CHAPTER 17 - VIBRATION ANALYSIS
WRITTEN FOR THE POWER TRANSMISSION DISTRIBUTOR ASSOCIATION HANDBOOK

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Introduction

We are all vibration analysts. When a sound (which is a vibration) reaches our ears it is converted to an electrical signal that is transmitted to our brain where it is processed. We compare this vibration signal to our library of sounds and act accordingly. Is it the roar of a nearby lion or a gentle rain? Does the gearbox sound “different”? Likewise, the vibration transducers we use are “ears” that convert a mechanical movement to an electrical signal that is digitized, so it can be compared to a known library of faults.

This process may be straightforward at times, but it can also be exceptionally complex. Because no two machines respond exactly the same, machines vary over time or with operating conditions, the vibration analyst’s job requires skills of a detective, the stamina of a draught horse and a healthy dose of physics and mathematics to recognize clues and deduce what is happening to a machine.

The ability to record and analyze machinery vibration data has existed for a considerable time. The first efforts were directed at improving the balance of machinery as speed increased. As steam turbines replaced reciprocating steam engines attention was directed at monitoring the fluid film bearings used on turbomachinery. The high cost of equipment coupled with the cost of downtime made it easy to justify efforts directed at preventing premature turbomachinery failure.

However, the cost of equipment using rolling element bearings was typically lower than turbomachinery. Only a select few who could justify the bulky expensive hardware and time required to gather and analyze machinery vibration data for critical equipment other than turbomachinery benefited and even then to a limited degree. Therefore, development of monitoring technology for equipment using rolling element bearings lagged that of fluid film bearings until the inexpensive personal computer arrived in the late 1970’s.

By the early 1990’s, vibration analysis programs using PC based equipment were considered essential to large industrial plants as the low cost personal computer and hand held data collectors enabled engineers and technicians to monitor all the equipment in a plant. Sophisticated analysis became a reality on the plant floor enabling appropriate corrective action to be taken before an unexpected failure occurred, saving companies billions of dollars annually.

Today, vibration analysts work in some or all these areas depending on the type of problem, time, money, available software, hardware and his or her skill.

Balancing - Vibration analysis started with efforts to reduce the unbalance of wheels and rotors that could be seen as equipment coasted to a stop. Today many shops specialize in balancing rotors.

Turbomachinery – Much of the early work in vibration analysis centered on expensive turbine driven equipment where the time and expense could be justified. Analysts working on turbomachinery will often have considerable expertise in balancing and rotor dynamics.

Machinery Fault Diagnosis -As electronics were developed, and it was possible to convert mechanical vibration into electrical signals, and match characteristic electronic signatures to machinery and bearing faults. Most of the vibration analysis today is focused on fault identification and correction before equipment fails catastrophically.

Structural and Rotor Analysis - With the development of the computer, it became possible to utilize complex mathematics to model and measure the response of machines and structures to the forces generated during operation. Originally developed in the aerospace, aviation and military, this technology is has become available to analysts in industry.

The vibration analysis process can be organized into four key steps:

1. Converting the mechanical response of a machine to an electrical signal.
2. Processing the electrical signal into a visual display so its fundamental components may be seen.
3. Comparing those components to the specific features of the machine.
4. Deciding if an abnormal condition exists and taking the appropriate action.

Throughout this chapter, various terms will be used that may not be familiar. Vibration analysis is no different than other technical disciplines in that it has its own “language”. If you are not familiar with the terms, definitions can be found at the end of this chapter.

It is not the intent of this chapter to make you a vibration analyst, but to give you an introduction to the language and tools used to analyze machines so you can understand the principals and use them to make decisions that will improve the reliability and ultimately the profitability of plants and mills.

Balancing

Balancing may be considered the root from which all vibration analysis techniques have grown. In the early part of the 20th century, engineers and technicians recognized that a wheel or rotor having a spot that always rotated toward the bottom as it coasted to a stop ran rougher than one that did not. It was quickly found that removing or adding weight on the opposite side of the "heavy spot" until it would stop at any position or was "balanced" made the machine run smoother. As the speed of equipment increased they devised a number of mechanical and electrical devices to measure and quantify the amount of unbalance. As this practical work was progressing, researchers were able to derive mathematical expressions and theories that described the behavior of rotors. The melding of the practical and theoretical work around balancing and the behavior of rotating shafts has evolved in to the field called rotor dynamics.

Because there are variations in the manufacture and maintenance of rotating equipment, weight is not always equally distributed about the center of a shaft. As a shaft rotates, the center of mass or weight does not match the axis of rotation and vibration occurs. Although the term weight is not correct in a strict engineering sense, we intuitively understand the term weight so it will be used in this description. Balancing is a process in which the center of weight (mass) is moved by adding or removing material until it matches the axis of rotation.

In Figure 1, a disk is mounted on a shaft. The weight of the disk is equally distributed around the shaft. When the disk and shaft rotate, there is no vibration.

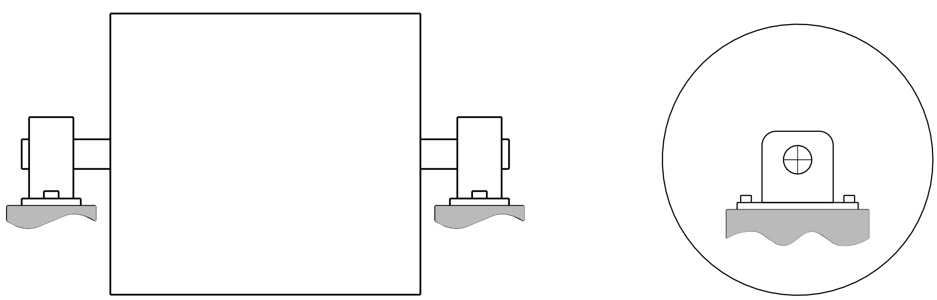


Figure 1
Rotor – No unbalance or vibration

If weight is placed on the disk, one side is heavier than the other and wants to fall to the bottom. When the shaft and disk rotate [Figure 2], the added weight pulls the center of the shaft toward it causing unbalance vibration.

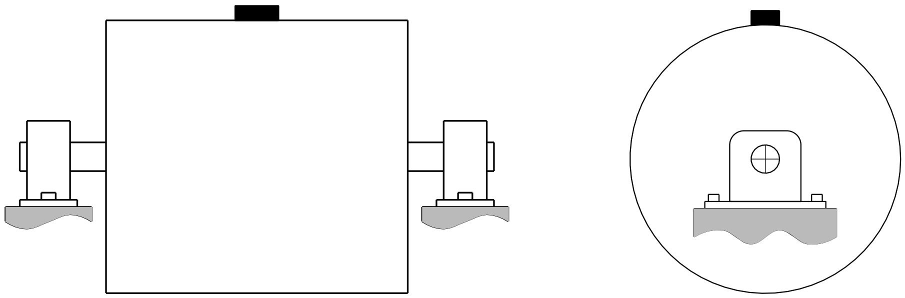


Figure 2
Rotor – Unbalance causing vibration

Similarly, if an equal amount of weight is placed on the disk exactly opposite of the original weight [Figure 3], the forces created by the two equal and opposite spinning weights as the shaft rotates are equal so they cancel each other and there is no vibration.

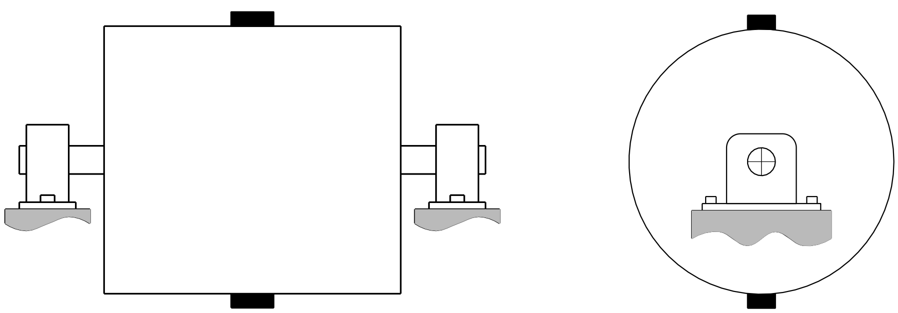


Figure 3
Rotor – Correction weight added, no unbalance

Because the early means of measuring the vibration caused by unbalance were mechanical, the displacement of the rotor was measured in inches. Displacement is simply the distance the center of a rotor moves from its axis of rotation while it is running. In the case of balancing, displacement is most often measured in thousandths of an inch (.001). Analysts often use the term mils when referring to displacement. One mil is .001". If a machine moving .006", this is the same as 6 mils.

Analysts use the term amplitude to describe how big the vibration is. In the case of our rotor the displacement amplitude is .006" or 6 mils.

It was quickly learned that the displacement of the unbalanced rotor went from a minimum position to a maximum as the rotor made one complete revolution. So in order to completely describe what is happening to the rotor, the terms period, frequency and running speed are used to measure the rate of rotation of the rotor.

We can combine the frequency and amplitude to create a graphical picture of the vibration caused by unbalance. A pivoted rod is attached to one of the bearing supporting a rotor [Figure 4]. At the other end of the rod is a pen that draws on a strip of paper that is moving past it. When the paper moves the pen draws a straight line until the rotor begins to turn. As the rotor turns the force created by the unbalance causes the rotor to move and the line on the paper records this movement. This record is called a time trace or time waveform.

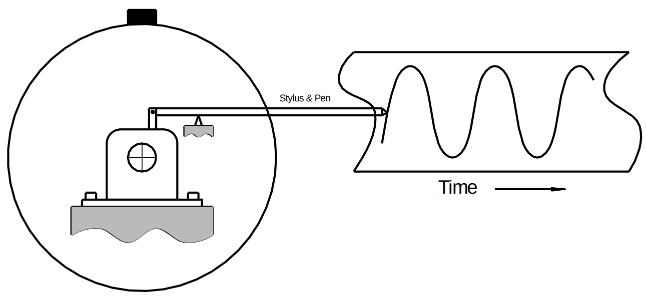


Figure 4
Pen connected to a rotor creating a time waveform

The trace of the movement on the paper allows us to measure the vibration via the displacement amplitude and the cycles or period of vibration. One cycle or period of vibration [Figure 5] in our unbalance example is the amount of time it takes the rotor or shaft to complete one revolution.

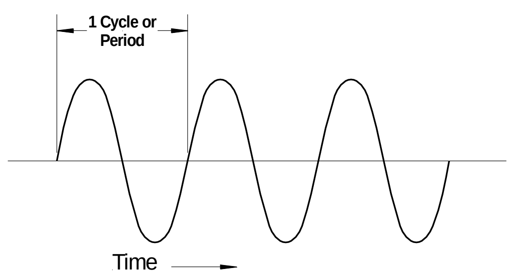


Figure 5
Time waveform

Vibration analysts use cycles to describe frequency in either cycles per minute (**CPM**) or cycles per second (**Hz**). The relationship between the two is:

1 Hz = 60 CPM

These can be easily converted between the two units by:

To convert from Hz to CPM…. multiply Hz by 60
To convert from CPM to Hz… divide CPM by 60

A motor running at 1775 rpm has a running speed of 1775 rpm or a frequency of 1775 CPM; these two are terms are directly interchangeable. However, if we use Hz then we have to divide 1775 by 60 to get a running speed of 29.583 Hz or 29.583 cycles per second.

Using the amplitude and frequency of the displacement, it is now possible to better understand what happens to a shaft or rotor in a machine while it is running

The time required to complete one cycle (period) of vibration is used to calculate the frequency of vibration. To calculate the frequency, the time required to complete one cycle (period) is divided into 1. If the time is measured in seconds, the calculated frequency is Hz or cycles per second. If the time is measured in minutes, the frequency is CPM or cycles per minute. These relationships can be easily remembered by using the “time / frequency” circle. In Figure 6, the time (T) of one cycle or period and frequency (F) are in the lower half of the circle. If the time is known, then it stays in the circle and is divided into 1 to calculate the frequency; likewise, if the frequency is known, it stays in the circle and is divided into 1 to calculate the period.

