing.

You may not have control over all of those steps, or any of those steps; however, an understanding of the potential problems can lead to future change.

It is not the intent of this course to teach better ways to run a maintenance department, or a company for that matter, it is simply to provide the bigger picture so that opportunities can be identified to improve the reliability of equipment.

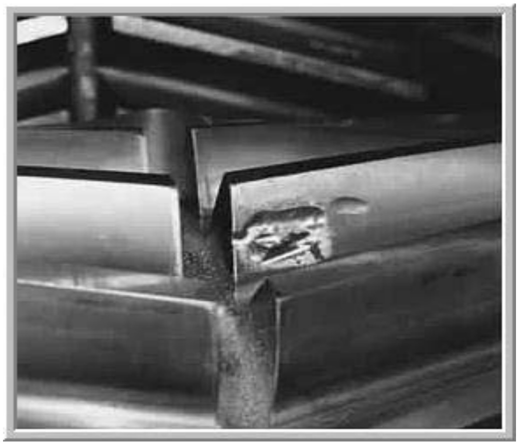


Figure 22-4

While the business world grows ever more competitive, often resulting in fewer staff, reduced budgets and greater production demands, the requirement to operate the plant more efficiently becomes that much more important, and that much more difficult. The aim is to improve equipment reliability, reduce maintenance costs, reduce energy usage, and to improve product quality.

Such changes will not happen overnight, and they will not happen unless everyone is onboard with the goals. From the person who does the motor rewinds, to the person who signs the checks, everyone needs to understand the benefits and requirements of precision maintenance.

It sure would be easier if all we had to do was buy a vibration data collector and some computer software to solve all our problems. It would be even easier if the machines never failed in the first place.

How Can You Achieve the Best Results?

The reality is that it takes a change in attitude throughout an organization, from the top down, before the full savings can be achieved. If management does not understand the benefits or what it takes to achieve them, the benefits will not be achieved.

Figure 22-5

If every person involved with condition monitoring does not understand and believe in the goals, success will be limited. And if other staff in operations, production, purchasing and engineering do not understand and believe in the goals, again success will be limited. It is no easy task to cause a wholesale change in company philosophy, but everyone must do what they can.

We can work to cooperate and inform others of our goals - bring people onboard. Nothing guarantees failure surer than the generation of antagonism between maintenance, operations and production. Over and over we have seen the same situation. A fault is detected, a work order is generated and now it is time to open the machine. One group of people hopes the machine has a real problem, to prove their point. Another group hopes there will not be a problem found, to prove their own point. These situations, all too common in industry, cause stress and greatly reduce the success of any program.

If everyone plays a part, then there is a greater likelihood of success. The solution is a change in philosophy that begins with upper management - and education. You and your colleagues should work to inform people in other departments about what it is you do, and what the benefits are, and how they can play a part.

Overall Equipment Effectiveness (OEE)

We have been discussing how we can do maintenance better to make the maintenance process itself more efficient. This includes reducing spare parts inventories, planning maintenance, reducing overtime etc. Now let's consider the second view of maintenance which is to increase plant profitability. We can think of it this way: The plant exists to create a product (or to provide a service such as in the case of the navy) and the machinery in the plant exists to create this product also. If the machines are down and the plant is down, then the product is not being produced.

Overall equipment effectiveness (OEE) gives us a way to look at the plant through this lens by asking:

- How much product could we produce in the ideal case?
- How much product are we actually producing?
- What is causing us to produce less than the ideal amount?
    - Unplanned downtime
    - Slow downs
    - Poor quality product / waste
- How do these losses affect our profitability?

This forces us to consider the purpose of the machines rather than the machines themselves and it gives us a clear way to measure the success of our maintenance strategy.

OEE is based on three measurements:

Availability (A) = actual production time / planned production time
World class = 90%
Performance (P) = actual run rate / planned run rate
World class = 95%
Quality (Q) = good product / total product
World class = 99.9%

OEE = A x P x Q
World class = 85%, a typical plant = 60%

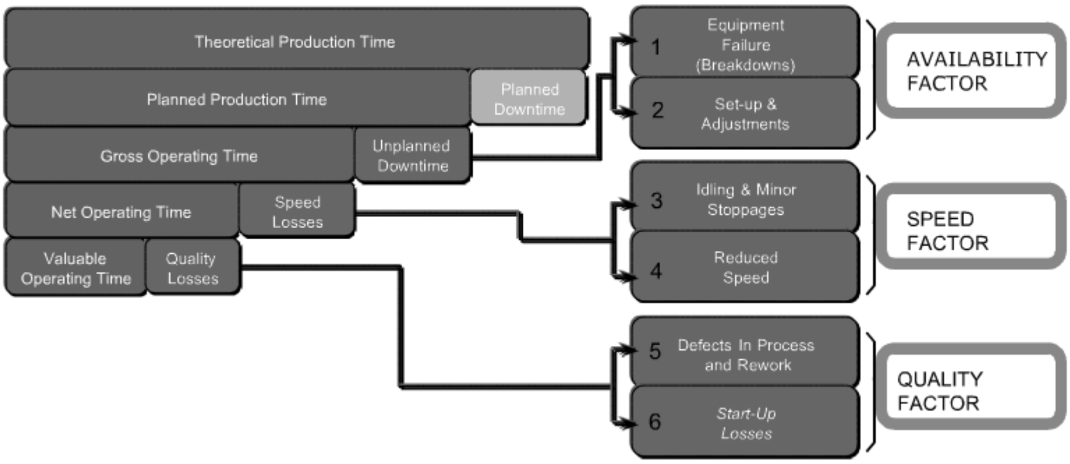


Figure 22-6 Understanding OEE components

Let's take a quick look at the plant in Figure 22-7. Towards the bottom left we can see the three components of OEE in blue and at the far right we see the proftability and earning power of the plant.

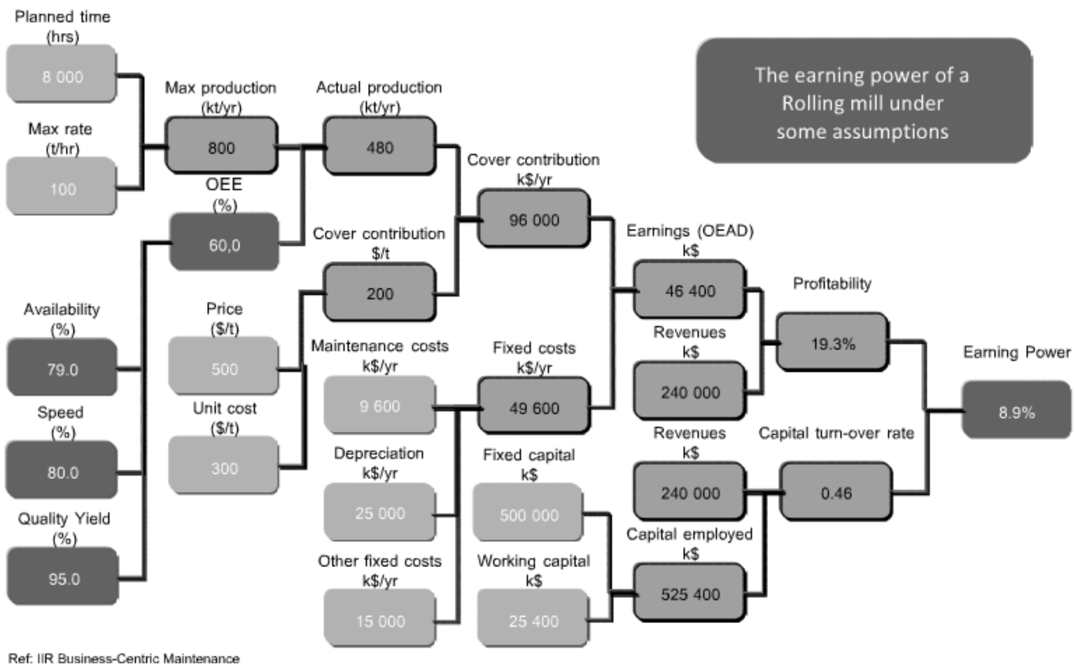


Figure 22-7 A plant in normal operation

In Figure22-8we have increased availability and production speed a bit in order to increase OEE by 5%, now looking at the earning power and poritability, we can see that this resulted in $7.2 million in additional earnings. The question then is how much money would we have had to invest in maintenance in order to achieve this 5% gain in OEE? How much does it cost to implement a world class vibration monitoring program?

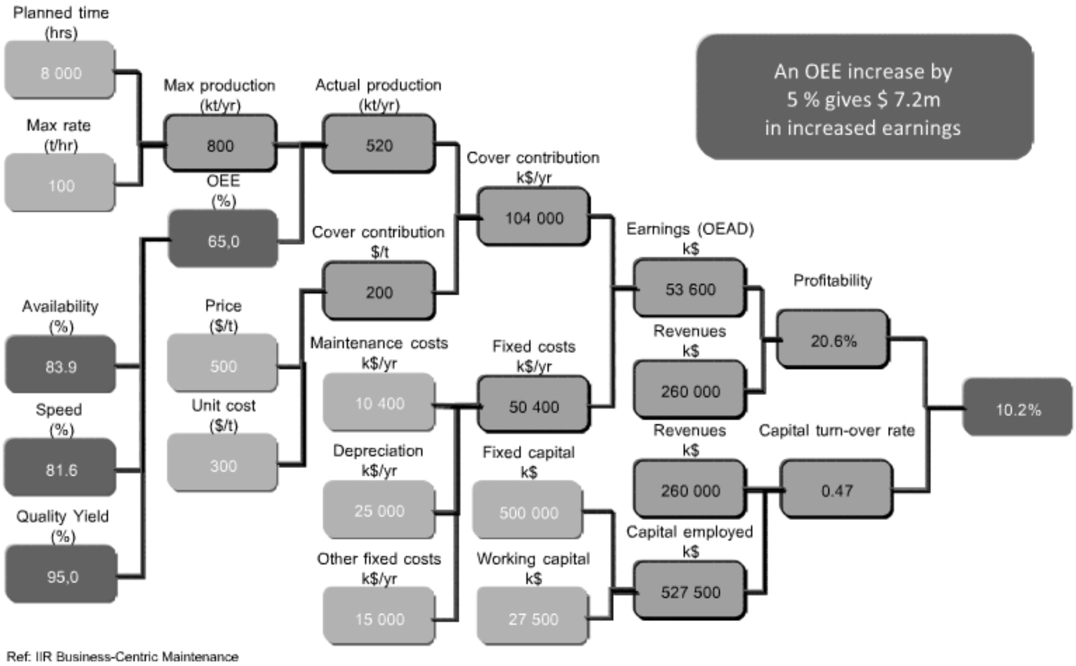


Figure22-8 A 5% increase in OEE gives $7.2 million in profits

The goal here is not to make you an expert in OEE, it is to demonstrate a point. Often we look to save money by cutting budgets. If we want to save money on maintenance, we fire a few people or we neglect to do work that needs to be done. This may result in a short term gain, but long term the machines will fail unexpectedly and it will cost us a great deal more to deal with it then. The author has observed real cases where plants attempt to save money on maintenanmce by not doing maintenance, then the genius who thought up that idea leaves to go work somewhere else and the people who remain have to deal with the consequences a year or two later when the plant starts falling apart!

Variable	Original	OEE Improvement	Price Increase	Maintenance Cost Reduction
Availability, %	79%	83.9%	79%	79%
Speed, %	80%	81.6%	80%	80%
Yield, %	95%	95%	95%	95%
OEE, %	60%	65% (+5%)	60%	60%
Price ($/t)	500	500	515 (+3%)	500
Maintenance Cost (k$/yr)	9,600	10,400	9,600	2,600 (-73%)
Profitability, %	19.3%	20.6%	21.7%	22.2%
Earning Power, %	8.9%	10.2%	10.2%	10.2%

Ref: IIR Business-Centric Maintenance

Figure 22-9 Increase OEE, increase product price or reduce maintenance budget?

We also talked about two views of maintenance. We can look at ways to make the maintenance work more efficient and save money that way, or we can look at maintenance in terms of increasing OEE and making the plant itself more efficient. Figure 22-9shows three strategies to increase the earning power to 10.2%. The first option is to increase the maintenance budget to 10,400 in order to achieve a 5% improvement in OEE. The second option is to raise the sale price of the product by 3%, the third option is to reduce the maintenance budget by 73% to 2,600.

When you look at it this way, it seems pretty obvious that saving money on maintenance will only get you so far – not to mention the future consequences of not doing maintenance well! It is much wiser to invest in increasing OEE. Therefore, one way to think about maintenance and maintenance strategies – and to measure the success of your efforts- is to set goals for increasing OEE and base your successes or failures on that indicator.

Breakdown maintenance

For many years (and in many plants still today), the philosophy has been to simply run the plant until a machine failed, deal with it and get up and running once again. If machines failed, they were repaired or a spare was used. Little thought was given to improving equipment reliability or predicting failures. The maintenance department was a huge cost sink, and that was considered a standard part of running the business.

Figure 22-10 For many years the maintenance philosophy has been just to keep the plant running and deal with the failures when they occur.

More recently the philosophy has changed. Now organizations recognize that it is worth the investment of time and money to change maintenance practices to be more proactive and to work to improve equipment reliability. Great cost savings have been realized by this approach, often termed “proactive maintenance”, “precision maintenance”, “reliability centered maintenance” or “reliability engineered maintenance”.

Figure 22-11 New philosophy...it is worth the time

There are a number of approaches an organization can take to maintaining rotating machinery, and often an organization will practice a number of different philosophies at once; allowing some machines to fail while being proactive about others. Understanding these approaches is very important.

Breakdown Maintenance

Breakdown maintenance is also known as "run to failure", "reactive" and "hysterical" maintenance among other things. It is the practice of allowing machines to fail rather than taking any preemptive action.

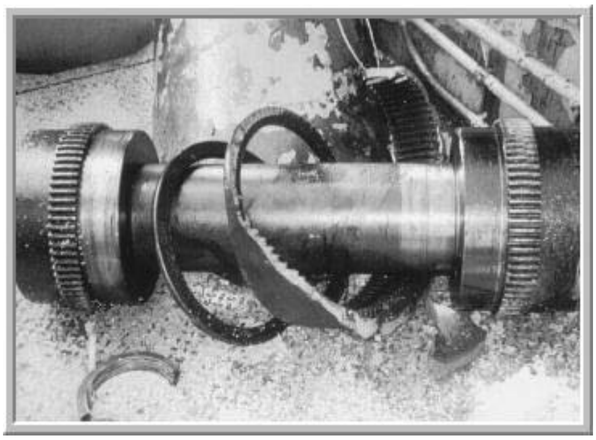


Figure 22-12 Run to Failure maintenance practices leads to very high maintenance costs.

The philosophy is:

"Fix it when it breaks"

This practice can lead to very high maintenance costs. Secondary damage to the machine (the costs that occur as a result of the primary failure), production downtime, the cost of keeping spares, and overtime labor are just a few of the significant costs that result from this approach.

