



Machinery Vibration Analysis

(Training Manual)

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1 Chapter (1) Introduction

1.1 Maintenance

Past and current maintenance practices in both the private and government sectors would imply that maintenance is the actions associated with the equipment repair after it is broken. The dictionary defines maintenance as follows: "the work of keeping something in proper condition; upkeep." This would imply that maintenance should be actions taken to prevent a device or component from failure or to repair normal equipment degradation experienced with the operation of the device to keep it in a proper working order. Unfortunately, over the past decade indicates that most private and government facilities do not expend the necessary resources to maintain equipment in a proper working order. Rather, they wait for equipment failure to occur and then take the necessary actions to repair or replace the equipment.

The main objective of maintenance is to

- Improve equipment availability/reliability.
- Increase equipment effective life time.
- Increase resource utilization.
- Increase productivity.
- Reduce operating cost
- Reduce total cost rate
- Increase profitability ratio

In another words, the maintenance targets are to:

- Avoid costly/unplanned downtime due to equipment failures.
- Achieve efficient machinery operation and eliminate the cost of unnecessary energy usage.
- Improve the accuracy of machines by keeping them clean and running in top form.
- Understand the equipment capabilities so that planned outputs can be optimized.

1.2 Maintenance Techniques

The following types of maintenance programs are performed on most types of equipment:

- Breakdown or run to failure maintenance
- Preventive or time-based maintenance
- Predictive or condition-based maintenance

- Proactive or design out maintenance.

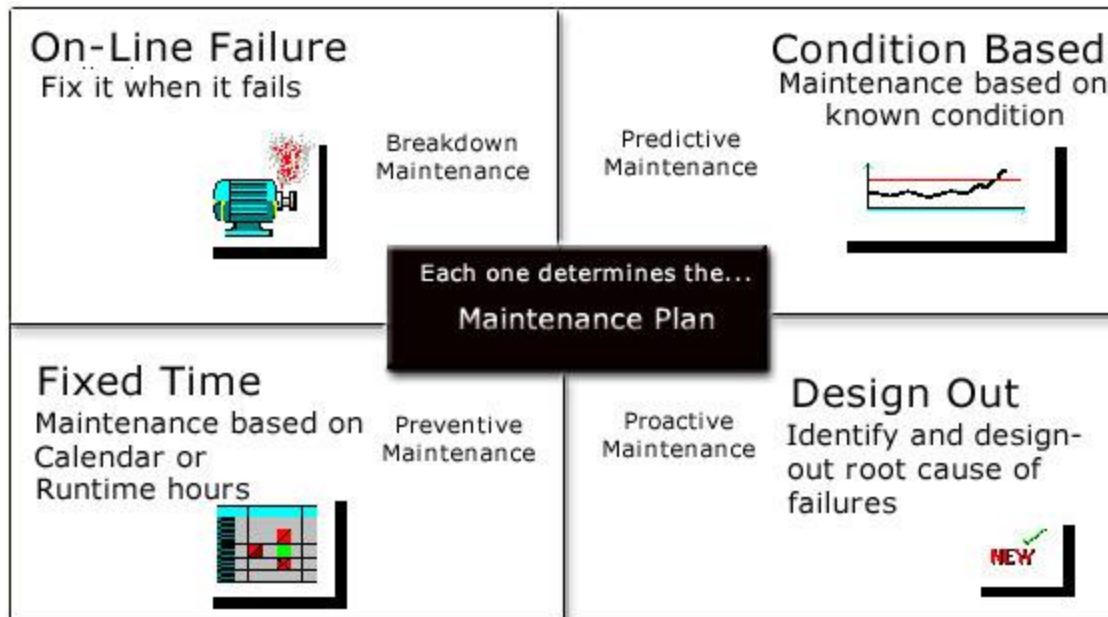


Figure 1.1 Maintenance technique.

1.2.1 Run-to-failure (breakdown) maintenance technique

- This method has no continuous activity associated with it. Essentially, no maintenance activity is performed on machinery until it fails or produces unacceptable product.
- It is the most expensive because it requires massive amounts of overtime and spare parts inventory.
- It generates the highest lost production rates.
- An unexpected failure can be dangerous to personnel and the facility.

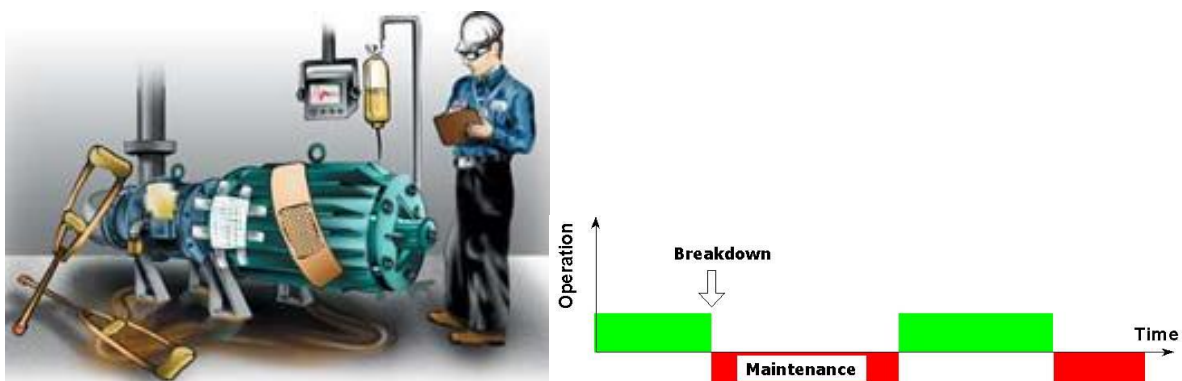


Figure 1.2 Run to failure maintenance technique.

1.2.2 Preventive maintenance technique

- From run to failure maintenance technique, we progress to periodic preventive maintenance which is sometimes called "historical" maintenance. This is where the histories of each

machine type are analyzed and periodic overhauls are scheduled to occur before the statistically expected problems occur.

- Preventive maintenance can be defined as follow: Actions performed on a time- or machine-run-based schedule that detect degradation of a component or system with the aim of sustaining or extending its useful life through controlling degradation to an acceptable level.
- In other words, preventive maintenance is a maintenance activity repeated at a predetermined frequency. The frequency may be based on calendar time or other occurrences such as service hours or number of starts. For example, someone may change the oil in a car every 3000 miles or every 3 months. Also it is a time-based system of parts replacement. That is, after a period of time has elapsed, parts will be replaced.
- Preventive maintenance activities are set up based on experience from past failures, equipment Characteristics, and vendor recommendations. Preventing past failures from recurring decides many procedures. If a pump has failed a number of times from alignment problems due to fasteners coming loose, a Preventive maintenance procedure might be set up to check fastener tightness for the pump each month.

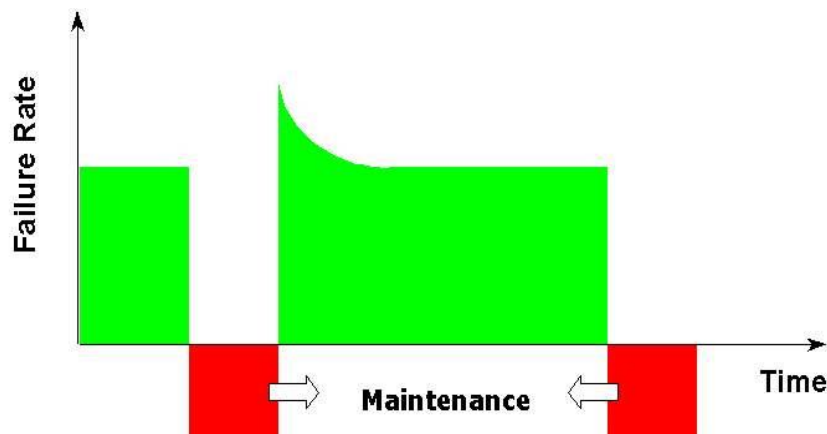


Figure 1.3 preventive maintenance technique.

1.2.2.1 Advantages of the preventive maintenance technique

- Cost effective in many capital intensive processes.
- Additional cost savings are realized because manpower and any heavy equipment are scheduled.
- Increased component life cycle.
- Generates Energy savings.
- Reduced equipment or process failure.
- Flexibility allows for the adjustment of maintenance

1.2.2.2 Disadvantages of the preventive maintenance technique

- It results in unnecessary replacing components in satisfactory condition.
- Sometimes, a machine in good condition is disassembled for preventive maintenance, parts replaced, and then returned to service in poorer condition because of errors during assembly.
- Catastrophic failures still likely to occur.
- Is more labor intensive
- Includes performing unneeded maintenance activities, which has the potential to result in incidental damage to components

1.2.2.3 Preventive maintenance tasks

- Checking and cleaning
- Inspecting
- Adjustments
- Lubrication
- Parts replacement and servicing
- Calibration
- Repair of components and equipment

1.2.3 Predictive maintenance technique

- Predictive maintenance can be defined as follow: Measurements that detect the onset of a degradation mechanism, thereby allowing casual stressors to be eliminated or controlled prior to any significant deterioration in the component physical state.
- It is maintenance actions based on machine condition data.
- Condition-based maintenance is related to predictive maintenance. In condition-based maintenance, the equipment is inspected and based on a condition, further work or inspections are done.
- Scientific application of proven predictive techniques increases equipment reliability and decreases the costs of unexpected failures.
- Predictive maintenance involves monitoring the machine's vibration characteristics or symptoms to diagnose its condition.
- The machine's condition also determines the required replacement parts.

1.2.3.1 Advantages of the predictive maintenance technique

- Minimizes or eliminates costly downtime - increases profitable uptime.
- Minimizes or eliminates catastrophic machinery failures - damage from catastrophic failure is usually much more extensive than otherwise would have been.
- Reduces maintenance costs.
- Reduces unscheduled maintenance - repairs can be made on time that least affect production.
- Reduces spare parts inventories - many parts can be purchased just in time for repairs to be made during scheduled machinery shutdowns.
- Optimizes machinery performance - machinery always operates within specifications.
- Reduces excessive electric power consumption caused by inefficient machinery performance - saves money on energy requirements.
- Reduces need for standby equipment or additional floor space to cover excessive downtime - less capital investment required for equipment or plant.
- Increases plant capacity.
- Reduces depreciation of capital investment caused by poor machinery maintenance - well maintained machinery lasts longer and performs better.
- Reduces unnecessary machinery repairs - machines are repaired only when their performance is less than optimal.
- Minimizes or eliminates the possibility that machinery repairs were the wrong repairs.
- Reduces repair time and unnecessary repairs
- Reduces penalties that result from late deliveries caused by broken down or poorly performing machinery.
- Reduces warranty claims due to poor product quality caused by poorly performing machinery.
- Reduces the possibility of accepting recently purchased new or used machinery with defects
- Increases machinery safety - injuries are often caused by poorly performing machinery.
- Reduces insurance rates because well maintained machinery increases safety.

1.2.3.2 Disadvantages of the predictive maintenance technique

- Increases investment in diagnostic equipment.
- Increases investment in staff training.

- Savings potential not readily seen by management.

1.2.3.3 Steps for implementing predictive maintenance

- Detection
 - Detection involves measuring and trending levels for each machine included in the program on a regularly scheduled basis. The objective is to reveal significant increases in the machine level to warn of developing problem
- Analysis or Fault Diagnosis
 - Analysis helps to pinpoint specific machinery problems by revealing their unique characteristics.
- Correction
 - Corrective action is taken after specific problem has been detected and identified by planning and scheduling all activities to ensure that machine downtime is kept to the absolute minimum.

1.2.3.4 Predictive maintenance programs

A variety of technologies can be used as a part of comprehensive predictive maintenance program. Since mechanical systems or machines account for the majority of plant equipment, vibration monitoring is generally the key component of most predictive maintenance programs. However, vibration monitoring cannot provide all the information required to perform a successful predictive maintenance. Therefore, other predictive maintenance programs have to be included.

Vibration analysis

Vibration frequency measurement provides information about entire machine operation and condition monitoring for machine fault vibration patterns created by:

- Wear on rotating elements - bearings, shafts, gears, linkages
- Broken/missing parts - gearboxes, electric motors, compressors, pumps, conveyor drives
- Lubrication deficiency
- Impeller/fan blade material build-up
- Improper parts assembly

We will discuss this technique in more details.

Thermography

Thermography is a predictive maintenance technique that can be used to monitor the condition of plant machinery, structures and systems. It uses instrumentation designed to monitor the emission of infrared energy, i.e. temperature, to determine their operating condition. By detecting

thermal anomalies, i.e. areas that are hotter or colder than they should be, an experienced surveyor can locate and define incipient problems within the plant. Infrared thermography employs computerized thermal measurement and imaging equipment to detect abnormal heating due to faulty conditions. An infrared survey can detect a problem before it manifests itself into a costly failure. Readings are taken as a part of the predictive maintenance routine and tracked over time. Failure shows up as a change in temperature. Infrared thermography helps to identify the following problems:

- Unbalanced or overloaded circuits
- Faulty electrical contacts or connections
- Improper grounding
- Bearing and coupling wear

Examples of what can be scanned:

- Electrical switch gear, breakers, bus connections, and contacts.
- Transformer connections.
- Motor and generator connections, windings, feeders and exciters.

Ultrasonic monitoring

Ultrasonic monitoring is used to detect high frequency signatures since the normal monitoring range for vibration analysis is (1Hz to 20,000Hz), but the ultrasonic technique monitors the frequency range between 20,000Hz and 100kHz. Leak (turbulent flow of liquids and gases through a restricted orifice) is mainly detected by the ultrasonic monitoring since it produces high frequency signatures that can be easily identified. In addition, ultrasonic monitoring can be used to detect early faults in elements of rotating bearings.

Oil analysis (tribology)

Oil analysis is used to determine foreign media in machinery lubricants where periodic check of oil samples will detect wear before failure occurs. Oil is the lifeblood of machinery oil analysis provides valuable information about the condition of the machinery. Degradation of hoses, filters, seals, air intakes, engine blocks, and oil can assist in performing inexpensive tests. By monitoring wear particles produced by everyday machinery operation, we can advise which components require replacement.

1.2.4 Proactive maintenance technique

- The latest innovation in the field of predictive maintenance is so called proactive maintenance.

- It uses a variety of technologies to extend the operating life of machines and to virtually eliminate reactive maintenance.
- The major part of a proactive program is root cause failure analysis which is the determination of the mechanisms and causes of machine faults.
- The goal of proactive maintenance is to apply advanced investigation and corrective technology toward solving the root cause of any machinery problem.
- In its ideal form, its goal is to eliminate failures in plant components.

1.2.4.1 Advantages of proactive maintenance technique

- Can be the most efficient maintenance program.
- Lower costs by eliminating unnecessary maintenance or overhauls.
- Minimize frequency of overhauls.
- Reduced probability of sudden equipment failures.
- Able to focus maintenance activities on critical components.
- Increased component reliability.
- Incorporates root cause analysis.

1.2.4.2 Disadvantages of proactive maintenance technique

- Can have significant startup cost, training, equipment, etc.
- Savings potential not readily seen by management [1].

Table 1.1 Comparison of maintenance techniques

	Breakdown	Preventive	Predictive
Method	Failure based	Time based	Condition Based
Advantages	No maintenance cost	Planned maintenance	Lower maintenance costs. - Fewer Machine failures . - Less repair downtime . - Reduced inventory - Increased production - Improved operator safety
Disadvantages	High spare parts inventory - High machine downtime - High overtime labor costs - Low production availability	-Failure may occur before scheduled maintenance -Maintenance may be performed unnecessarily . -Maintenance may cause failure .	-Initial investment in equipment .
Use	Used only on cheap and insignificant components	Used on all machines	Used on all machines

2 Chapter (2) Fundamentals of Vibration

2.1 Introduction

Vibration is the periodic, back and forth motion (oscillation) of an object. We encounter vibration in many different ways in our daily lives. Nearly all musical instruments utilize the periodic vibration of mechanical elements to make sound; for example, pianos and guitars use the vibration of a string and connected soundboard, clarinets use the vibration of a small reed, and trumpets use the vibration of the player's lips. Vibration also exists in nature. The motion of the tides is an example of a very low frequency vibration that is produced by the gravitational force of the moon and sun. This motion is an example of forced vibration (and resonance, in the case of the Bay of Fundy). A sudden gust of wind acting on a tall pine tree can also produce a periodic, low frequency vibration of the tree, an example of free vibration resulting from an initial impulse. The wind blowing on aspen leaves produces a continuous, periodic motion of the leaves, an example of self-excited vibration. Machines, because of internal and external forces, also vibrate.

Machinery vibration involves the periodic motion of rotors, casing, piping, and foundation systems, all at the same time. Usually this vibration is so small that sensitive equipment is needed to detect it. To illustrate the small size of machinery vibration, we can compare it to the diameter of a human hair. The average diameter is approximately $130\mu\text{m}$ (about 5mil). This is an unacceptable vibration level on some steam turbine generator sets that are the length of a house. Vibration in machines causes periodic stresses in machine parts, which can lead to fatigue failure. If the motion due to vibration is severe enough, it can cause machine parts to come into unwanted contact, causing wear or damage. Because of this, the control of vibration is an important part of machinery management, and the concepts underlying vibration must be thoroughly understood by the machinery professional.

This chapter will discuss the basic concepts of linear vibration, the vibration of machines, rotation, and the most commonly used vibration measurement units. Then, we will move onto a discussion of the concepts of free vibration at a natural frequency, forced vibration, and that most interesting marriage of the two, resonance. Finally, we will discuss self-excited vibration, where a system can internally transfer energy to produce vibration at a natural frequency.

2.2 The Basic Vibration Signal

A vibration transducer is a device that converts mechanical motion into an electronic signal. A displacement transducer can be used to measure the displacement, or position, of an object relative to the transducer. For most transducers this is, ideally at least, a one-dimensional measurement. If

the object is vibrating, the position of the object relative to the transducer will change in a repeating pattern over time. Figure 2.1 shows an object that is vibrating toward and away from a transducer. The different images show the evolution of the system over time. The transducer converts the position of the box on the transducer's sensitive axis to an output voltage, which is displayed at the top of the figure. The transducer output voltage, or signal, is proportional to the distance from the transducer to the object, the gap. This changing voltage signal represents the relative position of the vibrating object versus time.

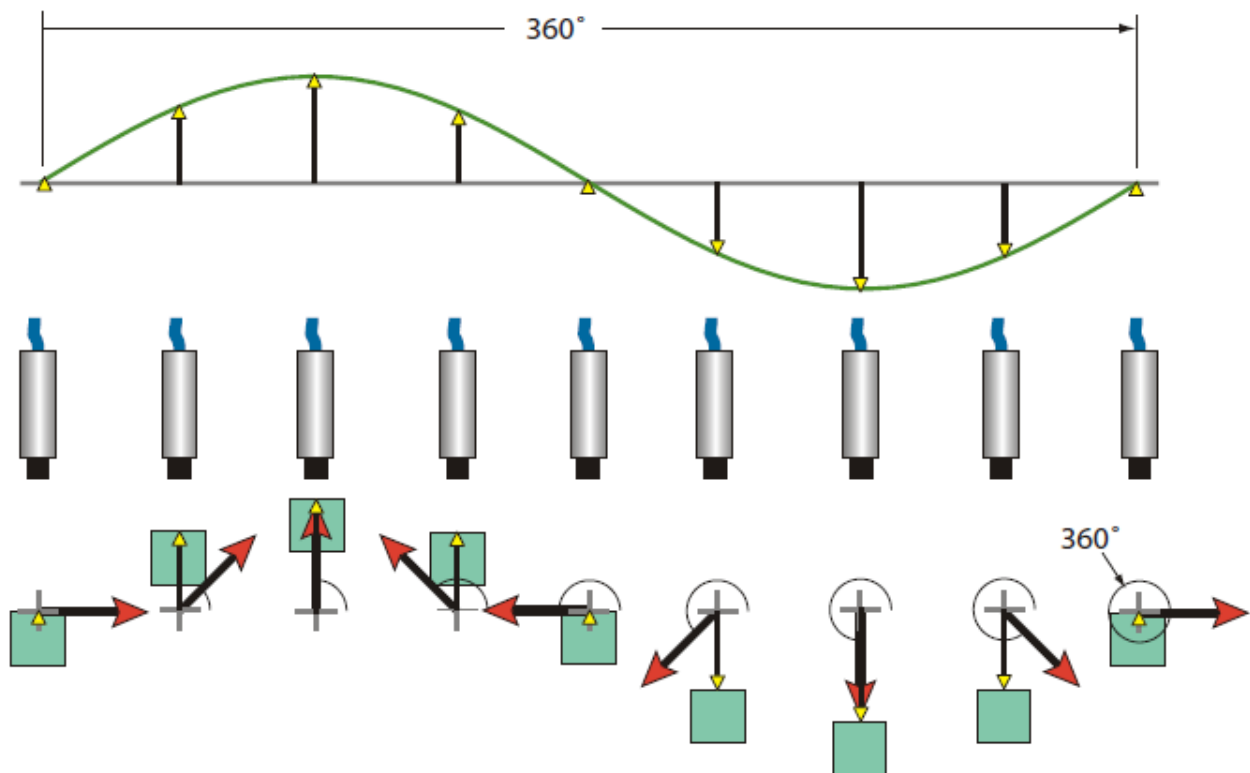


Figure 2.1. The relationship of a displacement vibration signal to the motion of an object.

There are two primary characteristics that we can measure on this signal, frequency and amplitude. This signal has a simple, sinusoidal shape, and it contains only one frequency. More complex (and typical) signals contain several frequencies of vibration with different amplitudes.

2.3 Frequency

Frequency is the repetition rate of vibration per unit of time. The vibration signal in Figure 2.1 has only one frequency. The frequency of this signal is found by measuring the amount of time it takes to complete one cycle of vibration (Figure 2.2). This length of time is called the period (T) and is shown in the figure. It has units of seconds per cycle of vibration. The frequency (f) has units of cycles/second, or hertz (Hz) and is the reciprocal of the period in seconds:

$$f(\text{Hz}) = \frac{1}{T} \quad (2.1)$$

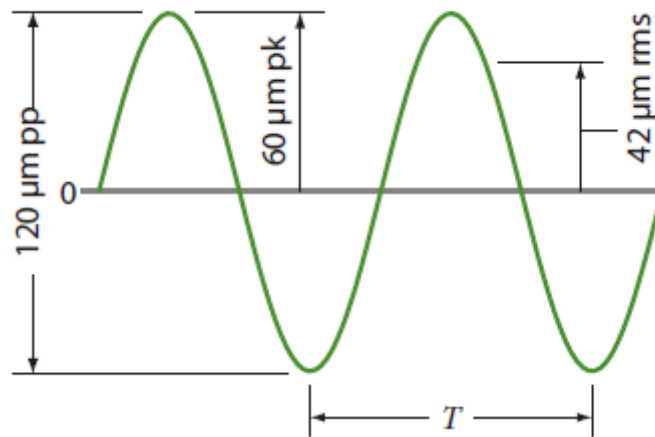


Figure 2.2. Three methods of amplitude measurement of a single-frequency (sine wave) signal.

In rotating machinery applications, we are often interested in expressing the frequency in cycles per minute, or cpm, so that the frequency can be directly compared to the rotating speed of the machine, measured in revolutions per minute, or rpm. The frequency in cpm can be calculated from the period using this expression:

$$f(\text{cpm}) = \left(f \frac{\text{cycles}}{s}\right) \left(60 \frac{s}{\text{min}}\right) = \frac{60}{T} \quad (2.2)$$

There are 2π radians or 360° in a circle; this concept can be extended to say that there are 2π radians or 360° in one cycle of vibration. Thus, the frequency can also be expressed in radians/second (rad/s):

$$\omega \left(\frac{\text{rad}}{s}\right) = \left(2\pi \frac{\text{rad}}{\text{cycle}}\right) \left(f \frac{\text{cycles}}{s}\right) = \frac{2\pi}{T} \quad (2.3)$$

The frequency ω (Greek lower case omega) is sometimes called the circular frequency. Several terms are commonly used to describe frequency ranges in machinery (Figure 2.3):

Synchronous

The same as rotor speed. The “X” is equivalent to a mathematical multiplication symbol. Thus, 1X can be read as “1 times rotor speed.”

Nonsynchronous

Any frequency except 1X.

Sub-synchronous

Any frequency less than 1X. This can include simple integer ratios such as $2/3X$, $3/4X$, etc., decimal ratios such as $0.48X$, $0.37X$, etc., or subharmonics (see below).

Super-synchronous

Any frequency greater than 1X. This can include simple integer ratios such as $3/2X$, $5/2X$, etc., decimal ratios such as $1.6X$, $1.8X$, etc., or super-harmonics (see below).

Super-harmonic

A frequency greater than 1X that is an integer multiple: for example, 2X, 3X, 4X, etc.