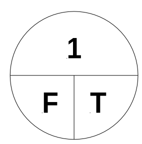


Figure 6
Time / frequency relationship

If a motor is running at frequency 1750 rpm, dividing 1 by 1750 equals 0.000571 minutes (the time it takes for one revolution) or converting to Hz by multiplying 0.000571 by 60 tells us one revolution takes 0.0343 seconds.

$$T = \frac{1}{F} = \frac{1}{1750} = 0.000571 \text{ minutes}$$

.000571 X 60 = 0.0432 seconds

Similarly, if it takes a motor 0.00087 minutes or 0.0522 seconds to complete one revolution, the frequency is 1 divided by 0.00087 minutes which equals 1150 cpm or 1 divided by 0.0522 seconds which equals 19.1667 Hz.

$$F = \frac{1}{T} = \frac{1}{0.00087} = 1150 \, \text{CPM}$$

$$\frac{1}{0.0522} = 19.1667 \text{ Hz}$$

Remember, it is important to keep the units (seconds / Hz or minutes) consistent when making these conversions.

At this point, a lot has been learned about the rotor, but we need to know where to put a weight to counteract the unbalance force. Unlike our example, we can't see the heavy spot so we don't know where the heavy spot is. We have to have some way to measure the location of a reference point on the rotor, so we will know where to place a weight to counteract the unbalance force caused by the weight.

A mark is put on the rotor [Figure 7] that triggers a pen to mark the chart created while the rotor is running. This mark called a key phasor is used to create a corresponding mark on the graph created at precisely the same time each revolution. This allows us to measure the distance or phase between the mark and the maximum vibration amplitude created by the unbalance.

Phase is measured in degrees. When the rotor has made one complete revolution or cycle, it has turned 360º. Now, each rotation of the rotor in the time trace can be measured in degrees on the time waveform chart and the heavy spot located.

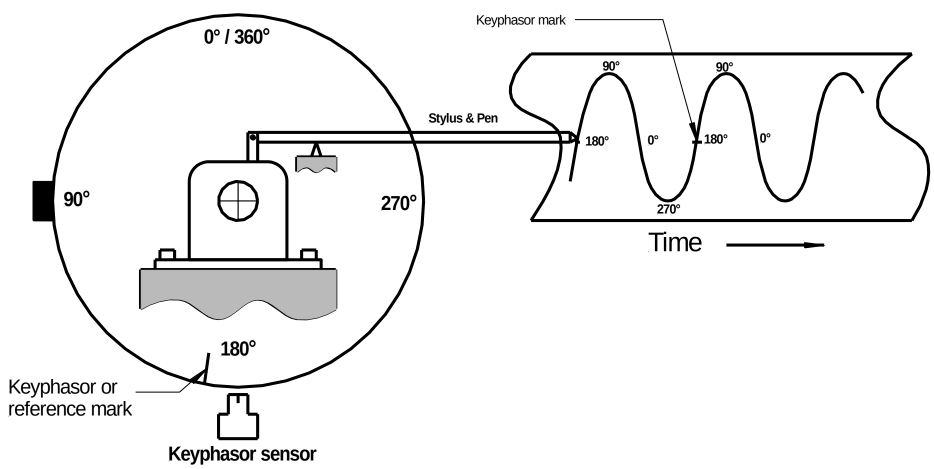


Figure 7
Rotor and trace in degrees showing key phasor reference

To balance the rotor, the location of maximum amplitude from the key phasor can be found using the time trace and a weight placed opposite of this location to reduce the unbalance.

The analyst uses sensors to measure the amount and location of the weight causing the vibration. An equivalent amount of weight may be added exactly opposite of the heavy spot or alternatively, material may be removed from the heavy spot.

The unbalance described in the preceding example is called mass unbalance or static unbalance. It is by definition a situation where the unbalance weight is concentrated in one plane at the center of the rotor [Figure 8]. It can be corrected simply by placing an equivalent amount of mass (weight) exactly opposite.

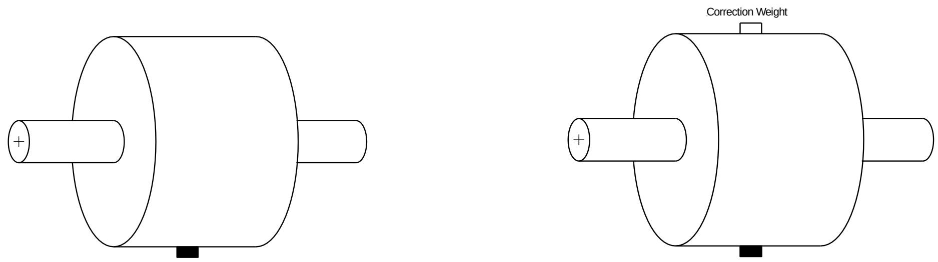


Figure 8
Mass or static unbalance

Another type of unbalance occurs when two unbalance weights are located at the opposite ends of the rotor 180º apart [Figure 9]. This is called couple unbalance because the two opposing weights cause the rotor to rock about its axis. To correct this type of unbalance weight must be added at both ends of the rotor.

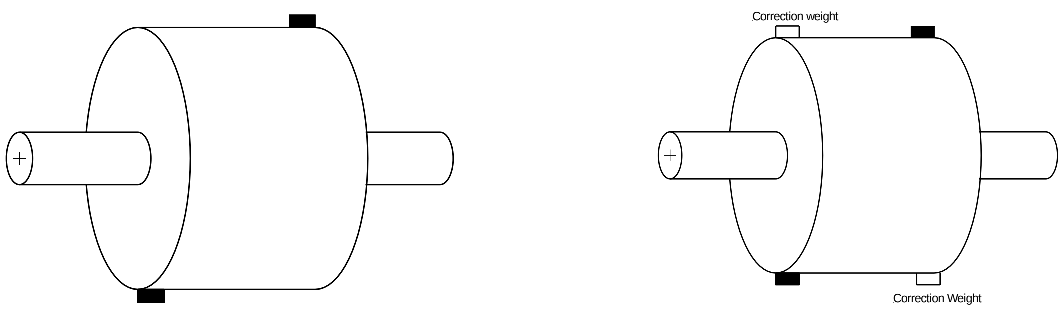


Figure 9
Couple unbalance

The most commonly occurring type of unbalance [Figure 10] is a combination of static and couple unbalance called dynamic unbalance. In this case, the weights are not located exactly in the center of the rotor nor are they 180º apart. Analysts use a variety of techniques (two plane, static / couple) to determine where correction weights should be placed.

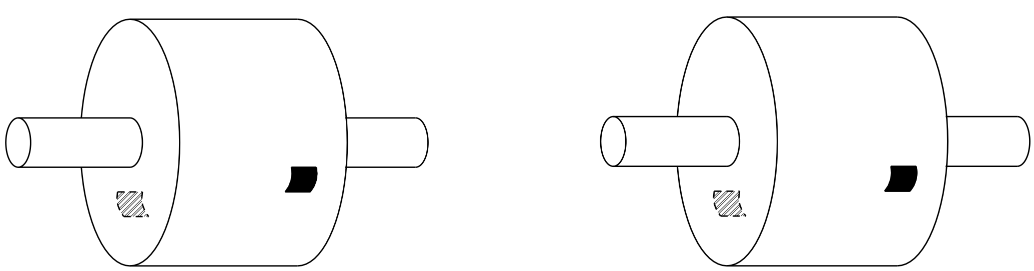


Figure 10
Dynamic unbalance

Balancing brings two significant benefits to a plant or mill: increased equipment life and reduced energy consumption. The force created by an unbalanced rotor increases as the square of its speed; if the speed is doubled, the force increases four times. Force created by unbalance travels through the bearings of a machine where it is dissipated in mechanical motion and heat. Unfortunately, when the force on a rolling element bearing is doubled the life of the bearing decreases by approximately 8 times. Power is directly related to force, so significantly more power is required to run an unbalanced machine. Power savings are important; ask any race car driver how important a balanced engine is!

Economics, design and precision do not allow us to balance every machine to perfection. A tolerance for unbalance must be set just as tolerances are set for fabrication of parts in a machine shop. The quality of balance may be specified using several different standards.

The first standard developed and perhaps the most common is the ISO standard. It specifies the amount of residual balance that is acceptable for the service conditions. The balance quality grades (G) range from the best G0.4 to the lowest G4000. The balance quality grades in Figure 10 are described in terms of the allowable amount of residual unbalance that is that is measured in units of mm/sec. This means the center of a shaft balanced to a G2.5 grade would vibrate at a velocity or speed of 2.5 mm/sec if it were suspended in space.

Here is where things get tricky. We just talked about a specification written in units of velocity, but the actual process of balancing requires the addition or removal of weight. Hence, the charts [Figure 11] converting the unbalance grade from units of velocity to something that relates to the physical changes to the rotor; namely the addition or removal of weight at some distance from the center of the rotor. The scales on the sides may be in written in units of $\frac{\text{oz.} - \text{in}}{\text{lb}}$ or equivalent. To the balance specialist, this relates the amount of weight that must be added or removed from a rotor and how far it must be from the center of the shaft per unit of rotor weight.

However, for the vibration analyst this cannot be related to the vibration amplitude without what is known as the influence coefficient. The influence coefficient is a function of the support stiffness and describes how sensitive a rotor is to an addition of weight. Because the stiffness of a shop balancing machine does not change, it isn't necessary to calculate it for every rotor. When rotors are balanced in the field, the supports for two identical pieces of equipment may be different so they will not respond the same to an identical weight. A trial weight is attached and the response of the rotor is measured so the influence coefficient may be calculated and the proper weight correction can be made.

To use the chart in Figure 11, find the operating speed across the X or horizontal axis at the bottom of the chart. Follow the line upward from that speed to the desired balance quality grade line that runs diagonally across the chart. The values on the vertical axis are the amount of residual weight permitted per unit weight of the rotor to achieve the required balance level. Where these lines intersect, follow a horizontal line to the left if you are measuring the correction weight in the same units used to weigh the rotor and to the right if the correction weights are in grams and the rotor weight is in lbs.