Control is lost when breakdown maintenance is employed. This is why it is often termed "reactive" maintenance. The plant reacts or responds to equipment failures rather than anticipating them, planning for them or avoiding them altogether. Component failure can occur at any time, thus affecting production and safety. If a trained ear detects a problem before failure occurs, secondary damage may be reduced, but there will still be loss of production when the repair is performed, perhaps due to scheduling conflicts or lack of spare parts.

Figure 22-13 Secondary damage is often more costly than the failed component.

Run to failure maintenance was standard practice through the 1950's and surprisingly this approach is still followed in many plants today.

A plant in reactive mode

When a plant is primarily using run to failure maintenance we can say they are caught in a reactive mode – they are always reacting to problems and situations. When visiting a plant in this mode one will find it to be dirty and disorganized. Employees will complain that they are over worked or always “too busy” to change how they do things. There will likely be a high rate of injuries, spills, fires and other damage caused by unexpected failures.

Because the plant “reacts” to problems instead of anticipating them and preparing for them, they are always behind and trying to catch up to meet production demands. They are not running the plant, the plant is running them and this is a frustrating environment to work in. Most of us prefer to work in a predictable environment in which we feel we are in control of the situation. A plant in reactive mode is clearly not that type of environment; Morale will likely be low.

There is a correlation between this maintenance strategy and psychology. Perhaps you know people in your life who are always stressed out, busy and over extended. They seem to go from one crisis to another, never getting ahead of the game.

Although breakdown maintenance is by far the least sophisticated of the maintenance strategies, there are still situations in which it is a viable option.

When to use breakdown maintenance

Machine is not critical, does not impact production, is highly redundant, is inexpensive to repair or replace and is unlikely to cause collateral damage, injury or other problems if it fails.

It is not cost effective to monitor the machine, there are no benefits of preventive maintenance actions

We will learn later in this section that a facility will employ each of the maintenance strategies to some extent depending on the asset in question. This is to say that there are in fact situations where breakdown maintenance is a viable option. As an example, consider your desk fan. If the fan fails, you will simply buy a new one. Your desk fan is not critical, it is easy to replace, it is inexpensive, it is unlikely to cause collateral damage or injury when it fails and it would not be cost effective to use some technology to monitor it. The light bulb in your desk lamp is another example.

There are also cases where a machine is being monitored and is known to be at risk of failure but a decision is made to continue to run the machine anyway in order to meet production demands. The hope in this case is that the machine will survive until just after a critical production period and the machine is run because the cost of repair (due to production demand as well as labor and parts costs) may exceed the risk of failure. To a large extent it all comes down to risk management.

Preventive Maintenance

Preventive maintenance is also known by a variety of terms: “planned maintenance”, “scheduled maintenance”, “calendar-based maintenance”, “historical maintenance" and possibly other terms.

The theory is that the life of a machine is limited, and the probability of failure increases as the machines age, so we will perform maintenance before the machine fails, thus avoiding failure and extending its life.

Theory: Perform regular overhauls so that the machine will not fail

The challenge is to estimate the life of a machine, and then perform the overhaul before it fails. Risk is balanced against cost. If the maintenance is put off too long, the machine may fail. If the overhaul is performed too early, it becomes too expensive, in labor, lost production and parts.

If the probability of failure is graphed with respect to time, one might expect to get a curve like Figure 22-16. The assumption would be that for a period of time the probability of failure remains low. At some point in the future the parts begin to wear and fatigue, so the probability of failure increases. See Figure 22-16.

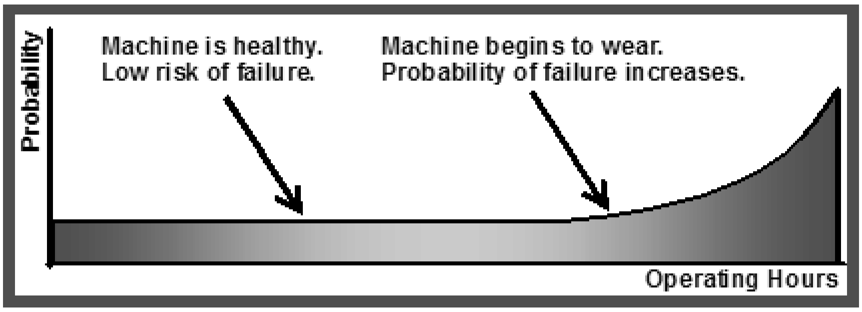


Figure 22-16 The assumption is that the probability of failure remains low for some period. But what about infant mortality?

But wait. What about "infant mortality"? The unfortunate reality is that there is a high probability of failure immediately after an overhaul due to poor lubrication (too much or too little), incorrect parts being installed, parts installed incorrectly, poor alignment and balance, and a host of other reasons.

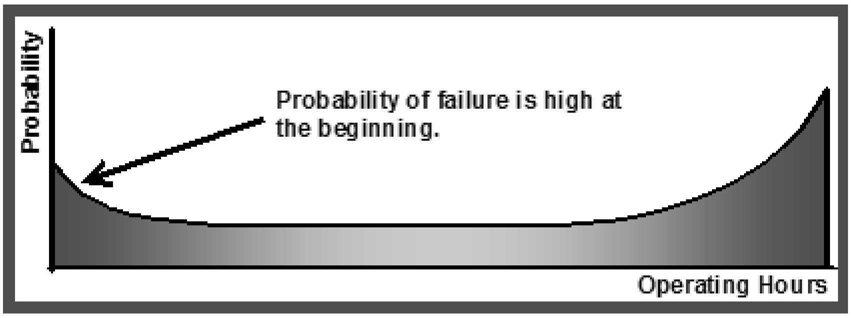


Figure 22-17 Probability of failure including Infant Mortality risks

Infant Mortality: - Machines often fail soon after overhaul. This new curve is called the “Bathtub curve”

Now the curve has to be adjusted to reflect "Estimated Life" versus "Probable Life". The probability of failure increases before it "should" fail, so the length of the flat section needs to change.

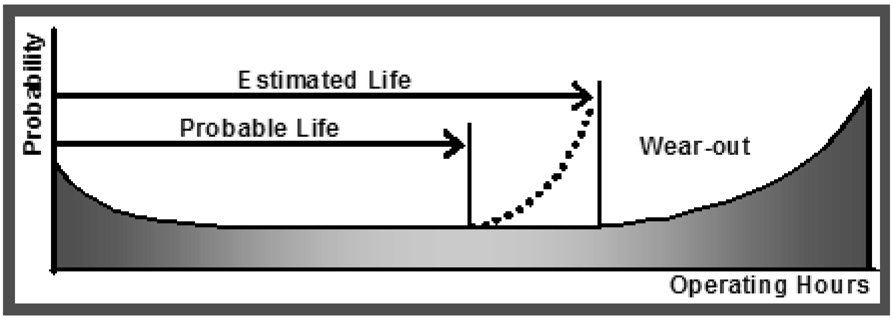


Figure 22-18 Probable Life and Estimated Life added to graph.

The reality of a failure rate is that it is random. Failure of a component occurs after a time that cannot be predicted.

The task is to schedule the maintenance within the "probable life" period. Unfortunately, we do not know what that period is, or how quickly the machine may fail once the wear-out phase begins.

So, we are left with a situation where we will typically perform maintenance far too frequently; performing work on machines that are running just fine. And to make matters worse, some machines will still fail when we expect the probability of failure to be low.

If maintenance is scheduled too frequently, the risk of Infant Mortality is increased. Plus the maintenance costs are higher because of the additional labor and parts required to perform the maintenance. And while the machine is being maintained, it is out of service resulting in reduced production.

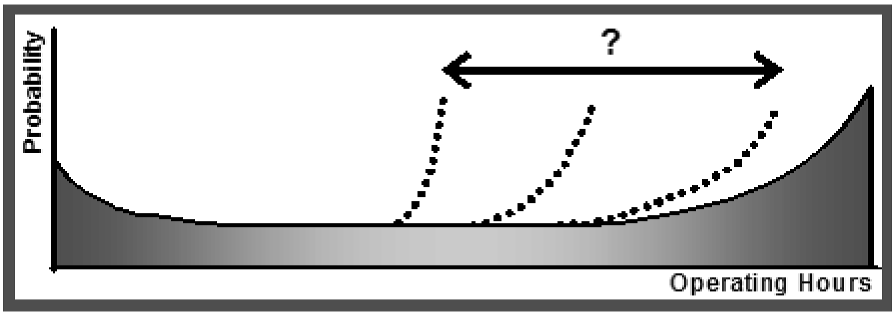


Figure 22-19 It is still a guessing game to determine when the machines should be overhauled. 2 months? 18 months? 6 years?

If maintenance is scheduled too infrequently, then there is a higher risk of unplanned downtime and a higher risk of catastrophic failure. A new curve of health versus time could be created as shown in Figure 22-20. The infant mortality section is added to the curve (and a machine should be closely monitored during this phase), but what about further down the track? At some point the health will begin to deteriorate, but the rate at which it degrades towards failure is unknown.

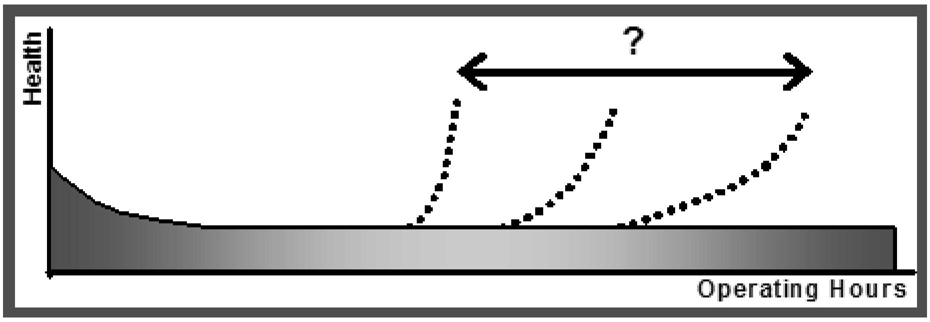


Figure 22-20 The new curve includes the Infant Mortality curve but the rate of degradation is unknown.

A study was performed some years ago (originally by United Airlines, but the results have been recorded by other organizations since) to examine exactly how the probability of failure does change over time. Two important facts came out of the study:

- The first is that the curve does not always follow the "bath-tub" shape portrayed thus far. In fact, the classic bathtub shapes accounted for only 6% of the machines. In most cases (68%) the shape was flat after the initial infant mortality period.

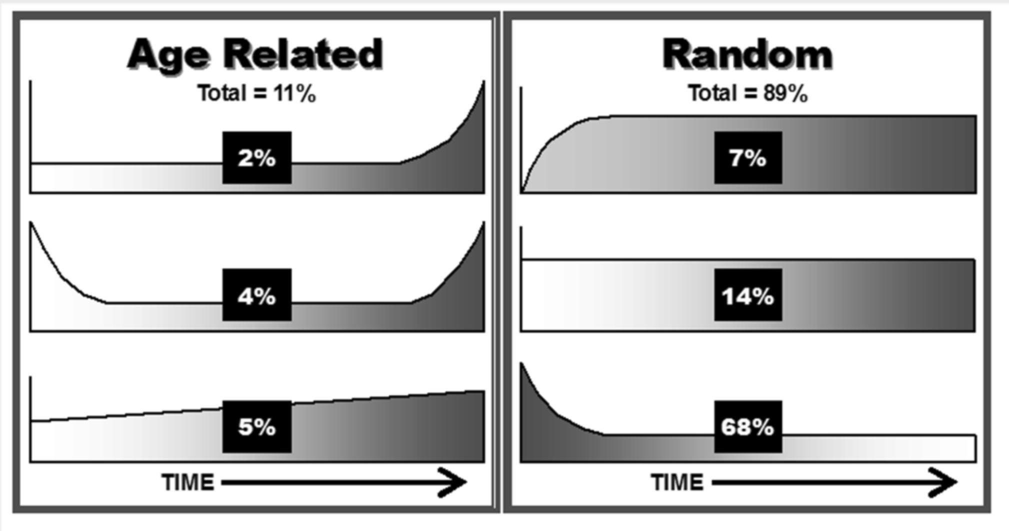


Figure 22-21 Failure curves for all plant equipment.

- The second point (which explains the flat curve) is that only 11% of failures were age related, while 89% were random. That means that they were as likely to fail after two months as they were in 22 months. Therefore, the concept of calendar-based maintenance was flawed.

A plant in preventive mode

When preventive maintenance is the dominant maintenance philosophy in a plant, machines will still fail due to the fact that 89% of failure modes are random, meaning that preventive maintenance actions will not catch these problems.

Because machines are still failing, the plant is still in reactive mode, however, they also have a huge amount of preventive maintenance work scheduled that they have to do. They probably won't be able to handle it all, so there will be stress, overtime and probably some resentment towards planned maintenance tasks that seem unnecessary (in light of all of the problems they have to react to) – these planned maintenance tasks are also costly in terms of manpower, spare parts and downtime – even if it is planned downtime.

That said, this philosophy still has a place in the overall maintenance strategy of the plant.

When to use Preventive Maintenance

- Machine or component wears out or degrades in known amount of time.
- Risk of infant mortality is less than risk of failure
- Cost of Preventive Maintenance action is less than the cost of failure.
- Condition monitoring is not a viable option

In a real plant, and even if the context of a single machine, all of the maintenance strategies will be used to some extent. Preventive maintenance is most applicable in cases where one really does know how long a machine or machine component will last before it begins to degrade. An example of this is a pump that is subject to a corrosive material or a cutting tool that can only cut a certain number of parts before the blade becomes dull and needs to be replaced. There are machines and machine components that do have known failure rates and where this maintenance philosophy is viable.

The next item to take into consideration is the risk of infant mortality. Sometimes it is best to leave well enough alone and not “fix” things that are not broken. In fact it is human intervention itself that is often the root cause of machine failures!

The cost of doing the preventive maintenance action should be lower than the cost of failure or the cost of replacing the machine. For example, it costs more to replace a light bulb based on its expected lifetime then to let it go to failure. Lastly, preventive maintenance is a viable strategy in cases where condition monitoring is not an option. For example, in our automobiles we change the oil every so many miles whether the oil needs to be changed or not (this is an example of preventive maintenance). In this case it is less expensive to change the oil then to test the oil to see if it is degraded or not. We also change the timing belt after so many miles / kilometers (another example of preventive maintenance) because belt failure could result in collateral damage, namely it could seriously damage the car’s engine. The automobile manufacturers presumably have enough data on the mean time between failures (MTBF) of timing belts to propose a conservative preventive maintenance schedule.

Predictive Maintenance

Predictive Maintenance is also known as “Condition Based Maintenance.” The philosophy is often described by the phrase:

“If it ain’t broke, don’t fix it”

Rotating machinery will usually give warning signs before failing. The vibration level and pattern will change. The temperature of some parts will increase. The wear in lubricated surfaces can be detected via the lubricant. The performance can change. The motor current will change. And there are other tell-tale signs.