Subharmonic

A frequency less than 1X that is an integer ratio with one in the numerator: for example, $1/4X$, $1/3X$, $1/2X$, etc.

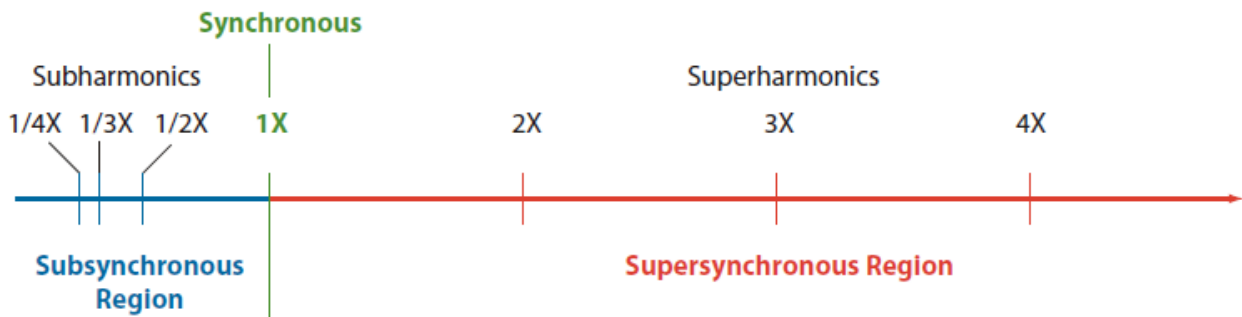


Figure 2.3. Machinery vibration frequency definitions.

2.4 Amplitude

Amplitude is the magnitude of vibration expressed in terms of signal level (for example, millivolts or milliamps) or in engineering units (for example, micrometers or mils, millimeters per second or inches per second, etc.). The amplitude can be measured using several methods. One is to measure the total voltage change from the minimum of the signal to the maximum of the signal. This method is used for displacement signals and is referred to as double amplitude, or peak-to-peak, abbreviated pp. In Figure 2.2, the peak-to-peak voltage change represents a total change in position of $120\mu\text{m}$ (4.7mil). Often, the machinery diagnostician simply needs to know how much the machine rotor is vibrating relative to the available diametric clearance in the machine. Peak-to-peak measurement makes this comparison relatively easy.

The single amplitude, or peak method, abbreviated pk, measures the voltage change from the middle of the signal to the maximum value of the signal. This method yields an amplitude that is one half of the peak-to-peak value. This method of measurement is commonly used for velocity and acceleration vibration signals, but is not well-suited to the measurement of displacement signals for the reason given above.

The root-mean-square method, abbreviated RMS, describes the amplitude of a continuously changing signal as a form of average. As the name suggests, it is calculated by taking the square root of the mean, or average, of the squares of the signal values. If, and only if, the signal is a sine wave (single frequency), the RMS amplitude will be equal to 0.707 times the peak amplitude, and it will be equal to 0.354 times the peak-to-peak amplitude. If the signal is not a sine wave, then the

RMS value using this simple calculation will not be correct. Most machine vibration signals are not sine waves. Instead, they contain a mixture of different frequencies.

2.5 Displacement, Velocity, and Acceleration

Displacement describes the position of an object. Velocity describes how rapidly the object is changing position with time, and acceleration describes how fast the velocity changes with time. Figure 2.4 shows an oscillating pendulum observed by a displacement transducer and plots of displacement, velocity, and acceleration. The displacement of the pendulum is measured relative to the vertical, rest position.

Note that the peak velocity occurs when the pendulum passes through the vertical position and is moving toward the transducer. As the pendulum reaches the end of its motion and is closest to the transducer (the maximum positive value in the displacement plot), the velocity is zero, momentarily. The pendulum starts moving back in the opposite direction with negative velocity. When the pendulum passes through the vertical position again, the velocity reaches its maximum negative value. When the pendulum reaches the opposite displacement extreme, the displacement is at the minimum on the displacement plot, and the velocity is again zero.

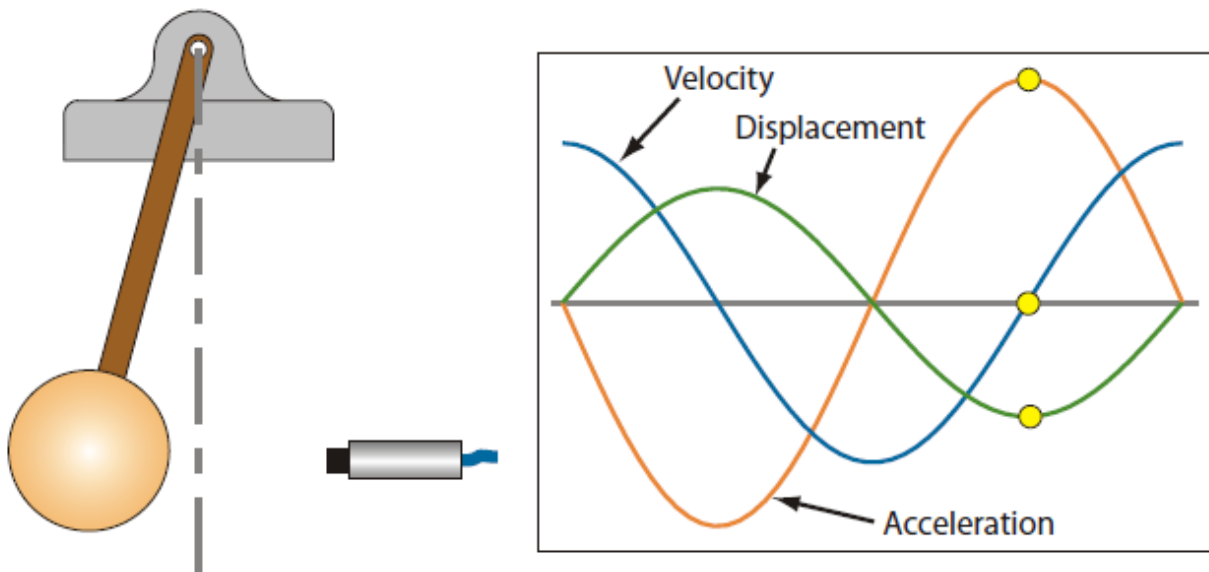


Figure 2.4. The motion of a pendulum. Curve of displacement (green), velocity (blue), and acceleration (orange).

Because the pendulum is driven by gravity, the acceleration is zero when the pendulum is in the vertical position. As the displacement approaches the positive peak near the transducer, the force due to gravity is acting in such a way as to reduce the velocity. Thus, the acceleration is negative, and it reaches its maximum negative value (the minimum on the plot) when the displacement is maximum. Then, as the pendulum starts back away from the transducer, the

negative acceleration becomes smaller, heading up toward the zero crossing on the plot. When the pendulum reaches the opposite extreme, gravity is trying to stop the pendulum and push it back toward the transducer. At this point, the acceleration reaches a maximum positive value on the plot. For single-frequency signals (sine waves) only, such as shown in the illustration, there is a simple mathematical relationship between displacement, velocity, and acceleration:

$$\begin{aligned} d &= A \sin(\omega t) \\ v &= A\omega \sin(\omega t + 90^\circ) = \dot{d} \\ a &= A\omega^2 \sin(\omega t + 180^\circ) = -A\omega^2 \sin(\omega t) = \dot{v} = \ddot{d} \end{aligned} \tag{2.4}$$

where d is the displacement, v is the velocity, a is the acceleration, A is the amplitude of the displacement (the maximum displacement possible, expressed as either PK, PP, or RMS), and ω is the frequency of vibration in rad/s. The numbers 90° and 180° represent the relative phase, or timing between the signals.

In Equations 2.4, velocity leads displacement by 90° ; that is, it reaches its maximum one quarter of a cycle, or 90° , before (the reason for the plus sign) the displacement maximum. Figure 2.4 shows a set of plots of Equations 2.4; note that the velocity maximum occurs when the displacement is zero and rising.

Acceleration leads displacement by 180° , and acceleration leads velocity by $180^\circ - 90^\circ = 90^\circ$. In the plot, the acceleration maximum occurs when the velocity is zero and rising, and the displacement and acceleration change in opposite directions. Important note: the phase angles here are based on a mathematical definition of phase, not on phase measured by instrumentation. As shown in Equations 2.4, the amplitude of the velocity is related to the amplitude of displacement by a factor of ω . Similarly, the amplitude of acceleration is related to the amplitude of displacement by a factor of ω^2 . This has important implications for transducer selection, because the amplitude of velocity and acceleration signals can become very small at low frequencies [2]. The displacement, velocity, and acceleration conversion are displayed in Table 2.1. Also, the relations between the form of amplitude measurement are shown in Table 2.2.

Table 2.1. Unit Conversion (British unit)

Sinusoidal Motion (zero-peak)

	Displacement (D) (in.)	Velocity (V) (in./sec)	Acceleration (A) (g)
Displacement (D) (in.)		$D = 0.159 V/f$	$D = 9.78 A/f^2$
Velocity (V) (in./sec)	$V = 6.28 fD$		$V = 61.4 A/f$
Acceleration (A) (g)	$A = 0.102 f^2D$	$A = 0.0163 fV$	

f = frequency, Hz

Table 2.2 The relation between amplitude measurement forms

Sinusoidal Wave Forms Multiplier x ($A = xB$)

		B			
		Peak	Peak to Peak	RMS	Average
A	Peak	1	0.5	1.414	1.570
	Peak to Peak	2	1	2.828	3.140
	RMS	0.707	0.354	1	1.110
	Average	0.637	0.319	0.901	1

2.6 Time Domain

Vibration data, plotted as amplitude versus time, are referred to a time domain data profile. Examples of simple and complex time domain are shown in Figure 2.5 and Figure 2.6, respectively. Time domain data are helpful in detecting some mechanical faults and sometimes it's so difficult to interpret the pattern since the time signal includes all kinematics and noise initiated in the machine during the measurement.

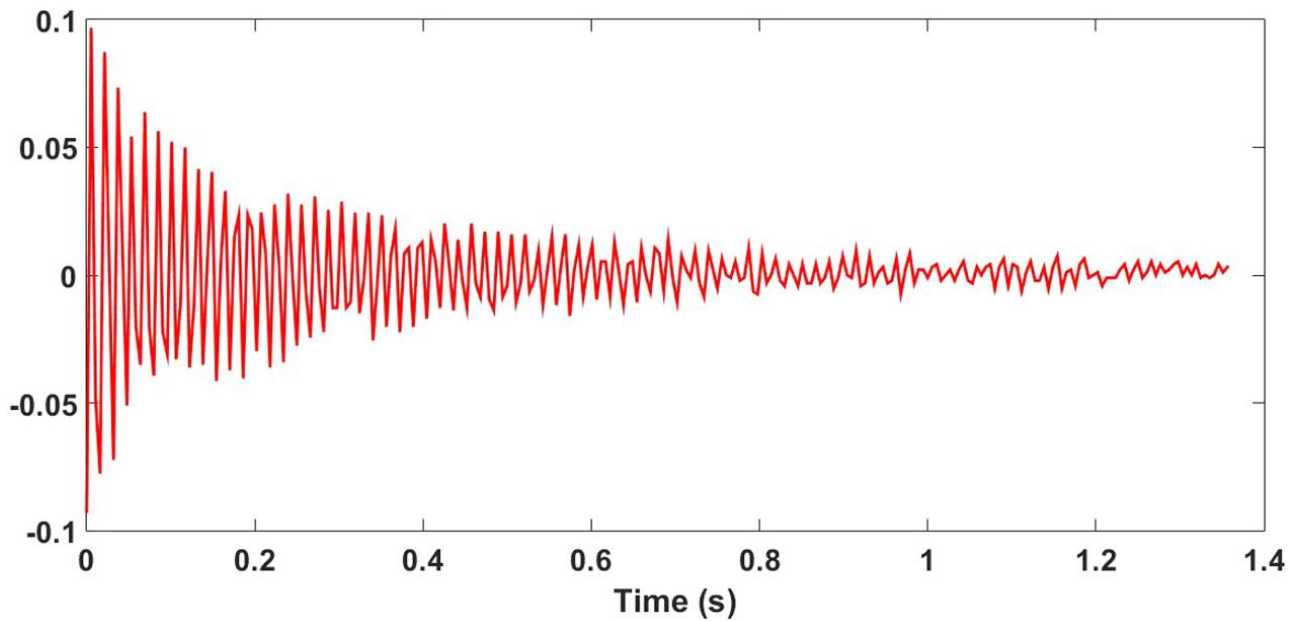


Figure 2.5. Example of simple time domain.

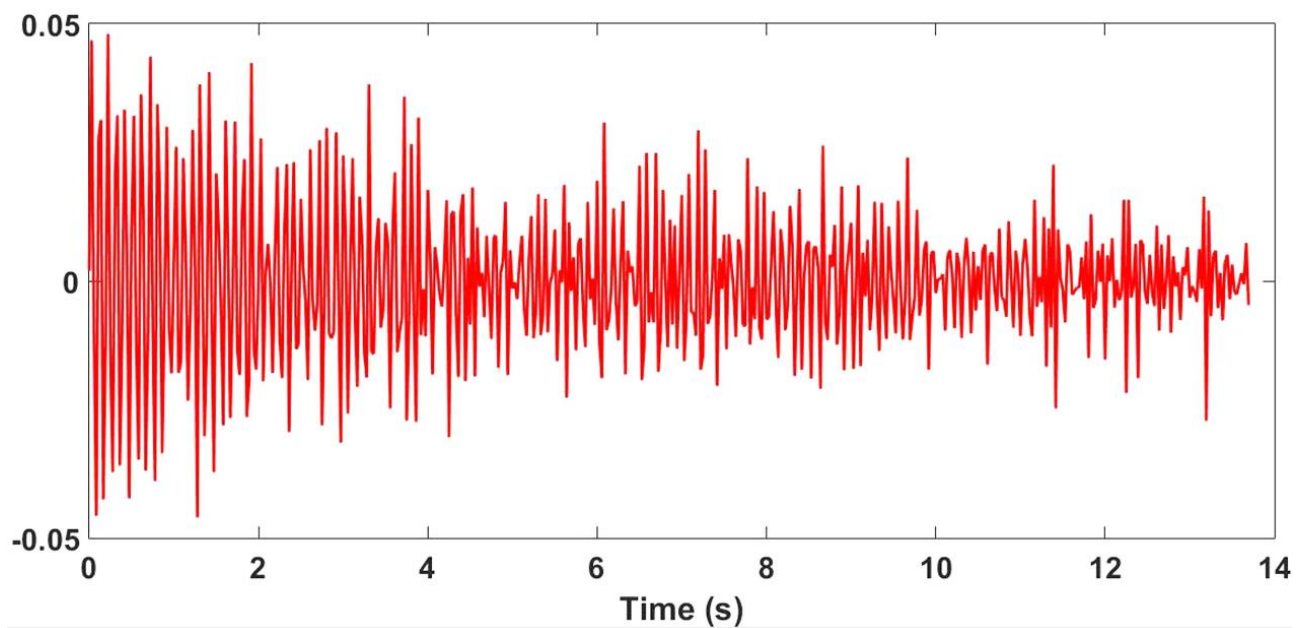


Figure 2.6. Example of complex time domain.

2.7 Frequency Domain

frequency domain displays how much of the signal exists within a given frequency band concerning a range of frequencies. The signature of frequency domain signal is displayed in Figure 2.7. A frequency-domain representation can include information on the phase shift that must be applied to each sinusoid to be able to recombine the frequency components to recover the original time signal.

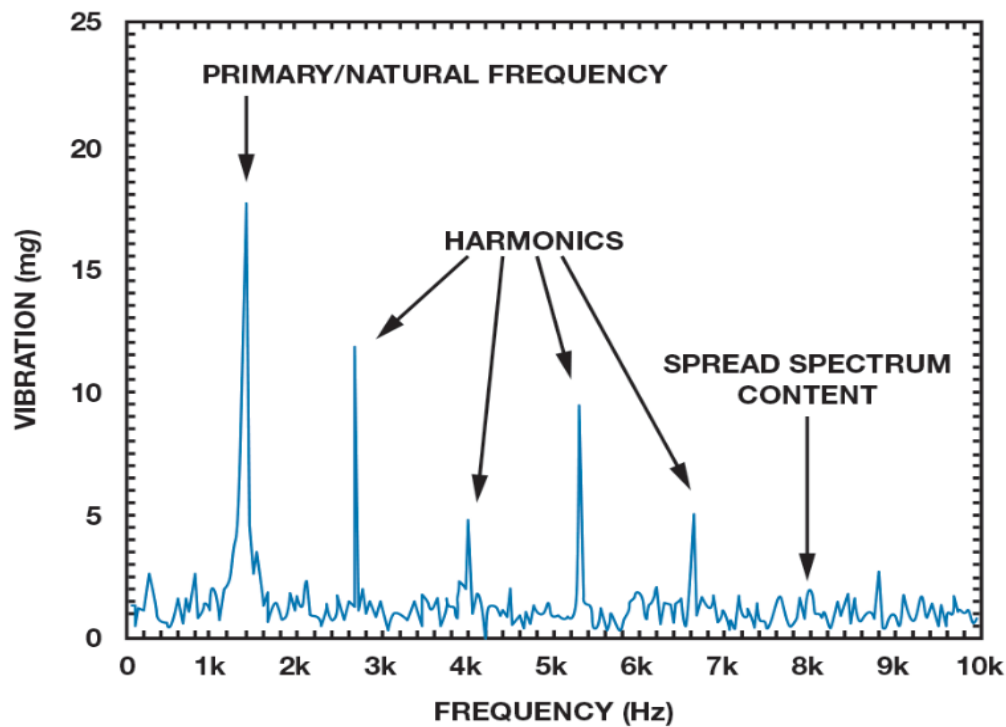


Figure 2.7. Typical frequency-domain vibration signature.

Frequency-domain data are obtained by converting time domain data using a mathematical algorithm called fast Fourier transform (FFT). FFT allows each vibration component of a complex machine-train spectrum to be shown as a discrete frequency peak. The frequency-domain amplitude can be the displacement per unit time related to a particular frequency which is plotted on the y-axis against the frequency on the x-axis. The relation between the frequency and time domain is illustrated in Figure 2.8.

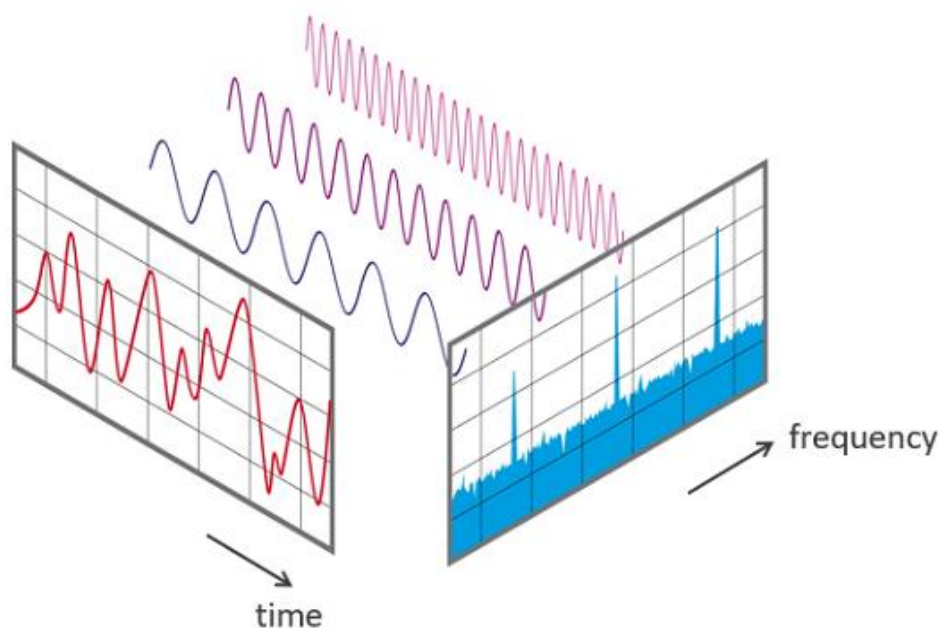


Figure 2.8. Relationship between time and frequency domain.

3 Chapter (3) Vibration Transducers

3.1 Introduction

At present in the industry like research and development, the ability of monitoring, measuring as well as analyzing the vibration is very important. Unfortunately, the suitable techniques for making a measurement system for vibration with precise repeatable are not always clear. There are some challenges related while measuring the vibration which includes a selection of suitable component, the configuration of the system, signal conditioning, analysis of waveform, and setup. This chapter discusses what is a vibration transducer, types, working principle, and applications.

3.2 Vibration Transducer

A transducer is a device that converts one type of energy, such as vibration, into a different type of energy, usually an electric current or voltage. Commonly used transducers are velocity pickups, accelerometers and Eddy current or proximity probes. Each type of transducer has distinct advantages for certain applications, but they all have limitations as well. No single transducer satisfies all measurement needs. One of the most important considerations for any application is to select the transducer that is best suited for the job. The various vibration transducers are discussed below.

3.3 Types of Transducers

There are different types of transducers based on the parameters of vibration source

3.3.1 Proximity Probe

The Proximity Probe, also called an "Eddy Current Probe" or "Displacement Transducer", is a permanently mounted unit. It includes all sensors that perform non-contact detection in comparison to sensors, such as limit switches, that detect objects by physically contacting them. Proximity Sensors convert information on the movement or presence of an object into an electrical signal. There are three types of detection systems that do this conversion: systems that use the eddy currents that are generated in metallic sensing objects by electromagnetic induction, systems that detect changes in electrical capacity when approaching the sensing object, and systems that use magnets and reed switches.

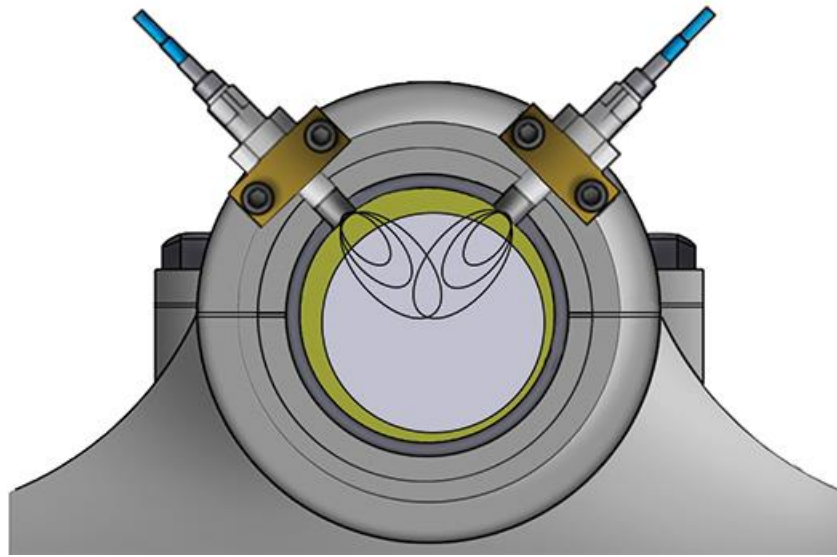


Figure 3.1. Proximity probe.

3.3.1.1 Inductive Proximity Sensors

The operating principle of inductive proximity sensors is based on a coil and oscillator that creates an electromagnetic field in the close surroundings of the sensing surface (see Figure 3.2). The presence of a metallic object (actuator) in the operating area causes a dampening of the oscillation amplitude. The rise or fall of such oscillation is identified by a threshold circuit that changes the output of the sensor. The operating distance of the sensor depends on the actuator's shape and size and is strictly linked to the nature of the material.

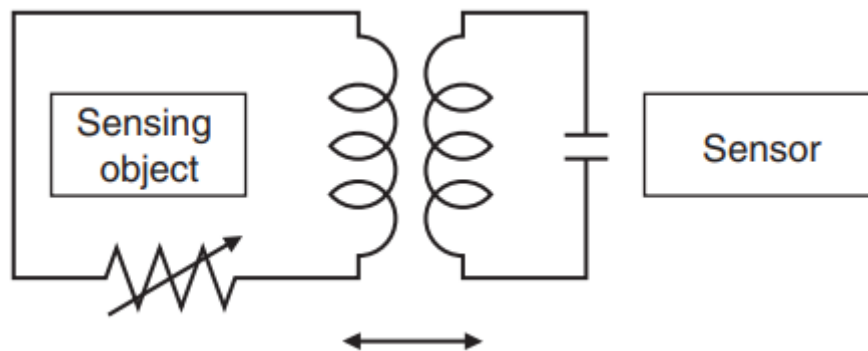


Figure 3.2. Schematic diagram of inductive proximity sensor.