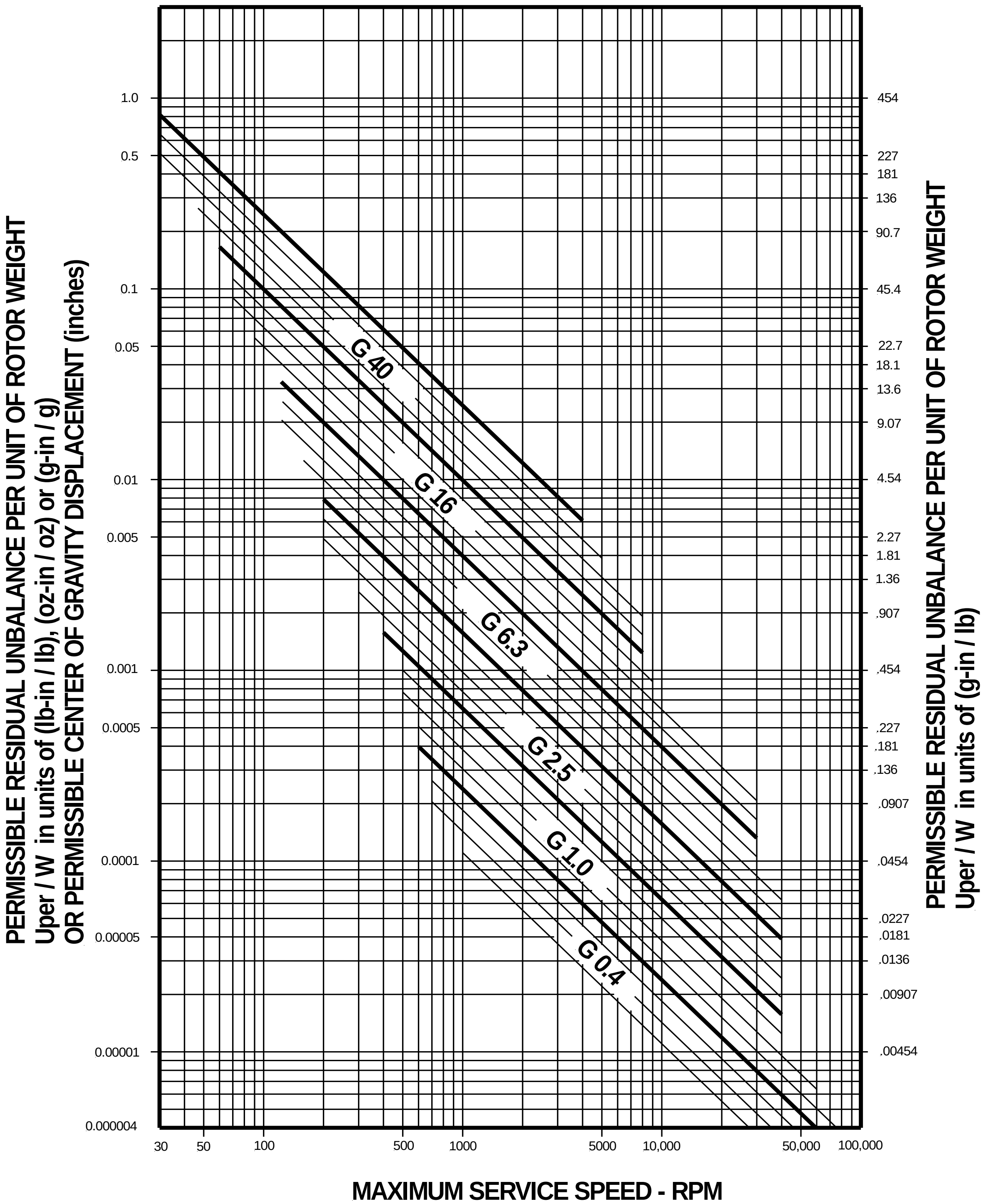


Figure 11
ISO Balance quality grades

When a balance specification is written using the ISO standard, it must specify the quality grade G1.0, 2.5, 6.3…. and how the residual unbalance is distributed between the correction planes. With the exception of low speed rotors that have a large diameter compared to the length, most rotors are balanced in more than one plane. Consequently, the balance tolerances must be divided between the number of correction planes. Figure 12 is a guide for the number of correction planes required. The length to diameter ratio is calculated exclusive of the shaft length.

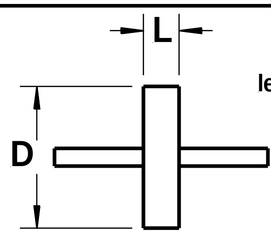
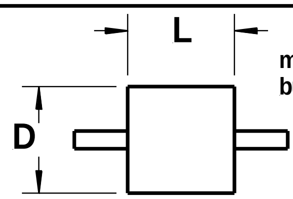
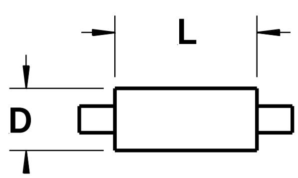
	SELECTION OF CORRECTION PLANES		
	SINGLE PLANE	TWO PLANE	MULTIPLANE
 L / D less than 0.5	0 - 1000 RPM	ABOVE 1000 RPM	NOT APPLICABLE
 L / D more than 0.5 but less than 2	0 - 150 RPM	150 - 2000 RPM OR ABOVE 70% OF 1ST CRITICAL	ABOVE 2000 RPM OR ABOVE 70% OF 1ST CRITICAL
 L / D more than 2	0 - 100 RPM	ABOVE 100 RPM TO 70% OF 1ST CRITICAL	ABOVE 70% OF 1ST CRITICAL

Figure 12

Determining the number of correction planes using the length to diameter ratio and service speed

The term “critical” also termed critical speed is used in Figure 12. A more complete discussion is included in the section on structural and rotor dynamics. For now, consider a critical speed as a point where vibration will increase dramatically as the speed of a rotor is increased. It is commonly encountered on large turbomachinery and high speed equipment.

Another commonly used balancing standard is found in the American Petroleum Institute (API) standard 610 "Centrifugal Pumps for General Refinery Service". This standard determines the amount of permissible unbalance by dividing 4 times the rotor weight by the rpm of the rotor as shown in the following formula.

$$U_{per} = \frac{4W}{N}$$

Where W = Rotor weight in oz.
N = Rotor speed in rpm
U_{per} = Permissible unbalance in oz.-in

Again because the API standard specifies the total amount permissible unbalance, it must be divided by the number of correction planes using the table in Figure 12.

Improvements in balancing equipment have made it possible to routinely reduce unbalance to levels that at one time were reserved for high speed precision machines allowing balancing specialists to produce better results in the same or less time. Also, since the ISO standard was first written in 1948, the precision of industrial equipment has evolved. Therefore, many industrial plants and manufacturers now routinely specify the ISO G1.0 or the API standard for process equipment that just fifteen years ago was often balanced to a lesser G6.3 grade. The API standard which is used in the petroleum industry will result in slightly lower residual unbalance than the ISO G1.0 standard. As the drive for greater reliability continues, ISO quality grades that were once considered acceptable for various classes of machines no longer meet today's requirements.

Machinery Fault Diagnosis

Before describing the different types of machinery faults and their characteristics, we need to introduce some more measurement tools used by the vibration analyst and look closer at how the mechanical vibrations are converted in to a display on the computer.

Up to this point, we have described vibration of a machine using the displacement or distance a machine moves and the velocity or speed it moves. It would be nice to describe all vibration in displacement terms, but physics gets in the way so we have to use different units of measurement to describe the vibration.

A slow moving machine does not have to move very fast to cause damage. For example, a paper clip flexed 1 inch does not have to be bent very many times before it breaks so we can bend it at a slower pace. Thus, displacement is typically used to measure vibration of equipment operating at relatively low speeds usually less than 600 cycles per minute (CPM) or 10 Hz.

If we bend another paper clip only a quarter of an inch, we must bend it many more times (faster) so it breaks in the same amount of time. So, we must also use other measurement units to describe vibration of a machine. In vibration analysis the term velocity is used to measure the speed at which our paper clip is bent.

Velocity is the most commonly used unit of vibration severity for the operating speed ranges of most industrial equipment. It is expressed in inches per second and usually abbreviated ips. Machinery operating from 600 CPM (10 Hz) to 60,000 CPM (1,000 Hz) is generally best measured using velocity.

Finally, as the speed of our equipment increases even more, there is simply not enough time for large changes in velocity and especially displacement. However, the forces involved can be large so a third type of measurement called acceleration is used. Acceleration is a measure of how fast the velocity changes. It is not the speed of the fall (velocity) that hurts, but the stop at the end (acceleration) when our velocity rapidly drops to zero.   Acceleration is measured in units of gravity called g's. Acceleration is generally the best indicator of vibration above 60,000 CPM (1,000 Hz).

Displacement, velocity and acceleration are mathematically related. Methods to convert from one unit to another can be found in vibration analysis textbooks.

Machinery fault diagnosis began as an offshoot of balancing as the instrumentation was developed. The analyst or reliability professional if often called to answer the question: how much vibration is too much? Various charts and tables were developed from empirical data as guides to help evaluate a machine's overall health and energy consumption.

One of the more familiar graphs is the vibration severity chart shown in Figures 13. The chart [Figure 13] defines categories ranging from extremely smooth to very rough, but does not tell us what is normal for a particular type of machine. An acceptable vibration level for a cooling tower fan would be unacceptable for a precision grinder.

Several features of the chart in Figure 13 should be noted: In terms of displacement (the left hand side), tolerable vibration levels decrease with increasing frequency. But, the allowable vibration when measured in velocity using the diagonal lines remains constant as the speed increases. Because the chart was designed for filtered readings, it should not be applied to overall vibration levels, but to the vibration amplitude at running speed. This and most other similar charts were constructed for typical machines with a casing to rotor weight ratio in the order of 5:1. Failure to observe these limitations can lead to incorrect conclusions.

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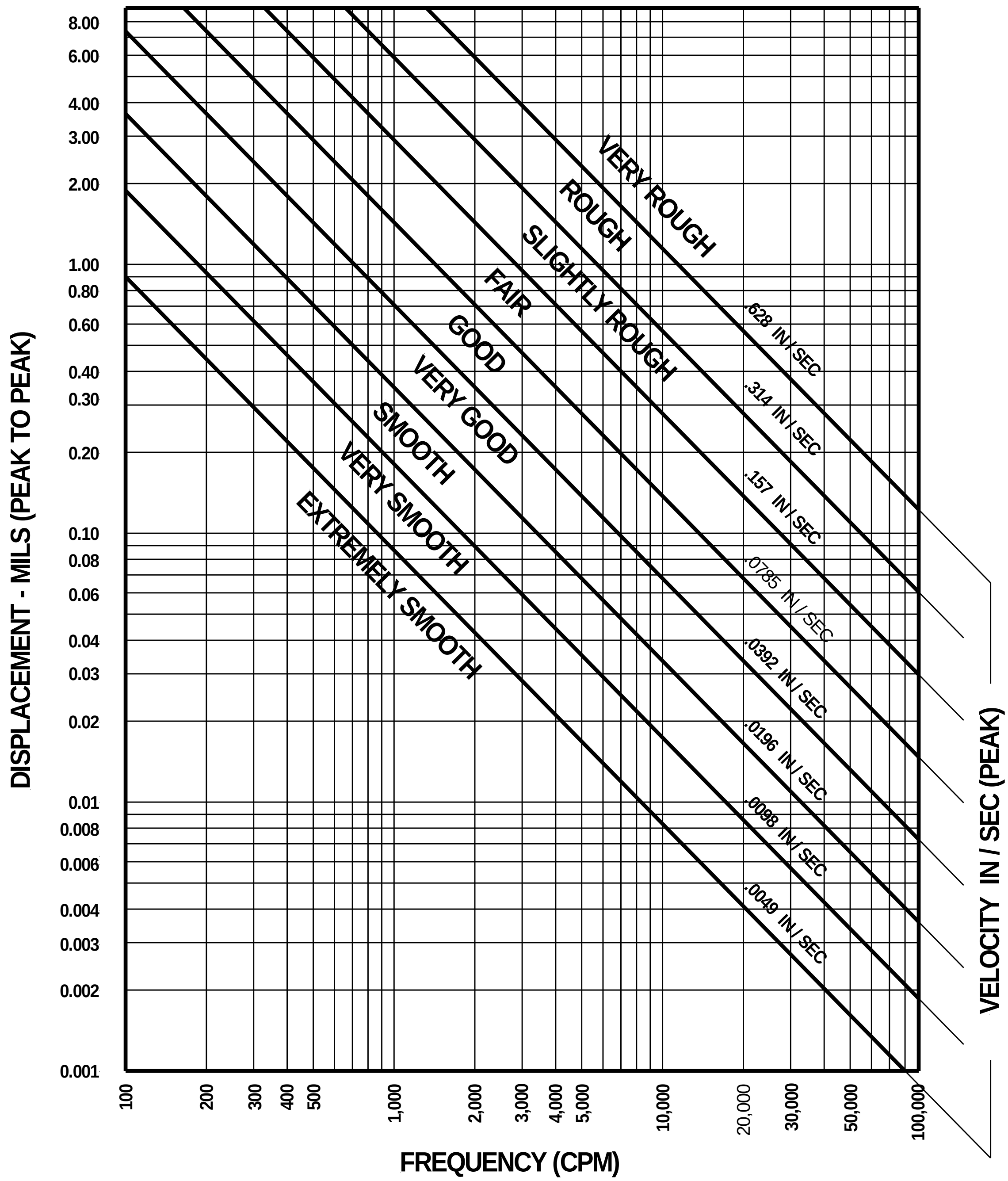

Figure 13
Vibration severity

A more recent chart derived from ISO standards [Figure 14] better addresses the type and mounting of machines using velocity measurements. Because frequency is inherent in

the velocity measurement, unfiltered readings may be used. The velocity values in this chart have been converted from the metric values to peak velocity values typically used in the US.