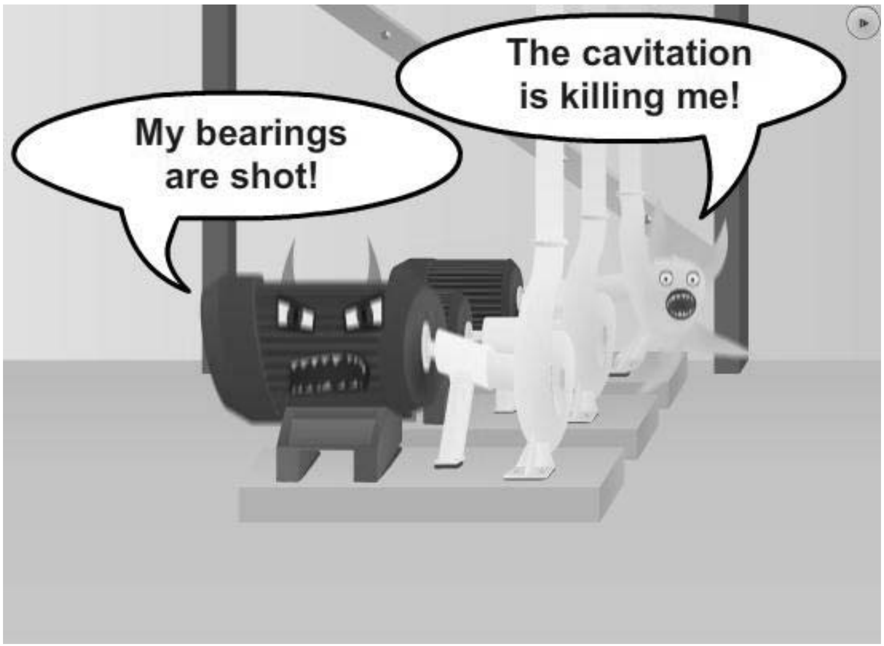


Figure 22-23 Machines warn you when they are about to fail

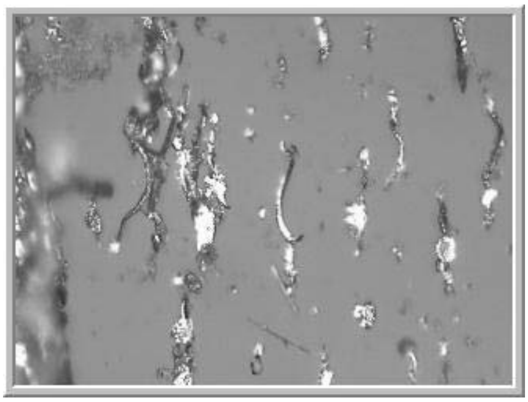


Figure 22-24 Wear Particles in Oil Sample

Various technologies are employed to monitor machines for signs of problems or wear, and then, when observed, the required maintenance is planned before the risk of failure is too high. This allows us to run the machine as long as possible before conducting maintenance and reduces the overall cost of maintaining the machine by removing unnecessary maintenance actions.

This process allows the repair to be made at a time that suits production and maintenance schedules.

In an Ideal World, the maintenance costs are reduced, and profit is increased because there is no downtime, no catastrophic failures, no secondary damage, reduced parts inventory and all work is planned.

In theory everything just stated is possible. However, that only assumes that all machines are fully monitored, and that all failures follow a convenient pattern giving us a few months' notice before failing.

The truth is that it is difficult and expensive to monitor every machine this way, and machines do not always give as much warning as you would like.

Condition Monitoring

The art of predictive maintenance is to monitor the machine with the appropriate technologies, frequently enough to detect the anticipated failure modes. Not unexpectedly, it is a financial issue. Financial interests must be balanced against the risk of failure.

It is a financial issue: Cost of monitoring versus risk of failure

A question needs to be answered… What are you willing to do to ensure that you know when a machine may fail? Each machine needs to be assessed and an appropriate strategy put in place. It is important to choose the correct technologies, and choose the correct monitoring rate (which may change during its life).

It may mean that a "run to failure" philosophy is implemented on some machines, because it is not economically justified to perform the condition monitoring.

Risk needs to be balanced against cost. It may be determined that certain machines will not be monitored at all. At the other end of the spectrum it may be necessary to install permanent monitoring systems, designed to keep a 24 hour watch on the critical machines.

It requires looking forward, assessing the obstacles ahead and making the right decisions. All the fancy electronics alone is not enough.

It also must be understood that condition monitoring only tells us the condition of the machine and allows us to plan our repairs. It does nothing to increase the reliability of the machines or extend their lives.

> *Monitoring the machines does not improve reliability – you do not stop them from failing, you do not extend their life, you are simply predicting their failure.*

If purchasing procedures are changed, if the machines are operated and maintained so that they became more reliable, with a longer life, then maintenance costs would be lowered even further and plant profitability would be higher. This is to say that condition monitoring is just part of the solution.

> *Condition monitoring helps you to look into the future – but you still have to take the appropriate action*

It is one thing to monitor a machine and understand its condition; it is another thing to change your maintenance processes and procedures in order to act on this information. One common cause of failed programs is exactly this disconnect. The vibration analyst monitors a machine every three months and he knows it has no mechanical problems but then a PM action comes up and someone else replaces the bearings in the machine! The opposite case is also common, where the analyst detects a problem but there is no procedure to act on the information, so the machine is run until it fails. In either case, one must re-write PM actions to take into account the existence of the condition monitoring technologies such that machines are repaired based on condition not the PM schedule. Predictive maintenance is about more than just adopting condition monitoring technologies, it requires a change in maintenance philosophy and procedures such that one basis repair decisions on the condition of the machine rather than the calendar. These decisions will also impact spare parts inventory strategies, worker scheduling, planned downtime work packages etc.

Figure 22-25 Seeing into the future is not enough, you also have to change course!

Plant in predictive mode

When predictive maintenance is the dominant maintenance philosophy in a plant, the condition of the rotating equipment is known, meaning that machinery should no longer fail unexpectedly - at least not frequently. This means we can plan repairs around the production schedule and we can plan the production schedule around our knowledge of the condition of the plant. In other words, we gain control over the production schedule.

The plant is no longer in a reactive mode, so most work is planned. This means there is less overtime and workers are not stressed out – they come in to work every day and they know what they need to do, there are few surprises.

Additionally, predictive maintenance will supersede many PM actions, meaning that a repair will not take place if the machine is not broken, thus, the overall number of maintenance actions will be reduced. This will reduce spending on unnecessary repairs in terms of man hours, spare parts and infant mortality problems. As one moves from preventive to predictive maintenance these cost savings should be documented to demonstrate improvements in the plant.

When to use predictive maintenance

When it is possible to determine the condition of the asset using some technology. The asset degrades over an unknown amount of time. It is more cost effective to monitor the asset and plan repairs than to allow it to fail or to overhaul it on a scheduled basis

Remember that we will use a combination of all of the maintenance strategies in our plant. Predictive maintenance has its place in cases where a machine or machine component does NOT wear out or degrade over a known amount of time as was the case for 89% of the machines in the study we just reviewed. Examples of this include most rotating machines with rolling element bearings. We don't know when the bearings will fail but they do let us know when they are damaged or failing.

The cost of monitoring the machine / asset must be less than the cost of it failing or of carrying out a preventive maintenance action including the risk of infant mortality.

Monitoring is often carried out on machines that are not redundant, where spare parts may be unavailable or costly, where a failure directly impacts production or can cause collateral damage including injury or death. Insurance premiums may be lower when a predictive maintenance program is in place since it mitigates these risks.

Proactive Maintenance

Proactive maintenance is a general term that includes a number of strategies and studies such as "Precision Maintenance", "Reliability Based Maintenance", and "Reliability Centered Maintenance" and "Root cause failure analysis". To be proactive is:

"To anticipate and solve problems before they become problems"

A predictive maintenance program gives a warning of bearing failure, for example, and then the replacement can be ordered and a repair scheduled. That's great, but why did the bearing fail? Knowing that answer and taking the required action to remove the cause, should enable the machine to last longer when it returns to service.

The term "proactive" is used because, rather than just waiting for the machine to fail, we take action ahead of time to reduce the chance of it failing. Now, that does not mean replacing bearings and seals as may be the case in preventive maintenance; instead we find the root cause of the failure and correct the problem. In order to do this, we must ask the question: "Why do machines fail?"

Why do machines fail?

The source of machine failure can start on the designers drafting board and end with poor maintenance practices and operating conditions. The way the machine is manufactured, the way it is specified, the way it is installed, the way it is operated and the way it is maintained all contribute to the ultimate life of the machine. A proactive strategy will consider every step of the process, from selecting high quality components from reliable suppliers to fine tuning the lubrication regimen and bearing installation procedures.

You may not have control over all of these steps or procedures, but what we will soon see is that a proactive maintenance strategy requires involvement from all employees at all levels. Everyone has a role to play.

It is not the intent for this course to teach better ways to run a maintenance department, or a company for that matter, it is simply our desire to ensure that you can see the bigger picture so that you can identify opportunities to improve the reliability of your equipment.

While the business world grows ever more competitive, often resulting in fewer staff, reduced budgets and greater production demands, the requirement to operate the plant more efficiently becomes that much more important, and that much more difficult.

The aim is to improve equipment reliability, reduce maintenance costs, reduce energy usage, and to improve product quality.

Such changes will not happen overnight in the best case, and they will not happen at all unless everyone is onboard with the goals. From the person who does the motor rewinds, to the person who signs the checks, everyone needs to understand the benefits and requirements of proactive maintenance.

Figure 22-26 Spare parts delivery

Reliability centered maintenance (RCM)

Reliability centered maintenance (RCM) is a sub set of Proactive maintenance, and an "RCM Analysis" is a study done on a piece of equipment in order to understand all of its failure modes and the consequences of any one component failing. With this information about the asset, one can design an optimal maintenance strategy. Condition monitoring may be carried out to monitor for some of the failure modes. Preventive maintenance actions may be carried out on other components that have a known time to failure, while some parts may be run to failure or allowed to fail.

As an example, think of your automobile. You let the headlights run to failure, you change the oil based on the number of miles / kilometers driven or on the number of months since the last oil change – i.e. on a preventive schedule; you frequently check tire pressure which is akin to condition monitoring. You have various sensors including an oil gauge installed that will alert you to changes in condition and a "Check Engine" light that alerts you before a catastrophic failure takes place. While driving you also monitor the condition by listening for strange noises, rattles and squeaks and you monitor how the car handles and feels as you drive. Regarding proactive maintenance, you look at different models of cars, research their recall history and quality problems, consider the manufacturer and finally choose a vehicle that is appropriate for its intended use – i.e. city driving, off road adventuring or hauling.

In other words, for one asset – your automobile – you actually employ a combination of maintenance strategies because different components will have different failure modes, some more critical than others. An "RCM Analysis" is a study of all of these failure modes – the goal being to prevent all failures from happening; especially those with serious consequences.

Precision maintenance

The term "Precision" maintenance is often used as it recognizes that today's machines are designed with tight tolerances and precision components. It has been said that if you hear someone using a hammer in a modern facility then the employee either does not have the right tool for the job or has not been trained adequately to do the job – for example if they are trying to hammer a bearing into a motor because it won't go in easily. This sort of approach leads to machine failure and is inappropriate in today's competitive environment.

A theme of precision maintenance is that every job has its tool and every tool has its place. The mop stands on this side of the utility closet, next to the bucket. The bucket is hanging on a nail labeled "bucket" and on the shelf above it sits cleaning fluid beneath a sign labeled "cleaning fluid." Everything has a precise purpose and a specific use and everything that is required is available and in the correct place so it can be found when it is needed.

Precision machinery also must be properly aligned and balanced and the proper lubrication must be used in the proper amounts.

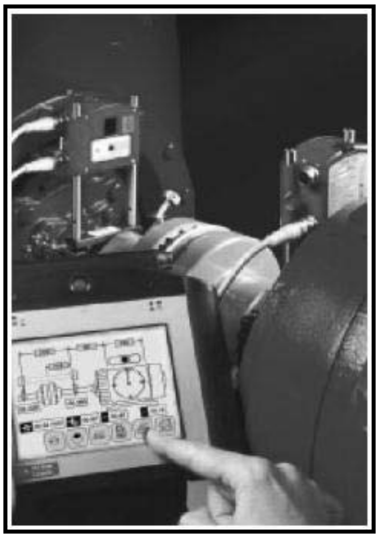


Figure 22-27 Precision Alignment and Balancing extend the life of the machine

Root cause analysis

Determining the root cause of a machine failure is often referred to as Root Cause Analysis (RCA) or Root Cause Failure Analysis (RCFA).

Remember that every effect has a cause, by removing the cause we remove the effect. Most people treat symptoms rather than causes, they take diet pills instead of eating less and exercising more; they apply acne medicine instead of eating healthy foods; they get rushed to the hospital with a heart attack instead of lowering their cholesterol intake. As a society we spend millions of dollars on cancer research but little money on reducing the production and discharge of toxins into the environment, we take greater risks drilling for oil rather than increase the efficiency of our engines and we work harder and acquire more debt rather than consume less.

When a machine fails

Ask the question: why did it fail?

Determine what has to change so it does not fail again.

Make sure the changes are made!

Determining the root cause of a failure can be an involved process. Consider the process the airlines go through after a major plane crash. They recover the black box, review instrument settings, weather conditions, pilot comments; they sometimes reconstruct the entire plane to understand which part failed and exactly how it failed. They then look to other planes in the same class to see if they have had similar problems. Once the root cause of the problem has been determined, steps are then taken to remove that root cause from other planes in the class or in the fleet so that the problem does not repeat itself. This is an excellent example of root cause failure analysis.

Typical root causes of failure in rotating machines are: lubrication problems, alignment, unbalance, resonance, improper operation, poor quality of the machine, improper specification of the machine, human error, overhaul or repair errors (i.e. infant mortality), bad bearings and poor electrical quality. By removing these typical causes, one increases the life and reliability of the equipment. But remember, it begins before the machine is even installed at the plant, from conception to design, to manufacturing, to specifying, installing, operating, to maintaining at each stage of the life of the machine one may introduce the root cause of its eventual failure.

Proactive maintenance components

Proactive maintenance is an attempt to anticipate problems and resolve them before they become problems. It is also an attempt to design problems out or remove the root causes of problems. We can say that being proactive is a general approach that consists of a wide variety of practices and strategies and utilizes various technologies where appropriate. Therefore, one can say that a plant in proactive mode will employ all of the other maintenance strategies where appropriate.