3.3.1.2 Capacitive Proximity Sensors

Capacitive proximity sensors detect changes in the capacitance between the sensing object and the sensor. The amount of capacitance varies depending on the size and distance of the sensing object. An ordinary capacitive proximity sensor is similar to a capacitor with two parallel plates where the capacity of the two plates is detected. One of the plates is the object being measured (with an imaginary ground) and the other is the sensor's sensing surface. The changes in the

capacity generated between these two poles are detected. The objects that can be detected depend on their dielectric constant, but they include resin and water in addition to metals.

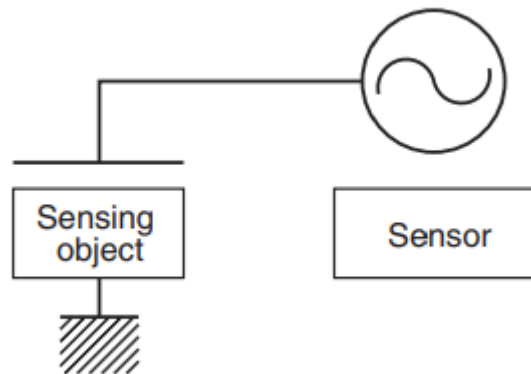


Figure 3.3. Schematic diagram of capacitive proximity sensor.

3.3.1.3 Features

- Proximity sensors detect an object without touching it, and they therefore do not cause abrasion or damage to the object
- No contacts are used for output, so the Sensor has a longer service life (excluding sensors that use magnets).

Proximity sensors are suitable for use in locations where water or oil is used.

- Proximity Sensors can be used in a wide temperature range.
- Proximity Sensors are not affected by colors [3].

3.3.1.4 Limitations

- Proximity Probes are difficult to calibrate absolutely.
- A variation in magnetic properties of shaft and geometric irregularities of shaft gives erroneous signal components.
- Have limited frequency range.

3.3.2 Velocity Transducers

Today, a vast majority of machinery vibration information is recorded and quantified in terms of velocity. Most vibration measurements in the process industries are analyzed in terms of inches per second (ips) in the United States, or mm/sec on the SI systems. Velocity readings are generally recommended for measurements in the 100 to 30,000 CPM (1.7 to 500 Hz) frequency band. Accurate and reliable vibration sensors are critical for successful machinery monitoring programs.

3.3.2.1 Piezo-Velocity Transducers

Piezo-Velocity Transducers (PVT) perform general purpose accelerometers and electrodynamic velocity pickups on slow speed equipment. In the 90 to 3600 CPM range, PVT internal integration provides greater signal fidelity than standard accelerometers. Solid state PVTs

are more reliable and measure broader frequencies than electrodynamic pickups. For many permanent sensor installations in paper mills, steel, and power generation facilities, a PVT sensor is the best investment. The PVT is essentially a piezoelectric accelerometer with an on-board velocity converter. The transducer employs a piezoceramic sensing element and dense seismic mass to produce a charge output proportional to acceleration. The high impedance charge signal is converted within the sensors to a low impedance voltage output and integrated to velocity [4]. Section views of compression and shear mode PVTs are shown in Figure 3.4

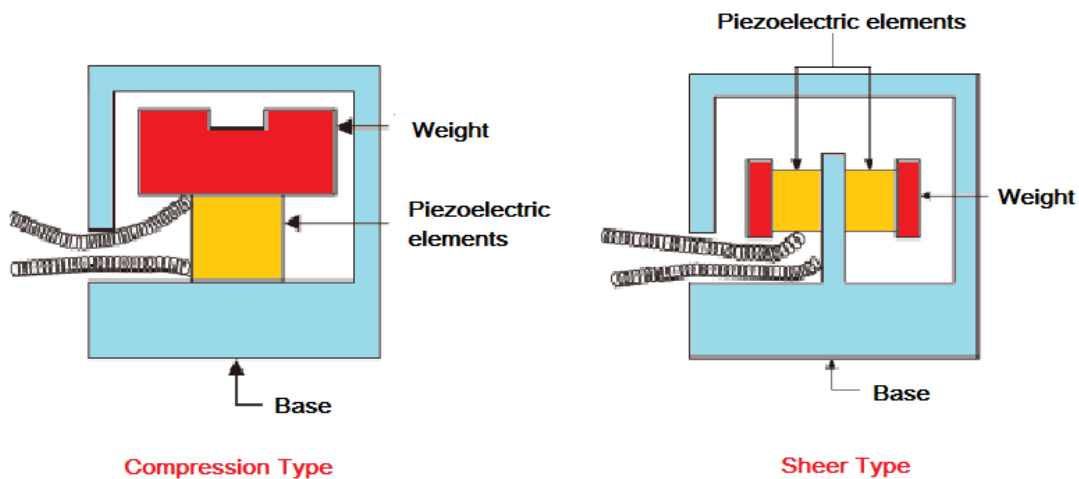


Figure 3.4. Two types of piezoelectric sensors.

3.3.2.2 Velocity Pickup Transducers

Velocity pickup transducer consists of coils, wire, and magnet and was arranged so that if the housing is moved, the magnet tends to remain stationary due to its inertia (see Figure 3.5). The relative motion between the magnetic field and the coil induces a current that is proportional to the velocity of motion and the unit produces a signal directly proportional to vibration velocity.

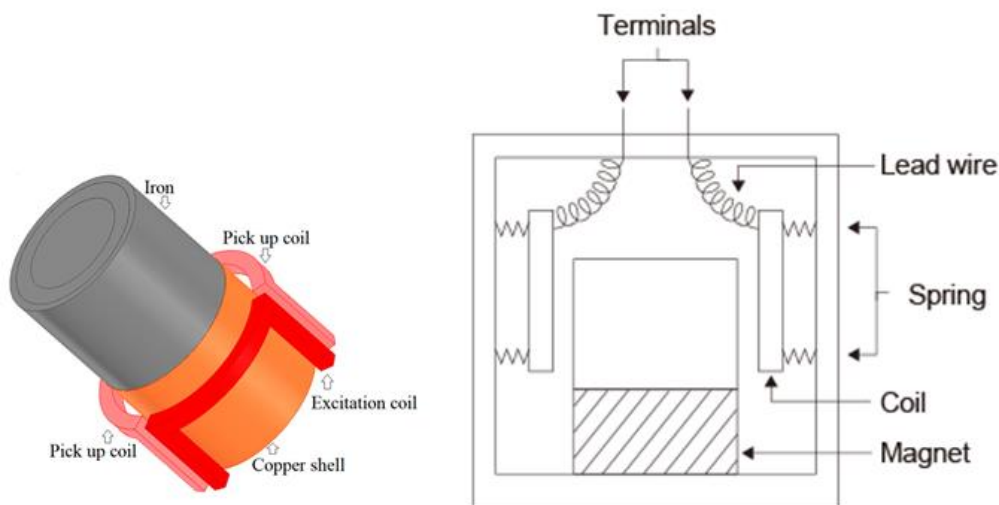


Figure 3.5 Velocity sensor: a) Geometry of two pickup coils, b) Finite element modeling.

3.3.2.3 Features

- Simple to install
- Good response in middle range frequencies
- Stand high temperature
- Do not require external power
- Lowest cost

3.3.2.4 Limitations

- Low resonant frequency and phase shift
- Cross noise
- Big and heavy
- Require electronic integration for displacement

3.3.3 Piezoelectric Accelerometers

Piezoelectric accelerometer is a self-generating device. The idea is that when a piezoelectric material is subjected to a mechanical stress (shear, compression, or bending), an electric signal generates across its faces and proportional to the stress (see Figure 3.6). Its frequency range is from below 1 to 10000 Hz. Certain natural and manufactured materials like quartz, tourmaline, lithium sulfate, and Rochelle salt generate electrical charge when subjected to a deformation or mechanical stress. The electrical charge disappears when the mechanical loading is removed. Such materials are called piezoelectric materials and the transducers, which take advantage of the piezoelectric effect, are known as piezoelectric transducers. The charge generated in the crystal due to a force F_x is given by:

$$Q = kF_x = kAP_x \quad (3.1)$$

Where k is called the piezoelectric constant, A is the area on which the force F_x acts, and P_x is the pressure due to F_x . The output voltage of the crystal is given by:

$$E = vtP \quad (3.2)$$

Where v is called the voltage sensitivity, and t is the thickness of the crystal. The values of the piezoelectric constant and voltage sensitivity for quartz are $2.25 \times 10^{-12} \text{C/N}$ and $0.055 \text{ volt-meter/n}$, respectively. These values are valid only when the perpendicular to the largest face is along the x -axis of the crystal.

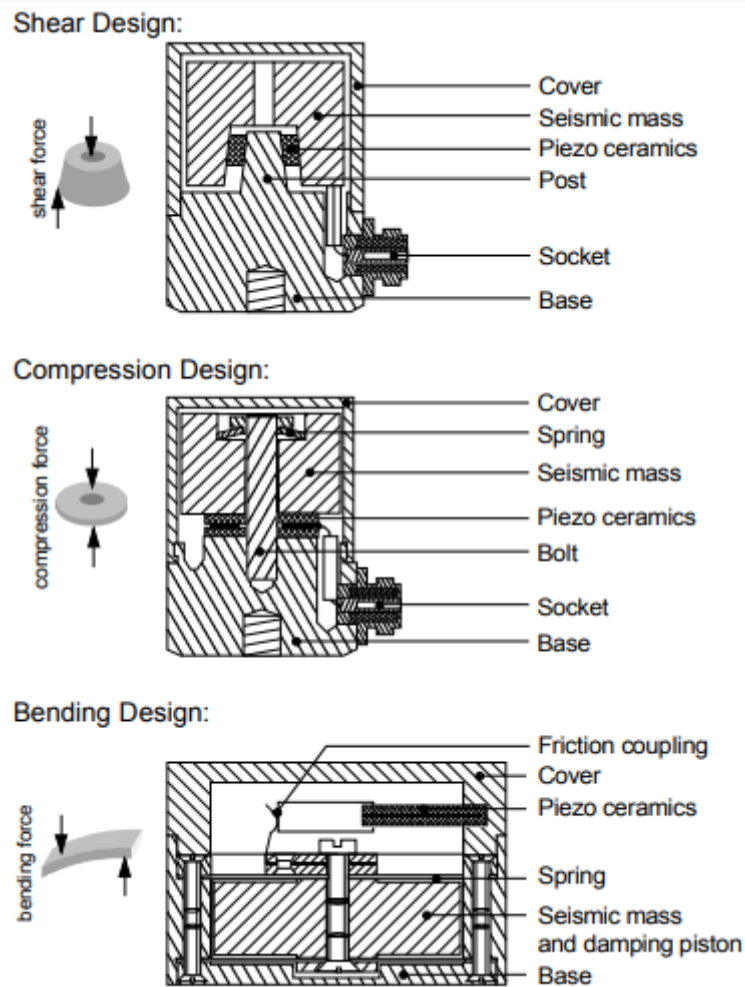


Figure 3.6. Piezoelectric accelerometer.

3.3.3.1 Piezoelectric Materials

The two basic piezoelectric materials used in vibration sensors today are synthetic piezoelectric ceramics and quartz. While both are adequate for successful vibration sensor design, differences in their properties allow for design flexibility. For example, natural piezoelectric quartz has lower charge sensitivity and exhibits a higher noise floor compared to the modern "tailored" piezoceramic materials. Now, most vibration sensor manufacturers use piezoceramic materials developed specifically for sensor applications. Special formulations yield optimized characteristics to provide accurate data in extreme operating environments. The exceptionally high output sensitivity of piezoceramic material allows the design of sensors with increased frequency response compared to quartz. Both quartz and piezoceramics exhibit an output during a temperature change (pyroelectric effect) when the material is not mounted within a sensor housing. Although this effect is much lower in quartz than in piezoceramics, the elements are isolated from fast thermal transients when they are mounted within the sensor housing. The dominant thermal signals are caused by metal case expansion strains reaching the base of the crystal. These erroneous signals are based on

the mechanical design rather than sensing material (quartz or piezoceramic). Some sensor designs isolate strains and minimize the thermal induced signals.

A typical piezoelectric transducer (accelerometer) is shown in Figure 3.6. In this figure, a small mass is spring loaded against a piezoelectric crystal. When the base vibrates, the load exerted by the mass on the crystal changes with acceleration and hence the output voltage generated by the crystal will be proportional to the acceleration. The piezoelectric accelerometer uses the characteristics of crystal material which responds to an applied force in accordance with Newton's second law (force = mass x acceleration). It produces a voltage and electric charge proportional to the applied force that is proportional to acceleration with a constant mass.

The piezoelectric accelerometer is very stable over long periods of time and maintain its calibration if it is not abused. The accelerometer can be damaged if it is subjected to excessive heat and dropped onto a hard surface. If it is dropped more than a few feet onto a concrete floor or steel deck, the accelerometer has to be re-calibrated to insure the integrity of crystal. The small crack reduces the sensitivity and affects the resonance, and thus the frequency response. It is a good idea to calibrate accelerometers about once a year if they are in service with portable data collectors.

The frequency range of the accelerometer is very wide, extending from very low frequencies in some units to several tens of kilohertz. The high-frequency response is limited by the resonance of the seismic mass coupled to the springiness of the piezo element. This resonance produces a very high peak in the response at the natural frequency of the transducer (near 30kHz for commonly used accelerometers). A rule of thumb is that an accelerometer is useable up to about 1/3 of its natural frequency as shown in the Figure 3.7.

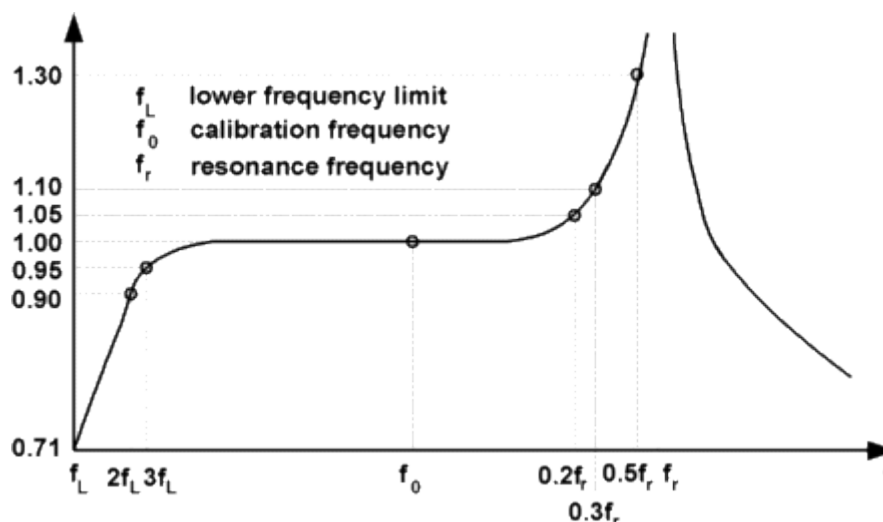


Figure 3.7. Accelerometer frequency response.

The resonant frequency of an accelerometer is strongly dependent on its mounting. The best type of mounting is always the stud mount -- anything else will reduce the effective frequency range of the unit. Today, most accelerometers used in industry are of internal integrated circuit preamplifier (ICP) as shown in Figure 3.6. This preamplifier is powered by a DC polarization of the signal lead itself, so no extra wiring is needed.

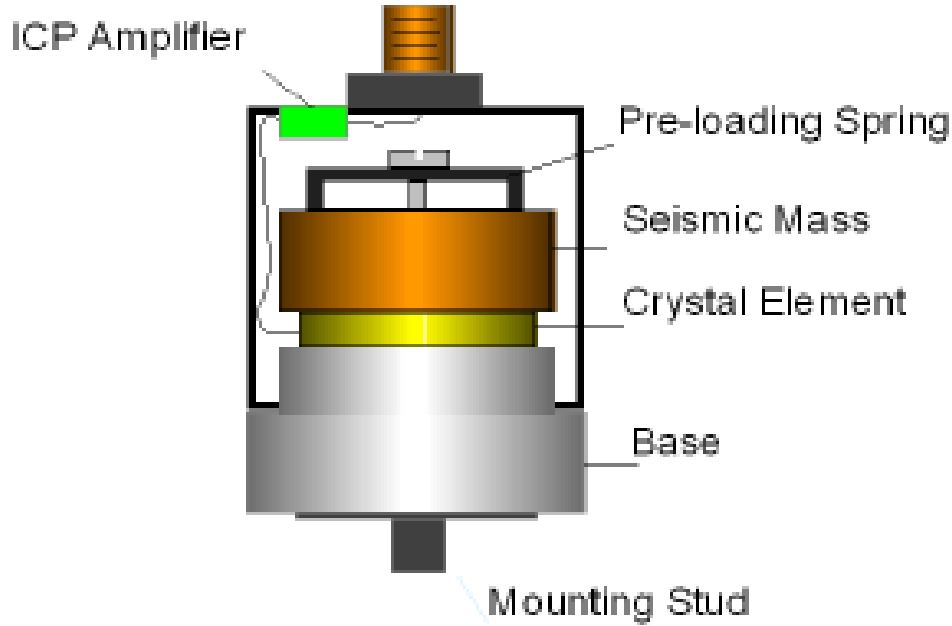


Figure 3.8. ICP accelerometer.

3.3.3.2 Accelerometer Mass

When the accelerometer is mounted on the test object, it will increase the mass of the vibrating system and the influence of the mechanical properties of the test object. The accelerometer mass should be less than the dynamic mass of the vibrating part onto which it is mounted. The effect of the accelerometer mass on the measured acceleration and system resonance frequency is as follow:

$$a = \frac{M+M_a}{M} a_a \quad (3.3)$$

$$F = \left(\sqrt{\frac{M+M_a}{M}} \right) F_a \quad (3.4)$$

$$e = \frac{a-a_a}{a} \times 100 = \frac{M_a}{M+M_a} \times 100 (\%) \quad (3.5)$$

Where a_a is the acceleration measured with accelerometer mounted; a is the acceleration without accelerometer; F_a is the resonance frequency with accelerometer mounted; F is the resonance frequency without accelerometer; M_a is the accelerometer mass (typical value: $0.7 \leq M_a \leq 500\text{g}$); M is the mass of the vibrating system.

3.3.3.3 Sensitivity

Sensitivity is accelerometer output per unit input in mv, (m/s^2) for voltage or in pc, (m/s^2) for charge. The sensitivity of condition monitoring accelerometers typically ranges between 10 and 500 mV/g; higher and lower sensitivities are also available. To choose the correct sensitivity for an application, it is necessary to understand the vibration amplitude levels to which the sensor will be exposed during measurements. As a rule of thumb, if the machine produces high amplitude vibrations (greater than 10 g RMS) at the measurement point, a low sensitivity (10 mV/g) sensor is preferable. The typical low sensitivity/high frequency accelerometers are used on equipment with geared drives. However, even lower speed machines may require less sensitive accelerometers if gear mesh frequencies correspond to structural resonance. If the maximum vibrations are less than 10 g RMS and the shaft speed greater than 600CPM (10Hz), a 100 mV/g sensor is probably the best choice. Low frequency accelerometers with sensitivities of 500 mV/g or higher should be used on precision spindles taming less than 600CPM. The sensitivity of different accelerometers and their natural frequencies are displayed in Figure 3.9. It was observed that as accelerometer size increases, its natural frequency decreases, its frequency range decreases, and its sensitivity increases.

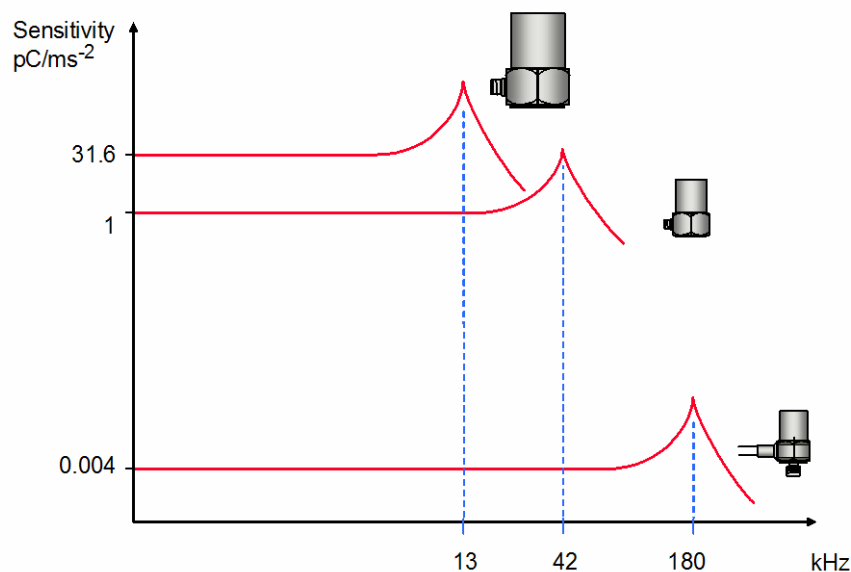


Figure 3.9. Sensitivity of different accelerometers and related natural frequency.

3.3.3.4 Dynamic range

The dynamic range determines the largest and smallest signals that can be simultaneously measured. The accelerometer gives an electrical output which is directly proportional to acceleration over a very wide dynamic range (approximately $1:10^8$). The lower end of the dynamic range is only limited by the noise in cables and preamplifiers. The lower dynamic limit is

determined by electrical noise from connecting cables and amplifier circuitry. The lower limit is about 0.001g. The upper dynamic limit is determined by the accelerometer's structural strength. For general purpose accelerometers, the upper limit is up to 500g.

3.3.3.5 Frequency range

Mechanical systems tend to have much of their vibration energy contained in the relatively narrow frequency range between 10 Hz to 1000 Hz but measurements are often made up to say 10 kHz because there are often interesting vibration components at these higher frequencies. Therefore, it is important to ensure that the accelerometer covers the range of interest. The frequency range over which the accelerometer gives a true output is limited at the low frequency by two factors. The first one is the low frequency cut-off of the amplifier which follows it. This is not normally a problem as the limit is usually well below 1Hz. The second factor is the effect of ambient temperature fluctuations with respect to sensitive accelerometer. This effect is minimal for modern shear type accelerometers. The upper limit is determined by the resonant frequency of the mass-spring system of the accelerometer itself. As a rule of thumb, if we set the upper frequency limit to one-third of the accelerometer's resonance(natural)frequency, we know that vibration components measured at the upper frequency limit will be in error by no more than + 12% (see Figure 3.10).

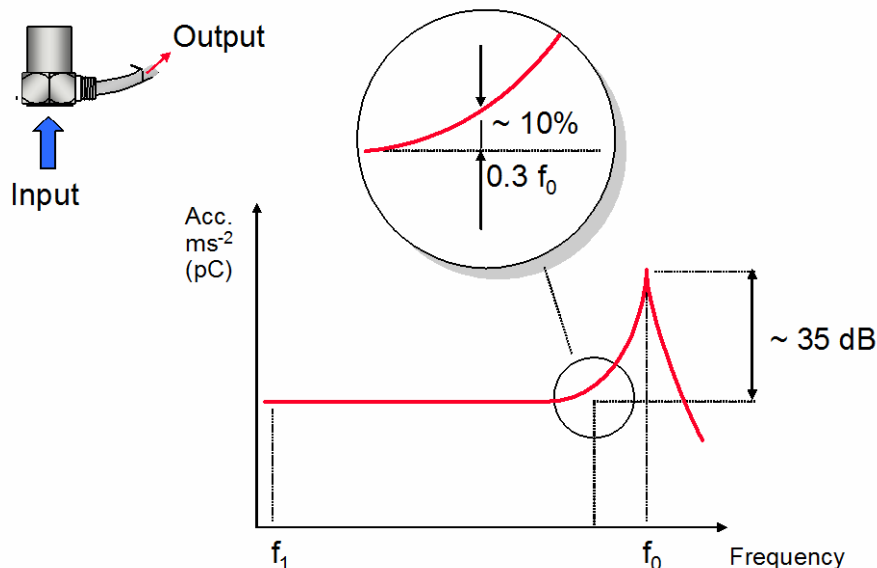


Figure 3.10. Accelerometer frequency range.

3.3.3.6 Accelerometer Types

A. Compression type accelerometers

This design is especially used for accelerometers which are intended for measurement of very high shock levels and other special purpose (see Figure 3.11)

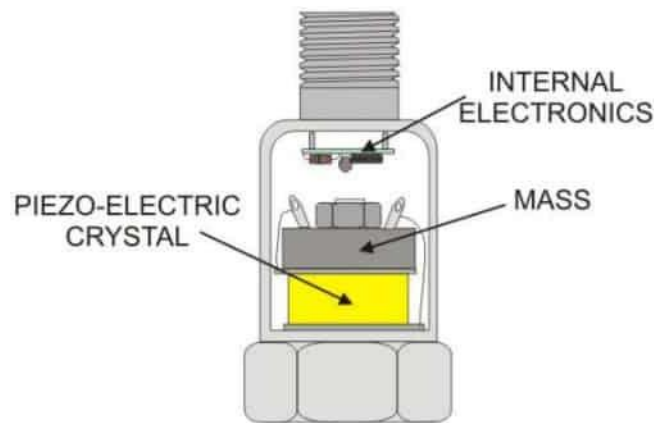


Figure 3.11 .Compression type accelerometer.