Vibration Severity	Velocity Range Limits & Machine Class Designations ISO Standard 2372-1974			
Inches / Second (Peak)	Small Machine Class I	Medium Machine Class II	Large Machines: Rigid Supports Class III	Large Machines: Flexible Supports Class IV
0.020	Good	Good	Good	Good
0.030	Good	Good	Good	Good
0.040	Good	Good	Good	Good
0.060	Satisfactory	Satisfactory	Good	Good
0.100	Unsatisfactory	Satisfactory	Satisfactory	Good
0.160	Unsatisfactory	Unsatisfactory	Satisfactory	Satisfactory
0.250	Unacceptable (Danger)	Unsatisfactory	Unsatisfactory	Satisfactory
0.400	Unacceptable (Danger)	Unacceptable (Danger)	Unsatisfactory	Unsatisfactory
0.620	Unacceptable (Danger)	Unacceptable (Danger)	Unacceptable (Danger)	Unsatisfactory
1.000	Unacceptable (Danger)	Unacceptable (Danger)	Unacceptable (Danger)	Unacceptable (Danger)
1.560	Unacceptable (Danger)	Unacceptable (Danger)	Unacceptable (Danger)	Unacceptable (Danger)
2.500	Unacceptable (Danger)	Unacceptable (Danger)	Unacceptable (Danger)	Unacceptable (Danger)
3.950	Unacceptable (Danger)	Unacceptable (Danger)	Unacceptable (Danger)	Unacceptable (Danger)

Machine Classes: ISO 2372-1974

I. 0.250 ips: Small machines, Electric motors up to 20 hp.
II. 0.395 ips: Medium machines, Electric motors 20 to 100 hp.
III. 0.623 ips: Large / Rigid support machines.
IV. 1.000 ips: Large / Flexible support machines / Turbomachinery.
V. 1.560 ips: Reciprocating / Rigid support machines.
VI. 2.500 ips: Reciprocating / Flexible support machines.

Figure 14
Current ISO standards for unfiltered overall vibration levels

In the early evolution of vibration analysis more complex diagnostics could be done, but it required expensive bulky swept filter signal analyzers and a substantial amount of time to collect and analyze data from even one machine. This limited more detailed analysis to the military (submarines, aircraft) and machines in industrial plants where the consequences of failure were extremely high. Unfortunately for the rest of the equipment, by the time a fault had reached a level where it would cause a change in the overall vibration level, the damage was well underway and failure usually occurred before corrective action could be scheduled.

The development of microprocessors and advent of the personal computer did for the vibration analysis field what the Model T did to automobiles. These now affordable tools enabled analysts to easily convert a complex time recording to a frequency format quickly and easily. This led to a dramatic increase in the understanding of faults found in rotating equipment and most importantly increased the time from when a fault was first detected to failure.

The cornerstone of this analysis is the conversion of the time trace or time waveform to a frequency spectrum. This conversion is very similar to a prism that breaks light into its various colors. Vibration analysts rely on the computer and software to convert a time waveform into its various frequencies using a Fast Fourier Transform or FFT. Just as a prism separates light into its different colors (frequencies) of varying brightness (amplitudes) the FFT breaks a complex vibration signal into its different parts.

A complex time waveform [Figure 15] can be separated into a series of simple sine waves using the FFT mathematical process. The complex waveform in this example has a low frequency component, and several higher frequency components. If we rearrange these similar to what a prism does to light using a FFT, we can see how the time waveform is separated into its component parts (bands of different colored light). Each of these parts or spectral peaks now has a frequency (color) and amplitude (brightness). We us the term “frequency domain” to describe the resulting graph when we convert a complex signal in time units to a series of simple signals in frequency units.

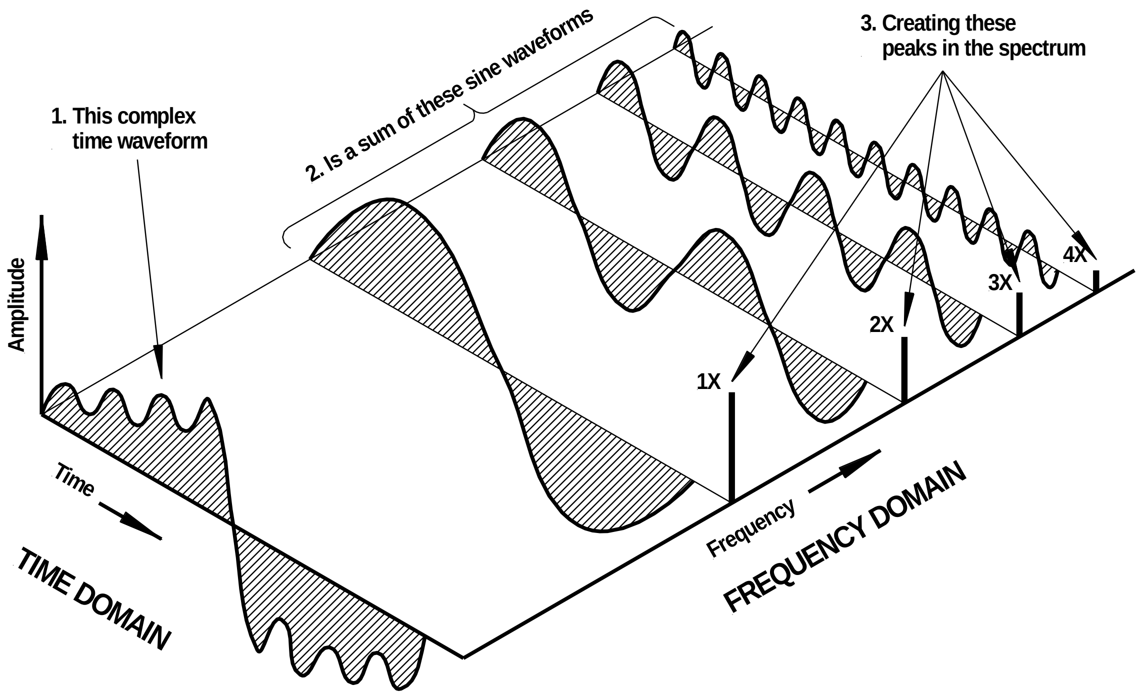

Figure 15
Conversion of a time waveform to a frequency spectrum using the FFT (Fast Fourier Transform)

Applying the FFT to the time waveform of a single reduction gearbox running at 1725 rpm with unbalance we have a spectrum that has two peaks: unbalance, and gear mesh [Figure 16]. The vibration analyst looks for peaks in the frequency spectrum; the presence or absence of peaks helps the analyst to determine the condition of the machine.

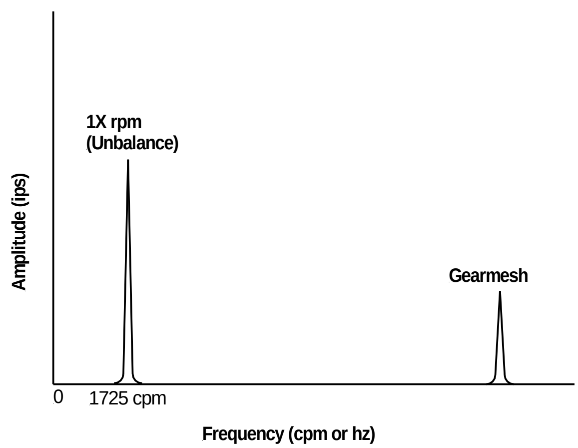

Figure 16
Gearbox spectrum showing unbalance & gear mesh frequencies

If a bearing develops a flaw on the outer raceway, this adds another peak to the spectrum shown in Figure 17. Through training and experience, the analyst knows that this peak should not be present and a bearing failure has begun to develop.

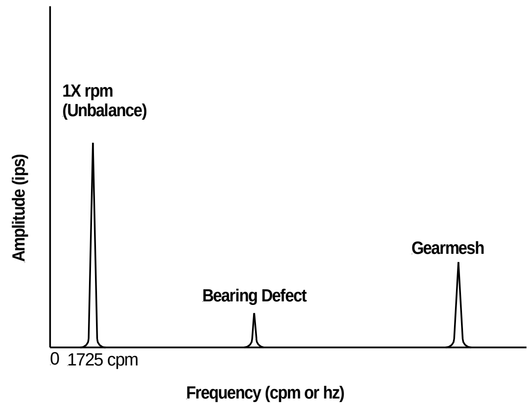

Figure 17
Spectrum showing unbalance, bearing defect & gear mesh frequencies

With these tools of frequency and the amplitude of displacement, velocity or acceleration, we can begin to accurately diagnose the condition of machinery using frequency spectrums. Combine this with the concept of phase discussed in the balancing section and it is possible to pinpoint a variety of machinery faults before they deteriorate into an unplanned failure.

Before listing all of the machinery fault spectrums, let's take a look at one of the more common faults to illustrate how this process works and to introduce two additional concepts that will add to your vibration analysis tool kit.

If we connect two machines together, using a flexible coupling and do not have the centers of the shafts perfectly aligned, our vibration spectrum looks like the one shown in Figure 18. In addition to a peak at the running speed of the machines (which could be measured in displacement, velocity or acceleration depending on the speed) there are additional peaks. These peaks are at multiples of the machine running speed.

If the two coupled machines are running at 1800 RPM (CPM), the first peak is at 1800 CPM, the second at 2X 1800 CPM or 3600 CPM the third at 3X 1800 or 7200 CPM. These peaks are called **harmonics** because they are whole number multiples of the running speed. The presence of harmonics is an important feature that can be used in the identification of many machinery faults.

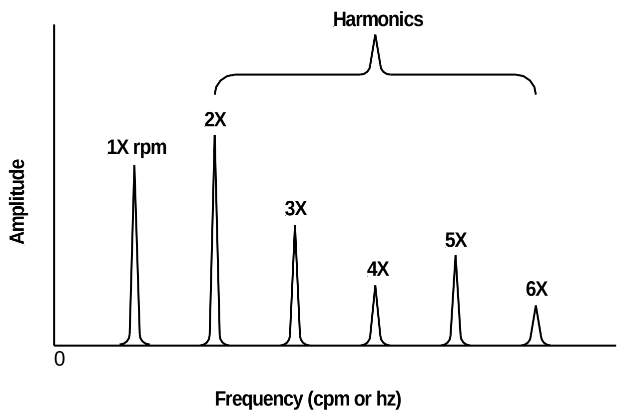

Figure 18

Misalignment spectrum showing harmonics of running speed

Another important feature that is often seen in vibration spectrums are sidebands. Sidebands are uniformly spaced lower amplitude peaks on either side of a peak in a spectrum. In Figure 19 the sidebands are located near the gear mesh frequency. A sideband is created when one frequency is modulated by another. In Figure 19 the contact of one gear with another generates a peak in the spectrum at a frequency of 1 X the shaft RPM times the number of gear teeth. If a gear running at 1,000 rpm has 47 teeth, the gear mesh frequency is 1,000 X 47 = 47,000 CPM. Because of small variations in the manufacture of the gear, there is a slight change in the amplitude of the gear mesh frequency each revolution. This creates sideband peaks in the spectrum spaced 1000 CPM away from the gear mesh frequency.

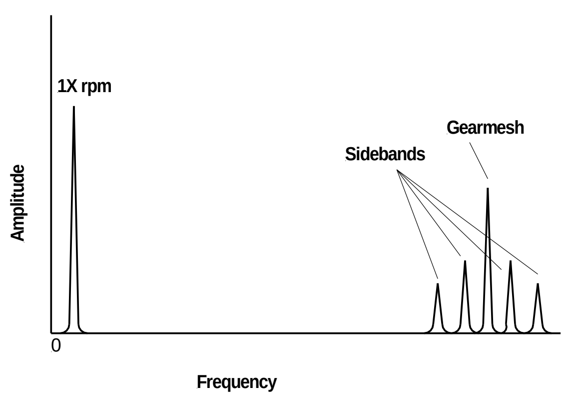

Figure 19

Gear mesh frequency with sidebands

With the tools we now have in our toolbox, we are ready to begin diagnosing some machinery faults. Fault diagnosis is a process of comparing a spectrum to one of the known faults. Although this sounds straightforward, it is like any other identification; what the analysts sees in the field does not match the "textbook" examples.