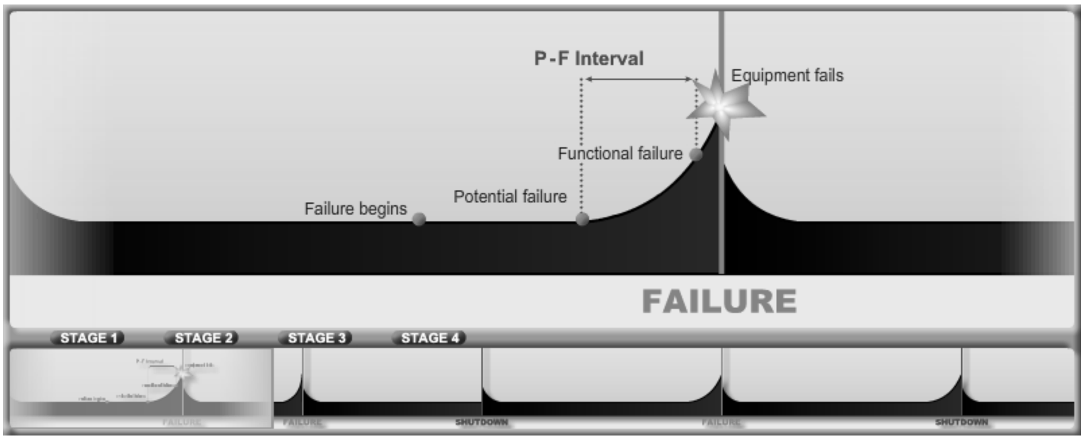


Figure 22-28 The P-F curve of a typical machine

If we return to the P-F curve for a typical machine we can see its life cycle. If we employ a run-to-failure maintenance strategy, the machine will simply follow this curve to its failure. If you remember the concept of preventive maintenance, if you know the time interval until the potential failure point is reached – if this is a known amount of time because the machine is subject to corrosion or it is a cutting tool that gets dull after a known period of time, then the idea is to repair or replace the part before it reaches the functional failure point on the curve. Unfortunately, any repair brings us back to the left edge of the graph where we have a higher risk of failure from infant mortality.

If you remember condition monitoring and predictive maintenance, these technologies and this maintenance strategy is employed when we do not know the amount of time it takes to enter into the functional failure part of the curve. In 89% of rotating machinery, the amount of time is unknown, so what we do instead is take a test to see if the machine is healthy or not. If it is not healthy we continue to monitor and schedule our repairs for a convenient time, if it is healthy, we leave it alone. This is great and extremely beneficial, but we have done nothing to increase the failure interval or the amount of time between commission or overhaul and the when the machine reaches its functional failure point.

Figure 22-29 The P-F curve of a reliable machine that goes through numerous shutdowns without need of overhaul

One goal of proactive maintenance is to remove the root causes of machinery failure in order to increase its functional lifetime. One still has to monitor the machine to detect changes in condition, but the goal is to make these changes in condition take much longer to occur. We will probably never get to 100% reliability, but that should still be the goal to strive for. If one considers accidents like the BP oil spill in the Gulf of Mexico in 2010 it will become obvious that 100% reliability *has* to be the goal in certain situations and if it cannot be achieved then permission will not be given to operate the facility.

Another contemporary example is the problem Toyota had in 2010 with its cars accelerating unexpectedly. This one lapse in reliability in a company who had a reputation for producing reliable vehicles has had an enormous effect on the company.

The world is changing; safety and environmental concerns are at the forefront of people's minds and reliability is now a household term. Global competition in most industrial sectors is fierce and the companies that will come out on top in the 21st century will be those who employ these proactive strategies, who are in control of their production schedules, who avoid environmental disasters, accidents and worker injuries and who can produce their products efficiently.

A plant in proactive mode

When proactive maintenance is the dominant maintenance philosophy in a plant, the condition of most of the plants assets is known because condition monitoring technologies are being employed. Machines should not fail unexpectedly - at least not frequently. By making improvements in every aspect of asset management, from specifying machines to purchasing, installing, operating, maintaining, overhauling and learning from mistakes, one removes the root causes of many failures thus increasing reliability and extending the life of the machines. There is less overall maintenance work that needs to be done and the work that is done is planned. There is ample time to complete all of the preventive maintenance tasks.

The people working in the plant are proud of their achievements; ask for a tour and they will gladly show you around and describe the problems they've solved and the obstacles they have overcome. Ask them what would happen if the machine over there failed and they will be able to give you a precise answer in terms of safety as well as impact to production. Not only do employees feel proud of their plant, they feel ownership of it and therefore take on the responsibility of reporting or solving problems.

The idea of continuous improvement will extend beyond the realm of plant machinery to the realm of plant employees, many of whom will take part in continuing education opportunities, health and fitness programs and other forms of personal growth. They will be given the right tools and training to do the job, and the jobs they do will be well defined and precisely undertaken. The plant will be clean and orderly with everything in its place. Because the plant is profitable employees will enjoy higher wages, better benefits and professional growth paths.

When to use proactive maintenance

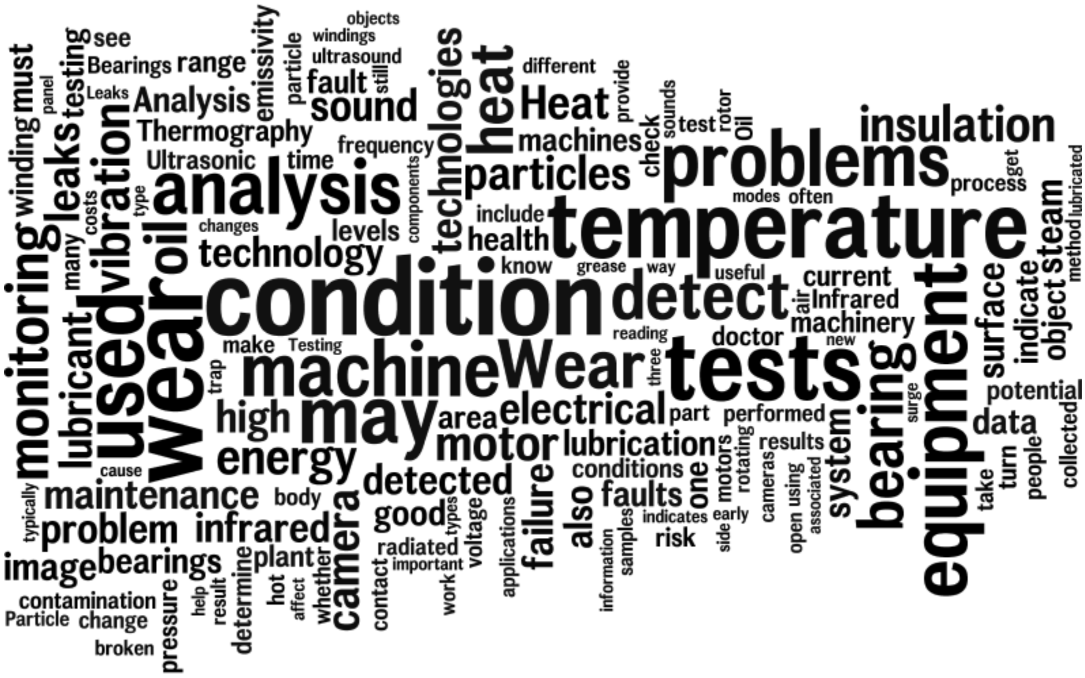
Proactive maintenance should be the goal of every industry that plans to remain competitive in the 21st century.

Proactive maintenance is a philosophy or practice that involves anticipating problems and solving them before they become problems. This is the general approach of proactive maintenance, but it includes a variety of practices and studies that will be used for varying degrees depending on the asset and its relation to the productivity of the plant as well as the consequences of it failing.

Technically we could say that making the decision to let your desk fan run-to-failure is a proactive decision if you went through the process of considering the ramifications of it actually failing. This does not require you to do an in depth RCM analysis of each component in your desk fan, but it does require to consciously choose the strategy you are employing and understand the consequences of your choice.

To be proactive versus reactive is something worth thinking about, not only at work but in all of your pursuits. Think about it for a moment. Are you in charge of your life? Do you have goals that you are actively pursuing or is most of your time spent reacting to the problems of the moment? Are you always busy? Does your life seem like a series of crisis coming one after the other? It is interesting to note that the term "proactive" is also used in psychology and the idea of "cause and effect" is a central concept in eastern philosophy. There is a direct relationship to how a facility operates and how the individual employees in the facility operate, in their minds, in their lives, in the way they handle problems and problem solving, in the way they accept responsibility; whether they are reactive or proactive.

As you go through the rest of this course, come back to the maintenance philosophies every once in a while and consider how the concepts you are learning and the technology you will become familiar with fit into these maintenance strategies. Again, we caution you that many people adopt new technologies but they do not change their strategies. They monitor equipment but then overhaul it during planned downtimes whether it needs it or not. The real benefits come from evolving ones maintenance strategy, the technologies are there to facilitate and enable this transition.



Chapter 23

Condition Monitoring Technologies

Objective:

- Understand Condition Monitoring technologies
- Understand the application of these technologies

Condition Monitoring is the art of monitoring plant equipment to determine its health or condition at a point in time.

Condition monitoring has two elements:

1. **Determine** whether the machine is running in a stable condition or if it is deteriorating. If the machine condition is deteriorating, then determine how fast it is deteriorating, and where it is in the life cycle.

2. **Convey** the conditions to the appropriate people so that decisions can be made regarding any maintenance or process changes that may be required.

Often the terms "condition monitoring" and "predictive maintenance" are used interchangeably. In reality, they do not mean the same thing. "Condition monitoring" is the act of determining the condition of a machine. Predictive maintenance involves taking action based on the condition.

Condition Monitoring:

"Monitor the condition of the machine.

If you know the condition you can plan your response."

Condition monitoring ≠ Predictive maintenance

We are going out of our way to make this distinction because so many facilities have adopted condition monitoring technologies but have failed to rewrite their preventive maintenance actions to take account of the new technology. One person collects and analyzes vibration data but someone else replaces the bearings because the PM action came up on the calendar and told them to do so! Or in another case, the analyst knows a critical machine is on the verge of failure, but a large production run is scheduled with no planned outage because the people scheduling the production do not know about the condition of the critical machine. There is no reporting infrastructure in place.

Condition monitoring

Rotating machines tells us about their mechanical condition in a variety of ways, and in most cases, machines will develop problems and wear gradually and give us plenty of warning before they finally fail. How do they tell us what is wrong with them? They get hot, they vibrate, they make noises, they shed particles into their lube oil, the put out high frequency sounds and they have different responses to electrical currents going through them. Fortunately, there are technologies available that can objectively measure all of these quantities in order to help us diagnose the mechanical condition of our machines.

Condition monitoring = Health monitoring

We can make an analogy between human health and machinery health in order to help explain the concepts. Doctors use different methods to determine the condition of the human body: temperature, blood pressure, heart pulse rate, even fluid samples. Several technologies are used to arrive at an accurate picture of our condition.

Figure 23-1 Various technologies are used to determine the state of our health

For plant equipment, the process is similar. Rotating machines try to tell us their condition through changes in **vibration, temperature, lubricants, sound, and motor curren**t. So, processes are checked…temperature, pressure, flow, speed, motor current, vibration, fluid and lubrication analysis, and more. When the information is gathered, a clear picture can be built of the machine's condition.

Rotating machinery is a lot like the human body. And the way many maintenance departments deal with machinery is akin to the way many people deal with their own health.

Some people lead a hard life - drinking, smoking, and working hard. These activities are not good for their body. They ignore the vital signs, and eventually the body gives out. They just have to hope that the fault is repairable! This is akin to ***breakdown maintenance***.

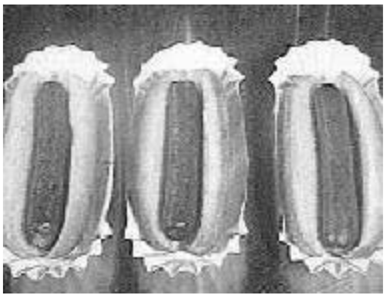


Figure 23-2

Then there are people who likewise may not take good care of themselves, but they will take a few vitamins now and again, and occasionally visit a "health farm" or fitness center to try and make up for all the wrongs. They periodically try to do the right thing, but they will still get sick from time to time (and they often feel worse after the visit to the health farm!). This is akin to ***preventive maintenance***.

Next we have the people who still don't take great care of themselves, but they do regularly go to the doctor. The doctor takes their blood pressure, a few samples of "bodily fluids", and listens to their heart. The doctor tries to detect if there are any problems, to determine the person's condition, and then administers the required drugs or other remedies before the person gets too sick.

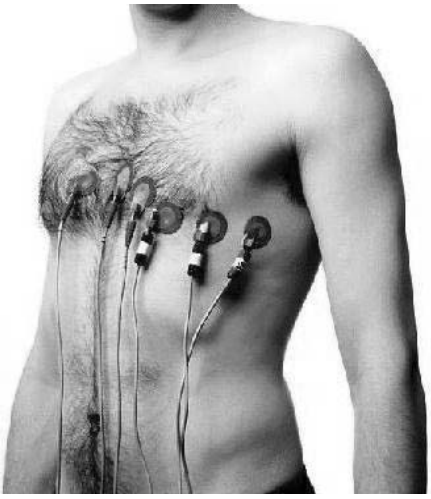


Figure 23-3 Several technologies are used to determine our health condition. In a similar way Predictive Maintenance uses various technologies to determine the machinery condition.

Occasionally a person will allow himself or herself to get sick, but they are going against their doctor's warnings. This is akin to ***predictive maintenance***.

And finally there are those healthy people. They eat the right foods, and they get plenty of exercise. They know what is good for their body, so they make the effort to do what they can to stay healthy. They still visit their doctor to go through the tests, just in case they do catch the odd virus, but by and large they work to stay healthy. Their employers consider them to be very reliable - they rarely miss a day of work. This is akin to ***precision (or proactive) maintenance***.

If these proactive people do actually get sick, regardless of how they take care of themselves, a good doctor will not only diagnose the illness and help get them back to good health, the doctor will also work with them to find out why they got sick in the first place - which may actually involve additional tests, and a review of their health and diet over the previous months or years. With this knowledge (and a willingness to make changes) they can make sure that they do not suffer from that illness again. This is called ***root cause analysis*** - the doctor gets to the root of the problem.

The philosophy that embodies all of these with the goal of making the machinery and equipment reliable is called ***Proactive Maintenance***. A plant with a proactive maintenance philosophy may incorporate all of the other maintenance practices to some extent to achieve the greatest reliability with the greatest profitability.

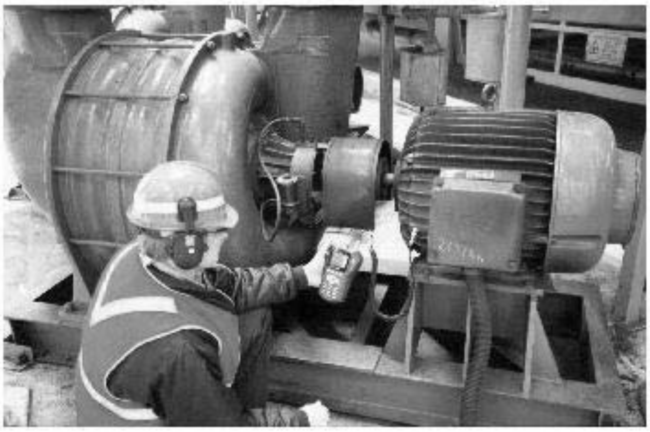


Figure 23-4

In our analogy, condition monitoring is simply the process of going to the doctor and having the tests done. The doctor will check our blood, listen to our heart, check our blood pressure, and look at our throats.