B. Planar Shear accelerometers

The main feature of planar shear accelerometer is its small size (see Figure 3.12). Therefore, it is used in places where the space is limited. Also, it is used to measure on very light structure.

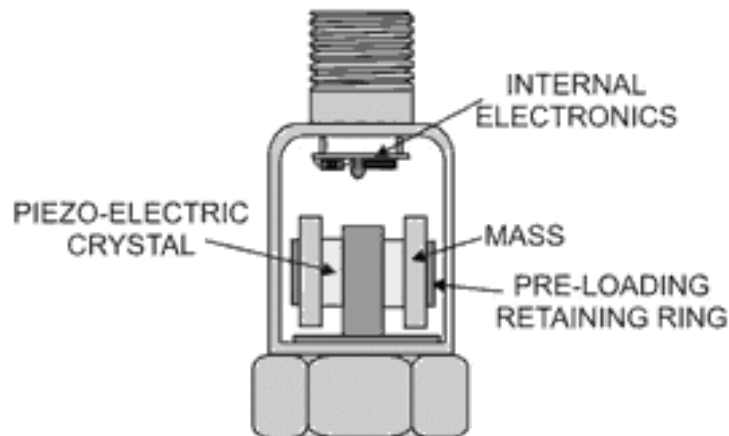


Figure 3.12. Planar shear type accelerometer.

C. Delta shear accelerometers

The delta shear accelerometer can be used for virtually any application. The advantage of the delta shear accelerometer is its excellent specifications and very low sensitivity to environmental influences.

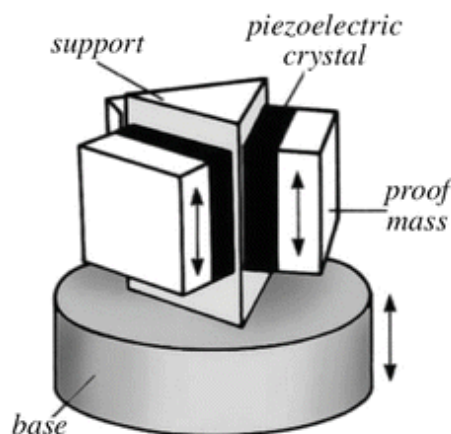


Figure 3.13. Delta shear type accelerometer.

4 Chapter (4) Concept of Vibration Measurement

4.1 Introduction

Excessive vibrations are often a product annoyance and a symptom of machinery in poor working condition. Low vibration means longer operating life, fewer maintenance problems, more comfort, and lower noise levels. Today, vibration measurements and analysis are the most widely used test, modeling, modification and diagnostic technique. For the study, understanding, and reduction of this phenomenon. Machinery condition measurement can tell us when the machinery condition starts to degrade. This allows maintenance to only occur when it is needed, rather than the regular schedule maintenance.

4.2 Types of Monitoring System

4.2.1 Intermittent Monitoring System

Intermittent monitoring system also called an off-line condition monitoring system is based on measuring vibration at selected time intervals in the field. Then, the analysis will be performed either in the field or later after data collection. Intermittent monitoring provides information at an early stage about incipient failure and usually is used where very early warning of faults is required, advanced diagnosis is required, measurement must be made at many locations on a machine, and machines are complex.

4.2.2 Permanent monitoring system

Permanent monitoring system also called an on-line condition monitoring system is based on measuring vibration continuously at selected points of the machine and comparing the measurements with the acceptable levels of vibration. In the permanent monitoring system, transducers are mounted permanently at each selected measurement point. Usually, permanent monitoring system used in critical applications where:

- No personnel are available to perform measurements (off shore, remote pumping stations, etc.).
- It is necessary to stop the machine before a breakdown occurs in order to avoid a catastrophic accident.
- An instantaneous fault may occur that requires machine shutdown.
- The environment (explosive, toxic, or high-temperature) does not permit the human involvement

4.3 Time Interval between Measurements

Selection of time interval between measurements requires knowledge of the specific machine. Some machines develop faults quickly and others run trouble-free for years, a

compromise must be found between the safety of the system and the time taken for measurements and analysis. The following rough rule of thumb is useful to select a time interval between measurements which is one-sixth to one-tenth the expected period between overhaul. In any case, measurements should be made frequently in the initial stages of condition monitoring program to ensure that the measured vibration levels measured are stable and no developed faults. When a significant change is detected, the time interval between measurements should be reduced sufficiently so as not to risk a breakdown before the next measurement. The trend curve will help in determining when the next measurement should be performed.

4.4 Mounting the Accelerometer

The method of mounting the accelerometer to the measuring point is one of the most critical factors in obtaining accurate results from practical vibration measurements. Sloppy mounting results in a reduction in the mounted resonant frequency which can severely limit the useful frequency range of the accelerometer. The main goal is to achieve close mechanical contact between the accelerometer base and the attached surface to which it is to be attached. The ideal mounting is by a threaded stud onto a flat, smooth surface as shown in Figure 4.1. A thin layer of grease applied to the mounting surface before tightening down the accelerometer will usually improve the mounting stiffness. The tapped hole in the machine part should be sufficiently deep so that the stud is not forced into the base of the accelerometer. The resonant frequency attained is almost as high as the 32kHz.

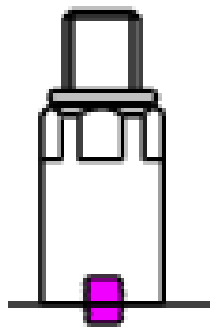


Figure 4.1. Threaded stud mounting.

A commonly used alternative mounting method is the use of a thin layer of bees-wax for sticking the accelerometer into place as shown in Figure 4.2. The resonant frequency is only slightly reduced (to 29kHz). Because bees-wax becomes soft at higher temperatures, the method is restricted to about 40°C. Bees-wax mounting method achieves acceleration levels up to 100 m/s² for clean surfaces.

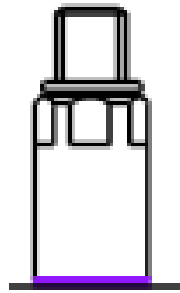


Figure 4.2. Bees wax mounting.

Where permanent measuring points are to be established on a machine and it is not wished to drill and tap fixing holes, pad can be used as shown in Figure 4.3. It is attached to the measuring point by means of a hard glue. Epoxy and cyanoacrylate types are recommended as soft glues can considerably reduce the usable frequency range of the accelerometer.

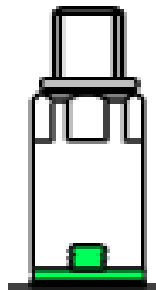


Figure 4.3. Adhesive pad mounting.

The permanent magnet mounting method is a simple attachment method where the measuring point is a flat magnetic surface (see Figure 4.4). This method reduces the resonant frequency of the accelerometer to about 7 kHz. The holding force of the magnet is sufficient for vibration levels up to 1000 to 2000 m/s^2 depending on the size of the accelerometer.

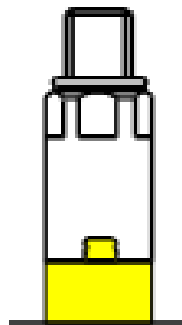


Figure 4.4. Flat magnet mounting.

A hand-held probe with the accelerometer mounted on top as shown in Figure 4.5 is very convenient for quick-look survey work, but can give gross measuring errors because of the low overall stiffness. A low-pass filter should be used to limit the measuring range at about 1000 Hz. The typical frequency response curves for different accelerometer mounting are displayed in Figure 4.6.



Figure 4.5. Hand held mounting.

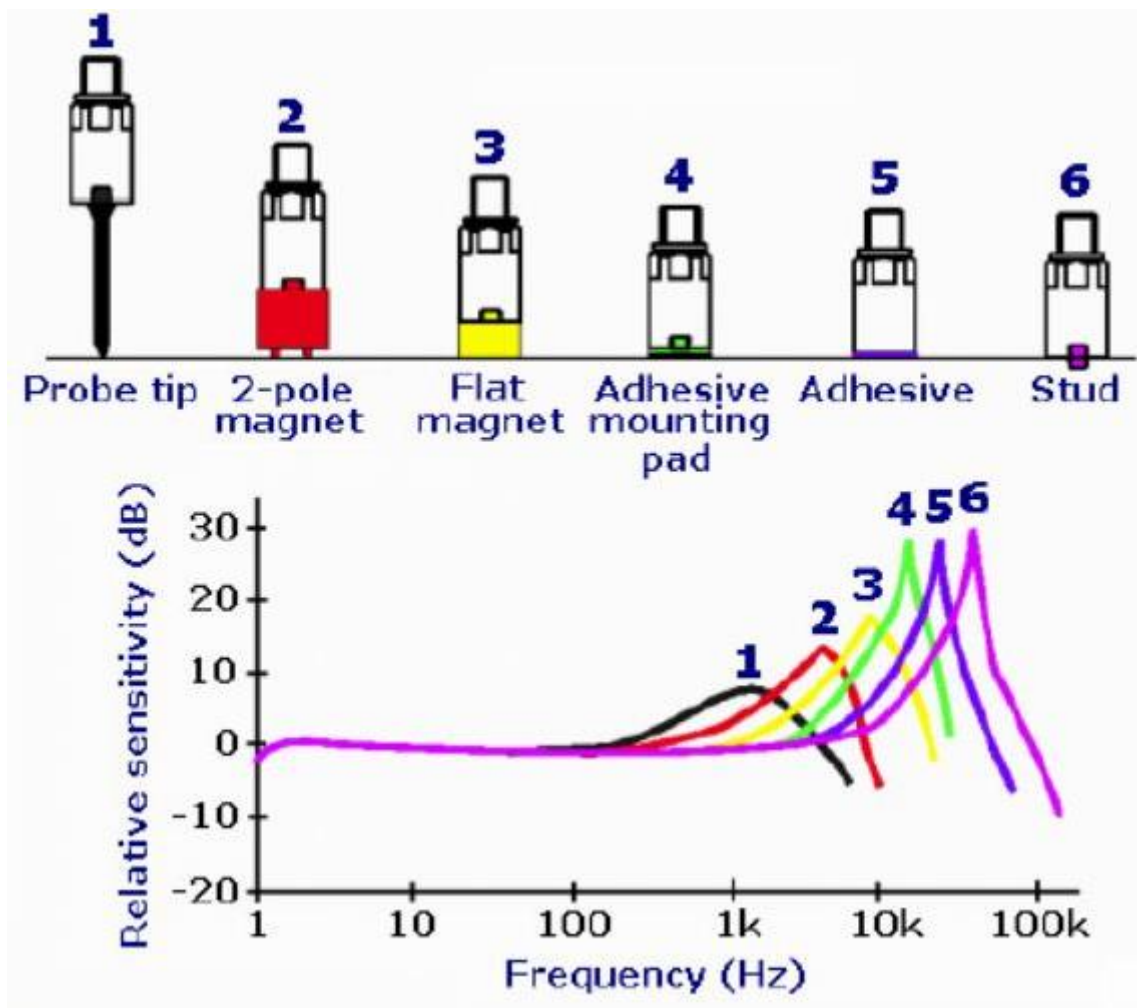


Figure 4.6. Frequency response for different accelerometer mounting.

4.5 The Influence of Environments on Accelerometers

Modern accelerometers and accelerometer cables are designed to have the minimum possible sensitivity to many external influences shown in Figure 4.7. Nevertheless, in more severe environments, special accelerometers are sometimes necessary.

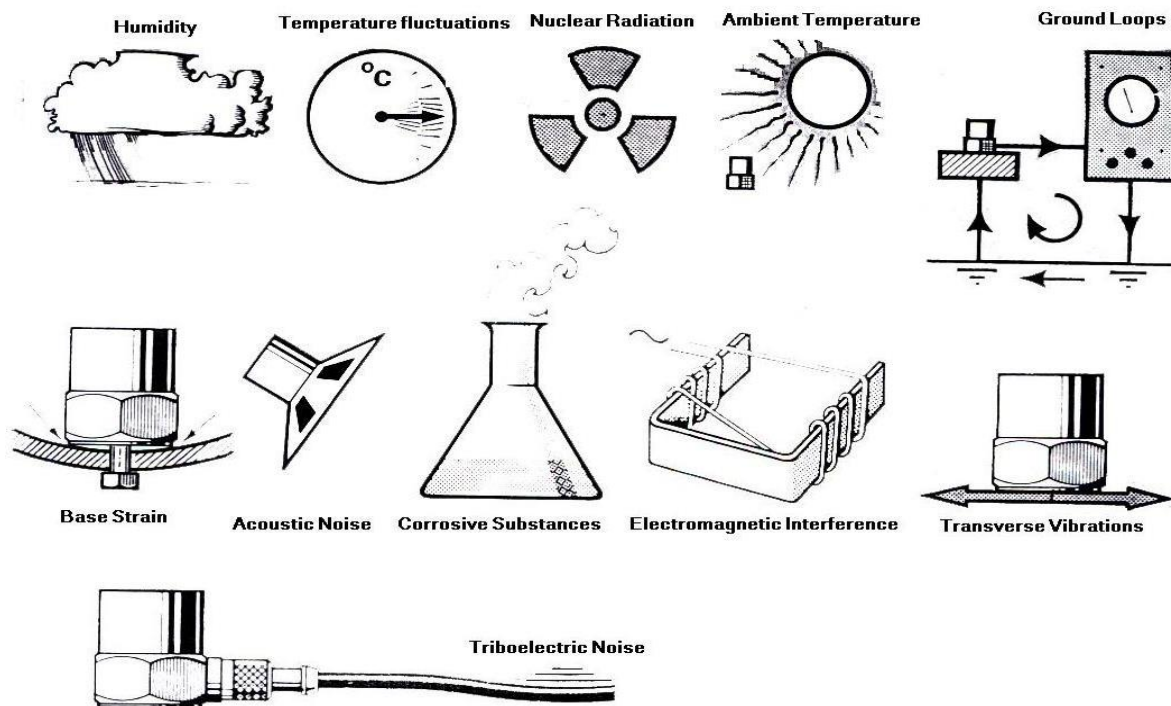


Figure 4.7. External influences of environments on accelerometers.

4.5.1 Base Strain

When an accelerometer is mounted on a surface that is undergoing strain variations, an output will be generated as a result of the strain being transmitted to the sensing element. Accelerometers are designed with thick and stiff bases to minimize this effect: Delta Shear types have a particularly low base strain sensitivity because the sensing element is mounted on a center post rather than directly to the accelerometer base.

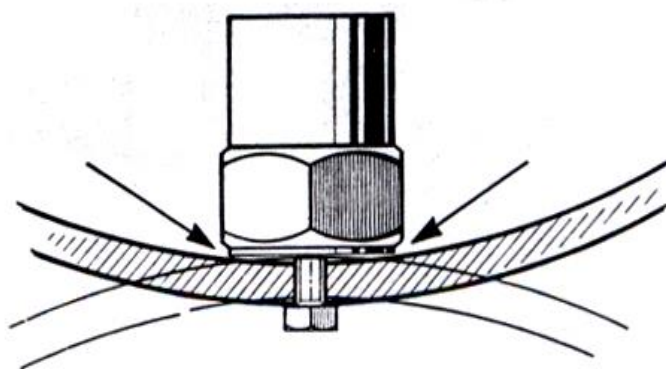


Figure 4.8. Influence of base strain.

4.5.2 Nuclear Radiation

Most accelerometers can be used under gamma radiation doses of 10kRad/h up to accumulated doses of 2 M Rad without significant change in characteristics. Certain accelerometers can be used in heavy radiation with accumulated doses in the excess of 100 M Rad.

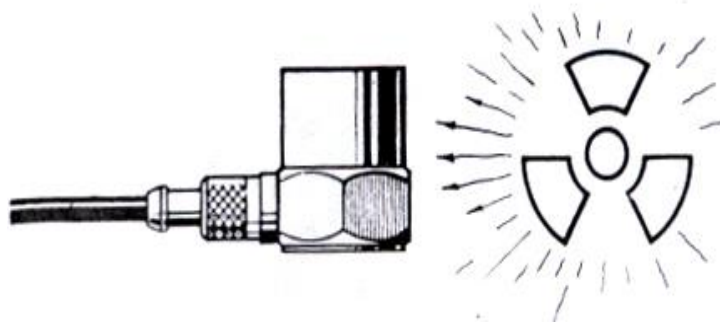


Figure 4.9. Influence of nuclear radiation.

4.5.3 Magnetic Field

The magnetic sensitivity of piezoelectric accelerometers is very low, normally less than 0.01 to 0.25 m/s² per k Gauss in the least favorable orientation of the accelerometer in the magnetic field.

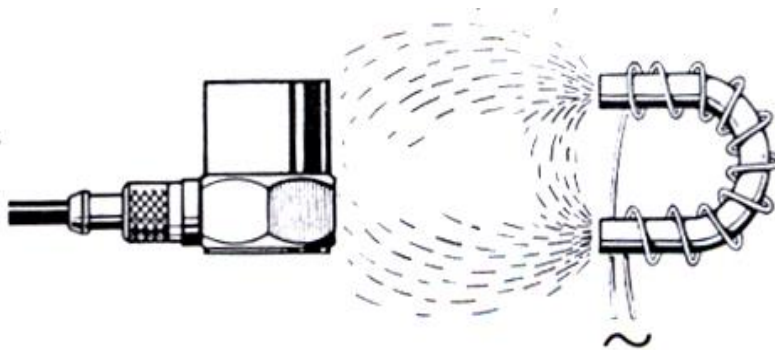


Figure 4.10. Influence of magnetic field.

4.5.4 Humidity

The accelerometer is sealed by epoxy bonding or welding to ensure reliable operation in humid environments. Teflon sealed accelerometer cables are recommended in case the measurement is performed in liquids for short duration. Also, the accelerometer connector should be sealed with an acid free room temperature vulcanizing silicon rubber or mastic. Industrial accelerometers with integral cables should be used for permanent use in humid or wet areas.



Figure 4.11. Influence of humidity.

4.5.5 Corrosive Substances

Special materials, which are resistant to most corrosive substances, are used in the construction of the accelerometer.

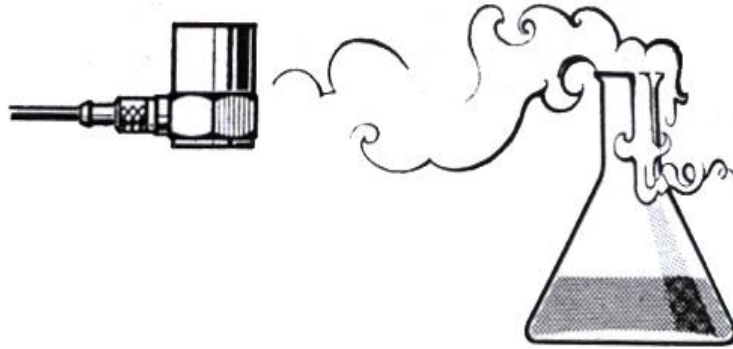


Figure 4.12.Influence of corrosive substances.

4.5.6 Acoustic Noise

Noise levels presented in machinery are not sufficiently high to cause any significant error in vibration measurements. Normally, the acoustically induced vibration in the structure, where the accelerometer is mounted, is far greater than the airborne excitation.

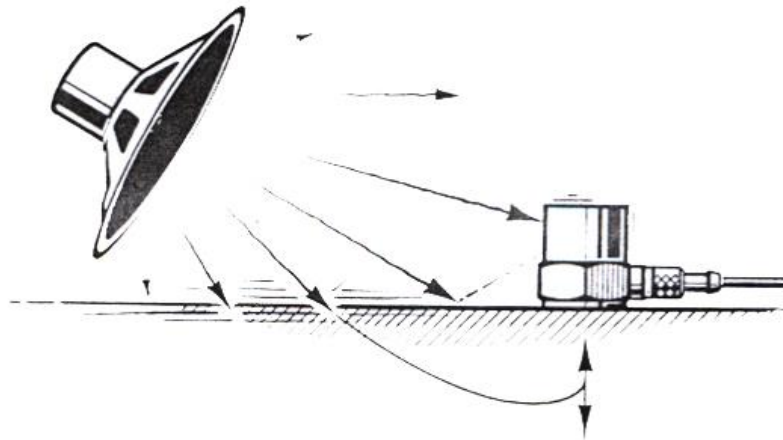


Figure 4.13.Influence of acoustic noise.

4.5.7 Transverse Vibrations

Piezoelectric accelerometers are sensitive to vibrations acting in directions other than coinciding with their main axis. In the transverse plane, perpendicular to the main axis, the sensitivity is less than 4% of the main axis sensitivity (see Figure 4.14). As the transverse resonant frequency normally lies at about 1/3 of the main axis resonant frequency, it should be considered where high levels of transverse vibration are present.

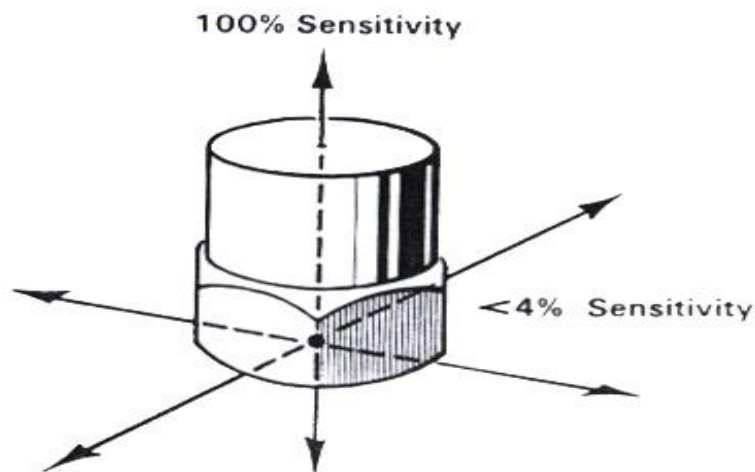


Figure 4.14. Influence of transverse vibrations.

4.5.8 Temperature Transients

Temperature transients (rapid fluctuations) can cause an electrical output from the accelerometer, but this effect has been considerably reduced in the delta shear accelerometer. The charges developed on the piezoelectric material due to temperature transients are mainly developed on surfaces normal to the polarization of the piezoelectric material which are not measured. This is the reason why the delta shear accelerometer is a good choice.

4.5.9 Sensitivity Change due to Temperature

Typical general purpose accelerometers can tolerate temperatures up to 125°C. At higher temperatures the piezoelectric ceramic will begin to depolarize so that the sensitivity will be permanently altered. Such an accelerometer may still be used after recalibration if the depolarization is not too severe. For temperatures more than 125°C, accelerometers with a special piezoelectric ceramic are available. Alternatively, a heat sink and mica washer can be inserted between the base and the measuring surface and the stream of cooling air can provide additional assistance.

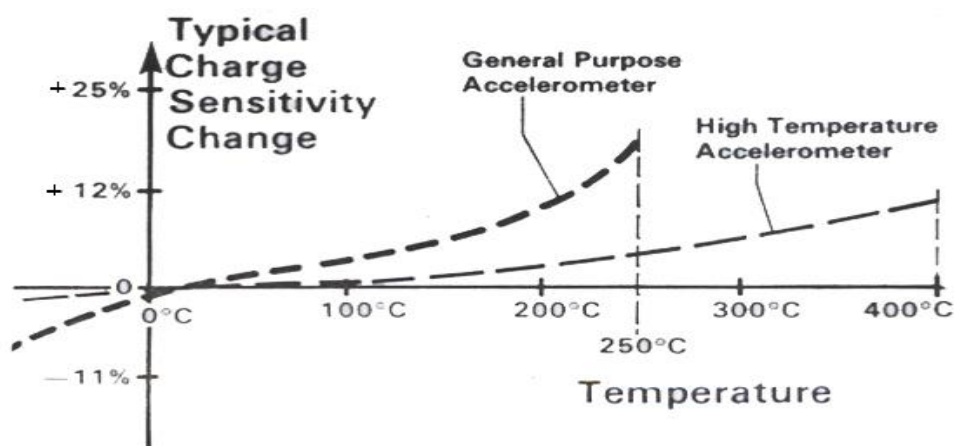


Figure 4.15. Sensitivity change due to temperature.

4.5.10 Earth Loops

If the accelerometer is fixed to a test object which is connected to the ground, an earth loop will exist causing noise to superimpose the vibration signal from the accelerometer. This situation can be avoided by mounting the accelerometer by the aid of an isolating mounting method. Another way of avoiding earth loops is to use an accelerometer which has its piezoelectric material isolated from the housing.

4.5.11 Triboelectric Noise

Triboelectric noise is often induced into the accelerometer cable by mechanical motion of the cable itself. It originates from local capacity and charge changes due to dynamic bending, compression, and tension of cable layers. This problem can be avoided by using a proper graphited accelerometer cable and taping or gluing it down as close to the accelerometer as possible.