The following section does not include all of the faults; one comprehensive reference to other faults may be found references such as the "Vibration Fault Guide" published by Full Spectrum Diagnostics.

Unbalance

Unbalance occurs when the rotation about the center of mass does not match the geometric center of rotation determined by the bearings. Depending on the type of unbalance (static, couple or dynamic) the axis of mass and geometry may or may not intersect. However, in all of these cases the forces created by the unbalance result in a spectrum [Figure 20] with a high 1X peak. Confirmation of unbalance is done by taking another measurement 90 degrees apart on the same bearing. There should be a 90 degree phase shift with a margin of ± 30 degrees. To determine if the unbalance is static, couple or dynamic, additional phase measurements must be taken between the bearings. Comparison of phase angles between these enables the analyst to complete the diagnosis.

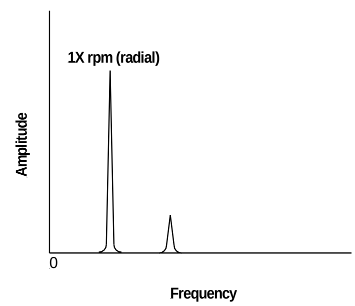

Figure 20
Unbalance spectrum

Misalignment

Misalignment occurs when the centerlines of two rotating shafts have a parallel, angular or an angular offset at the point where power is transmitted [Figure 21]. As the shafts try to find a common centerline, the abnormal forces produce excessive vibration levels and eventually cause premature failures of bearings, shafts, couplings, seals, etc.

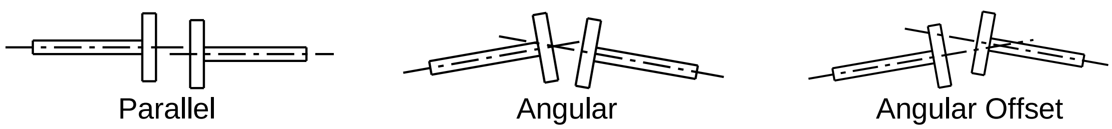

Figure 21
Types of shaft misalignment

In the majority of situations, misalignment is indicated by an excessive 1X RPM amplitude, an excessive 2X RPM and harmonics of the running speed [Figure 22]. The amplitudes and number of harmonics depend on the amount of misalignment as well as the type of coupling used. Comparison of the phase angle between the horizontal, vertical and axial readings at 1X RPM is made to determine the type (horizontal, vertical, angular or a combination) of misalignment.

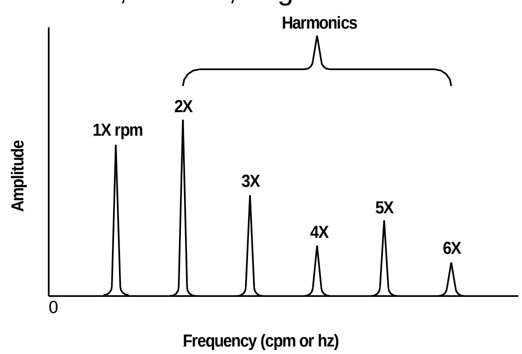

Figure 22
Misalignment spectrum

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Looseness

A basic looseness condition often called "Type A" [Figure 23] has a dominant peak at 1X RPM and can be misdiagnosed as unbalance. The greatest response is typically localized at a single machine component measured in the vertical direction. This type of looseness is typically found at connections between machine components or bases.

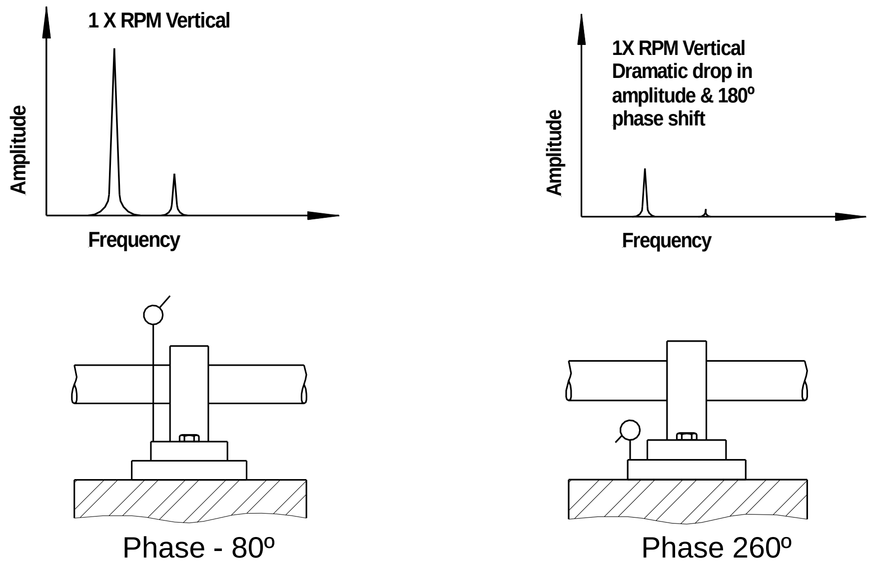


Figure 23
Type A Looseness spectrum

Looseness is verified by phase analysis across the machine joints such as bolted connections, welds and grout lines. The phase does not shift 90º in a horizontal to vertical comparison that typically signifies unbalance. But, the phase will shift across the loose joint by 180º $\pm$ 30° and a dramatic change in magnitude will occur.

Another type of looseness called "Type B" [Figure 24] is not as easy to diagnose. It can produce a spectrum that looks similar to misalignment with a high 2X RPM peak and frequently fractional sub-harmonics. If severe enough, it may also create alternating high and low harmonic peaks. Phase is erratic phase and measurements across the suspected failed interface are used to diagnose this type of looseness. This looseness is usually caused by cracks in housings or pedestals, loose pedestal bolts or faulty mechanical isolators.

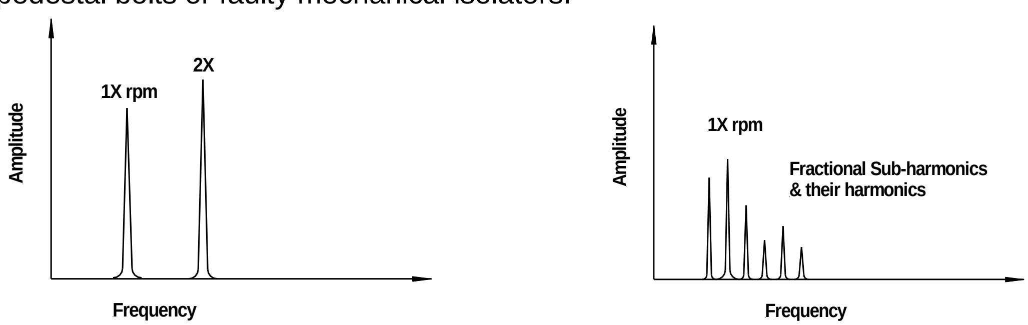
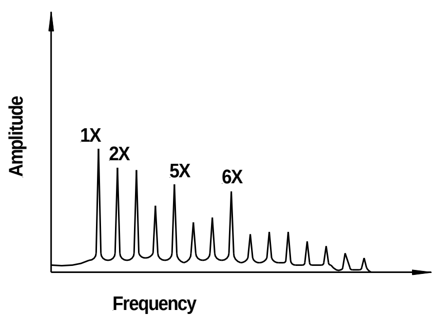


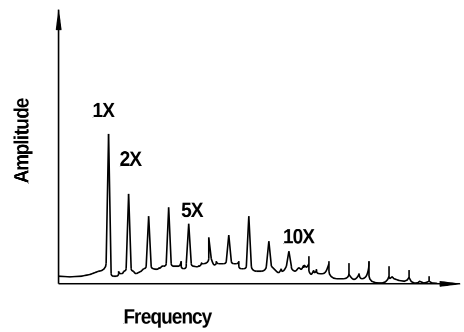
Figure 24
Type B mechanical looseness

Type C mechanical looseness [Figure 25] is caused by poor fit between a coupling, hub, impeller or antifriction bearing and its shaft or housing. It is recognized by numerous 1X RPM harmonics and a raised noise floor in the spectrum. A time waveform can be used to verify this type of looseness: look for truncation of peaks, sharp repetitive impact peaks and random patterns between peaks. Acceleration amplitudes less than 2 g's usually indicate misalignment while those above 2g's indicate looseness.

Phase readings will be unstable and the 90º phase shift from horizontal to vertical commonly found in unbalance will not occur.



Harmonics & "looped" noise floor



1X Harmonics and raised noise floor

Figure 25
Type C mechanical looseness

Rolling element bearings

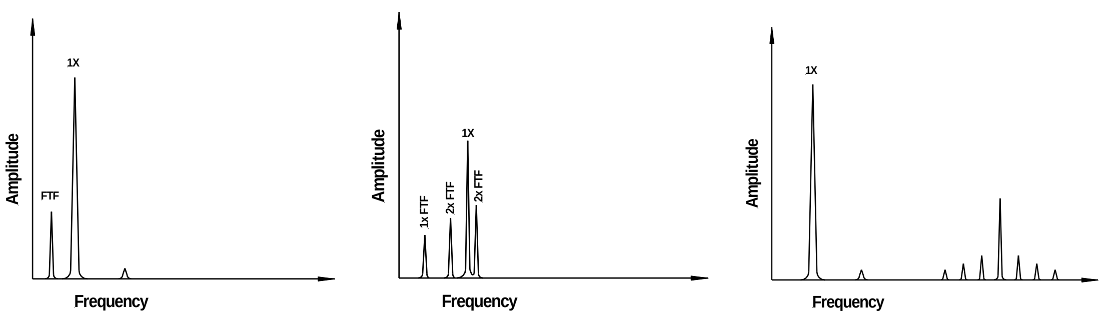
Defects that occur in rolling element bearing spectrums are usually caused by damage and occasionally normal wear. The resulting defect peaks that occur in a vibration spectrum are directly related to the bearing geometry (raceway pitch diameter, rolling element diameter, number of rolling elements and thrust angle).

Four coefficients determined by the geometry can be calculated and used to determine the location of the defect: inner raceway, outer raceway, rolling element and cage. These coefficients are available from all of the major bearing manufacturers.

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

These coefficients are multiplied by the 1X RPM to determine the defect frequencies in the spectrum. These non-synchronous peaks (a non-synchronous peak will <u>not</u> be a whole number multiple of running speed) are one of the important features that allow an analyst to identify rolling element bearing defects.

The primary function of a bearing cage is to guide and separate the rolling elements. The Fundamental Train Frequency (FTF) is always lower than the RPM of the shaft or housing (outer ring rotation). For inner ring rotation the FTF defect frequencies fall in the range of 0.33X – 0.48X RPM; for outer ring, 0.52X – 0.67X RPM. This type of defect [Figure 26] may be present as an individual peak, harmonics or sidebands around another spectral peak.