Maintenance analogy

In the case of rotating machinery, we can listen to the vibration, check the temperature of the bearings, check the lubricants, check the sound, check the motor current, and test the flow and discharge pressure.

When we are through with our tests, we hope to be able to build a clear picture of the machine's health condition.

If we take care of the machines, they will be more reliable. They must be:

- Correctly specified
- Checked before installation
- Correctly installed
- Precision balanced, precision aligned.
- Properly lubricated.

If we monitor the condition ***and take appropriate action*** we can reduce the risk of catastrophic failure. The benefits include:

- Reduced downtime
- Reduced labor costs
- Reduced parts inventory
- Greater profit

Condition monitoring: the whole picture

In determining the condition there are three points that should be considered.

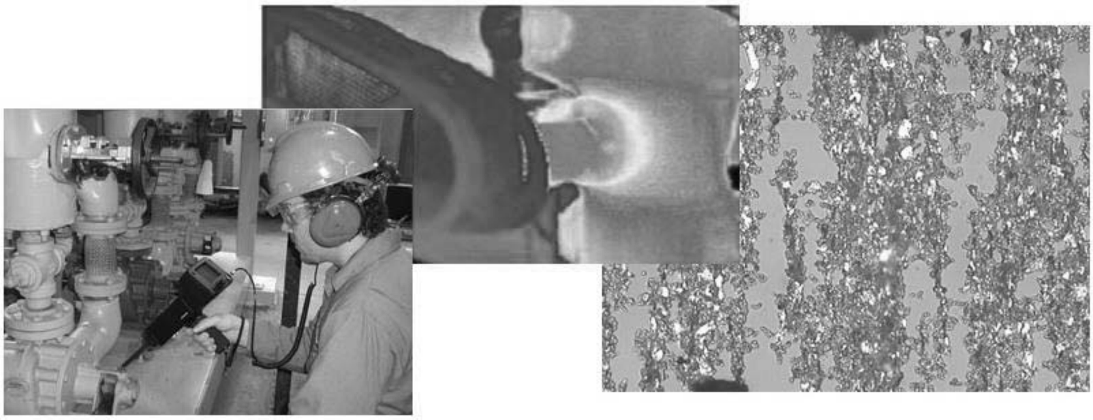


Figure 23-5 Various technologies let us see the condition inside a machine

First, the best picture of a machine's health is arrived at when we perform all the tests and correlate the results. One test alone may be misleading or may give only part of the picture. In fact certain tests will not give us any information regarding some fault conditions. For example, wear particle tests on an oil sample won't indicate whether the machine has an unbalance problem. Vibration analysis won't be able to indicate whether there are contaminants in the lubricant. The technologies must be combined to provide **Integrated Condition Monitoring**.

The **second point** is that certain technologies and tests can provide information to indicate situations that can cause problems for a machine, not that they necessarily are causing problems. Examples:

Resonance per se is not a fault. All machines have resonances. The fact that a resonance at running speed can result in excess vibration and may destroy the bearings is a problem.

Contaminants in a lubricant is not necessarily a problem. The fact that the contaminants can damage bearings and other components is the problem.

The **third point** is that the best diagnosis will be achieved when there are previous results to compare to and data is available to use as a reference. Doctors know what a person's blood pressure should be, and what cholesterol levels are safe. Some guidelines do exist for vibration analysis and other technologies (ISO standards, for example) however for the most part we must rely on comparisons with previous data taken from the machine.

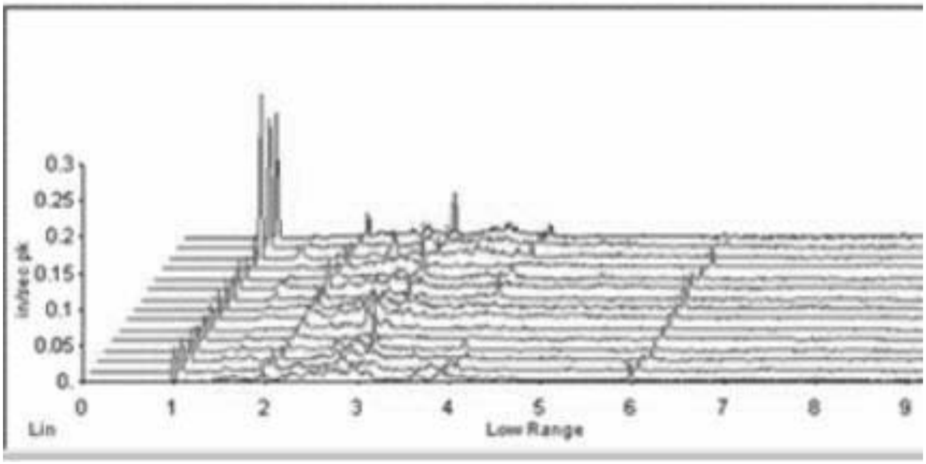


Figure 23-6 Compare to previous measurements to see changes in condition

Vibration Analysis

All rotating machinery like fans pumps, motors, turbines and compressors will vibrate. The level of the vibration and the pattern of the vibration indicate the condition of internal rotating components.

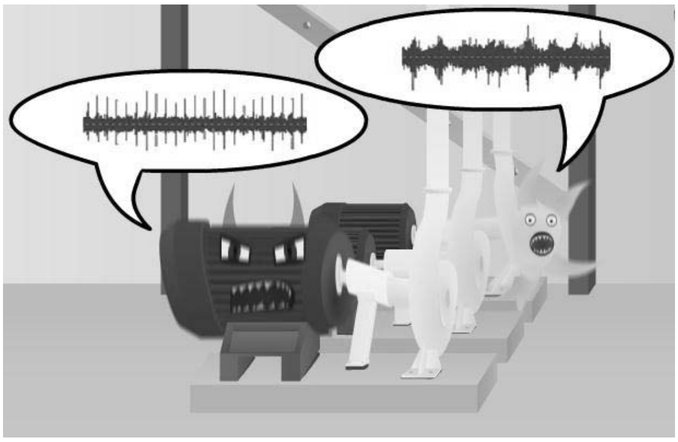


Figure 23-7 Our machines tell us when they have problems

If we use electronic instruments to measure the vibration, those levels can be monitored and the pattern studied. To a large extent, if the levels increase, and the patterns change we can not only detect that there is a problem, but we can diagnose the type of problem.

A number of different types of problems can be detected with vibration analysis. The vibration pattern can indicate a misalignment condition or an unbalance condition. The pattern can point to a rolling element bearing problem or a journal bearing problem.

Fault conditions detectable with vibration analysis include:

- Bearing problems – both journal and rolling element bearings
- Unbalance
- Misalignment
- Looseness
- Soft foot
- Electrical faults
- Eccentric rotors
- Belt and coupling problems
- Gear mesh
- Broken rotor bars

Figure 23-8

Vibration analysis utilizes a special sensor mounted to a bearing housing that is sensitive to movement. A “snapshot” of the vibration is captured in a portable data collector and transferred to a computer for analysis. The snapshot data is generally collected on a monthly basis except for critical machinery which may have permanent sensors mounted for continuous monitoring.

The “snapshot” data is studied to determine whether a problem exists, and the severity of the problem.

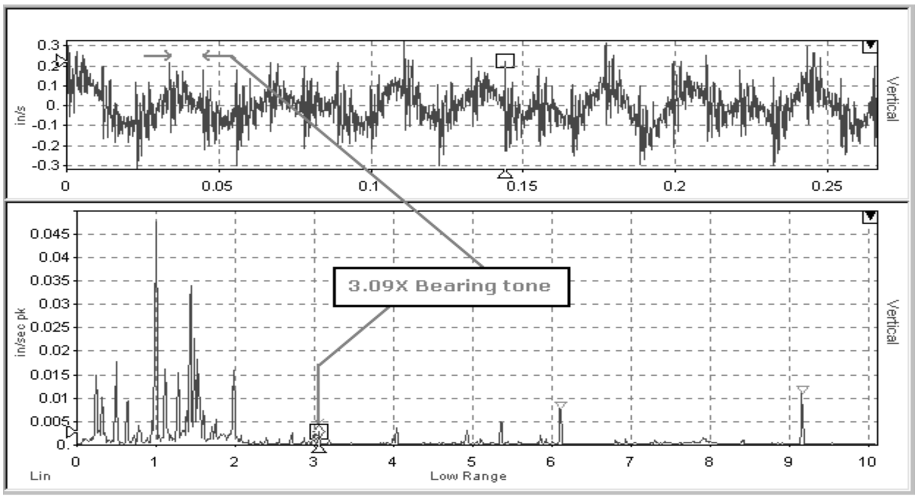


Figure 23-9

Data is collected from more than one location, and in more than one direction. A machine vibrates up and down (vertically), side to side (horizontally), and end to end (axially). Different faults reveal themselves in different ways and in different axes.

Sensors can be permanently mounted so that data can be collected at junction boxes in safe environments.

Online monitoring

For a machine that is critical to the process, and machines located in remote or hazardous environment (such that routine measurements cannot be taken), sensors will be mounted permanently on the machine, and a monitoring system will monitor the vibration levels to give an early warning of a fault condition.

Online monitoring may be set up for the most critical machines. They will typically have a protection system installed so that if vibration levels reach predefined alarm levels, the machine will shut down. This type of system is often used for turbine/generators.

Acoustic Emission (Airborne Ultrasound)

Rotating equipment and other plant assets emit high frequency sounds that provide clues to potential problems. Ultrasound testing is a useful technology for a variety of applications.

Finding air leaks	Excellent
Finding steam leaks in steam traps	Excellent
Detecting lubrication problems	Good
Adding the correct amount of lubricant	Good
Detecting electrical faults (arcs, coronas)	Good
Finding flow related problems in pipes and valves	Very limited success

Table 23-1

How it works

The human ear can only detect sound in the sonic frequency range of 20 Hz to 20,000 Hz. Sounds above this range are referred to as "Ultrasonic", meaning above human hearing capability.

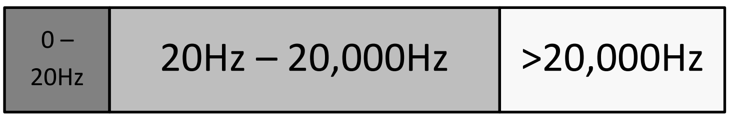


Figure 23-10 Sound is categorized into 3 regions; Sub-sonic range, Sonic range, and Ultrasonic range.

The ultrasound sensor is used to measure the signal and heterodyne (demodulate) it to a frequency range within the human hearing range.

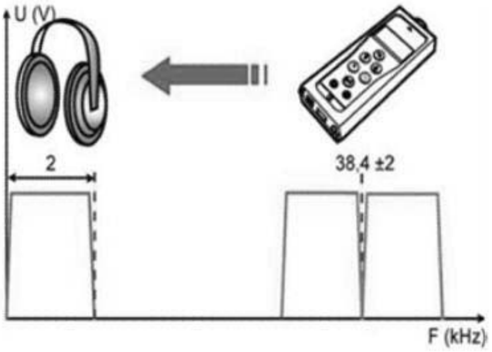


Figure 23-11 Ultrasound frequencies are converted to an audible range

A few considerations in using ultrasound technology:

- The sounds are directional and the sensor in the "gun" is very directional. Therefore the sources of the sounds can be pinpointed (but may easily be missed).
- The volume depends on the distance from the source.
- Sounds travel through air, liquid, and solid objects, but not through a vacuum.
- There are two modes of detecting ultrasonic sounds – airborne, and direct contact. The typical airborne sensor or "gun" has an open end which is good for scanning areas in close proximity. For distances, a concave dish similar in style to a satellite dish is used. The dish reflects the sound to the sensor in the center.

Figure 23-12 A "dish" captures sound from a distance

A typical "direct contact" system makes contact with the surface of the equipment to provide better transmission of the high frequency sound.

Ultrasound data can be listened to via headphones, which is very useful when searching for leaks and detecting faults in noisy environments.

Figure 23-13 A Direct Contact probe uses a "stinger" to capture the sounds

Note: The sound can also be measured and displayed in db units in waveform and spectra displays.

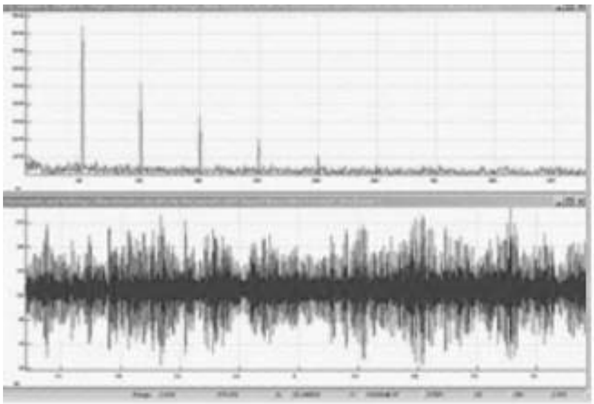


Figure 23-14 Ultrasound spectrum and waveform

Air Leaks

Air leaks are the most expensive utility leaks in manufacturing. Turbulence from leaks creates white noise with a strong ultrasonic component. Ultrasonic instruments pinpoint pressurized gas and vacuum leaks regardless of ambient background noise. Finding and correcting leaks can save hundreds of thousands of dollars annually.

A compressed air maintenance program includes a complete inspection of the airlines three to four times a year. Leaks are tagged when detected so that action can be taken. Regular inspections ensure that new leaks are found and confirm that tagged leaks from previous surveys were repaired.

Know the system – familiarize everyone with the supply side, the demand side, the number of compressors, operating pressures and any additions to the system since it was installed. Be sure the equipment user is properly certified and trained by a reputable trainer. Find, tag, repair, and re-check the leak area with the ultrasonic detector. Remember that the person making the repair may not be the same person who tagged it. It is also possible that a new leak was created while the old one was being repaired. Document everything.

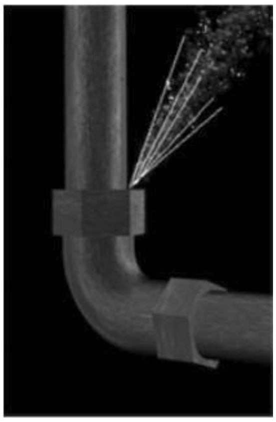


Figure 23-15 Air Leaks emit high frequency

Boiler, Heat Exchanger, and Condenser Leaks

Scan for external pressure or vacuum leaks in Boilers and Heat Exchangers and condensers All pipe connections, flanges, seals, and access doors should be inspected as part of regular PM's. Listen for the same sound that is associated with compressed gas and air leaks. Tube leaks in condensers and heat exchangers can be checked using either the pressure method, the vacuum method, or the bisonic transmitter method. Choose the method that suits your application best to save valuable inspection time.