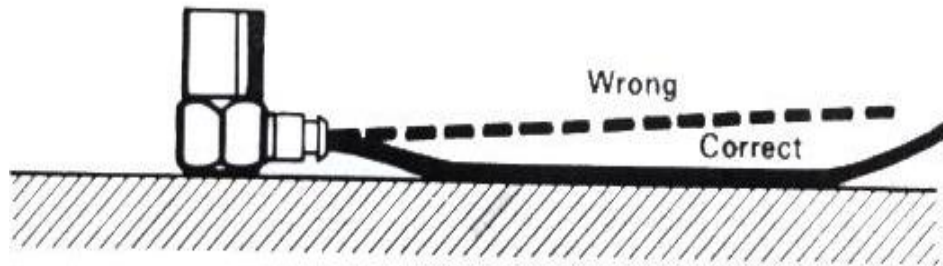


Figure 4.16. Influence of accelerometer cable movement.

4.6 Accelerometer Mounting Directions

Vibration is a result of dynamic forces in machines which have moving parts connected to machine structures. Different machine parts vibrate with various frequencies and amplitudes. These vibration data are collected from points on bearings, preferably in two directions, radial measurements on bearing housing and axial measurements on the ends of bearing housing where the bearings take up axial forces (see Figure 4.17). The axial direction (A) is parallel to the rotational axis of the machine. The radial directions (R) is perpendicular to the rotational axis of the machine. The same concept of axial and radial directions is applied to the vertical mounting machines as shown in Figure 4.18. The accelerometer should be mounted so that the desired measuring direction coincides with its main sensitivity axis. Accelerometers are also slightly sensitive to vibrations in the transverse direction, but this can normally be ignored as the transverse sensitivity is typically less than 4% of the main axis sensitivity.

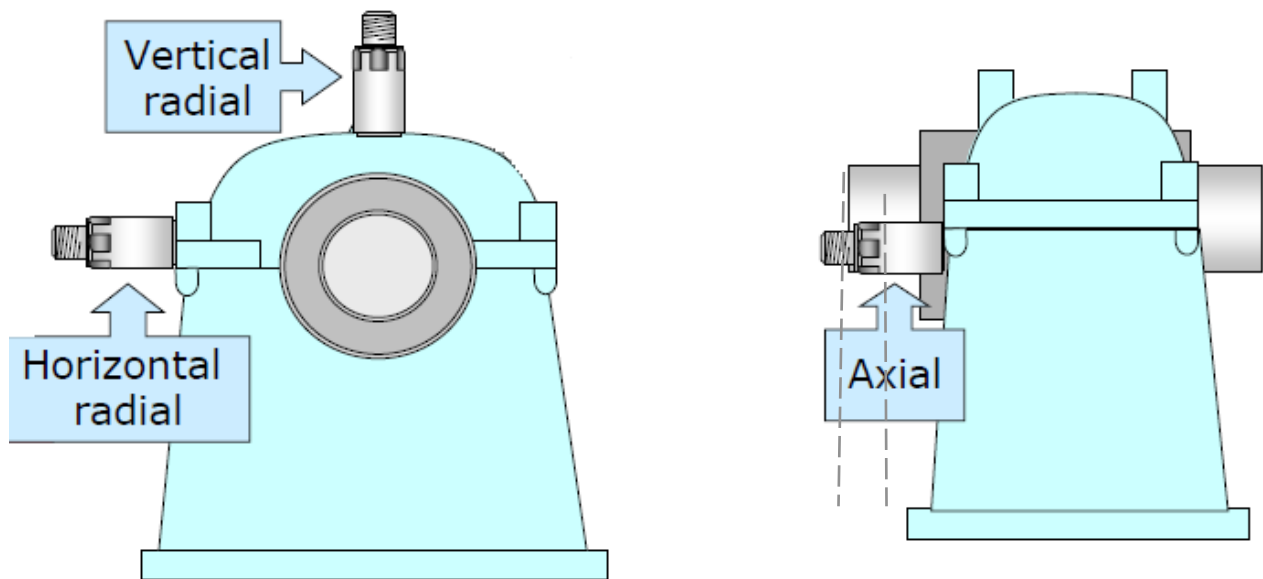


Figure 4.17. Vibration measurement direction.

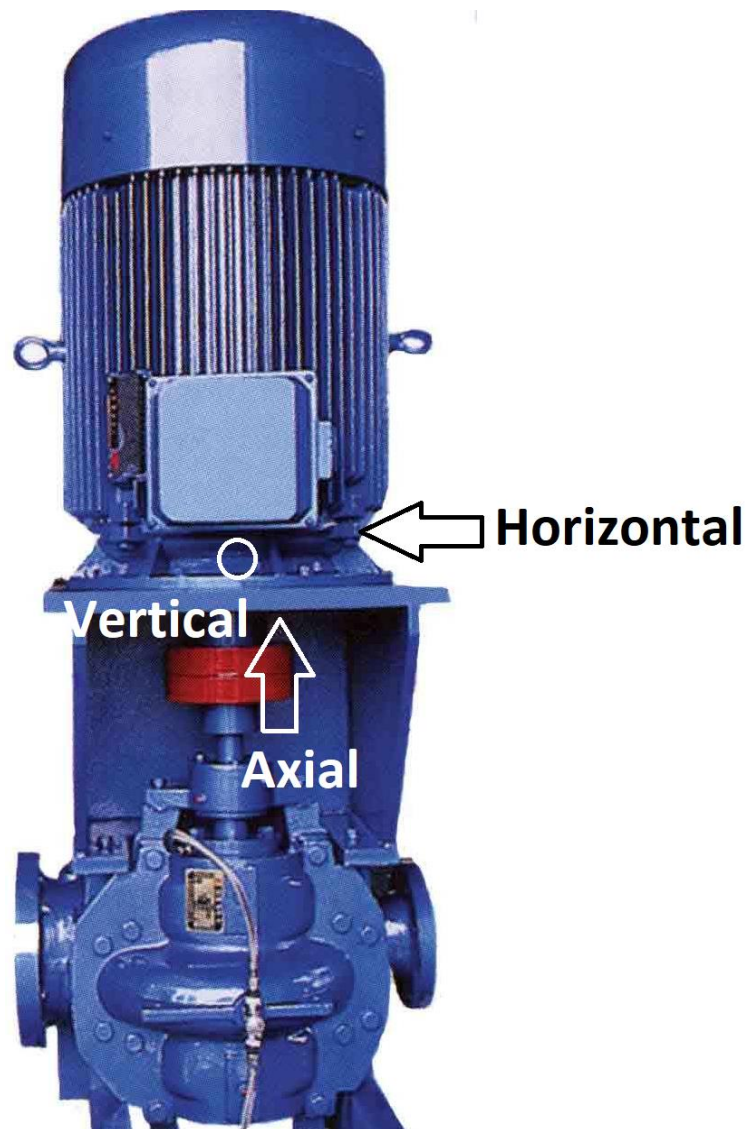


Figure 4.18. Measurement directions of vertical machines.

Vibration measurement locations are defined by the rigid member such as bearing housings, bearing pedestals, and machine casings. Each component has at least two measurement points. For example, motor-pump machine shown in Figure 4.19 has two measurement points for motor and pump. The point nears coupling is called drive end and the other point is non drive end.

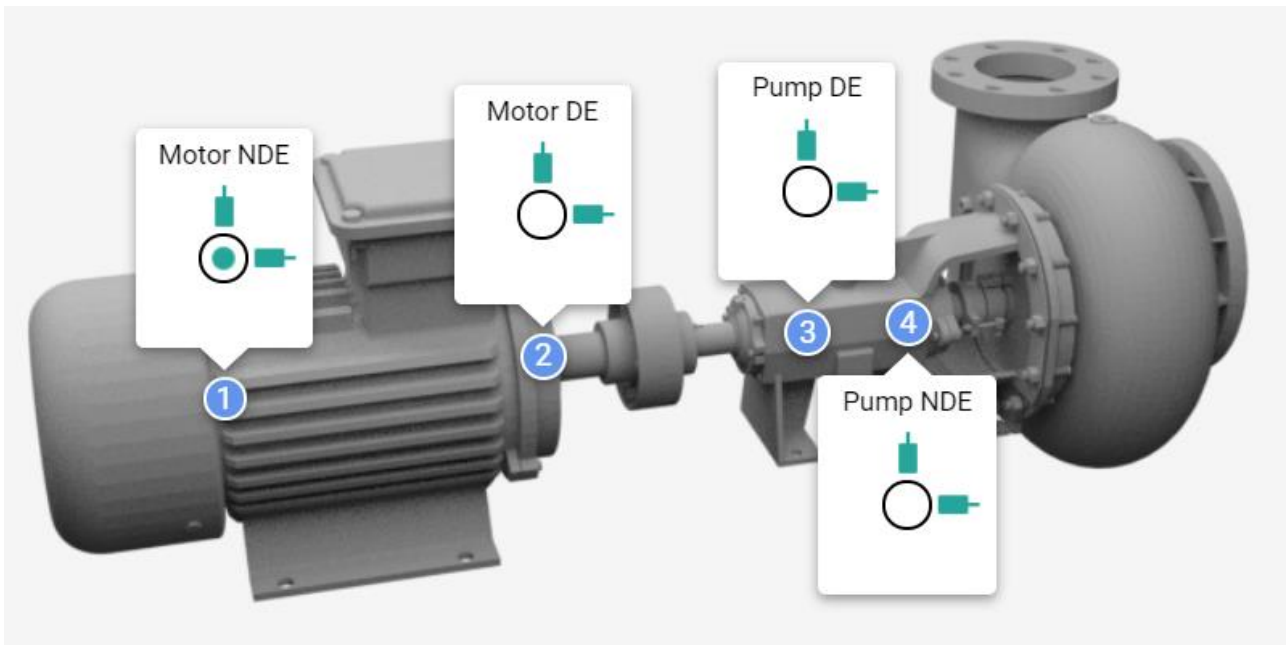


Figure 4.19 .Vibration measurement locations for a typical horizontal machine.

5 Chapter (5) General Fault Analysis

5.1 Type of Vibration Signals

There are three types of vibration signals as shown in Figure 5.1; periodic (i.e., machinery shaft speed), random (i.e., varying), transient (i.e., pump cavitation due to improper system line-up).

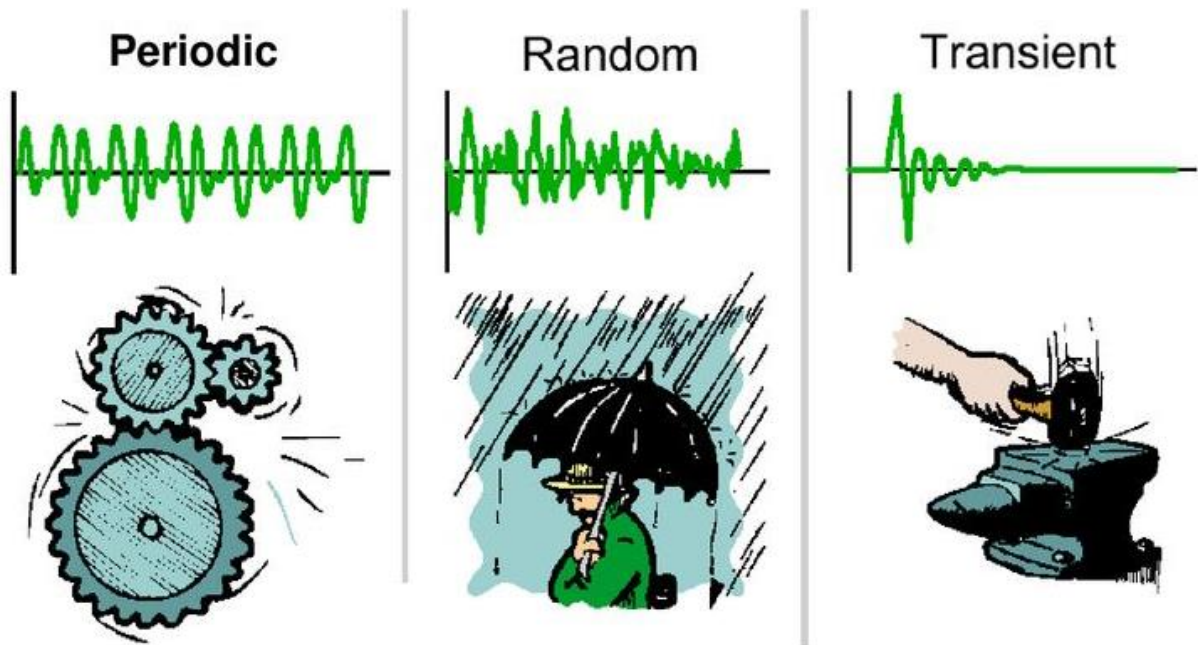


Figure 5.1. Basic types of signals.

5.2 Interpretation of Vibration Data

The key of using vibration signature analysis for predictive maintenance, diagnostic, and other applications is the ability to differentiate between normal and abnormal vibration profiles. Vibrations are usually found in rotary machines due to shaft rotation, rotors contact with bearings, gear-mesh, and else. However, specific problems with machinery generate abnormal vibrations such as rotor unbalance, loose bolts, misaligned shafts, and worn bearings. Predictive maintenance utilizes vibration signature analysis based on the following facts:

1. All common machinery problems and failure modes have distinct vibration frequency components that can be isolated and identified.
2. A frequency-domain vibration signature is generally used for the analysis since it includes discrete peaks representing the specific vibration source.
3. There is a reason for each frequency component in the vibration signature of machine train.
4. When the machine vibration signature is compared over the time, it will be similar until some event changes the vibration pattern.

While the increase of amplitude may indicate degradation of machine train, there are other parameters have significant effect on change the amplitudes of vibration signature such as the load and operating condition. In addition, it is important to note that a lower amplitude does not necessarily indicate an improvement in the mechanical condition of the machine train. Therefore, it is important to understand clearly the source of all amplitude variations.

5.3 Vibration monitoring

Most vibration based predictive maintenance programs rely on one or more monitoring techniques. These techniques include broadband trending, narrowband trending, or signature analysis. The following events are essential in machine condition monitoring:

- Date of time based maintenance
- Initial fault indication and tentative diagnosis
- Diagnosis fault and predicting date of breakdown
- First alarm level (warning)
- Planned repair effected
- Second alarm level (shutdown)
- Machine breakdown

5.4 Monitoring Techniques

5.4.1 Trending

Trend monitoring of vibration level is a more useful maintenance tool than a one-time survey of the absolute magnitudes of features only. A slight linear increase of feature values turns to exponential when the point of failure is approaching. The time of failure depends strongly on machines and the stress caused by operating conditions. The slope of the feature values or different order derivatives can be used in control in order to reduce the stress imposed on process equipment. Since the remaining useful life can be estimated, the effects of the operating point on overall equipment effectiveness can be calculated. Examples of healthy and faulty machine trending are displayed in Figure 5.2 and Figure 5.3, respectively.

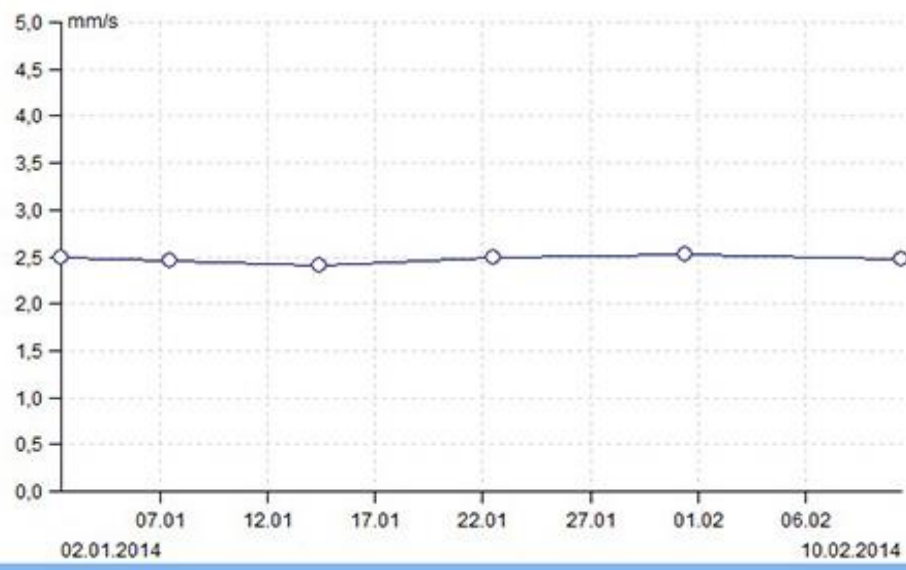


Figure 5.2.Example of healthy machine (constant trending).

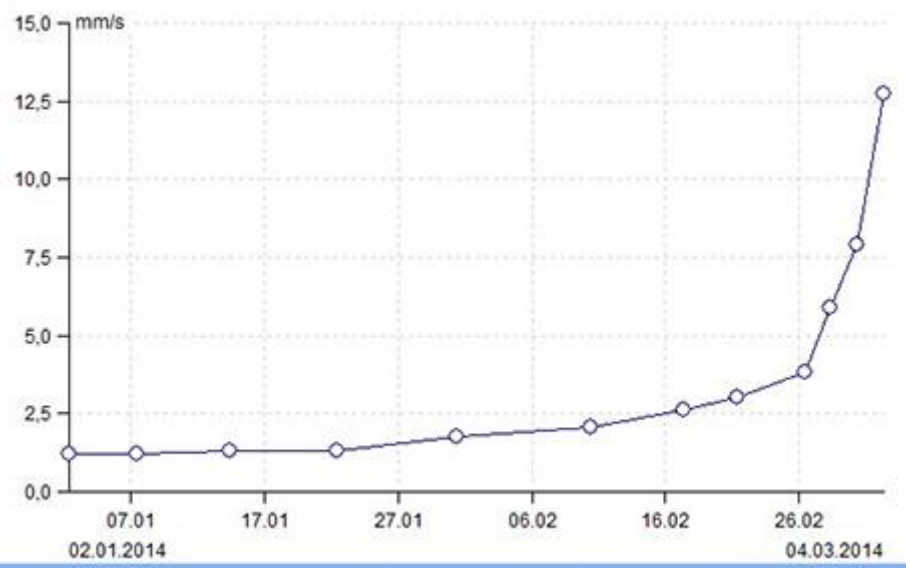


Figure 5.3.Example of faulty machine (linear increase trending).

5.4.1.1 Broadband Trending

Broadband trending acquires overall or broadband vibration readings from selected points on the machine train. This data is compared to either a baseline reading taken from a new machine or to vibration severity charts to determine the relative condition of the machine. Broadband measurement between 10 and 1000Hz is usually used in this type of analysis. Broadband or overall RMS data present the total vibration level for the machine at a specified measurement point where the data was acquired. It does not provide any information about the individual frequency components or machine dynamics that created the measured value.

5.4.1.2 Narrowband trending

Narrowband trending, like broadband, monitors the total energy for a specific bandwidth of vibration frequencies. Unlike broadband, narrowband analysis utilizes vibration frequencies that represent specific machine components or failure modes. This method provides the means to quickly monitor the mechanical condition of critical machine components, not just the overall machine condition. This technique provides the ability to monitor the condition of gear sets, bearings, and other machine components without manual analysis of vibration signatures.

5.4.2 Comparative Analysis

Comparative analysis directly compares two or more data sets in order to detect changes in the operating condition of mechanical or process systems. Vibration signatures such as time domain and frequency domain signals are used in in this type of analysis. The comparison is performed with baseline data, known machine condition, or industrial reference data. It is important to use the same units of frequency and amplitude to make a correct comparison.

5.4.3 Baseline Data

Vibration measurements are collected for each machine train after new installation. These data set can be used as a reference or comparison data set for all future measurements. However, such data set must be representative of the normal operating condition of each machine-train. There are three criteria use the baseline comparisons: reset after maintenance, proper identification, and process envelope.

5.4.4 Signature Analysis

Signature analysis provides visual representation of each frequency component generated by the machine train. Vibration signatures are helpful to determine the required maintenance. Most predictive maintenance based vibration programs use some forms of signature analysis in their programs. However, the majority of these programs rely on comparative analysis rather than full root cause techniques.

5.5 Quantifying the Vibration Level

The overall vibration level, which describes the vibration severity, can be quantified using RMS for velocity measurements. RMS is the most relevant measurement of overall vibration based on international standards organization (ISO). After taking measurement values of vibration velocity levels in mm/s (RMS), they are evaluated based on ISO standard. In general, ISO 10816-3 is used to evaluate the majority rotary machines.

Figure 1 Vibration Severity ISO 10816-3 : 2009						
Vibration Severity			Goodness judgement for machine group			
Value of Velocity -boundary-			Group2		Group1	
0.71	mm/s		A	A	A	A
1.4			B	B	B	
2.3						
2.8			C	B	B	B
3.5						
4.5			D	C	C	C
7.1				D	D	
11.0				D	D	D
Base state of Machine			Rigid	Flexible	Rigid	Flexible

Example of Goodness A:Good B:Satisfactory C:Unsatisfactory D:Unacceptable

Group 1 Huge Machine Output, 300kW~50MW

Group 2 Middle Machine Output, 15kW~300kW

Figure 5.4.Vibration severity ISO 10816-3.

6 Chapter (6) Understanding Data Collection

6.1 Introduction

Vibration data collection is one of the most important steps in analyzing issues caused by machinery vibration. If done correctly, your analysis, recommendations and reporting become much more accurate. It involves the ability to safely collect vibration data from a given machinery under normal operating condition. This is useful for the analysis of the health condition of the equipment and to provide corresponding recommendations to correct any noted anomalies. To correctly and safely collect the vibration data, we need to understand how this process is implemented. In this chapter, you will find the methods of collecting data and the sequence of processing this data in the collector.

6.2 Methods of Vibration Data Collection

6.2.1 Automated Method

In this method, accelerometers are permanently mounted on machines and terminated into data acquisition Units, which collect and process data at designated intervals (every minute, hourly, daily, weekly, etc.), depend on the criticality of the machine. As shown in Figure 6.1, this method has a simplified layout with less personnel involvement in the day-to-day operations of the process. Data collected can be made available instantly online for all stakeholders to view.



Figure 6.1 .Schematic diagram of automated collection sequence.

6.2.2 Manual Method

This entails the use of trained personnel to download routes manually from software on a computer onto a data collector. The data collector is carried around the plant (from one machine to the other) and used to collect data from each designated location on a machine. Data is collected on the average interval of once per month, or quarter, while under normal operating condition. After data has been collected, the device is connected to a computer and the routes uploaded back into the software for analysis. As depicted in Figure 6.2, this method relies on a lot of manual involvement by trained personnel to ensure

the process is efficient. Data collected is commonly exported and sent by email to other stakeholders who may not have access to the local computer where the software is installed.

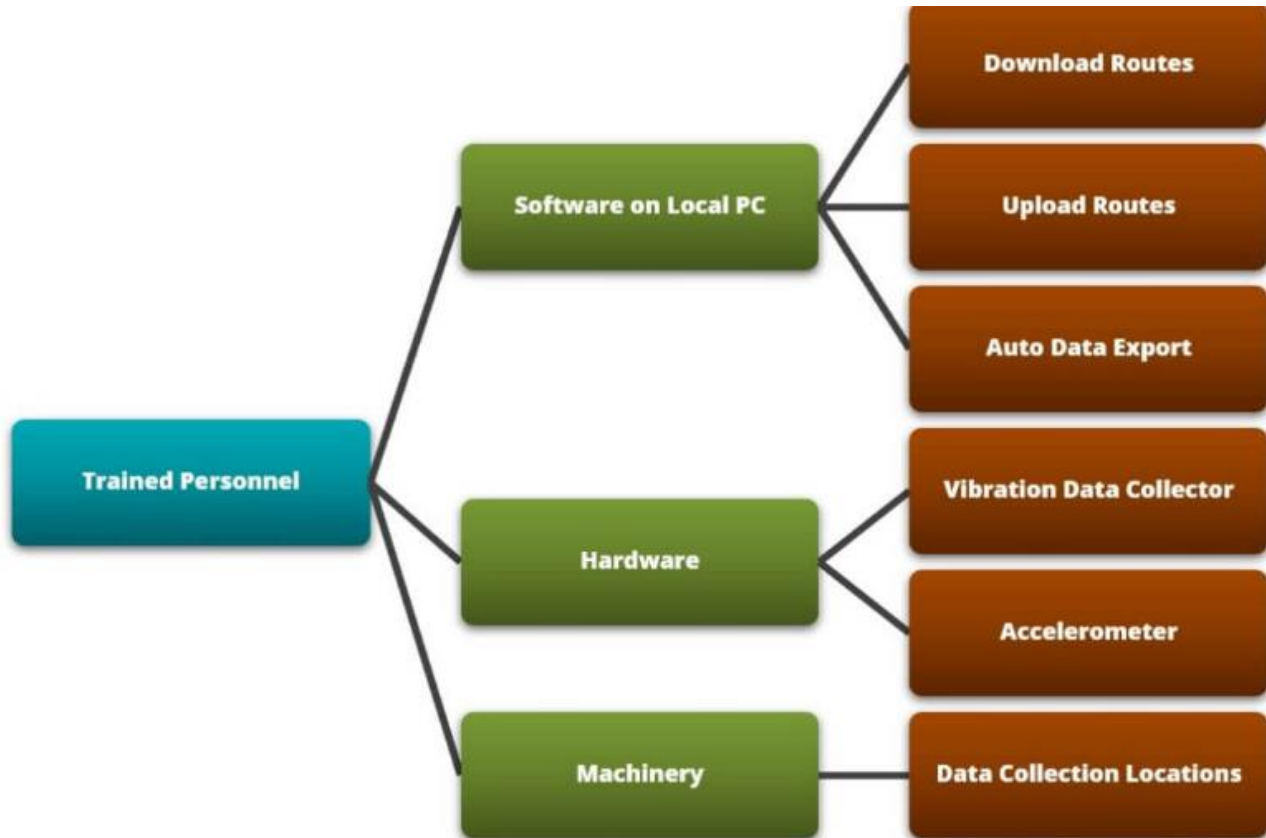


Figure 6.2 .Schematic diagram of manual collection sequence.