FTF by itself   FTF harmonics   FTF sidebands on another bearing defect

Figure 26
FTF defect - velocity spectrums

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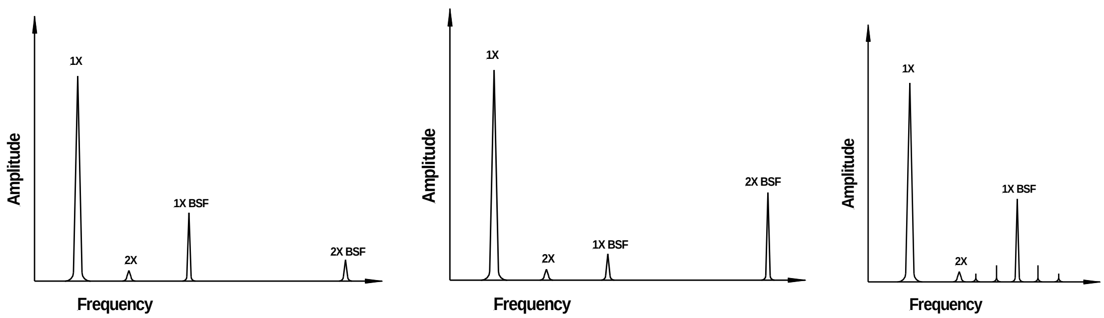
The rolling elements in an antifriction bearing are manufactured to match the geometry of the inner and outer ring raceway so they can smoothly and efficiently transfer the load between the raceways. A single defect on a rolling element [Figure 27] will contact both the inner and outer raceways every revolution typically generating a dominant 2X BSF frequency.

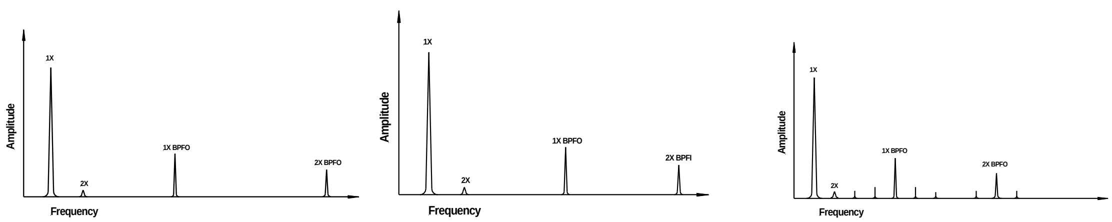


Dominant 1X BSF | Dominant 2X BSF | BSF with FTF Sidebands

Figure 27

BSF fault - velocity spectrum

The raceways of the inner and outer rings in an antifriction bearing are designed with curvatures that match the rolling element shape. When a rolling element in a bearing passes over a defect in one of the raceways, a peak is created in the spectrum [Figure 28]; for a given bearing, the frequency of the outer ring peak will always be lower than the inner ring peak.



BPFO fault & harmonic | BPFO & BPFI faults | BPFI with 1X RPM sidebands

Figure 28

Raceway faults - velocity spectrum

Because damage can and many times does occur on more than one component of a bearing, combinations of spectral peaks caused by the different components can be found in a spectrum from a single bearing. However, rolling element bearing deterioration follows a fairly predictable pattern. Generally speaking, most bearing failures progress through four stages each of which has distinguishing characteristics.

The signal processing algorithms used to monitor rolling element bearings have continued to evolve over the past decade to the point where it is possible to detect rolling element bearing deterioration well in advance of failure with a high degree of accuracy.

Up to this point, velocity spectrums have been used to illustrate the various types of bearing defects. However, the earliest indication of bearing distress most often begins in the high frequency ultrasonic or acoustic emission frequency range(s). The damage begins either on or just below the surface(s) and may not be visible to the unaided eye. As the damage progresses, the high frequency energy increases in response to impact, inadequate lubrication film, developing surface damage, or subsurface faults.

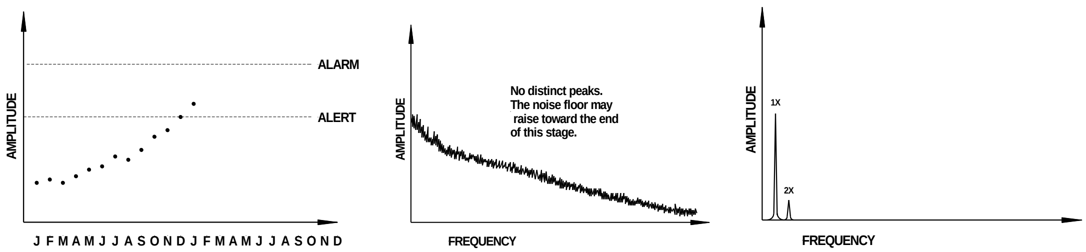
An ultrasonic demodulated spectrum is often used to detect this early stage of deterioration. Demodulation is a signal processing technique used to remove the low frequency signal from an incoming signal and process the remaining signal into a form that can be displayed in a low frequency spectrum. This removes all of the information caused by what are considered low frequency events like unbalance, misalignment, looseness, motor defects, bent shaft etc. The remaining very high frequency but low amplitude signal caused by bearing defects is analyzed by constructing an envelope around them and taking an FFT of the envelope.

Various signal processing algorithms are used to accomplish this: PeakVue (CSI), GSE Spectrum (ENTEK / IRD), Acceleration Enveloping (SKF) are some of the few available. These systems use a variety of filters and signal processing algorithms to separate the low amplitude high frequency content from the high amplitude low frequency content allowing the vibration analyst to detect defects far in advance of failure. So, when a defect is first found one using one of these algorithms, typically about 20% of the L_{10} bearing design life remains.

This advanced detection is not without issues. Because the high frequency content of the vibration is used, mounting of the accelerometer is important to prevent signal loss and consistent data. For good consistent results, the accelerometer mounting surface must be flat, clean, and free of paint. If it is not, the high frequency content of the vibration can be lost or data will not be consistent making trending difficult if not impossible.

Stage I - (Figure 29)

If a bearing is removed at this stage, there will be no visible indications of damage. Typically, there are no visible defect frequencies in the velocity spectrum; however, low level peaks may be visible in the acceleration spectrum. The overall ultrasonic trend has started to climb, but distinct peaks are not visible in the ultrasonic demodulated spectrum.



Overall ultrasonic trend

Ultrasonic demodulated spectrum

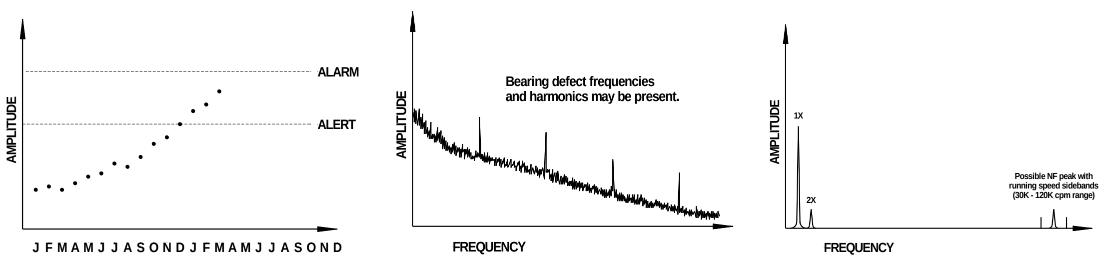
Velocity spectrum

Figure 29
Stage I trend & spectrums

Stage II – (Figure 30)

As the bearing enters Stage II, the damage is still not visible to the unaided eye. Damage to the surfaces may visible under magnification. The overall high frequency energy continues to trend upward and the first evidence of defect peaks begins to appear in the demodulated spectrum. Also, bearing defect peaks may begin appearing in the acceleration spectrum as well.

The impact signatures at this stage usually remain buried in the “noise floor” of time waveforms from machinery operating above 600 RPM. The time waveforms of machinery operating below 600 RPM may have peaks spaced at the inner raceway (BPFI) or the outer raceway (BPFO) frequency(ies).



Overall ultrasonic trend

Ultrasonic demodulated spectrum

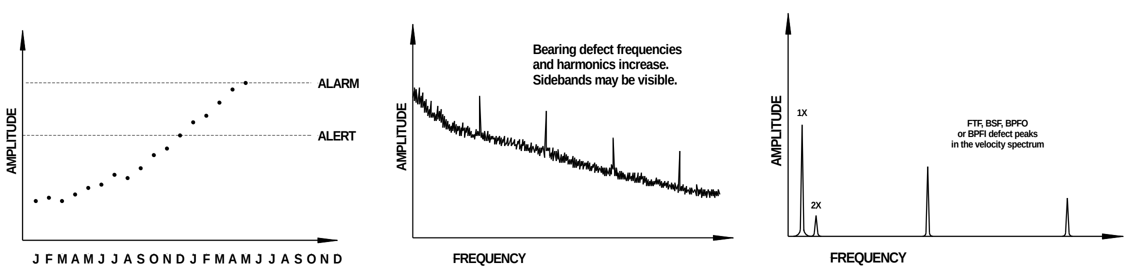
Velocity spectrum

Figure 30
Stage II trend & spectrums

At this stage only about 10% of the L_{10} bearing design life remains. The interval between data collection should be reduced depending on the criticality of the equipment.

Stage III – (Figure 31)

Defects in the bearing are visible to the unaided eye at this stage. The overall high frequency energy continues to trend upward and the bearing coefficient peaks and harmonics become more pronounced in the demodulated spectrum. As damage progresses, the peaks may have sidebands at $\pm$ 1X RPM or another component defect frequency such as the fundamental train (FTF) or ball spin (BSF).



Overall ultrasonic trend | Ultrasonic demodulated spectrum | Velocity spectrum

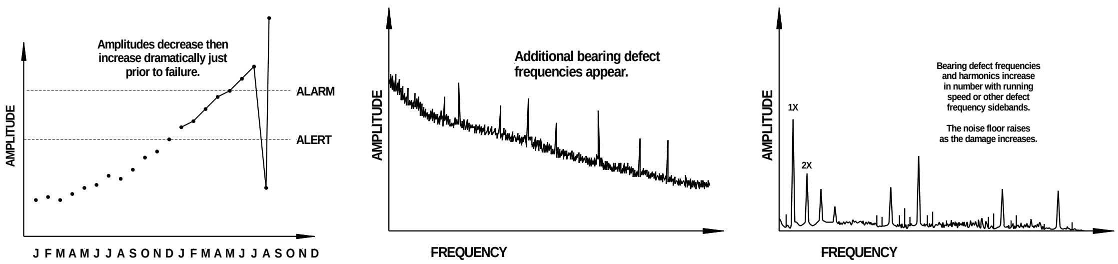
Figure 31
Stage III trend & spectrums

The bearing defect frequencies typically begin to appear in the <u>velocity</u> spectrum at this stage. As the damage spreads, more peaks appear in the form of defect harmonics, additional defect frequencies and depending on the damage, $\pm$ 1X RPM sidebands. The extent of damage should not be judged by the amplitude of defect peaks, but by the number of defect frequencies, their harmonics and the presence of sidebands no matter how small. The bearing should be scheduled for replacement at the earliest opportunity. Bearings at this stage typically have less than 5% of the L_{10} design life remaining.

Stage IV – (Figure 32)

A bearing in this stage is courting disaster. Damage is typically widespread because the faults originating on one component having spread to others. The overall high frequency energy continues to trend upward in the earliest part of this stage but may drop dramatically as the damage becomes more widespread. Bearing defect frequencies and often sidebands increase in number in the high frequency spectrum.

As the defect frequencies, harmonics and sidebands continue to increase in number, they may shift in frequency as the geometry of the bearing is destroyed and in some cases disappear into a noise floor that is continuing to grow.



Overall ultrasonic trend | Ultrasonic demodulated spectrum | Velocity spectrum

Figure 32
Stage IV trend & spectrums

Equipment with a bearing in this condition should not be operated. Total failure is imminent and damage usually included other components. A bearing at this stage typically has less than 1% of the L_{10} design life remaining.

We have described four common of the more than 16 faults that may be encountered by a vibration analyst. Other faults an analyst may encounter include:

- Rotor eccentricity
- Bent shaft
- Drive belt misalignment
- Journal bearing faults
- Gear train faults
- Aerodynamic and hydraulic problems
- AC Motors
- DC Motors
- Fluting
- Beat frequencies
- Barring and corrugation

There are many excellent sources of information on the characteristics of these and other faults that may be used to aid diagnosis. Some of the sources of information are listed at the end of this chapter.

Hardware and Software

Up to this point, the focus has been on recognizing characteristics of machinery faults. Before this diagnosis can happen, the mechanical vibration (motion) of the machine must be converted from mechanical movement to a computer display.