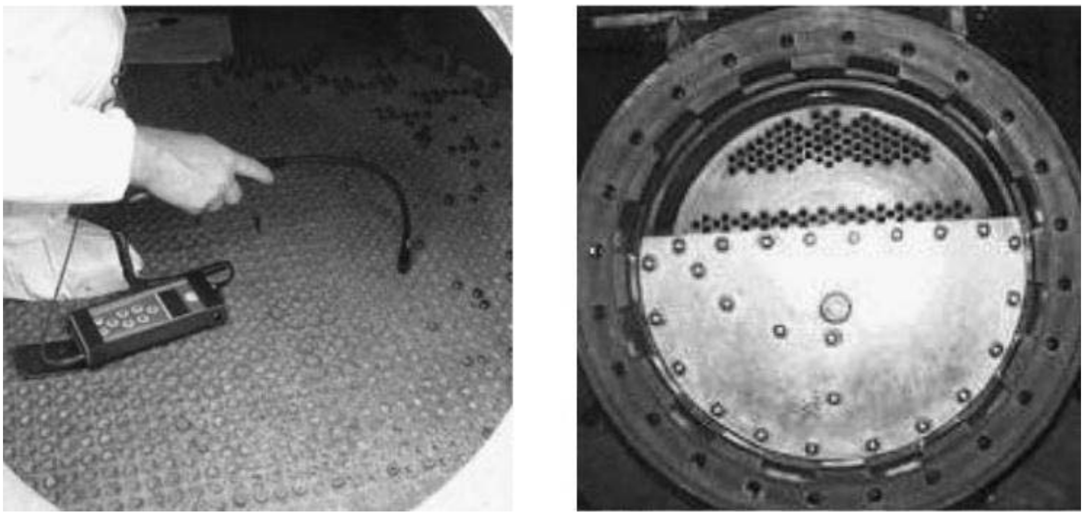


Figure 23-16 Sweep the microphone across the area to detect the leak.

Sweep the microphone across the area to detect the leak. Sweep side to side and up and down.

Detecting Faulty Steam Traps

Steam traps open occasionally. This opening and closing produces a sound that is very distinct from the normal flow. A direct contact probe is ideally suited for listening for correct operation. There should be a steady sound interrupted by the distinct opening and closing of the valve. If the sound is a continuous fluttering sound, it is not operating correctly. A consistent rushing sound indicates the trap is stuck in the open position.

If the steam trap is stuck open, it wastes hundreds or thousands of dollars. It generates excess steam in the system which creates back pressure and causes failure in other steam traps.

Figure 23-17 Checking a steam trap with a Direct Contact probe

When the steam trap is stuck shut, it produces a water hammer sound. In cold climates the line may freeze causing the piping or trap to rupture.

There are four common types of steam traps, and all work to remove impurities from the steam system.

The four types are known as:

- Inverted Bucket
- Float and Thermostatic
- Thermostatic
- Thermodynamic or Disk

Traps work on one of three operative modes.

- Change in Density
- Change in Temperature
- Change in Velocity

Ultrasonics and electrical problems

Ultrasonic monitoring can be used to detect arcing, nuisance corona, destructive corona, tracking, and line bushing conditions such as may be found in:

- Motor control centers
- Breaker Panels
- Power Lines
- Connections
- Insulation breakdown

Figure 23-18 Ultrasonics is a good choice for monitoring distribution lines

Ultrasonics is one of the top choices for electrical inspections of transmission and distribution lines, Substation inspections, Switch gear, Transformers, Corona, Arcing and Tracking, Radio and television interference faults. Sounds associated with frying, popping, buzzing, and humming are characteristic of these faults.

Bearing Faults and Lubrication

Rolling element bearings produce friction as the internal rolling elements turn against the raceways. This friction produces sound in the ultrasonic range that can be detected with ultrasound equipment. Add more grease to the bearing and the amount of friction goes down and so does the quantity of ultrasound being detected.

Over lubricating bearings can damages seals, build internal pressures and cause premature failure. In fact, over greasing is a major root cause of bearing failure! Over greasing an electric motor can push lubricant into the windings causing shorts and more severe damage.

Figure 23-19 Direct Contact probe is useful for detecting bearing problems

Under lubrication of bearings is also responsible for a large number of failures. Too little lubrication results in internal friction which generates frequencies above 30 kHz that can be detected with ultrasonics.

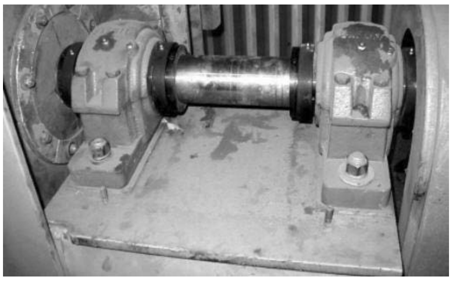


Figure 23-20 Under lubrication generates Ultrasonic energy

If too much lubrication is a problem and too little lubrication is a problem how do we know how much is the right amount? Ultrasonic technology can be used to monitor the bearing as grease is added to it. The frictional sounds can be monitored through headphones as the grease is pumped into the bearing and the noise reduces substantially as the grease reaches the bearing. In other words, you can hear when the bearing is sufficiently lubricated.

At this point, an RMS dB reading can be collected from the bearing. The level indicates the amount of ultrasonic energy the nearing emits when properly lubricated. Similar RMS readings can be captured in the future on the same bearing and trended. A rule of thumb is that when the RMS level increases by 10 dB from its baseline, it is time to grease the bearing again. Thus ultrasound technology can be used to define a condition based lubrication regimen. Training is required to ensure that this method is used correctly.

In the image below you can see the change in ultrasonic energy as the bearing is being lubricated. This is a time waveform of the ultrasound signal.

Figure 23-21 Capture of Ultrasonic energy during lubrication

Mechanical Fault Detection

Ultrasonics can be used to detect early bearing wear in rotating equipment such as Gearboxes, pumps, motors, and compressors. The values can be trended for changes which can be an early warning indicator.

As rolling elements in a bearing strike defects in the races, high frequency pulsations are created by the impacts. These can be detected with ultrasound equipment and provide a very early warning of very minor damage in the bearings.

Ultrasonic monitoring should be an integral part of the condition monitoring program

Benefits:

Ultrasonic equipment is not expensive and is not difficult to operate.

- It is often used to detect a fault while other technologies are used to follow-up and determine the severity.
- It is very direction which enables the pinpointing of particular faults, especially leaks.
- It is useful in high noise environments.
- Can be used in peak production hours
- Integrates with other predictive maintenance (PdM) technologies

Infrared Thermography

Infrared Thermography is the study of *radiated energy* using a thermal infrared imaging system.

Thermography is a popular technology applied to rotating and non-moving equipment in the plant. It involves the study of temperature as increased wear, steam leaks, and electrical arcing (to name but a few conditions) result in a change in temperature.

Excessive heat is an indicator of problems or potential problems in plant equipment including moving and stationary parts and equipment such as electrical panels, boilers, transformers, and electrical power transmission conductors, insulators and switchgear. Infrared Thermography is an ideal, non-intrusive technology for detecting these problems.

The technology uses sensors that are sensitive to the radiated electromagnetic energy associated with heat. The device translates the detected level of radiated energy into a temperature based on information entered by the user. Two types of devices are commonly used in our industry: spot radiometers and infrared cameras.

Spot radiometers sum up the energy in a small area and display a temperature reading. Radiometers often use a laser beam to help you to target where the measurement is to be taken. It should be understood that this reading is not based purely on the temperature at that point; the further you are from the target, the larger the area used to determine the temperature. You should visualize a cone radiating from the device – the greater the distance to the target, the greater the measurement area.

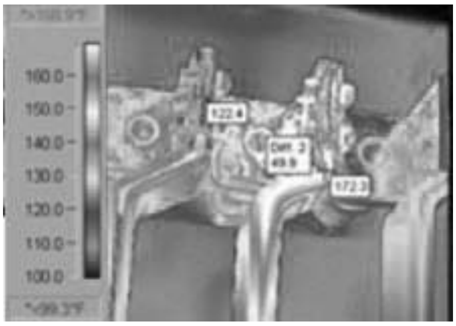


Figure 23-22 Thermography image of bus bar connections. White area is the hottest.

It is important to understand that the actual temperature indicated may be incorrect as surface type, air flow, and other factors affect the accuracy.

An infrared camera can create a thermographic image. The instrument used to "see" radiated electromagnetic energy is generally referred to as a camera. These cameras use special sensors to detect the heat which is displayed in a visual image similar to a photograph. Many thermographic cameras also have a standard photographic camera built in so that a photographic image can be compared to the thermographic image. Most thermography systems include a software program for transferring the images to a computer for analysis and printing reports.

Note: the infrared camera does not "see" temperature. Temperature is calculated from inputs by the user in the camera or software.

Figure 23-23 Infrared Thermography instruments include Spot Radiometers, still cameras, and movie cameras.

The instruments range from a "spot radiometer" used to detect temperature in an area, to still cameras and movie cameras which can record the changing temperatures. The price ranges from inexpensive to expensive with the more expensive models typically having a higher resolution image, an ability to zoom in to the area of interest, more field options, and additional software options.

Temperature comparisons

The typical infrared camera has the ability to adjust the sensitivity so that the color scale shows the hottest area as "white hot." This "white hot" area could be 50 degrees f to several hundred degrees depending on the application. **The primary usefulness in most applications is the relative temperature rather than the absolute temperature.**

Figure 23-24 Thermographic and photographic image of overheated bearing

It is possible to adjust the image to make any part of it glow "white hot." A scale indicates the relationship between color and temperature. Although the scale indicates temperature, comparing temperature at selected points provides the best indication of severity. Temperatures are calculated based on inputs the user provides to the camera.

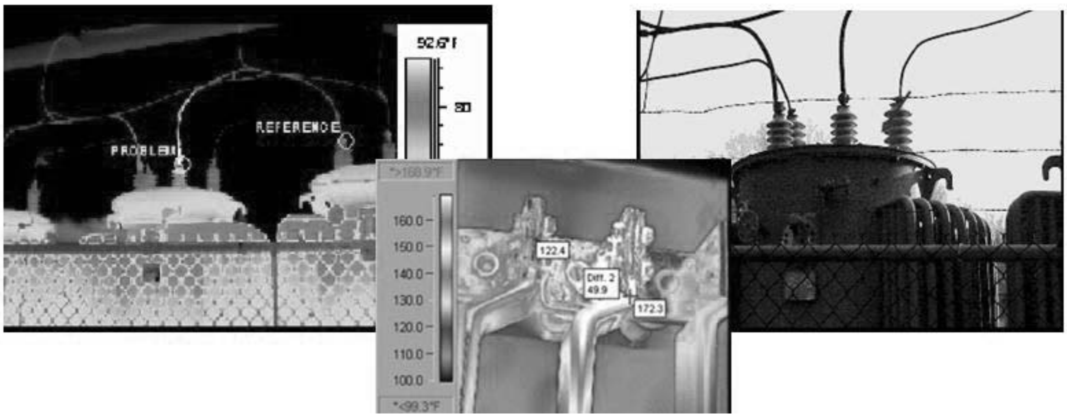


Figure 23-25 In most applications we are interested in relative temperatures

Infrared Thermography is typically used in the following applications:

- Mechanical
- Machines, pipes, bearings, belts
- Electrical
- Overhead lines, transformers motors, control panels
- Steam Systems
- Piping, steam traps
- Refractory plant

A few things to know…

The infrared camera looks at **radiated electromagnetic** energy from the first 1/1000” (one mil) of the surface.

The infrared camera does NOT “see” temperature. Temperatures are calculated from inputs the user provides to the camera or computer. An IR camera cannot see through the surface of a tank, switchgear cabinet, bearing housing, or any other structure. If the source of high temperature (hot liquid, electrical short, bearing wear) causes the surface to become hot, then the infrared camera will be able to detect the difference in temperature. It is therefore necessary to open cabinets to detect electrical faults – unless special windows are used.

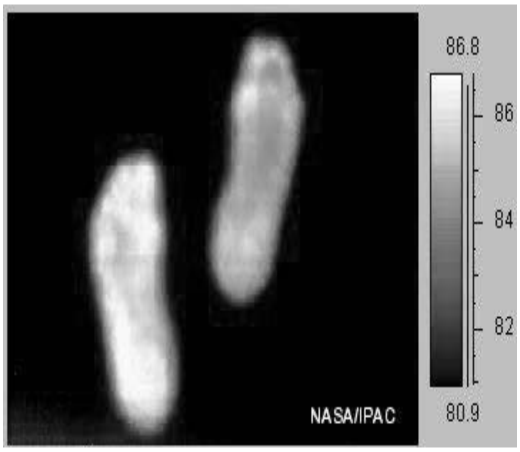


Figure 23-27 IR image of footprint after standing a few seconds

Heat Transfer

Heat is transferred in three ways. These can affect the accuracy of the temperature reading of the object.

Radiation or Emission – is the type of heat transfer where the heat is transferred from the surface of an object as an infrared energy.

Conduction - A type of heat transfer mainly through a solid object

Convection - A type of heat transfer where the heat is transferred by the heated part of gas or liquid moving upward.

See Figure 23-28. These issues must be understood to correctly measure temperature.

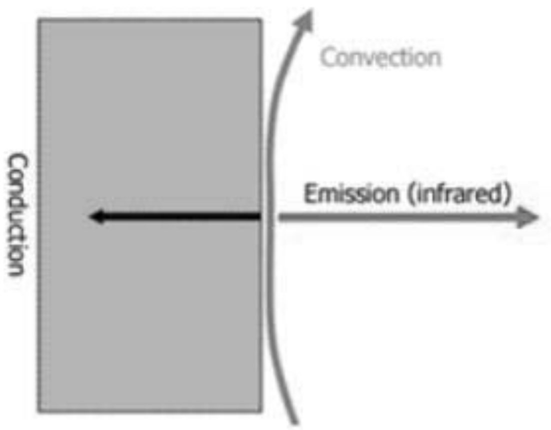


Figure 23-28 Heat is transferred via conduction, convection, and radiation (emission)

Emissivity - How does it work?

The infrared camera detects 3 forms of radiation:

- Emitted Heat
- Reflected Heat
- Transmitted Heat

These three must be understood and considered in order to obtain accurate temperatures. All objects emit, reflect, and transmit heat differently. Therefore it is important to know the emissivity rating of an object.

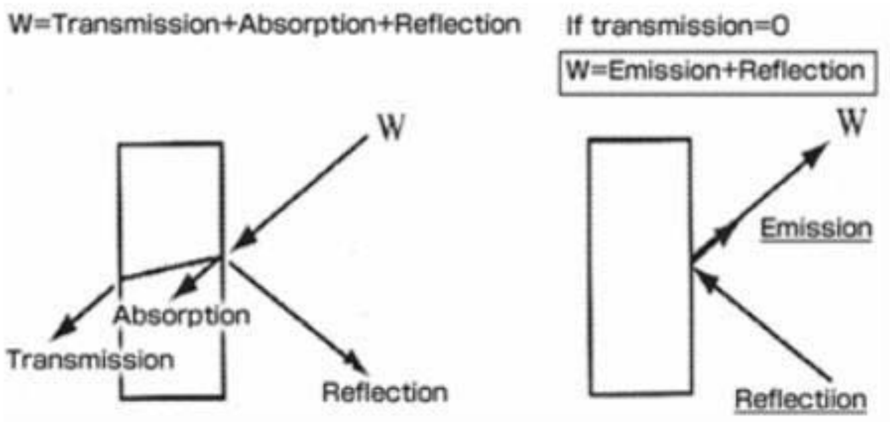


Figure 23-29 Relationship of Emitted Heat, Reflected Heat, and Transmitted Heat

Emitted Heat – Emissivity

Emitted heat is the heat radiated from an object as infrared energy.