6.3 Analog to Digital Converter

The vibration waves collected by transducers are analog signals. Analog signals have to be converted into digital values for further processing (see Figure 6.3). This conversion from an analog signal to a digital signal is done by an Analog to Digital (A/D) converter. The A/D conversion is essentially done by microprocessors based on binary numbers system (the power of 2). A 12-bit A/D converter provides 4096 (2^{12}) discrete intervals whereas a 16-bit A/D converter provides 65536 (2^{16}) intervals. The greater the number of intervals, the better the amplitude resolution of the signal. A 12-bit A/D converter would result in a resolution of 0.025% of the full scale, whereas a 16-bit A/D converter would yield a resolution of 0.0015%. It is thus possible to collect a signal with large and small amplitudes accurately.

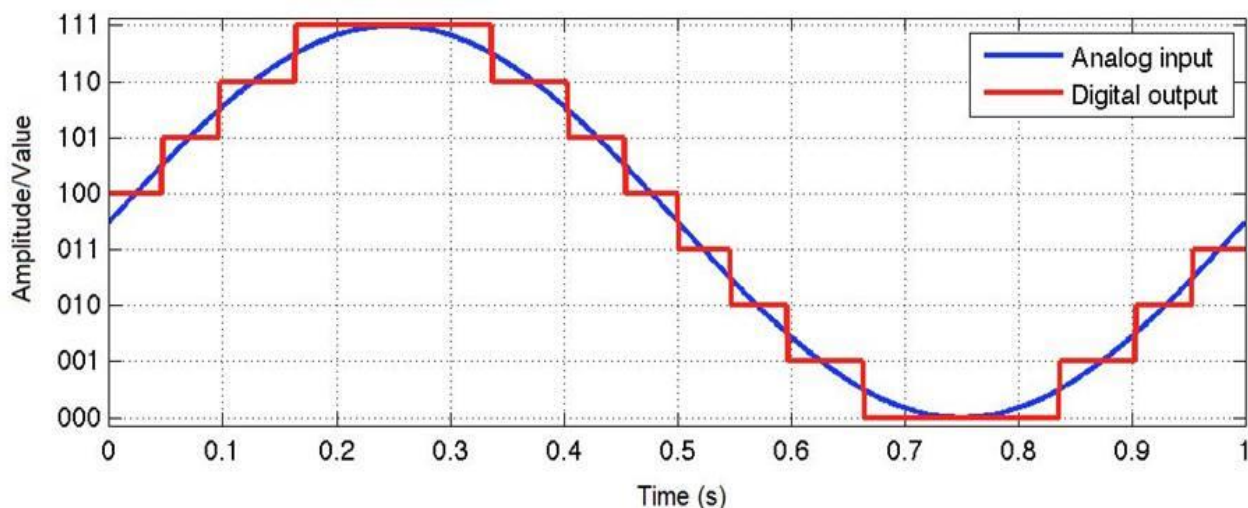


Figure 6.3. Analog to digital converter.

6.4 Sampling rate

Sampling is the process of recording the amplitude of a wave at given instants, and then generating a curve from the recorded points. Thus, the collected discrete sampled data points (digital) are used to reconstruct the wave which was originally in an analog form. The reconstructed digital wave has to look similar to the original wave or in other words, take samples so that the digitized wave is an exact replica of the original analog wave. The answer lies in the Nyquist sampling theorem, which states that:

‘If we are not to lose any information contained in a sampled signal, we must sample at a frequency rate of at least twice the highest frequency component of interest.’

Figure 6.4 (left) displays examples for low and high sampling rate. It was observed that when the sampling rate is too low (under sampling rate), another wave can be generated with a different frequency rather than the original wave frequency. The sampling rate needs to increase to make a correct representation for the original wave as shown in Figure 6.4 (right). The phenomenon of a lower frequency wave due to under sampling is called aliasing. All data collectors/analyzers have automatically selected built-in sampling rates and fitted with an antialiasing filter. The antialiasing filter is a low pass electronic filter which allows low frequencies to pass and prevents higher ones. The filter removes all vibrations in the analog signal that have frequencies greater than half the sampling rate. It is automatically tuned to the proper values as the sampling frequency is changed (this occurs when the frequency range of the analyzer is changed by the user). It is very important to note that filtering has to occur before digitization of the analog commences.

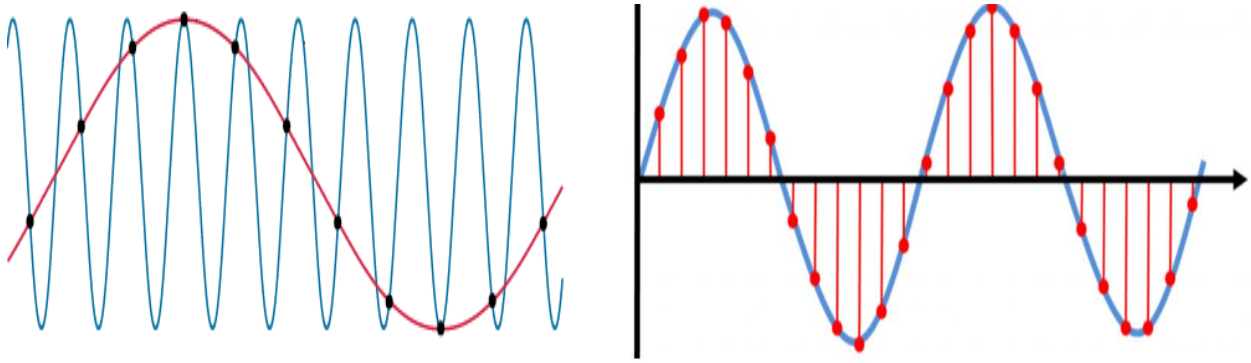


Figure 6.4. Example of low sampling rate (left), high sampling rate (right).

6.5 Lines of Resolution

Since the digitized time wave forms are sometimes difficult to interpret, another form of data is presented based on Fast Fourier Transform (FFT) technique. The FFT is a spectrum of amplitude vs frequency. The resolution is the number of lines (bins or LOR) that are used to display the frequency spectrum. If we have N samples in the time record, FFT produces a spectrum with $N/2$ lines. So, a 2048 sample time record would produce a 1024-line spectrum. But due to the nature of anti-alias filters, we actually only keep 800 lines. The accepted rule is that the number of lines is equal to the number of samples in the time record divided by 2.56. The LOR could be 200, 400, 800, 1600, 3200, 6400 and 12800. It can now be deduced that when resolution is high, there is a better distinction between frequency peaks. Another important parameter is the required maximum frequency (F-max) which has a direct effect on the sampling rate since the sampling rate equals 2.56 of maximum frequency. The time required for collecting the data varies inversely with F-max. The higher F-max, the quicker the FFT can be displayed. As a general guideline, the following advises are provided to select F-max value:

- For general rotating machinery such as pumps, fans, blowers, and motors, set the F-max to 20x or 40x where x is the running speed.
- When measuring vibrations on gearboxes, the F-max setting should be at least three times higher than the gear mesh frequency, where the gear mesh frequency is the number of teeth of pinion and gear times their respective running speeds.
- If the analysis on the machine is conducted for the first time, it is advisable to take two spectra, one by selecting F-max at 10x and the other at 100x. This is to ensure that no important frequencies are lost in the low or high frequency zone. Once the range of the suspicious frequencies is noted, F-max could be selected accordingly.

7 Chapter (7) Signal Processing

7.1 Introduction

Since the captured data from the transducer are continuous analog signals, some methods of signal processing have to be performed for further analysis such as windowing, averaging, filtering, and else. In this chapter, windowing and its types are presented. Then, the averaging and forms of averaging are included. Finally, the concept of anti-aliasing filter is identified.

7.2 Windowing

In signal processing, a window function is a mathematical function that is zero-valued outside of some chosen interval, normally symmetric around the middle of the interval, usually near a maximum in the middle, and usually tapering away from the middle as shown in Figure 7.1. In actual practice, the segment of data within the window is first isolated, and then only that data is multiplied by the window function values. Thus, tapering, not segmentation, is the main purpose of window functions. The reasons for examining segments of a longer function include detection of transient events and time-averaging of frequency spectra. The duration of the segments is determined by requirements like time and frequency resolution. But that method also changes the frequency content of the signal by an effect called spectral leakage. There are different types of window functions which will be discussed in the following sections.

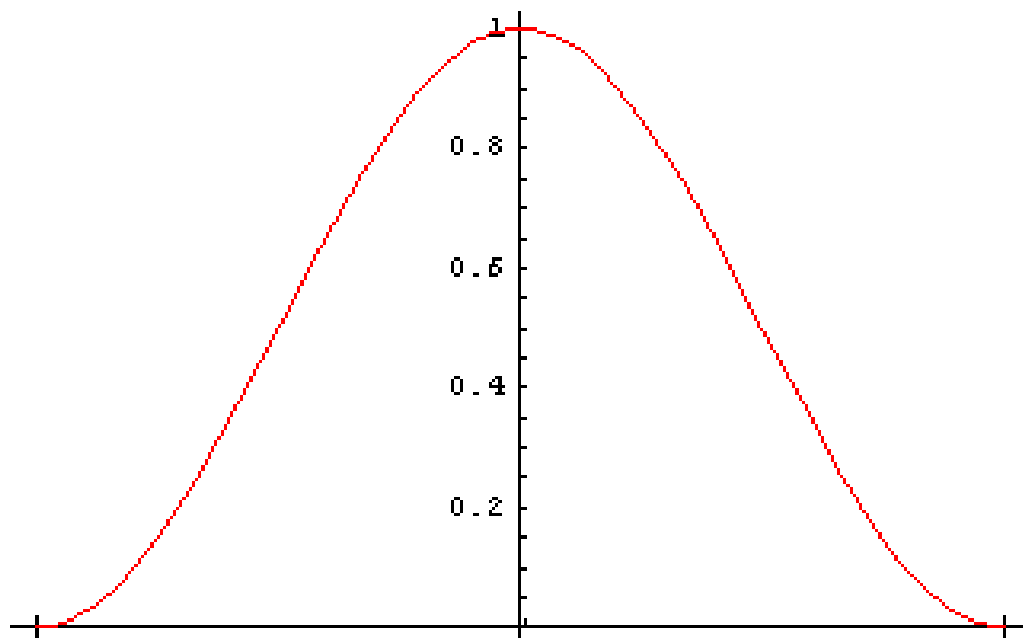


Figure 7.1. A form of window function.

7.2.1 Rectangular Window

Even if you use no window, the signal is convolved with a rectangular-shaped window of uniform height, by the nature of taking a snapshot in time of the input signal and working with a discrete signal. This convolution has a sine function characteristic spectrum. For this reason, no window is often called the uniform or rectangular window because there is still a windowing effect.

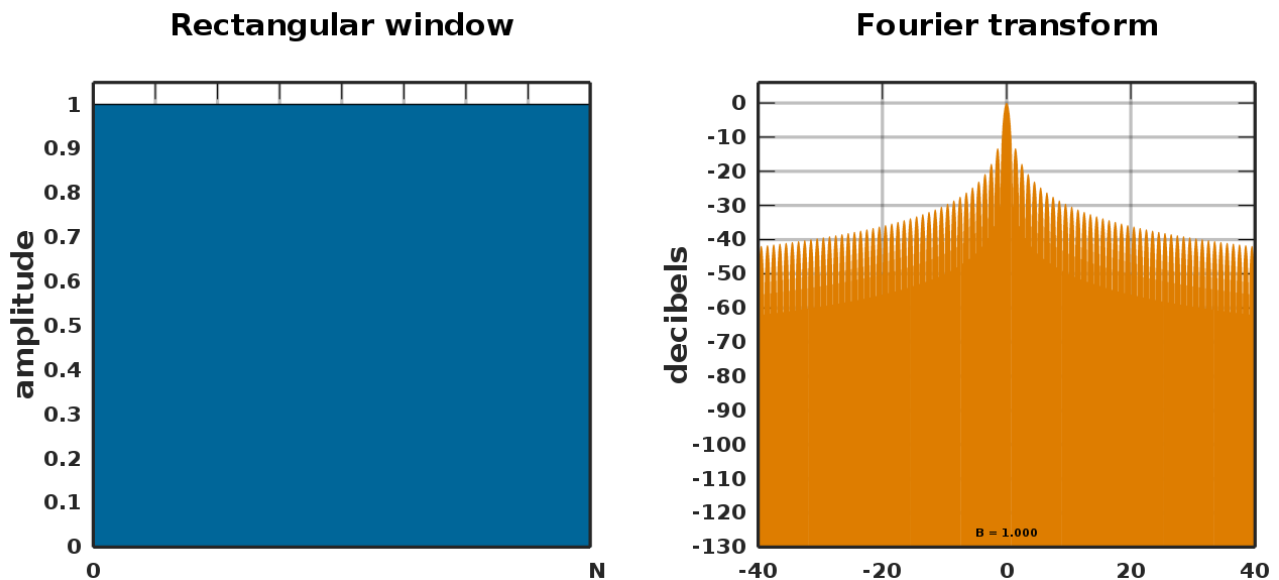


Figure 7.2. Rectangular window function.

7.2.2 Hann and Hamming Window

The Hann and Hamming window functions both have a sinusoidal shape. Both windows result in a wide peak but low side lobes. However, the Hann window touches zero at both ends eliminating all discontinuity. The Hamming window doesn't quite reach zero and thus still has a slight discontinuity in the signal. Because of this difference, the Hamming window does a better job of cancelling the nearest side lobe but a poorer job of canceling any others. These window functions are useful for noise measurements where better frequency resolution than some of the other windows is wanted but moderate side lobes do not present a problem. The Hann window is the most commonly used window in machinery analysis.

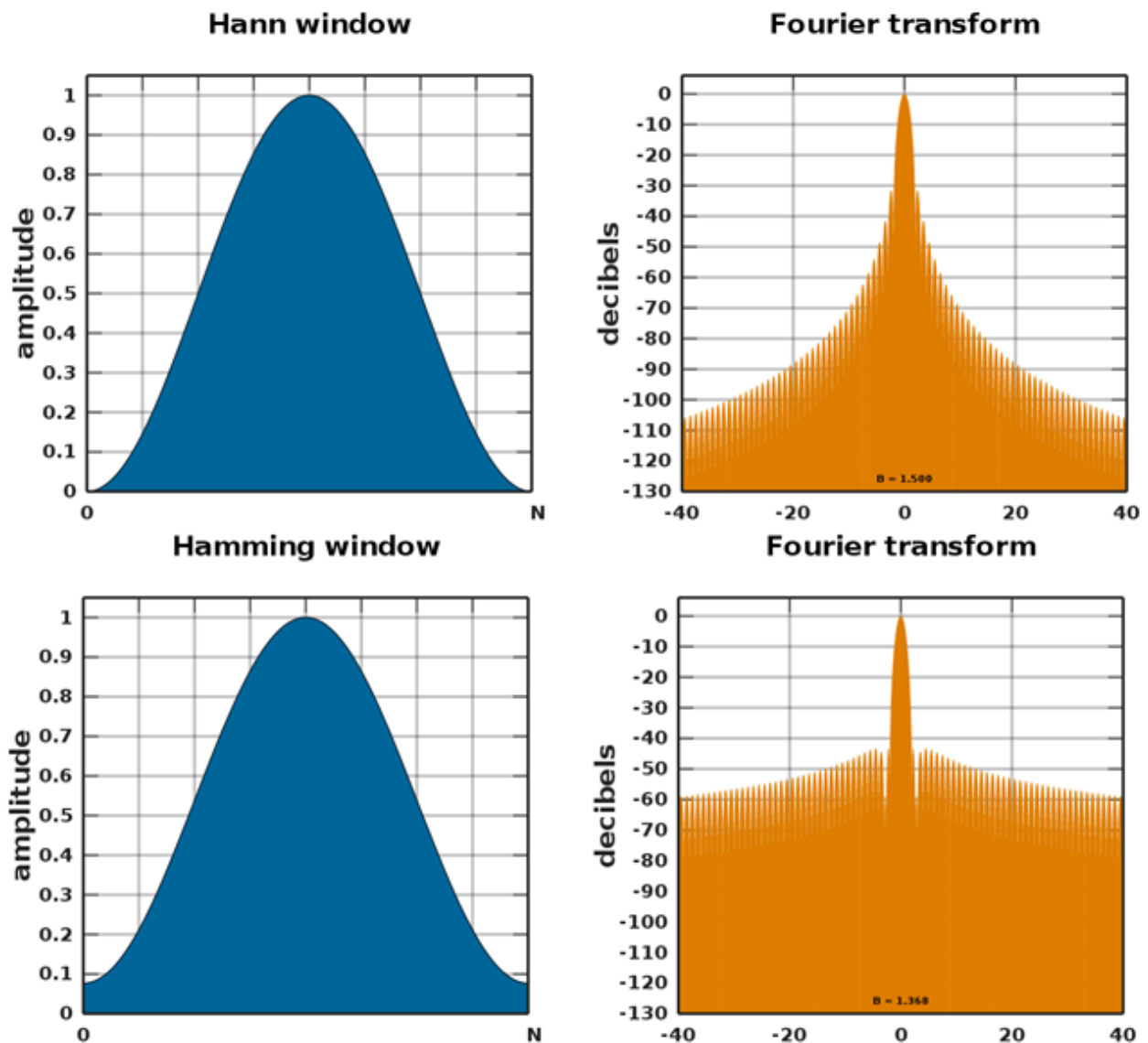


Figure 7.3. Hann and hamming window functions.

7.2.3 Blackman-Harris Window

The Blackman-Harris window is similar to Hamming and Hann windows. The resulting spectrum has a wide peak, but good side lobe compression. There are two main types of this window. The 4-term Blackman-Harris is a good general-purpose window, having side lobe rejection in the high 90s dB and a moderately wide main lobe. The 7-term Blackman-Harris window function has all the dynamic range you should ever need, but it comes with a wide main lobe.

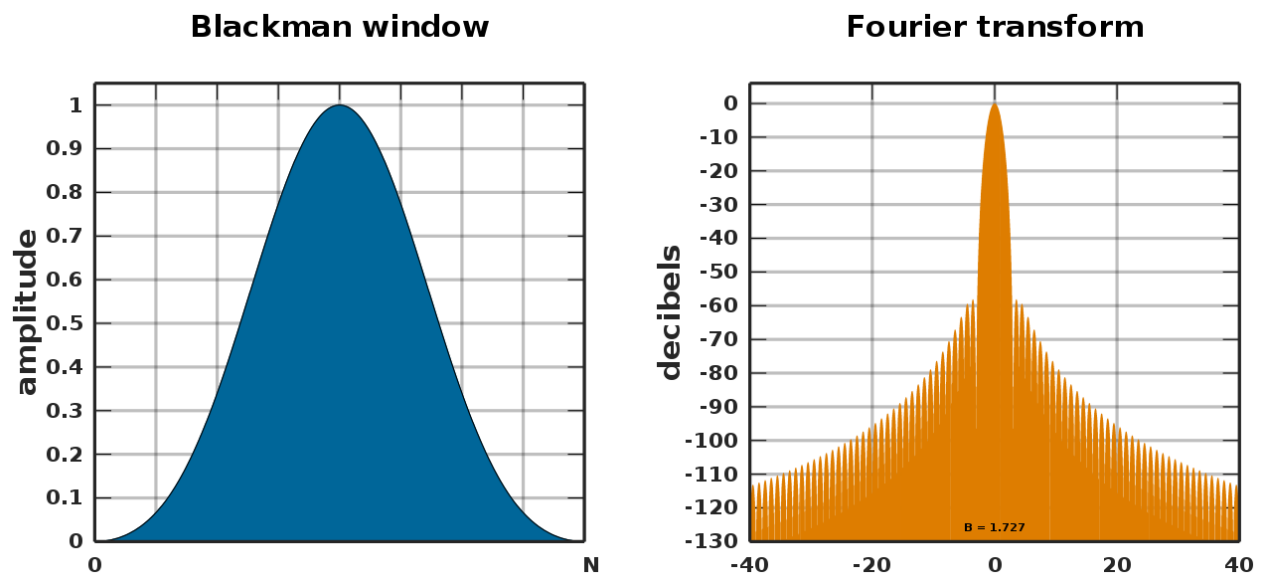


Figure 7.4. Blackman-Harris window function.

7.2.4 Flat Top Window

The flat top window is sinusoidal as well, but it actually crosses the zero line. This causes a much broader peak in the frequency domain, which is closer to the true amplitude of the signal than with other windows.

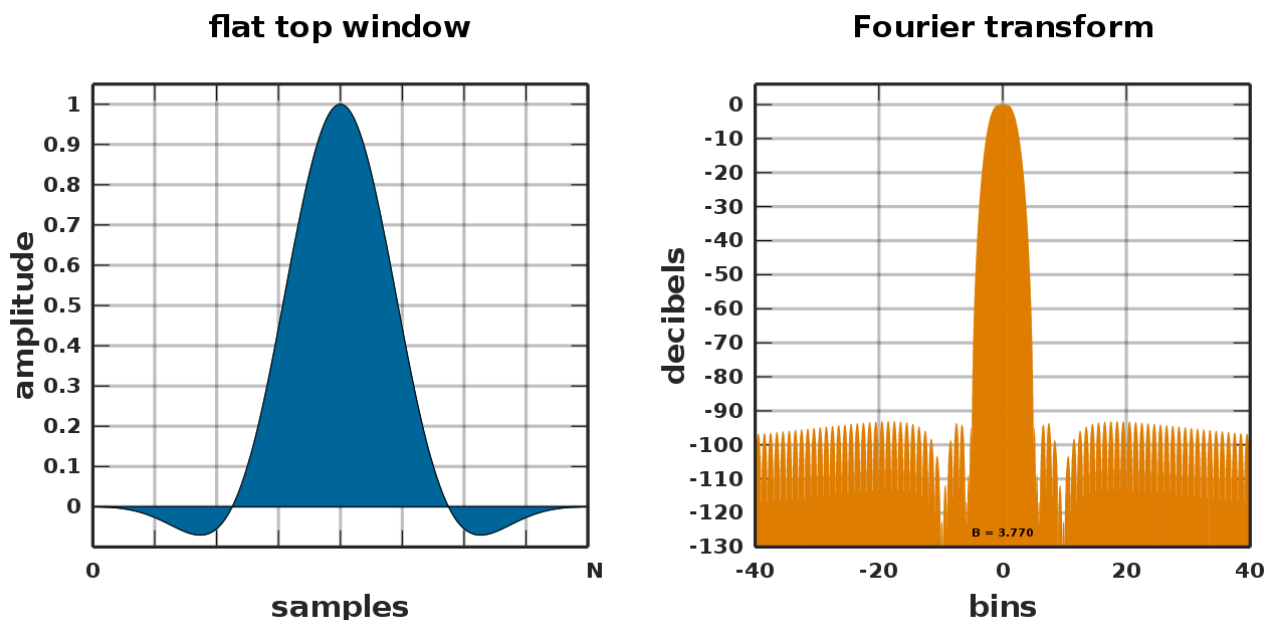


Figure 7.5. Flat top window function.

When application of the flat top window is compared to when no window (rectangular window or uniform) is applied, a broader peak is observed in the FFT. The Hann window also results in broadening of the peak, but to a lesser extent than the flat top. Discerning between two very close frequencies becomes very difficult due to the broadness of the peaks. When the intent is

to identify the presence of a signal component (a peak) at a specific frequency, it is best to apply a rectangular window to do the analysis. But, if the magnitude of the peak is important, the flat top window is clearly the best.