In most industrial plants, this process begins with a transducer called an accelerometer that is mounted on the equipment. The typical accelerometer shown in Figure 33 is a device with a small weight mounted on a small quartz crystal or ceramic. When force is applied to the quartz or ceramic by the weight as it vibrates, a charge is generated in it. This charge is amplified by an internal amplifier in the accelerometer producing an output voltage proportional to the amount and frequency of the vibration.

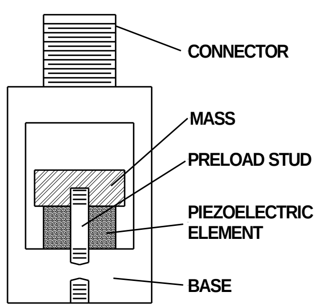


Figure 33
Piezoelectric accelerometer

Accelerometers must be supplied with a constant current source for operation of the internal amplifier. Modern devices such as data collectors and signal analyzers provide what is called ICP (Integrated Circuit Piezoelectric) power for accelerometers. In turn, the output of most industrial accelerometers received by the data collector or signal analyzer is specified in millivolts per unit of gravitational force or mv/g. A wide variety of accelerometers are available; selection depends on the frequency range desired, sensitivity, temperature and room available for mounting.

Velometers are also commonly used for vibration measurements. These are an accelerometer with additional electronics in the housing that integrate the signal from acceleration to velocity. The output from these devices is specified in millivolts per unit of velocity or mv / (in/sec). These devices avoid some of the signal processing “noise” that occurs when the signal from an accelerometer is converted into other units in a data collector or other external device.

Velocity pickups were commonly used but have been replaced by the accelerometer and velometer. Figure 34 shows the construction of a velocity pickup using a permanent magnet with springs in a damping fluid surrounded by a pickup coil. Vibration causes the magnet to move through the coil generating a voltage proportional to the vibration. Because the pickups contained moving parts, performance was limited by the physics involved and their life was limited.

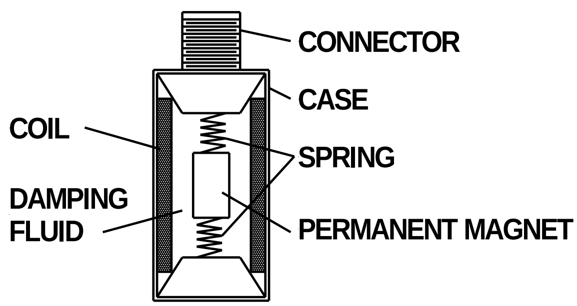


Figure 34
Velocity pickup

Proximity probes are used to detect motion of equipment in several specialized cases. A proximity probe like the one shown in Figure 35 is made by inserting a pickup coil in the end of a small housing. A high frequency alternating current is supplied to the coil creating a magnetic field. As the shaft or roll moves it causes a change in the magnetic field producing a signal proportional to the motion. Proximity probes are used on fluid film bearings in turbomachinery where it is important to measure motion of a relatively flexible shaft in a fluid film bearing inside a heavy housing. Because a proximity probe does not contact the shaft directly, it is sensitive to the surface finish of the shaft surface as well as residual magnetism. The output of proximity probes is usually specified in millivolts per unit of displacement or mv/mil. Because proximity probes require gap calibration for each installation this limits them to permanent installations.

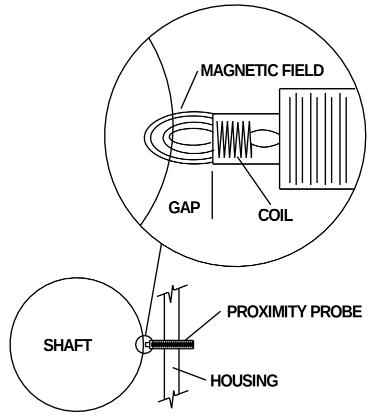


Figure 35
Proximity probe

The analog signal generated by a transducer must be digitized so it can be processed by the computer software. This task is accomplished in the A/D (Analog to Digital) section of a data collector or analyzer in very much the same manner music is digitized in today's world.

The digital data can be processed using a variety of digital signal processing algorithms that have replaced various analog filters making analysis of vibration data much easier.

Structural and Rotor Analysis

Structural and rotor analysis tools are used by engineers and vibration analysts to determine how a structure or rotor responds to the forces applied to it during operation and how it interacts with its surroundings. These tools share many of the same engineering principles, but also have some features that distinguish them from each other.

Structural analysis is focused on structures and equipment that does not rotate. This can include: bridges, automobile chassis, equipment housings and foundations. Rotor analysis or, as it is more commonly called, Rotor Dynamics, focuses on the rotating shafts in the equipment and their interaction with the machine that supports them.

Although both are commonly applied to the design of new machinery, structural analysis is more widely used to solve problem in industrial plants. The study of rotors tends to be focused on the design of rotors especially turbomachinery and how they interact with bearings, fluid flows and other forces applied to them.

Understanding structures and rotors is essential. We have likely seen video of the Tacoma Narrows Bridge failure in the 1930's that was attributed to an incomplete understanding of the mechanisms that cause destructive vibration. But, we fly on aircraft equipped with high speed turbofan engines that have been extensively analyzed during design and testing that have an outstanding record of reliability.

When a musician's tuning fork is struck, it rings with a very distinct sound at a single frequency. This is a natural frequency. If the length of the forks is changed, the frequency will change. Likewise, if the thickness of the forks is changed the frequency will also change. Finally, a tuning fork made from aluminum will have a different natural frequency than one made from brass with the exact same dimensions.

Similar to the tuning fork, every machine or structure has one or more natural frequency(ies). These are a consequence of the material, geometry and how it was mounted either to another machine or on its foundation. The natural frequencies are often very directional because the characteristics (apparent mass and stiffness) are different in different directions.

In the case a piano, the vibration of the different length strings gives the pianist a wide range of natural frequencies to create music. If an unbalanced variable speed motor was placed on the piano, when the speed of the motor matched the natural frequency of a particular string, that string would vibrate. Analysts call this condition resonance. However, in machinery, when a natural frequency is at or near the operating speed of a machine, music is not created. The resulting resonance can destroy a machine. Ordinarily, natural frequencies do not cause problems unless they are near the operating speed of equipment or near a harmonic of its running speed such as the one created when the blades of an impeller pass by the discharge opening of a pump.

As a child, we learned very quickly that if we pumped a swing at a rate that matched the rate at which it wants naturally move would get us much higher often to the consternation of our parents. The frequency at which we pumped the swing is called a forcing frequency. In our industrial world, if a pump impeller has 6 vanes and operates at 1,800 RPM, it has a forcing frequency at 6X RPM created by the vanes or 6 X 1,800 = 7,200 CPM.

If this pump has a natural frequency of 7,200 CPM the vibration level would be amplified dramatically (resonance). The increase in vibration amplitude is limited only by the amount of damping (energy dissipation capability) of the machine or structure.

Two characteristics of resonance help in diagnosis. It is highly directional and the phase at the natural frequency shifts 90º as the forcing frequency reaches it. In Figure 36 as the forcing frequency transitions through the peak, another 90º phase shift occurs for a total of 180º as the forcing frequency passes through the resonant peak.

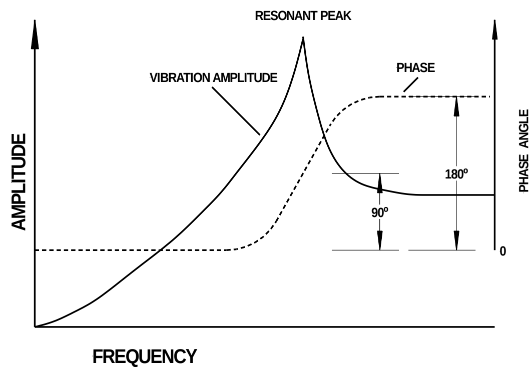


Figure 36
Resonance

The analyst has three means to control resonance: change the stiffness, mass or damping of the structure. Increasing stiffness raises the natural frequency while increasing the mass decreases the natural frequency. Often attempts made to increase the stiffness also increase the mass resulting in no change of the natural frequency.

Damping will not change the natural frequency appreciably, but it can be used to dissipate energy and reduce the effects of resonance. Unfortunately, in most industrial equipment, the size and amount of damping needed make it impractical. Thus the analyst is left with changes in stiffness or mass to change a natural frequency and reduce resonance.

The concept of natural frequency and resonance is further complicated by the shape the structure takes as it vibrates. The mode shape [Figure 37] of the structure is a map of the displacement of all the locations on a structure at a natural frequency. If a machine or structure has more than one natural frequency and possibly resonant frequency, each of these will have a distinct mode shape different from the others.

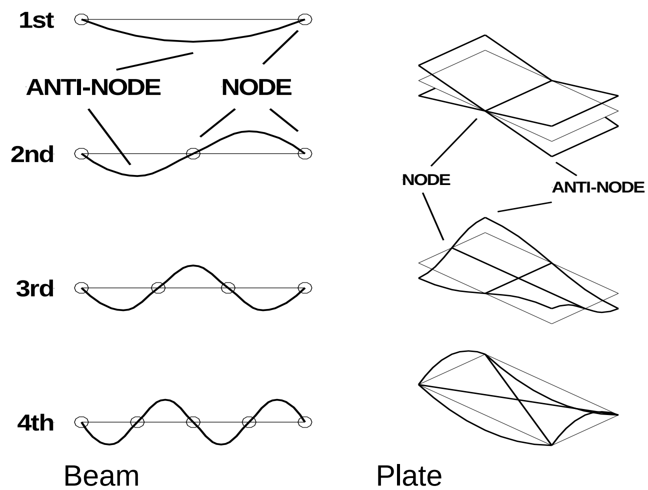


Figure 37
Bending modes of beams & plates

Mode shapes are characterized by locations called nodes and anti-nodes. Nodes are located on a structure where the displacement is at a minimum. Conversely, anti-nodes are located where the displacement is at a maximum. If an analyst does not know the location of these, and changes the stiffness or mass at a node, the effect on the resonance response will be minimal. Anti-nodes on the other hand typically represent optimum locations for modifications that can reduce the resonance response.

The question facing the analyst becomes: Is the vibration the result of a resonant condition, machine fault or both. Two closely related tools are used to aid in the diagnosis: Modal Analysis and Operating Deflection Shape (ODS).

There are two subsets of modal analysis. One called Finite Element Analysis (FEA) uses a computer model for analysis while the other experimental modal analysis utilizes data taken from the actual machine or structure for analysis.

Finite Element Analysis (FEA) is the process of defining a structure in terms of its physical characteristics (dimensions and material properties) and simplifying assumptions to determine its mode(s) of vibration. Each mode has a natural frequency, a shape associated with that frequency and a damping factor. Software is available that allows construction of a virtual model [Figure 38] that can be modified in the computer so the effects of geometry, material properties and varying the simplifying assumptions may be tried before making changes to the actual structure. The results are displayed graphically in the computer model.

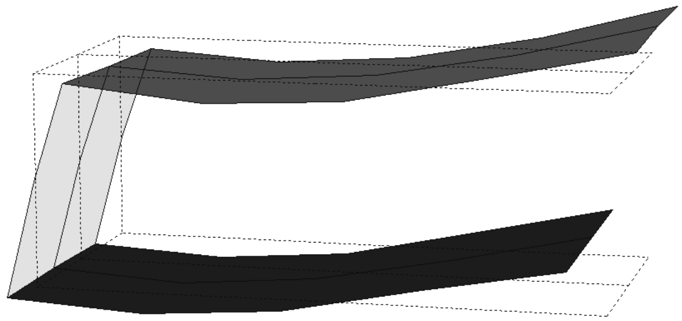


Figure 38
Finite element analysis model showing deflection of a bracket

Experimental modal analysis requires measurements from the actual machine or structure. This is done by exciting the machine or structure with one or more mechanical shakers or an impact hammer. The frequency and amplitude of the response is measured at multiple points. Software is used to create a model of the machine or structure which is then animated using the data collected from the field.