- The 'black body' has an emissivity value of 1.0. The black body has an anodized black coating.
- All other bodies have an emissivity value of less than 1.0.
- All objects emit, radiate, and transmit heat differently; therefore it is important to know an object's emissivity.
- Examples of high emissivity objects are water, snow, ice, vegetation, glass, paper, soil and minerals. Oxidized copper = .0.68, polished copper = 0.02
- Examples of low emissivity objects (.2 and below) are polished metals (copper, aluminum, steel, silver, chrome)
- Tables of emissivity values for various objects are available for inputting emissivity values into the camera or computer.

Note: Emissivity is the term used to describe the ratio of the energy emitted by an object compared to the energy emitted by a 'black body' at the same temperature.

The infrared camera looks at radiated electromagnetic energy from the first 1/1000" of the surface. The source of heat may come from inside the object (tank, machine, cabinet), but the detector only sees the energy emitted from the surface. Therefore it is necessary to open cabinets to detect electrical faults – unless special windows are used or the cabinet door gets hot.

The content and color of paint can affect the emissivity.

Figure 23-30 The paper is all the same temperature. The color affects the Emissivity.

Reflected Heat

Heat can be from a source other than the object being measured and may drastically affect the accuracy. Other sources could be intense light, sunlight, heat from a nearby object or machine. The reflected heat could be at such an angle that the camera is looking directly at the reflected heat rather than the heat actually emitted from the object. For example, an infrared image of a shiny tank on a sunny day may reflect sunlight directly into the lens and thereby affect the accuracy of the reading.

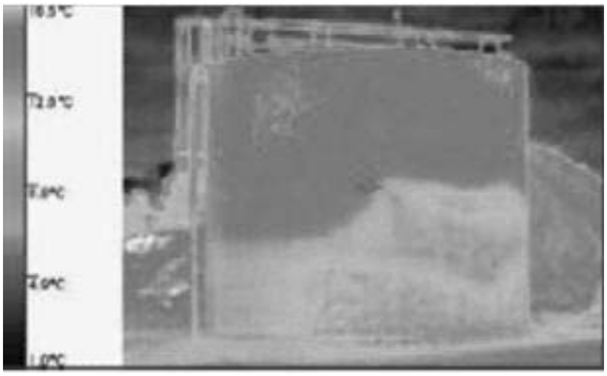


Figure 23-31 Storage tank in sunlight

Transmitted Heat

Heat from another source can be absorbed into an object and actually transmitted out again. This affects the actual true temperature and the radiated electromagnetic energy

Generating Reliable Measurements

To generate reliable measurements:

- The emissivity must be greater than 0.6
- The background must be known or controlled
- There must be controlled or no wind

Unreliable measurements are generated when:

- Emissivity is less than 0.6
- Radiation is emitted poorly
- Reflects background radiation
- Extreme, unknown or changing background

Warning: If there is wind, even relative temperatures will be inaccurate.

Electrical Applications

Infrared Thermography is especially useful in many electrical applications. In substations and distribution lines watch for influences of sun and wind.

Loose connections and overloading can be detected.

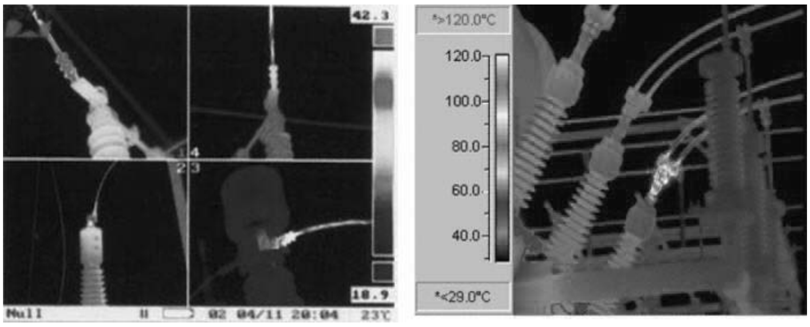


Figure 23-32 Loose connections and overloading can be detected. Overcome the effects of sun by scanning at night.

Bus Bar Connections may be loose causing excessive heat, or the loads may be unbalanced. These conditions can be detected when the panel is open or there is a window in the panel cover. See Figure 23-33.

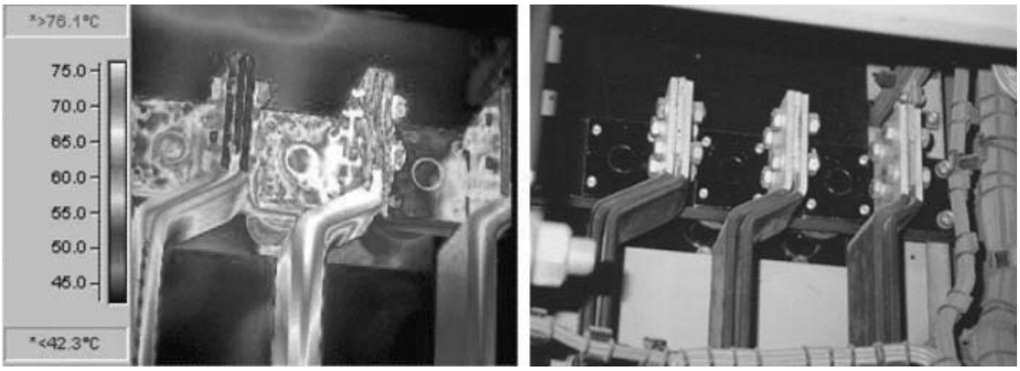


Figure 23-33 Loose or overloaded connection.

Fuses and fuse blocks can have similar problems which can be detected. Candy striping due to broken strands is very apparent using a thermographic image.

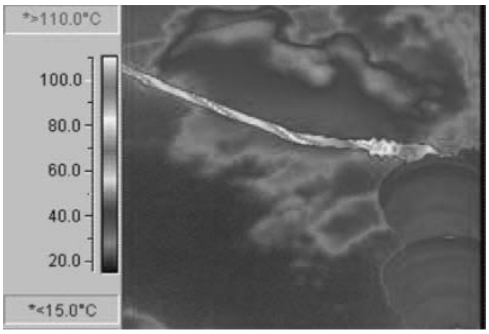


Figure 23-34 Candy striping due to broken strand. Light colored strand is carrying the load.

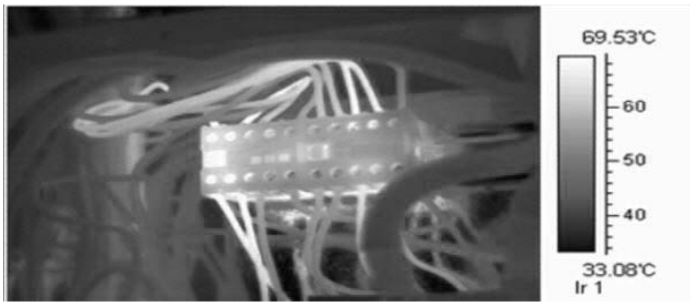


Figure 23-35 Hot terminal block

Mechanical Applications

Many mechanical conditions can be detected using Thermographic imaging including:

- winding problems in motors See Figure 23-37
- cooling issues,
- belt problems
- overheated bearings See Figure 23-37
- abnormalities in pumps, pipes, and compressors

Note: IR is not a good early-warning indicator of bearing wear.

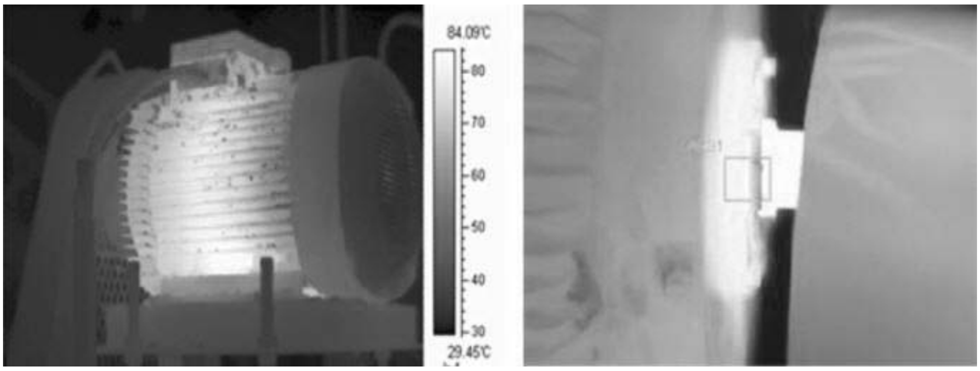


Figure 23-36 Thermographic and Photographic image of failed steam trap. Note the delta temperature of 20 degrees.

Steam Systems

Leaks and blockages in steam systems are easily seen with Thermographic imaging. It is also a good way to check Steam Traps to verify whether they are functioning properly. One key is to check the relative temperature across the trap from the inlet to the outlet. Figure 23-36 shows a delta temperature of 20 degrees.

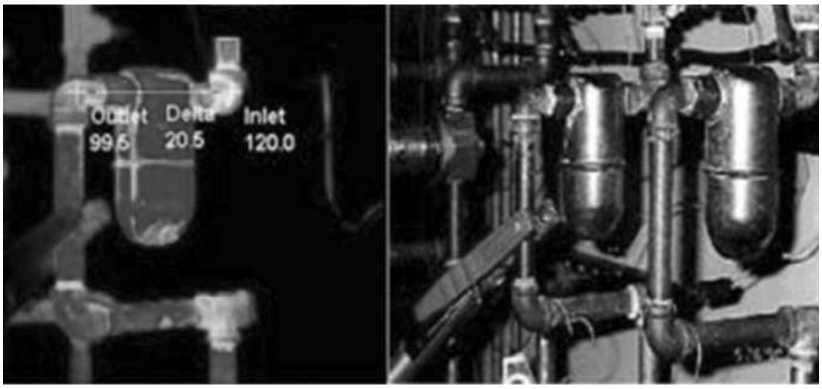


Figure 23-37 Overheated winding in motor; Overheated bearing

Refractory Plant

Boilers, kilns, and other refractory equipment often have insulation problems that are detectable with Thermographic imaging. It can also provide and indicator of blockages or buildup that may prevent the system from functioning at an optimum. Hot areas in the lime kiln in

Figure 23-38 indicates areas of heat loss which affect the process efficiency.

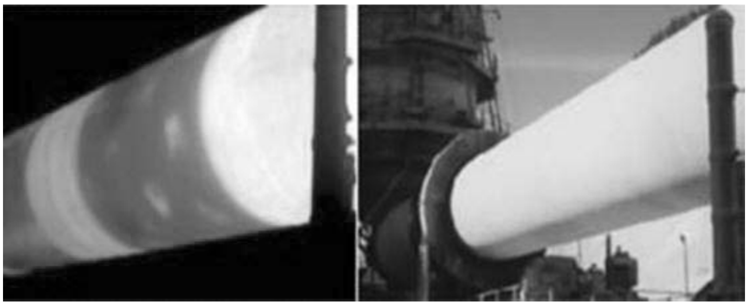


Figure 23-38 Lime kiln with hot areas indicating poor insulation.

Infrared Thermography is a technology that has many applications in and around a plant. It has useful applications in processes, in support equipment and systems such as electrical power, piping, tank levels, leaks, and more. The price of IR equipment has gone down quite a bit in recent years so the technology is now affordable to most plants.

Electric Motor Testing

Electric Motors are the main equipment component in most plants. It is imperative to know their condition in order to ensure uninterrupted processes and to schedule downtimes rather than have surprise downtimes.

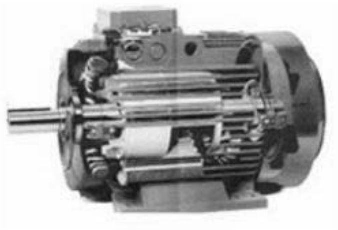


Figure 23-39

Mechanical problems of motors can be detected with Ultrasonics, Infrared Thermography, and Vibration monitoring. However, there are special tests that can detect the electrical condition. The tests fall into two categories:

1. Static / off-line tests
2. Dynamic on-line tests

Static / Off-line Tests

Static or off-line testing is usually performed once a year or during outages with the motor shut down. Off-line testing is also used as a quality assurance tool when first receiving reconditioned or rewound motors from the motor shop before they are stored or returned to service. Testing these incoming motors provides proof the motor shop is doing its job properly and becomes the new base-line for future trending. Off-line equipment can also be used as a troubleshooting tool. Any time a problem has occurred the motor involved should be tested for insulation integrity. Overload situations, contamination issues and voltage problems can compromise the insulation.

Off-line testing includes:

- winding resistance
- meg-ohm
- polarization index
- high potential
- surge testing.

The tests should be performed in that sequence with modern, state-of-the-art test equipment. Equipment is manufactured today that can adequately reproduce “real world” experiences without causing damage to the motor’s insulation system. It is important to test motors at voltage levels and conditions they will see in their normal, day-to-day operation.

Motor Circuit Analysis - MCA Traditional Test Methods. - Most of the traditional test methods require a significant voltage application in order to work. The purpose is to stress the insulation system by forcing a reaction of the insulation dipoles or to force a potential across a resistive or capacitive fault.

Meg-ohm or Insulation to ground testing - Polarization Index; Resistance Testing; and, Surge comparison tests. Insulation to Ground Testing (Meg-Ohm meters)

PdMA – MCE - Static

Figure 23-40

A DC potential is placed across the motor winding conductors and ground. The applied potential is set and a value of current (leakage) crosses the insulation boundary. This value is converted to resistance, usually in meg-ohms. It is, in effect, a method of measuring leakage across the insulating boundary, but only between the surfaces of the conductors and ground. As the insulation dipoles are only excited with DC, some time is required for them to polarize. Standards normally indicate a winding charging time of about 1 minute and, as insulation resistance is directly affected by temperature and moisture, normalization for temperature.

Polarization Index - The polarization index (PI) test is a measurement of leakage at one minute then at ten minutes. The results are shown as a ratio of the ten minute to one minute reading. It is assumed that a fault will polarize slowly (high ratio) or rapidly (low ratio) due to contamination and changes to the circuit capacitance.

High Pot or High Potential Resistance Testing - Resistance tests use a low voltage DC output and a bridge. The primary purpose is to detect high resistant joints, loose connections, broken connections (or conductors) and direct shorts.