7.3 Averaging

Signal averaging is a signal processing technique applied in the time domain, intended to increase the strength of a signal relative to noise that is obscuring it. By averaging a set of replicate measurements, the signal-to-noise ratio (SNR) will be increased, ideally in proportion to the square root of the number of measurements. There are different types of signal averaging which will be discussed in the following sections.

7.3.1 Linear averaging

Each FFT spectrum collected during a measurement is added to one another and then divided by the number of additions. This helps in obtaining repeatable data and tends to average out random noise. This is the most commonly used averaging technique.

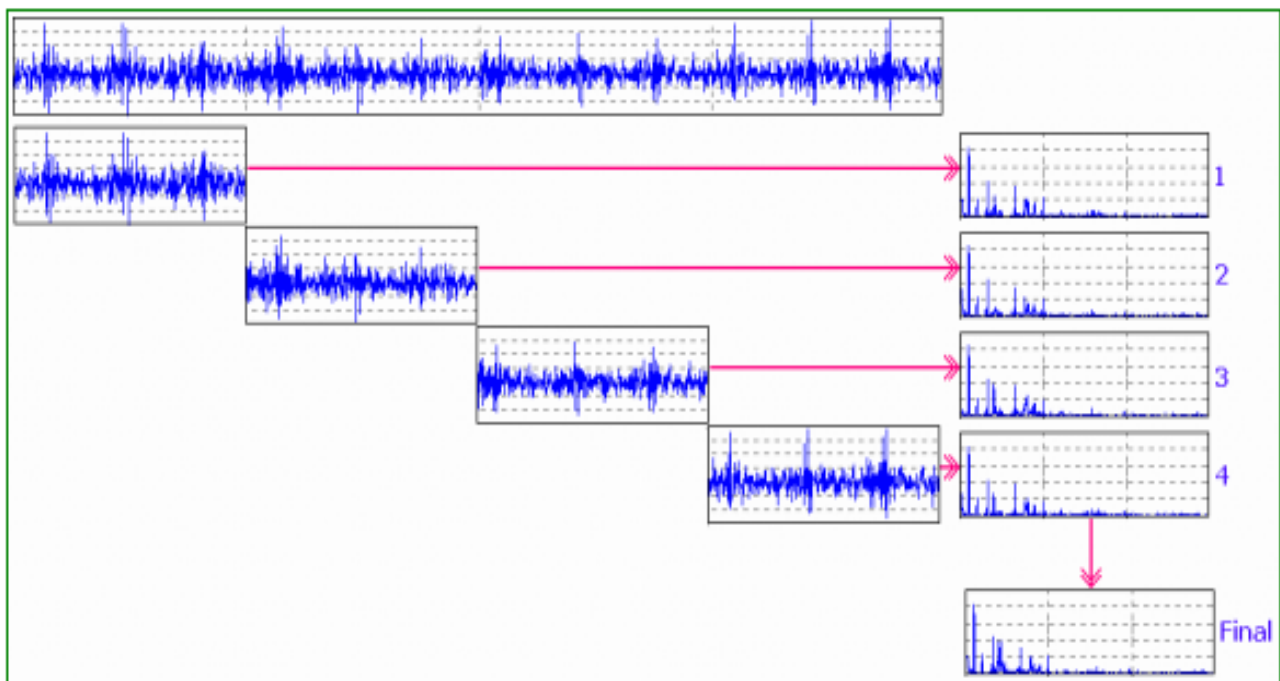


Figure 7.6. Linear averaging.

The start and end of the time record is windowed and pushed to zero. You could say that the vibration information at the start and end of the record is being wasted. So what if we overlapped the waveforms? Instead of using an entirely new time record for each average, we could use part of the previous record. This illustration shows just how much time is saved. We have not needed to use the data shaded gray. That means less time spent next to the machine.

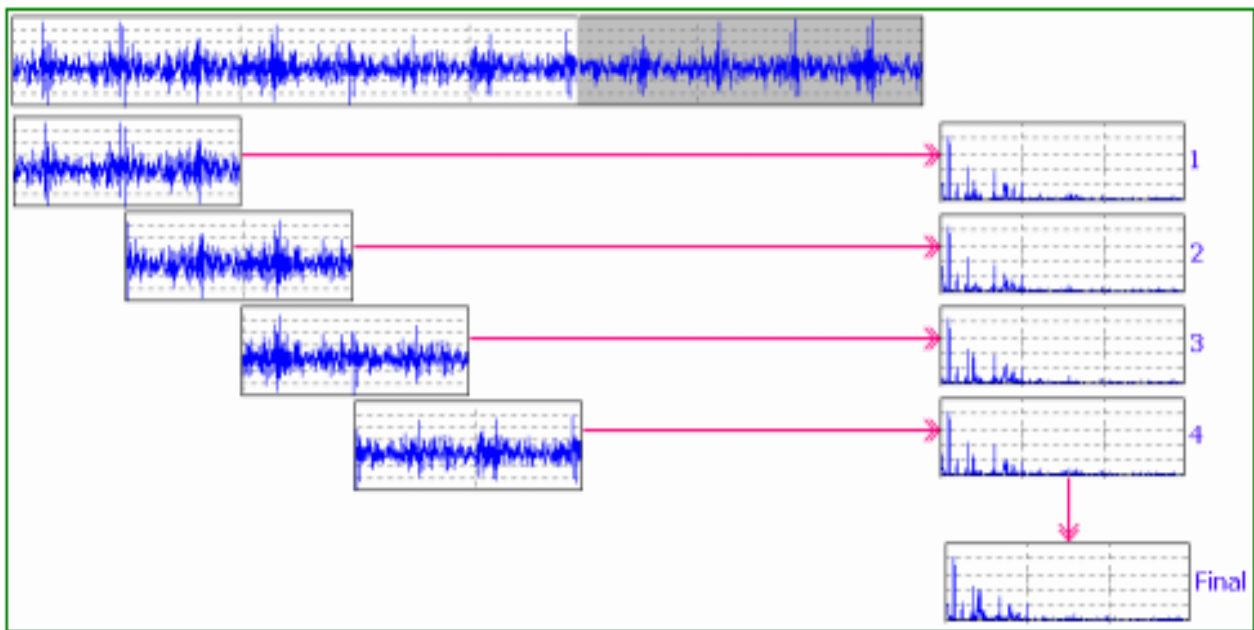


Figure 7.7. Overlap averaging.

7.3.2 Peak Hold Averaging

Peak hold develops an envelope of the highest spectral line amplitude measured for any average. This technique is used for viewing transients such as impact and run up / coast down tests.

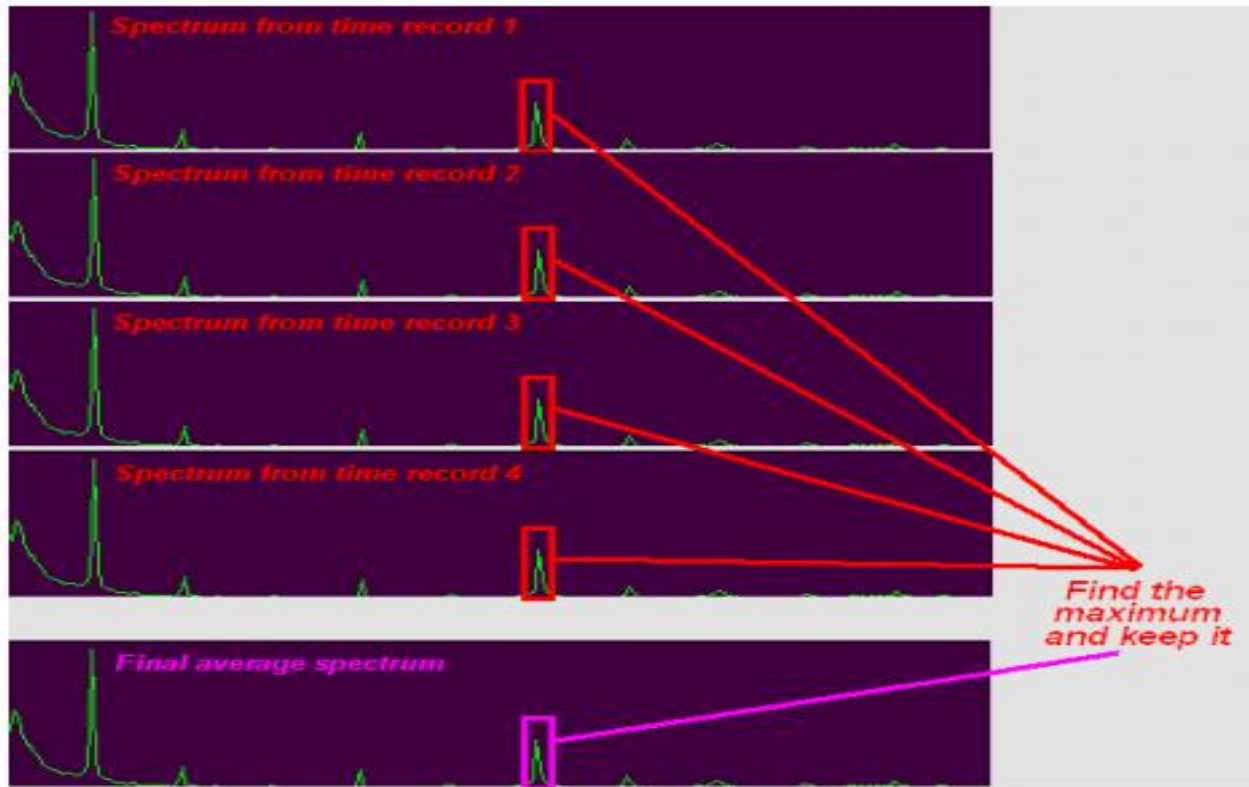


Figure 7.8. Peak hold averaging.

7.3.3 Time Synchronous Averaging

Time synchronous averaging (TSA) is a signal processing technique that extracts periodic waveforms from noisy data. The synchronizing signal is usually a pulse generated by a photocell or an electromagnetic pickup at a reference position on the shaft circumference. The TSA is well suited for gearbox analysis, where it allows the vibration signature of the gear under analysis to be separated from other gears and noise sources in the gearbox that are not synchronous with that gear.

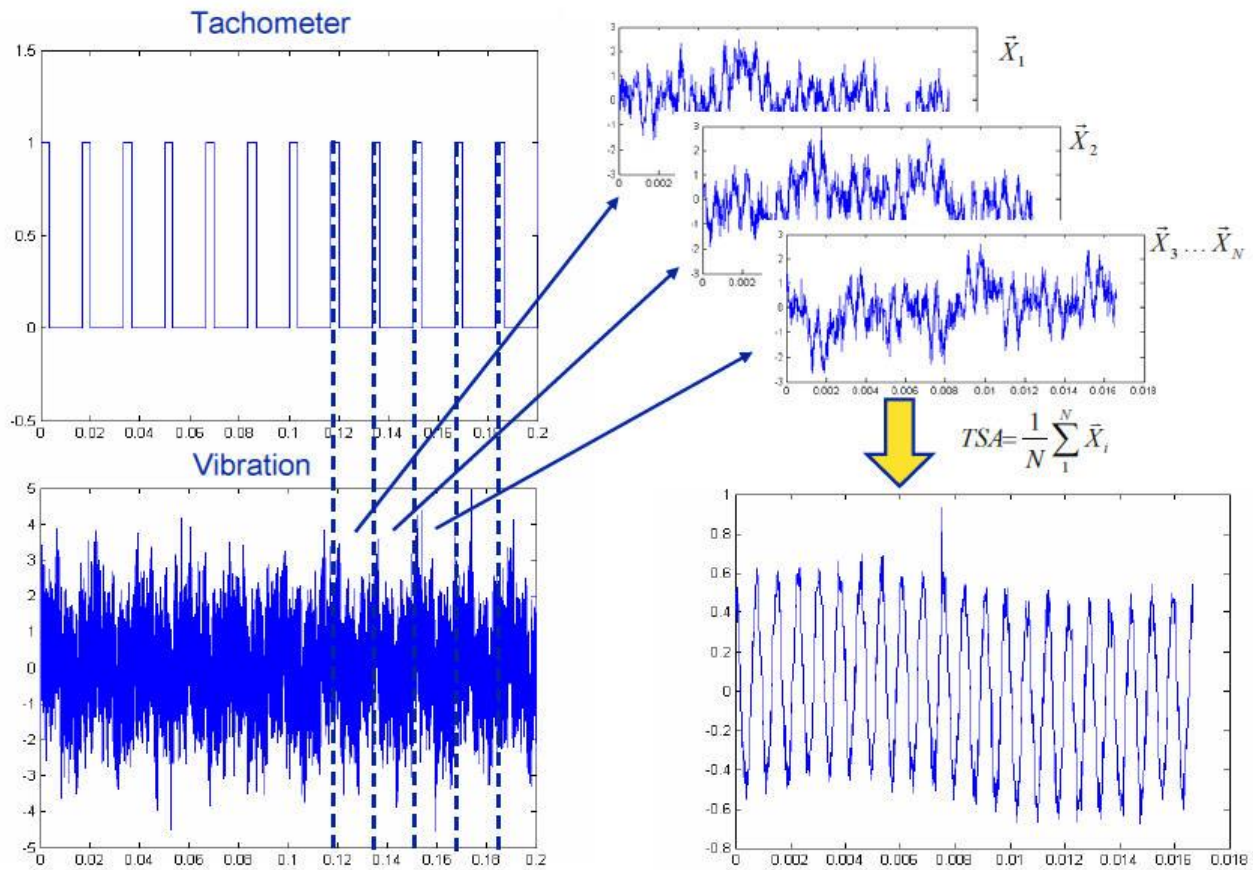


Figure 7.9. Peak hold averaging.

7.4 Anti-aliasing Filter

According to the Nyquist sampling theorem, the sampling rate should be at least twice the maximum frequency component of the signal of interest. In other words, the maximum frequency of the input signal should be less than or equal to half of the sampling rate.

How do you ensure that this is definitely the case in practice? Even if you are sure that the signal being measured has an upper limit on its frequency, pickup from stray signals (such as the power line frequency or from local radio stations) could contain frequencies higher than the Nyquist frequency. These frequencies may then alias into the appropriate frequency range and thus give you erroneous results.

To be sure that the frequency content of the input signal is limited, a low pass filter (a filter that passes low frequencies but attenuates the high frequencies) is added before the sampler and the A/D converter. This filter is an anti-alias filter because by attenuating the higher frequencies (greater than the Nyquist frequency), it prevents the aliasing components from being sampled. Because at this stage (before the sampler and the A/D converter) you are still in the analog world, the anti-aliasing filter is an analog filter.

An ideal anti-alias filter passes all the appropriate input frequencies (below f_1) and cuts off all the undesired frequencies (above f_1). However, such a filter is not physically realizable. In practice, filters look as shown in illustration (b) below. They pass all frequencies $< f_1$, and cut-off all frequencies $> f_2$. The region between f_1 and f_2 is known as the transition band, which contains a gradual attenuation of the input frequencies. Although you want to pass only signals with frequencies $< f_1$, those signals in the transition band could still cause aliasing. Therefore in practice, the sampling frequency should be greater than two times the highest frequency in the transition band. This turns out to be more than two times the maximum input frequency (f_1). That is one reason why you may see that the sampling rate is more than twice the maximum input frequency.

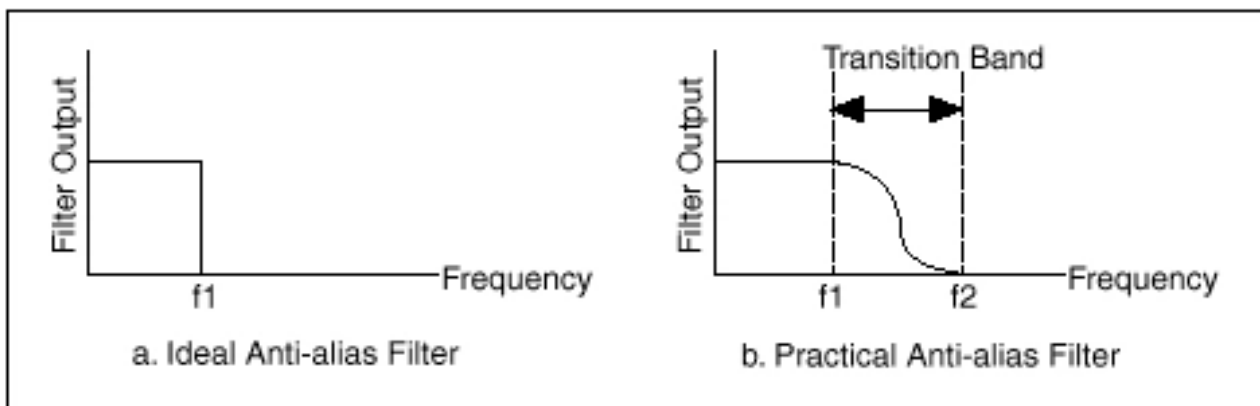


Figure 7.10. Difference between a) ideal and b) practical anti-alias filter.

8 Chapter (8) Signal Analysis

8.1 Introduction

It is highly desirable to be able to detect all types of faults during the operation of rotating machinery. Such faults range from vibrations at very low frequencies (sub synchronous components indicating looseness, oil whirl, faulty belt drive, etc.), to vibrations at very high frequencies (tooth-meshing frequencies, blade-passing frequencies, frequencies of structural resonance excited by faulty rolling-element bearings, etc.). Such detection should be applicable to all of machines in the plant which operate from very low to very high speed. This requires to select some kind of signals each one has exclusive characteristics to identify and detect certain types of faults. In this chapter, different vibration analysis techniques are presented such as time domain analysis, frequency domain analysis, phase analysis, envelope (demodulation) analysis, and Cepstrum analysis. Summary of vibration evaluation techniques is displayed in **Table 8.1**.

Table 8.1. Summary of vibration analysis techniques.

Technique	Application	Description	Fault/machine
Time domain analysis	Waveform visualization for identification of distortion	Amplitude vs. time	Rubbing, impact, clipping, cracked teeth
Frequency domain analysis	Direct frequencies, natural frequencies, sidebands, beats, sub harmonics, sum and difference frequencies	Amplitude vs. frequency	General Fault diagnosis
Phase analysis	Detection of developing cracks in shafts Balancing	Phase vs. frequency at varied machine location	General Fault diagnosis
Cepstrum analysis	Identification and separation of families of harmonics Identification and separation of families of sidebands	Inverse Fourier transform of the logarithmic power spectrum	Rolling element bearings, bladed machines, gearboxes
Envelope analysis	Signal analysis	Amplitude demodulation observation of a low frequency amplitude modulation happening at high frequency	Rolling element bearing, electrical machines, gear boxes

8.2 Time Domain Analysis

The time base plot is the most fundamental graphic presentation of machinery dynamic data. It shows how a single parameter (most often displacement, velocity, or acceleration, but also any other dynamic measurement) from a single transducer changes on a very short time scale, typically a fraction of a second. This is in contrast to trend plots, which display the value of a slowly

changing parameter (for example, axial position) over a much longer time scale, typically hours to months.

A time base plot represents a small slice of time in the vibration history of the machine. Usually, the amount of time involves only a few revolutions of the rotor. During this short length of time (about 17ms for one revolution of a 3600 rpm machine), the overall behavior of the machine is not likely to change significantly. However, unfiltered time base plots can clearly show a change in machine response if sudden events occur in the machine or if the machine is rapidly changing speed (such as an electric motor startup).

Time base plots have several important uses. They have the advantage in being able to clearly display the unprocessed output from a single transducer. This allows us to look for noise on the signal or to detect the presence of multiple frequency components. An important use of a time base plot is to identify the presence and timing of short term transient events.

Multiple time base plots can allow us to establish timing relationships at different axial locations along the machine train. Or, the time base plots from a pair of XY transducers can be used to determine the direction of precession of the rotor shaft.

The time base plot is a rectangular (Cartesian) plot of a parameter versus time (Figure 8.1). Time is on the horizontal axis, and elapsed time increases from left to right; events occurring later in time will be to the right of earlier events. Because of the time scales encountered in rotating machinery, the elapsed time is typically displayed in milliseconds (ms). The measured parameter, converted from voltage to engineering units, is on the vertical axis (in the figure, this is $1\mu\text{m}/\text{div}$).

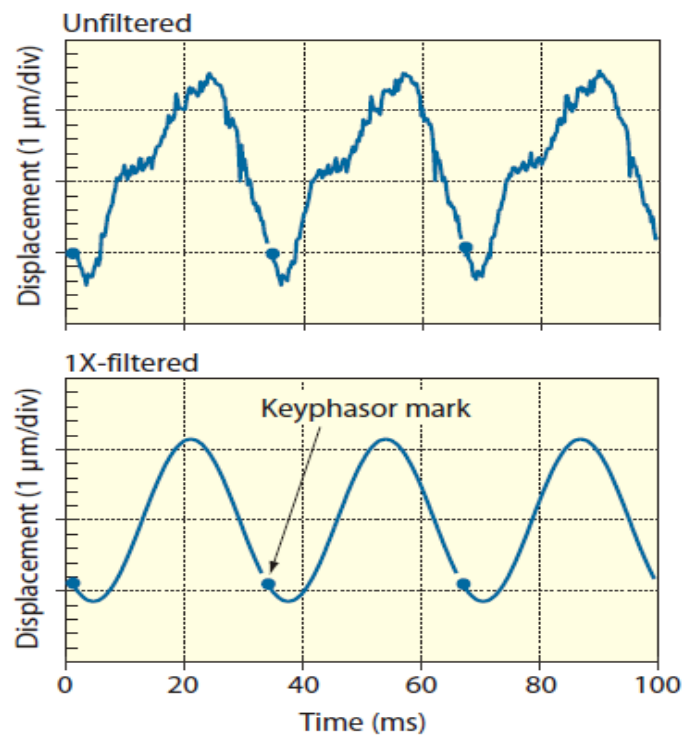


Figure 8.1. Unfiltered and filtered time base plot.

In unfiltered time base plots, digitally sampled signal voltages are first divided by the transducer scale factor to convert them to equivalent engineering units. Then, the converted values are plotted on the time base plot. The resulting waveform describes the instantaneous behavior of the measured parameter from one moment to the next.

Filtered time base plots are constructed from the amplitude and phase of vibration vectors. The plot is synthesized by computing a sine wave with the correct frequency, amplitude, and phase. This synthesis process assumes that conditions in the machine don't change significantly over the period of time represented by the synthesized waveform. This is usually, but not always, a correct assumption.

Compensation of time base plot is a process to remove unwanted signal content (noise) that is unrelated to the machine behavior that we want to observe. This noise, electrical and mechanical runout (glitch), bow, etc., can partially or completely obscure the dynamic information. Shaft scratches or other surface defects create a pattern of signal artifacts that repeats every revolution. It can be very useful to remove this noise to better reveal the important dynamic information. One type of compensation is slow roll compensation of vibration vectors. Most often, we wish to remove the effects of any 1X slow roll response that may be present in the signal so that we can see the 1X response due to unbalance.

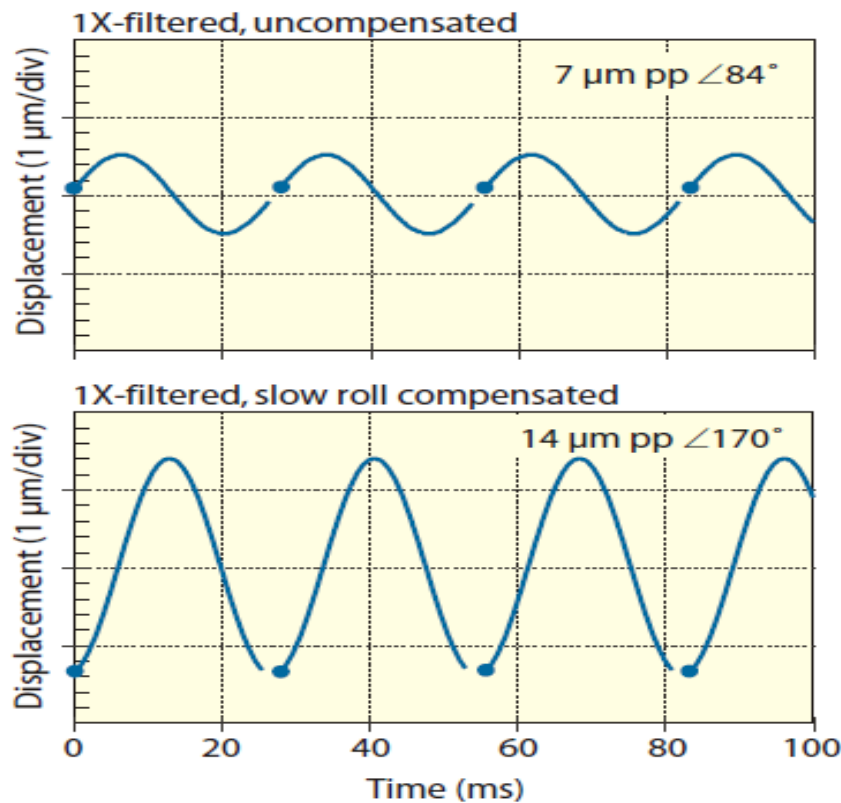


Figure 8.2. Slow roll compensation of filtered time base plot.