The results of the finite element analysis and the experimental modal analysis may be compared. Adjustments may be made to the experimental model so the results agree with the actual response of the machine or structure. Then when changes are made to the experimental modal model in the computer, the analyst is confident changes to the physical machine or structure will create the same change in response.

A close cousin to modal analysis used to diagnose a variety of machinery faults is Operating Deflection Shape (ODS). Modal analysis and operating deflection shape use computer models to create what is seemingly the same type of animated response of a machine. However, there are some important differences.

The data for an operating deflection shape is taken while the machine is in operation in contrast to modal data that must be collected while the machine is not running. The only peaks that can be analyzed and animated are those created from the forces that are generated by the operation of the machine. In comparison, data for modal analysis is gathered by either hitting the structure / machine or shaking it with equipment that measures the amount of force applied. This is necessary for the analyst to calculate the natural frequencies, mode shapes and damping. Consequently, modal analysis can be used to describe all the natural frequencies and damping while operating deflection shape does not.

Operating Deflection Shape analysis is useful for reducing a large amount of data into a visual model like that shown in Figure 39 that allows one to easily see the interaction and response of different machine components. Faults that may escape the notice of the analyst when looking at numerical data become visible. The animation also aids the analyst who can show how a machine is moving instead of trying to describe phase angles, frequencies and amplitudes to a person who does not have experience with vibration analysis.

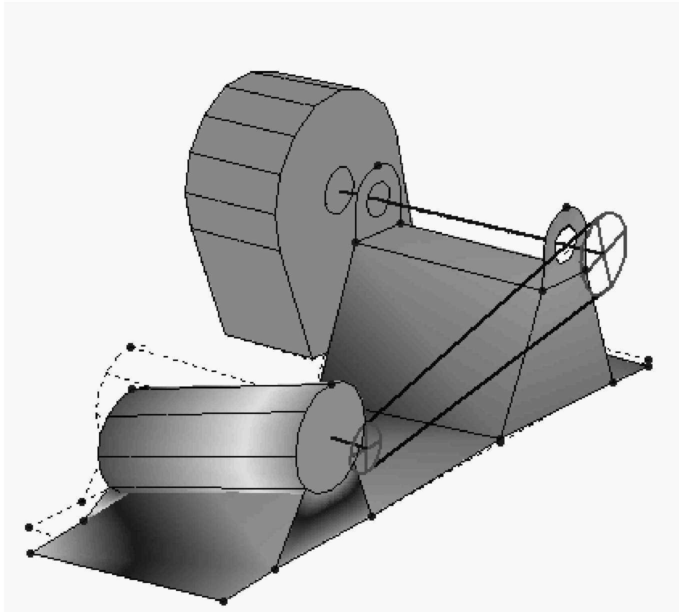


Figure 39
Operating Deflection Shape model of a centrifugal fan showing a loose base

Key Terms

Acceleration

The rate of change in velocity of an object with time. See g's

Acoustic emission

As materials are subjected to events that result in a sudden change or movement, they emit energy in the form of high frequency vibrations. These are usually due to a defect related condition. The technique of listening to these "acoustic emissions" is used to locate defects as they occur, providing early warning of an impending failure. Typically, acoustic emission systems operate in the 1 kHz to 2 MHz range. In the case of bearing fault detection, these ranges may be considerably lower depending on the manufacturer of the equipment. See Ultrasonic or acoustic emission.

Amplitude

The maximum value of a vibration caused by a machine. It may be expressed in units of displacement (mils – peak to peak), velocity (ips - peak), or acceleration (g's rms).

Anti-node

A point of maximum vibration amplitude in the natural frequency mode shape of a structure. It may be a point, line or surface depending on the construction of the structure or machine.

Axis of rotation

The geometric center of a shaft / rotor. An axis of rotation about which all parts of a rotor are symmetric.

Center of mass

An axis about which the mass of a rotor is equally distributed.

Correction plane

Also called "balancing plane" is a plane perpendicular to the shaft axis of a rotor in which correction for unbalance is made.

CPM

A unit of frequency - Cycles Per Minute. See frequency

Critical speed

Occurs when the speed of a rotating shaft matches its natural frequency. A resonant condition of a rotating shaft

Damping

A reduction of vibration by dissipating energy. Types of damping include viscous (oil), coulomb (friction) and solid (elastomer).

Displacement

A quantity that specifies the change in position of a body or particle with respect to a reference frame. Vibration displacement is usually measured in units of mils, one mil equals .001"

Experimental modal analysis

Analysis of a structure to determine under a given loading condition how (shape) and how much (amplitude) will it vibrate. The analysis may determine the natural frequencies, mode shape and damping by collecting vibration data from multiple points on a structure while it is excited by a shaker or impulse. The data is imported into a computer model where the resulting animations and spectrums may be used to determine what changes must be made to reduce or eliminate unwanted vibration.

Fast Fourier Transform or FFT

A mathematical process to change a complex signal from one form (time waveform) into another (frequency) that can be displayed as a series of sine waves.

Finite Element Analysis (FEA)

A machine or structure is represented by a geometrically similar model consisting of multiple, linked, simplified representations of discrete regions—i.e., finite elements. Equations of equilibrium, in conjunction with physical characteristics of the elements are applied to each element, and a system of simultaneous equations is constructed. The equations are solved for unknown values using the techniques of linear algebra or nonlinear numerical schemes to determine how a machine or structure will respond to the forces applied to it

Forcing frequency

The frequency at which a force is applied to a machine or structure. The force may be a result of a machine fault, or generated as a result of normal operation (i.e. hydraulic or impact).

Frequency

The number of cycles occurring during a given time period. It may be expressed in Hz (cycles per second), CPM (cycles per minute) or RPM (revolutions per minute). Frequency is the reciprocal of a period or cycle.

Frequency domain

Graphical display of a complex signal as various vertical lines in a graph with amplitude on the vertical scale and frequency on the horizontal scale. This representation is also called a spectrum of a signal.

g's

A measure of acceleration amplitude related to the acceleration due to the earth's gravity (386 in/sec/sec).

Harmonic

A sinusoidal vibration whose frequency is an integer multiple of its fundamental frequency.

Hz

A unit of frequency – Cycles Per Second. See frequency.

Influence coefficient

Tells how much weight at a given radius corresponds to a given vibration change. It also includes an angle which defines the difference between the phase indicated on the instrumentation and the actual location of the heavy spot on the rotor. It is also used to describe the effect of a change on one balance plane on another balance plane in multi-plane balancing.

IPS

A measure of frequency – Inches Per Second.

Key phasor

A mark or key on a shaft used to trigger a reference pulse for phase measurements,

L_{10} bearing design life

The number of hours a population of bearings will run before 10% of them have failed.

Mil

A unit of vibration displacement equal to .001". See displacement.

Mode shape

The vibrating shape of a structure from an applied force at a specific rotational speed or frequency. Each mode is defined by the modal frequency (natural frequency), modal damping (amplification) and mode shape (three dimensional deflection of the structure).

Natural frequency

The free vibration frequency(ies) and mode shape(s) of a structure or machine that exist independently of others in the same structure or machine. The natural frequencies are determined by the mass stiffness, geometry and mounting.

Node

A point of minimum vibration amplitude in the natural frequency mode shape of a machine or structure. It may be a point, line or surface depending on the construction.

Operating Deflection Shape (ODS)

The shape of a machine or structure as a result of the forces applied to it during operation. It is usually displayed as an animated computer model. These animations are very useful for diagnosing machinery faults. It does not allow determination of resonance or natural frequencies.

Oz – in / lb

A description of rotor unbalance per unit of rotor weight used in ISO balance tolerances. Other units may also be used: lb-in / lb, oz.-in / oz., g-in / g or g-in / lb.

Period

The time required for one complete rotation of a shaft or single cycle oscillation of an event. A period is the reciprocal of frequency.

Phase angle

The timing difference between two signals. It may be measured by the shift between two signals or by the time shift between an event and the detection of a reference signal (key phasor). It is usually expressed in 0 – 360 degrees.

Plane

A surface perpendicular to the axis of rotation.

Resonance

The amplification of a natural frequency by a matching forcing frequency. A highly directional amplitude increase and 90° phase shift occur at resonance.

Rotor dynamics

The study of vibrations related to equipment rotors, structures, bearings and all of the forces acting on them.

Running speed

The rotating speed of a shaft in a machine. Measured in RPM (Revolutions Per Minute) which is identical to CPM

Sideband

Two or more spectrum peaks located at equally spaced intervals on both sides of a carrier frequency peak. They are low frequency in comparison to the carrier frequency and are created when a low frequency event modulates a high frequency event such as an eccentric low speed gear modulating the high frequency gear mesh.

Spectrum

An array of vibration amplitudes displayed in accordance with frequency. It is also called the frequency domain, a signature or FFT.

Sub-harmonic

A sinusoidal vibration with a frequency that is a fractional multiple of its fundamental frequency, i.e. ½ RPM, 1/3 RPM, ¼ RPM……

Time trace or time waveform

A plot of vibration amplitude vs. time showing the amount of vibration and the amount of time required to complete one or more cycles. It is useful for diagnosing certain equipment faults especially those involving impacts.

Ultrasonic or acoustic emission frequency range

Ultrasonic vibration frequencies are generally considered to be those in the range from approximately 5,000 to 30,000 Hz; this range may vary slightly between manufacturers. Acoustic emission frequencies are in the range of 1 kHz to 1 MHz.

Ultrasonic demodulated spectrum

A vibration spectrum created by using filters and a mathematical algorithm to remove the low frequency components from a vibration signal and identify the very high frequency but low amplitude repetitive signals. The demodulation techniques are very useful to identify lubrication and rolling element bearing problems.

Unbalance
A condition when vibration is caused by unequal weight distribution in a rotor.

Unbalance – Couple
Occurs when the principal mass axis intersects the shaft axis at the center of gravity. More practically, a condition when the vibration in a rotor is caused by two heavy spots located 180° apart on opposite ends of the rotor.

Unbalance – Dynamic
A combination of static and couple unbalance when the shaft axis and the principal mass axis do not coincide or touch. It is the most common unbalance condition.

Unbalance – Residual
Unbalance remaining after a rotor has been balanced. It is expressed in units of weight at a given radius per unit of rotor weight in balance tolerance charts. See oz.-in / lb.

Unbalance – Static or Mass
A condition when the center of gravity axis is displaced parallel to the shaft axis. It can be corrected with a correction weight exactly opposite the heavy spot.

Velocity
The rate of change in position of an object with time. See IPS.

Summary

From its earliest beginnings, the art and science of vibration analysis has been a valuable tool to extend the life of machinery, improve product quality increase production and reduce costs. It has evolved into four fields of practice: balancing, turbomachinery analysis, equipment fault detection and structural / rotor analysis.

Typical machinery faults have been identified and catalogued so the analyst can compare data to a known library of faults to correctly identify them and recommend corrective action.

A variety of transducers may be used to convert the mechanical vibration into an electrical signal that can be digitized for processing in a computer. This has reduced the size and cost of data collection and processing equipment to a point where analysis once reserved for large very expensive equipment or military applications may be used routinely in industrial plants.

Development of specialized software has fostered the growth of structural and rotor analysis to help reduce resonant vibrations and identify machinery faults more easily.

From quieter cars to much better equipment life, vibration analysis has provided tremendous benefit to our industrialized world.

Chapter Objectives

When you have finished Chapter 17, “Vibration Analysis,” you should be able to do the following:

1. Identify the areas for which vibration analysts work today.
2. List the four key steps in a vibration analysis process.
3. Describe the relationship between time and frequency.
4. Tell the differences between three types of unbalance.
5. Use vibration severity charts to gage the health of a machine.
6. Describe the mathematical process used to change a time signal into a frequency spectrum.
7. Describe the vibration characteristics of some common machinery faults.
8. Identify the features that distinguish modal analysis and operating deflection shape analysis.
9. Describe the characteristics of natural frequency and resonance