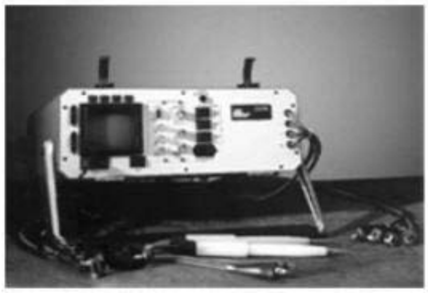


Figure 23-41 Baker Instrument used for Static tests

Surge Comparison Testing - This is an older method of evaluating windings for shorts. A series of steep-fronted higher voltage pulses are sent from the instrument to the stator. The higher voltages force the dipoles in one direction leaving the ability to detect a reactive fault as creating enough potential to cross the barrier (Paschens law) either being shut down after partial discharge occurs or an arc is drawn. Both methods of detecting cause a change to the properties of the insulation at the point of defect either accelerating the fault or completing the fault. In order to force slight defects, a greater potential must be applied, stressing the complete insulation system. Due to the steep fronted pulses, the applied voltage is normally impressed only on the first 2-3 turns in the first coil of each phase.

The situation is quite different for detecting the breakdown of the turn insulation in a winding (parallel or phase) having many coils. The breakdown of the turn insulation in a single coil in a winding of many coils produces a very small relative change in the characteristics (L, C, R) of total load impedance seen by the surge generator. Hence the change in the VFW [voltage wave form] shape produced by the breakdown of the turn insulation somewhere in a winding of many coils is relatively very small. Therefore the surge tests may not reliably verify the presence of one shorted turn in a single phase winding or three phase winding in a machine.

"The surge tests on windings in a machine may possibly lead to wrong conclusions. Perfectly intact windings may appear to have a turn short. More importantly, a turn short induced by the surge test by breaking down the weakened turn insulation may not be detected. In such a case, the stator winding would likely fail after the machine is put back into service. "In view of the above facts, caution is advised in surge testing of the turn insulation in complete windings. These tests carry very significant risks, which should be carefully considered. Such caution is more important for diagnostic tests on machines in service as such tests are carried out quite infrequently in contrast to frequent tests on new, or refurbished, or repaired machines in a manufacturer's plant."[8] As shown, traditional testing has specific flaws in the ability to detect, and the ability to detect defects in a non-destructive manner.

Dynamic On-line Tests

On-line tests enable testing at the motor and at the panel while the motor is in service. The tests view the current and voltage spectra depending on the test. The data is treated like vibration data.

Figure 23-42 Dynamic On-line Tests can be done at the electrical panel.

Online Tests that can be performed include:

- Winding shorts between conductors or coils
- Winding contamination
- Insulation to ground faults
- Air gap faults, including eccentric rotors
- Rotor faults including casting voids and broken rotor bars.
- Vibration which detects broken rotor bars, air gap eccentricity, eccentric rotor
- Current Analysis with a current clamp – broken rotor bars
- Flux Coil – uneven flux field.

Current spectra – Current spectra can be collected on each power leg at the panel using a current clamp. See Figure 23-43. One general comparison that can be made is the amperage draw from each leg. These should be approximately the same for proper load balancing. An exceptionally high or low load indicates problems.

Figure 23-43 Current Clamp

The current spectrum can indicate the probability of broken rotor bars and other defects such as uneven air gap or a bowed rotor.

The flux coil is used on the motor rather than at the panel. It collects data of the flux field generated by the motor. The data can be viewed as a spectrum. It can indicate potential problems in the windings.

Figure 23-44 Flux Coil used at the motor

The various on-line tests tend to be less destructive than off-line tests and have the added benefit of keeping the motor in service and avoiding starts and stops. The on-line tests can typically be performed by certified in-house personnel while many of the off-line tests are normally performed by outside personnel such as a motor shop.

Oil Analysis

Oil is the life-blood of rotating equipment. Rotating machinery needs correct lubrication. But it is surprising how often the incorrect lubricant is used, or the lubricant is contaminated. The result is increased wear and equipment failure. There is also an economic issue - the lubricant is expensive, both to purchase and dispose of.

Figure 23-45

Too often perfectly good lubricant is changed out, at great expense. So testing is performed on the oil and grease. The tests indicate:

- whether the lubricant is still able to perform its job (is the additive pack OK, etc.)
- whether there are any contaminants such as water or dirt
- whether there are any metals or other elements, which may give an early warning of wear

Oil analysis tests and what they measure

Samples are collected routinely for analysis. They may be sent to an outside lab or an in-house lab. Various tests on the oil include:

Test	Measures…
Oil Bath 40c and 100c	Viscosity
R. D. E. Spectroscopy	Elemental Concentrations
FT – IR (Infrared)	Degradation, contamination, additive depletion
Total Acid	Acid Levels
Total Base	Base Levels
Water Crackle Karl Fisher	 Concentrations to 200ppm Concentrations to 10ppm
Particle Count	NAS & ISO Cleanliness

Table 23-2

Strengths of Oil Analysis

- Detects normal wear particles up to 6-10 microns.
- Determines lubricant additive depletion
- Detects fluid contamination

Weaknesses of Oil Analysis

- Does not detect the onset of abnormal wear – wear particles in excess of 10 microns
- Does not detect the sources of wear (bearings, gears, seals, rings, etc)
- Does not provide information regarding machine condition.

Additional tests that can be performed

- Visual analysis
    - Vents/breathers: old or blowing vapor
    - Sight glasses: check levels and color
    - Leaks: oil or process fluid
    - Moisture: water separation in oil samples
    - Color: dark oil samples indicate oxidation
- Smell
    - Acrid smell: oil may have been heated to a high temperature

Oil analysis provides good information about the condition of the oil but not necessarily about the condition of the machine. Fortunately, there is another related technology that looks oil, not to determine the condition of the oil itself but to relate particles in the oil to mechanical wear and failure modes in the machine. This technology is called Wear Particle Analysis (WPA) or Ferrography.

Wear Particle Analysis

Ferrographic wear particle analysis is a machine condition analysis technology that is applied to lubricated equipment. It provides an accurate insight into the condition of a machine's lubricated components by examining particles suspended in the lubricant.

By trending the size, concentration, shape, and composition of particles contained in systematically collected oil samples, abnormal wear-related conditions can be identified at an early stage.

Figure 23-46

Wear particle analysis complements vibration analysis by providing, in some cases, earlier fault detection and is less susceptible to the limitations imposed by slowly rotating or reciprocating machinery.

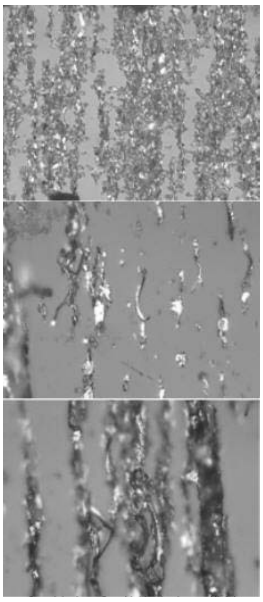


Figure 23-47 The different wear particle types indicate specific problems

Although it is possible to purchase laboratory equipment and perform tests in-house, most industries rely on external commercial laboratories for their testing. The oil samples must still be collected on-site in a controlled manner, but they are then sent off-site to a laboratory. Test results are typically available electronically for integration into the condition monitoring program.

Wear Particles are typically divided into six types. They each have particular characteristics and causes.

Abrasive Wear

Abrasive Wear is the result of hard particles coming in contact with internal components. Such particles include dirt and a variety of wear metals. Introducing a filtration process can reduce abrasive wear. It is also important to ensure vents, breathers, and seals are working properly.

Adhesive Wear

Adhesive Wear is generated when two metal surfaces come in contact allowing particles to break away from the components. Insufficient lubrication or lubricant contamination normally causes this. Ensuring the proper viscosity grade lubricant is used can reduce adhesive wear. Reducing contamination in the oil will also help eliminate adhesive wear. Cavitation occurs when entrained air or gas bubbles collapse. When the collapse occurs against the surface of internal components, cracks and pits can be formed. Controlling foaming characteristics of oil with an anti-foam additive can help reduce cavitation.

Corrosive Wear

Corrosive Wear is caused by a chemical reaction that actually removes material from a component surface. Corrosion can be a direct result of acidic oxidation. A random electrical current can also cause corrosion. Electrical current corrosion results in welding and pitting of the wear surface. The presence of water or combustion products can promote corrosive wear.

Cutting Wear

Cutting Wear can be caused when an abrasive particle has imbedded itself in a soft surface. Equipment unbalance or misalignment can contribute to cutting wear. Proper filtration and equipment maintenance is imperative to reducing cutting wear.

Figure 23-48 Cutting wear

Cutting wear appears as long, curly strips of material with aspect ratios ranging from 5:1 to 50:1 (length to width.) It is never considered to be normal.

Fatigue Wear

Fatigue Wear results when cracks develop in the component surface allowing the generation and removal of particles. Leading causes of fatigue wear include insufficient lubrication, lubricant contamination, and component fatigue.

Sliding Wear

Sliding Wear is caused by equipment stress. Subjecting equipment to excessive speeds or loads can result in sliding wear. The excess heat in an overload situation weakens the lubricant and can result in metal-to-metal contact. When a moving part comes in contact with a stationary part sliding wear becomes an issue.

Oil Analysis vs. Wear Particle Analysis

Figure 23-49 shows the relationship between standard oil analysis capabilities using Spectroscopy and Wear Particle Analysis. Spectroscopy only recognizes particles up to 6 microns and ignores the larger abnormal wear particles. For this reason it is a good practice to have Wear Particle analysis performed on oil samples.

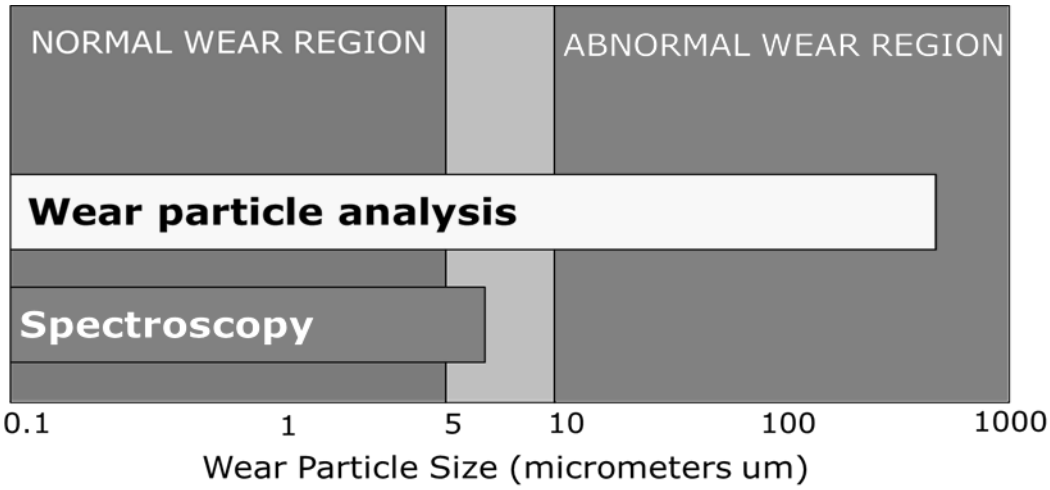


Figure 23-49 Normal oil analysis does not see abnormal wear particles.

Wear metals

The following table is a useful tool in finding possible sources of wear particles.

Wear Metal	Possible Origin
Aluminum	Bearings, Blocks, Blowers, Bushings, Clutches, Pistons, Pumps, Rotors, Washers
Chromium	Bearings, Pumps, Rings, Rods
Copper	Bearings, Bushings, Clutches, Pistons, Pumps, Washers
Iron	Bearings, Blocks, Crankshafts, Cylinders, Discs, Gears, Pistons, Pumps, Shafts
Lead	Bearings
Nickel	Bearings, Shafts, Valves
Silver	Bearings, Bushings, Solder
Tin	Bearings, Bushings, Pistons

Table 23-3

Wear particle analysis is a powerful tool for non-intrusive examination of the oil-wetted parts of a machine. It can detect particles from 1 micron to 350 microns. The analysis considers the particle shape, composition, size distribution, and concentration. The results aid in determining operating wear modes within the machine, resulting in specific maintenance recommendations. Wear Particle Analysis detects abnormal wear. The standard oil analysis detects normal wear particles up to 6 microns.

Figure 23-50

Selecting the Best Technology

Before selecting the technology it is best to step back and perform a review of all the plant equipment. Issues that must be considered are:

- reliability requirement
- the importance to the process
- whether there is redundant equipment
- physical accessibility and location
- hazards

All of these issues are financial issues. Everything must be justifiable financially. If the time and effort required to monitor a machine cannot be justified, don't monitor the machine.

Figure 23-51

Consider the **history of the machine**. Consider its reliability and failure modes along with ways to detect the failure modes. What is the impact of failure? Will there be losses due to downtime and secondary damage? What are the associated costs of parts and labor? What are the production downtime costs and capital costs involved? Is there a spare unit?

Risk analysis

Once you understand the consequences of a machine failure you can look into all of the ways to prevent or avoid that failure from happening. Somewhere in that analysis a balance will be struck between the cost of preventing the failure and the risk and associated costs of the failure occurring. Where this balance is struck and the line is drawn is basically the amount of risk you are willing to take.

For example, there is a risk associated with driving an automobile. Wearing a seatbelt is proven to greatly reduce the likelihood of serious injury if you should get into an accident. The cost of installing and wearing a seatbelt is low. Additionally, adding airbags to the vehicle will do even more to mitigate the risk. The cost is slightly higher, but not prohibitive by any means. A case could be made that if everyone drove 5 miles an hour, the risk of serious injury or death by automobile accident would be reduced significantly, but what are the costs? Most people would say the costs are too high and they are willing to live with some risk. This is essentially the same process one goes through when thinking about condition monitoring and CM technologies.

It should be noted that the risk levels may change during the year. At times of high demand, or adverse weather conditions (summer for cooling plant, winter for power generation, for example), the monitoring frequency and repair plan may need to change. In the same way that the risk of driving your car changes when the weather is bad or when your car is in need of new brake pads and tires.

Then consider the technologies required to detect future problems. Can you be successful with vibration? Should other technologies be included: infrared, wear particle analysis, etc.

	Technology						
Application	Vib	Lube	Wear	MCA	IR	US	Vis
Generator	☺	☺	☺	☹	☺	☺	☺
Turbine	☺	☺	☺	☹	☺	☺	☺
Pump	☺	☺	☺	☺	☺	☺	☺
Elec. motor	☺	☺	☺	☺	☺	☺	☺
Diesel eng.	☺	☺	☺	☺	☺	☺	☺
Fan	☺	☺	☺	☺	☺	☺	☺
Gearbox	☺	☺	☺	☹	☺	☺	☺
Cranes	☺	☺	☺	☺	☺	☺	☺
Elec. Circ.	☹	☹	☹	☺	☺	☺	☺
Transformer	☹	☺	☹	☺	☺	☺	☺

Table 23-4

The following table shows which technologies are good for specific fault types.

	Vib	Lube	Wear	MCA	IR	US	Vis
Wear	☺	☹	☺	☹	☹	☺	😐
Heating	☺	☺	☺	☹	☺	☹	😐
Impact	☺	☹	☺	☹	☹	☺	😐
Corrosion	☹	☺	☺	☺	☺	☺	😐
Fatigue	☺	☺	☺	☹	☹	☹	😐

Table 23-5 From Keith Young, paper in Maintenance Technology, June 1995