Slow roll compensation is primarily applied to eddy current displacement transducer data because these transducers have a significant output at slow roll speeds. At these speeds, output from velocity and acceleration (seismic) transducers is extremely low, and there is usually no measurable slow roll signal. For this reason, slow roll compensation is rarely, if ever, performed on seismic transducer data. Filtered time base plots can be slow roll compensated using a 1X, 2X, or nx slow roll vector. The slow roll vector is subtracted from the original vibration vector, and the new, compensated vibration vector is used to synthesize the filtered waveform. The end result is a filtered time base plot that is slow roll compensated.

8.3 Frequency Domain Analysis

Vibration vectors are filtered to multiples of running speed, 1X, 2X, 3X, etc., and provide us with information about the behavior of a machine at one of those particular frequencies. This information is presented to us as the amplitude and phase of the vibration at that frequency. However, machines can vibrate at many different frequencies simultaneously. These frequencies can be related or unrelated to running speed and include both sub-synchronous and super-synchronous frequencies. Since these frequencies are associated with the operating condition of the machine, the machinery diagnostician must have some way to determine the frequency content of a vibration signal in order to make an accurate diagnosis.

Vibration frequencies sometimes appear as a series of harmonics. The series consists of the lowest frequency in the series, called the fundamental, and a number of frequencies at integer multiples of the fundamental. In a typical series, the amplitude of higher order frequencies will decline rapidly. To avoid confusion, we will define a harmonic as any frequency that is an integer multiple of the fundamental. The first harmonic is the fundamental, the second harmonic has a frequency of twice the fundamental, the third harmonic has a frequency of three times the fundamental, etc. Often, the term harmonics will be used as a general term to indicate integer multiple frequencies that are above the fundamental. The fundamental vibration frequency of a series can be any vibration frequency. Often, the fundamental is $1X$, but it can also be any sub-synchronous or super-synchronous frequency. For example, a series could be based on a fundamental at $\frac{1}{2}X$ and include $1X$, $\frac{3}{2}X$, $2X$, $\frac{5}{2}X$, etc. In this series, the $\frac{1}{2}X$ is the fundamental or first harmonic, $1X (= 2 \times \frac{1}{2}X)$ is the second harmonic, $\frac{3}{2}X$ is the third harmonic, etc. Such a series can be generated by a $\frac{1}{2}X$ rub.

In the past few years, an important new tool, the full spectrum plot, has been developed that uses the signals from a pair of orthogonal, shaft relative, vibration transducers. The full spectrum plot compares to a conventional spectrum plot in the same way that the orbit plot compares to a time base plot. The full spectrum plot contains much more information than the spectrum plot, including vibration precession direction and orbit ellipticity. It is so important for machinery diagnostics that we now refer to the conventional spectrum plot as the half spectrum plot.

Spectrum plots are used to identify the frequency components that are present in complex vibration signals and to trend changes in the amplitude of frequency components. These frequencies include running speed, multiples of running speed, line frequency electrical noise, gear mesh frequencies, gear defect frequencies, rolling element bearing frequencies, and vane and blade pass frequencies. Rotor system natural frequencies that are excited will also show up on the spectrum plot. Sub-synchronous frequencies that are often associated with fluid-induced instability, compressor rotating stall, compressor surge, or rub, and super-synchronous frequencies that are often associated with rubs and shaft cracks can also be identified.

8.3.1 Half Spectrum Plot

Let's start with a machine operating at a constant speed. Because of a combination of unbalance and other effects, the machine is vibrating in a complicated way. The unfiltered vibration signal from a transducer on this machine will reveal all of the complexity of the machine vibration that falls within the bandwidth of the data collection system. A filtered signal from this transducer

will be a sine wave at the filter frequency with some amplitude and phase. Figure 8.3 shows a complex, time base vibration signal (red) and a series of sine waves (blue) that add up to produce the time base signal. Using the Fourier transform, the frequency, amplitude, and phase of these sine waves (called components) can be computed from a digital sample of the original time base signal. The phase for each signal is measured with respect to the trigger signal that starts the sampling process at time t_0 . The Fourier transform output is equivalent to the output of a series of band-pass filters that have been set to integer multiples of the lowest frequency signal, f_1 . (In practice, the lowest displayed frequency of a spectrum plot will usually be below the lowest measured vibration frequency.)

The amplitude scale can be either linear or logarithmic. Logarithmic scaling is useful when there is a need to compare signals with both very large and very small amplitudes. This scaling will clearly display all signal components and the noise floor. However, when applied to rotating machinery work, logarithmic scaling makes it more difficult to quickly discriminate between significant and insignificant vibration components. Linear scaling has the advantage of showing the most significant components; weak, insignificant, and low-level noise components are greatly reduced in scale or eliminated. Because of its advantage for machinery work, linear scaling will be used for the spectrum plots in this book.

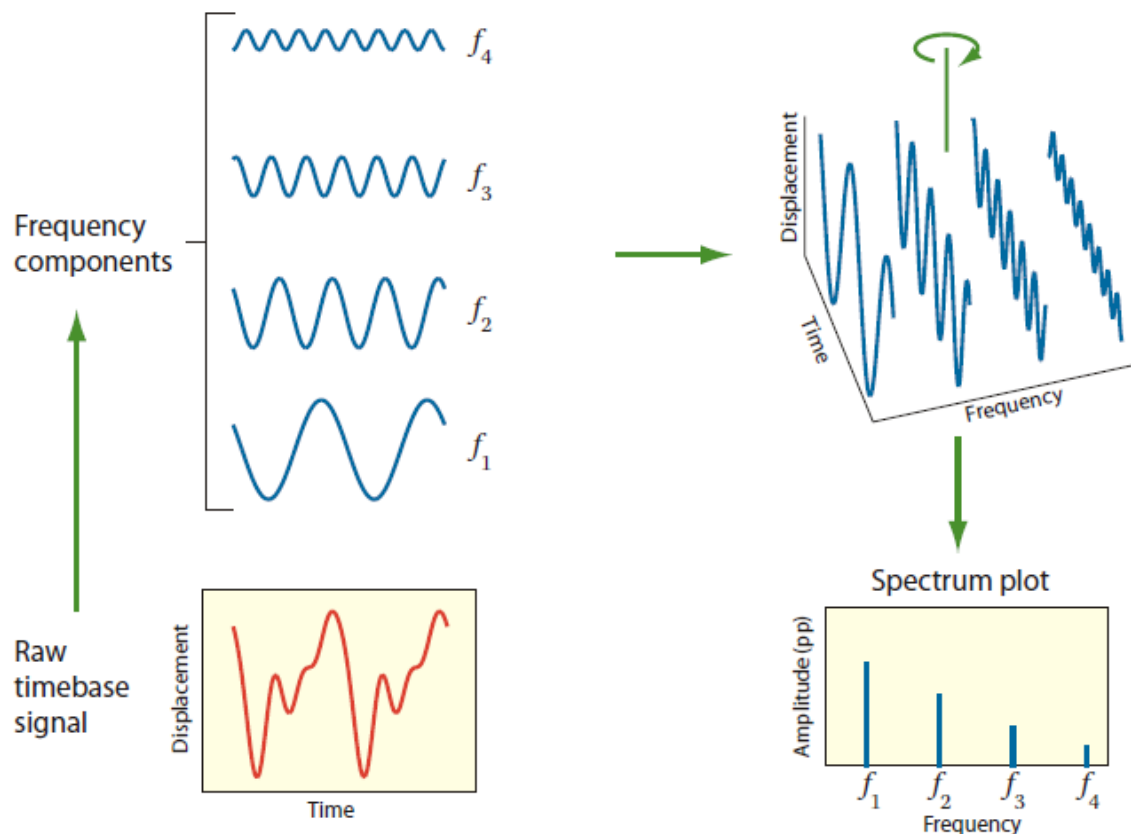


Figure 8.3. Time and frequency domain for a complex signal.

The frequency scale can be displayed in several frequency units. Most spectrum analyzers display the frequency in hertz (Hz). This is useful when comparing machine vibration frequencies to line frequencies, such as in induction motor or steam turbine generator diagnostics. Some software packages can display units of Hz, cpm, or orders of running speed. Cpm is a very convenient unit when working with machinery, because it is easy to compare a frequency in cpm to the running speed of the machine in rpm. Spectrum plots that are displayed in orders of running speed (1X, 2X, etc.) are also easy to interpret.

A digitally calculated spectrum consists of discrete frequency bins, or lines, of finite width. The width of these lines, the resolution of the spectrum, is an important consideration. The maximum resolution of a spectrum is determined by the ratio of the spectrum span (the range of displayed frequencies) to the number of spectrum lines that are displayed:

$$\text{Resolution} = \frac{\text{Span}}{\text{No of Lines}} \quad (3.1)$$

The spectrum plot is a collection of these lines, arranged side by side. The width of each line is equal to the resolution of the spectrum. For example, a 400 line spectrum with a span of zero to 200 Hz will have a resolution of

$$\text{Resolution} = \frac{200 \text{ Hz}}{400 \text{ Lines}} = 0.5 \text{ Hz/Line}$$

Thus, each frequency line will, ideally, represent only the spectral energy in a 0.5 Hz (30 cpm) wide band from 0.25 Hz below to 0.25 Hz above the center frequency of the line. Accuracy in the displayed amplitude and frequency of a spectrum line will depend on where the actual vibration frequency is with respect to the center frequency and which window function is used.

The limited resolution of spectrum plots means that there is always an uncertainty associated with any frequency we wish to measure. In the example above, a frequency actually located at, for example, 99.75 Hz, is displayed at 100 Hz. A spectrum plot with poor resolution will have a corresponding large uncertainty in the measured frequency. Even good resolution spectra may not be able to discriminate between vibration frequencies of exactly ½X and 0.49X, an important distinction for malfunction diagnosis. Higher resolution (zoomed) spectra can help, but orbits with Key phasor dots can sometimes be superior to spectrum plots for making this kind of discrimination.

Noise can be a problem in spectrum plots. The Fourier transform of a spike is a series of spectrum lines extending to very high frequency. Thus, anything that produces a sharp corner in the signal will produce a series of spectrum lines. Sharp corners can result from shaft rebound at a rub contact point or from an inadequate sampling frequency (causing a corner where a smooth transition really exists), among other things. Spikes or steps in the signal can originate from electrical noise

problems or from scratches on the shaft. Spectrum plots are calculated from *uncompensated* waveforms, which may contain significant slow roll or glitch content. In general, the appearance of spectrum lines as a series of harmonics should be viewed with caution. Use time base, orbit, or cascade plots (below) to validate the data.

8.3.2 Full Spectrum

The half spectrum is a spectrum of a single time base waveform. The full spectrum is the spectrum of an orbit. It is derived from the waveforms from two, orthogonal, shaft relative transducers, combined with knowledge of the direction of rotation. The information from the two transducers provides timing (phase) information that allows the full spectrum algorithm to determine the direction of precession at each frequency. Because the timing information is critical, the two waveforms must be sampled at the same time.

The full spectrum is calculated by performing an FFT on each transducer waveform. The results are then subjected to another transform that converts the data into two new spectra that represent frequencies of precession, one spectrum for X to Y precession and one for Y to X precession. The last step uses the direction of rotation information to determine which of the spectra represents forward and which represents reverse precession frequencies. When this process is completed, the two spectra are combined into a single plot, the full spectrum plot (Figure 8.4).

Figure 8.5 shows the relationships among time base waveforms, half spectra, the orbit, and the full spectrum. The Y and X time base waveforms and their half spectra are at the top. The two waveforms combine to produce the orbit at bottom left. The data used to generate the half spectra are further processed to produce the full spectrum at bottom right. Note that you cannot generate the full spectrum by combining the two half spectra. In the full spectrum plot, the spectrum of forward precession frequencies is on the positive horizontal axis and the spectrum of reverse precession frequencies is on the negative horizontal axis. Thus, for each frequency, there are two possible spectrum lines, one forward, and one reverse. The relative length of the spectrum lines for each frequency indicates the shape and direction of precession of the orbit filtered to that frequency.

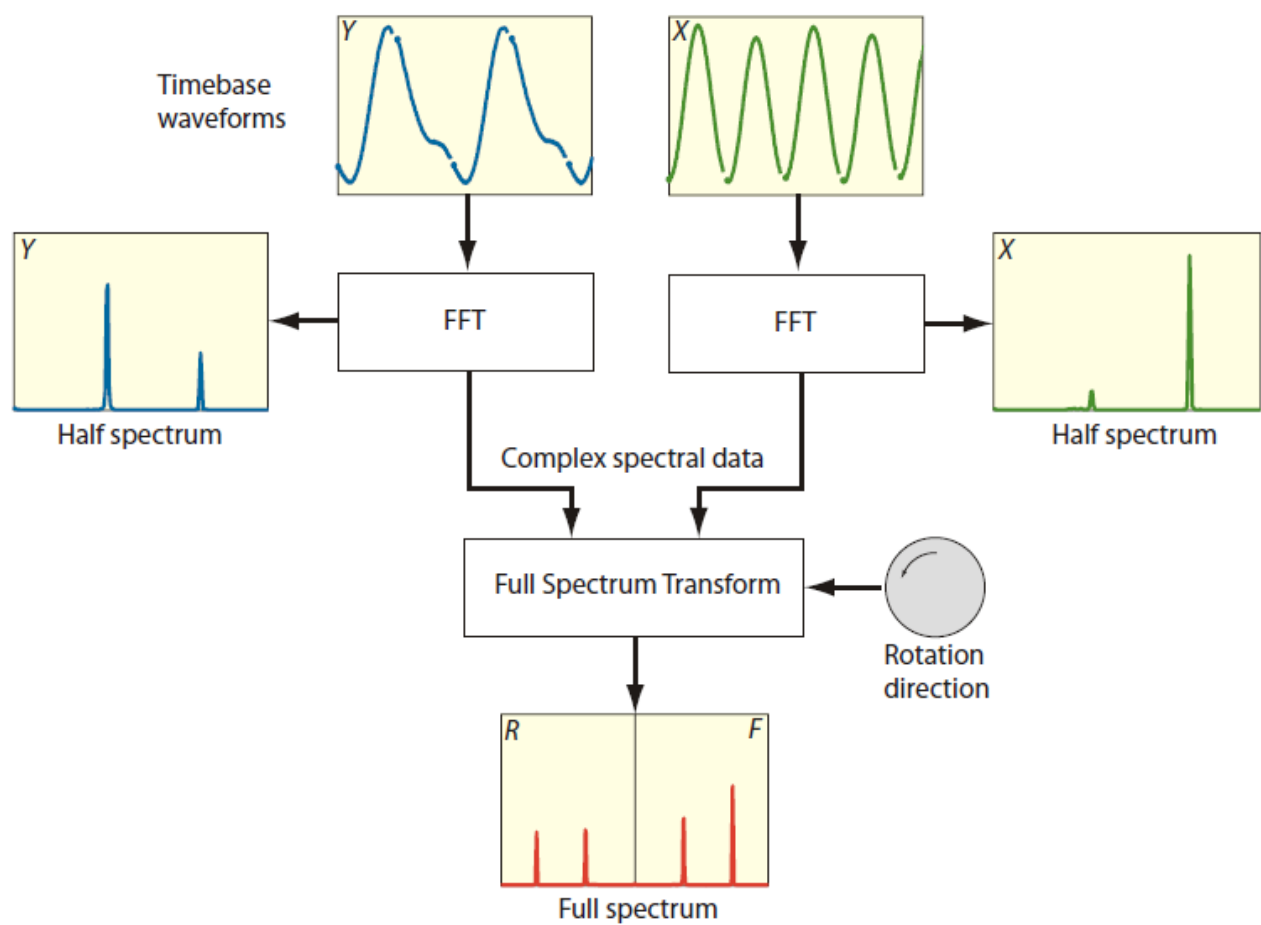


Figure 8.4. Calculation of the full spectrum.

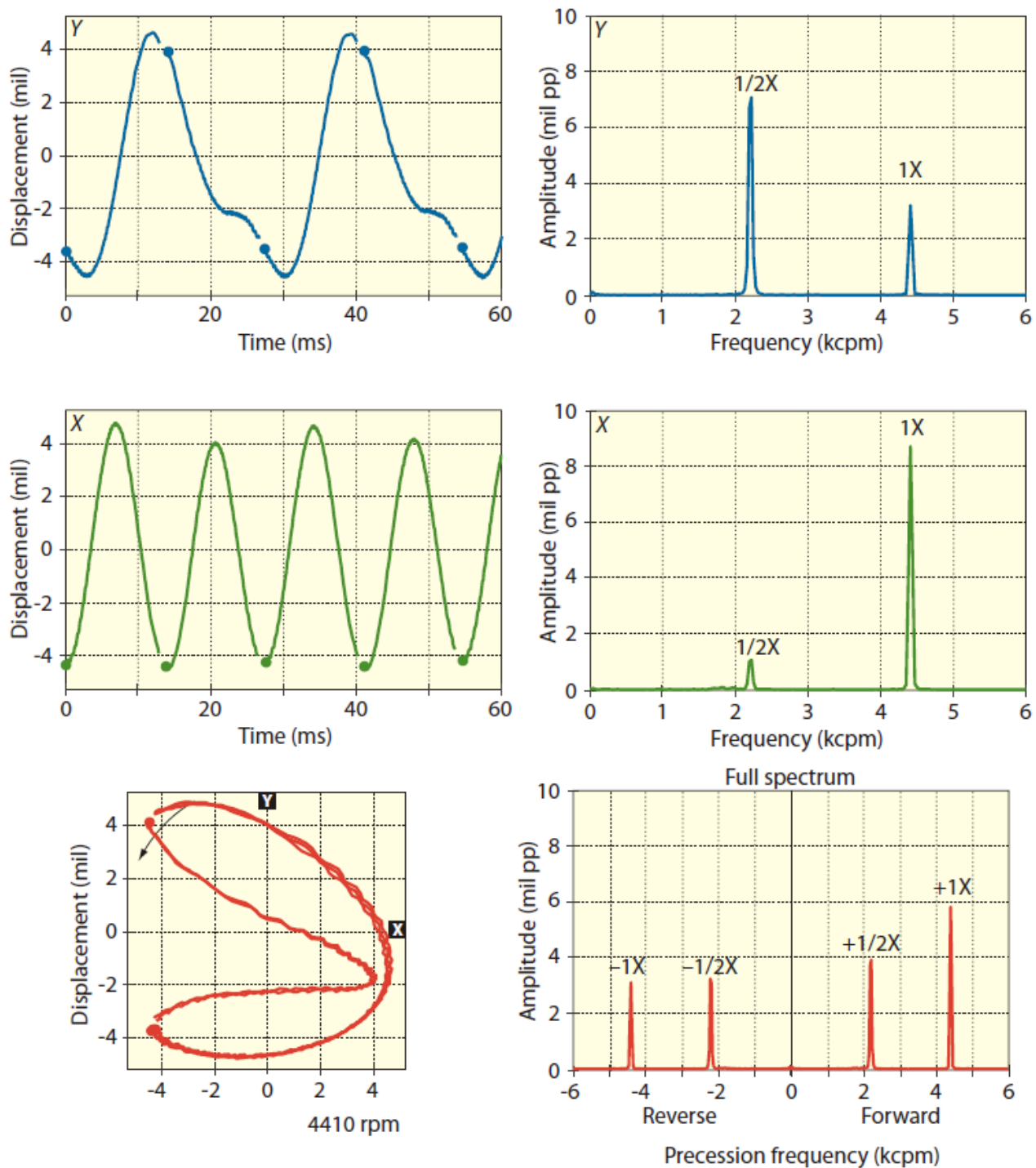


Figure 8.5. Time base waveforms and their half spectra, the orbit and its full spectrum.

8.4 Phase Analysis

We have seen that vibration can be measured with two parameters, frequency and amplitude. These are good and useful measurements. However, vibration never occurs in isolation; there is usually a root, or fundamental, cause of vibration in a machine. The machinery specialist needs to identify the root cause of any vibration problem, and it is often difficult to do this on the basis of frequency and amplitude alone. More information is needed. One piece of information that can be very useful is the timing difference, or phase, between events. If we know the timing between a root cause and its effect, we can use our knowledge of rotor behavior to deduce the possible root causes of what we can measure. This gives us a powerful tool for the diagnosis of rotating machinery.

Phase is another name for the relative timing between two events in different signals. For example, in Figure 8.6, two similar vibration signals are shown. These two signals reach the positive peaks at different times. This timing difference is referred to as a phase difference. In machinery applications, the phase difference of equivalent events on different vibration signals is called relative phase. Absolute phase compares the timing of an event on the vibration waveform to a different type of reference signal, produced by a once-per-turn marker on a shaft. Both methods are commonly used, and both have their particular applications. Even though phase compares the timing of events, it is expressed in units of degrees of the vibration cycle. In machinery instrumentation, the timing difference is usually expressed as a fraction of the complete 360° vibration cycle. This is a positive number with increasing time delay and is called positive phase lag. For example, in Figure 8.6, the green signal lags the blue signal by 60° .

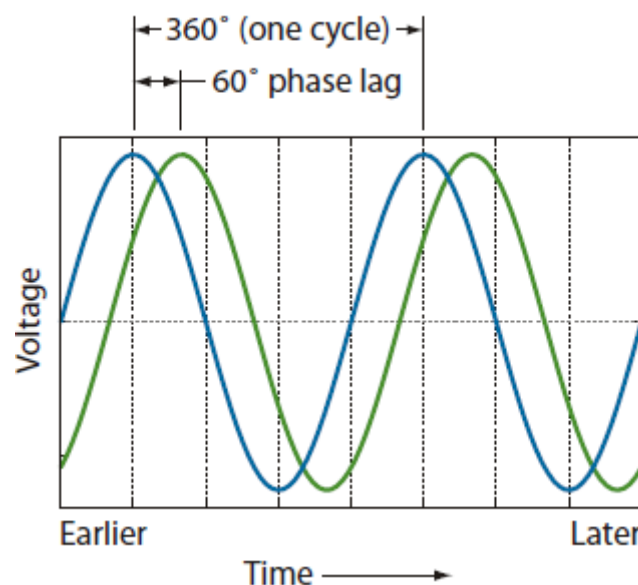


Figure 8.6. Relative phase difference between signals.

8.4.1 Why Is Phase Important?

In an automobile engine, there may be spark and fuel, but if the timing is off, the engine may not run. In rotor behavior, the *timing* (phase) is just as important as vibration amplitude and frequency for effective diagnosis of machine behavior. It is also necessary for efficient balancing of machinery. Balancing requires us to know the angular location of the unbalance (the *heavy spot*). We deduce this location by using phase measurement of the vibration response of the machine combined with our knowledge of rotor behavior. Without phase information, we would have to perform many more runs to calculate an initial balance solution. When rotors vibrate, they deflect away from the machine centerline. When the vibration is 1X, the point on the shaft which is on the outside of the deflected shaft is called the high spot (Figure 8.7). The timing of the rotor high spot passage under a transducer (the positive peak of displacement) provides important information about rotor behavior. It can be compared to the timing at different axial positions in the same machine. The amplitude and phase information can be combined to produce a picture of the deflection shape, or mode shape, of the rotor at running speed. Phase can also be used to document mode shape of the casing or structure in a similar manner.

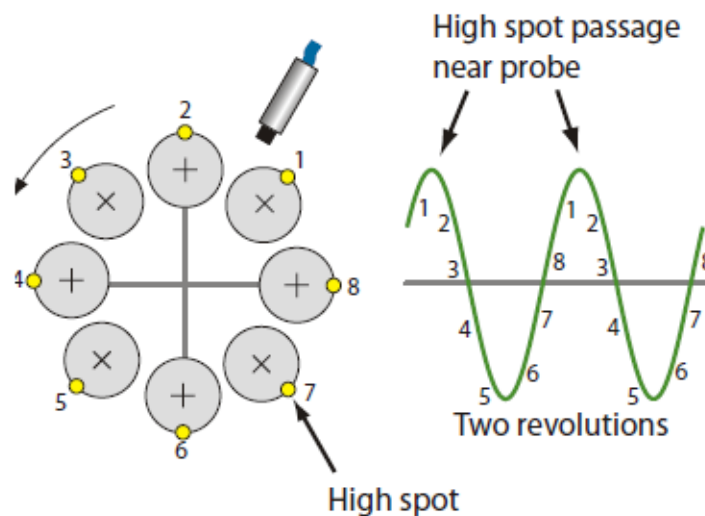


Figure 8.7. The high spot of vibration on a rotor shaft.

The vibration at the source of a machine problem always happens earliest in time. As vibration propagates away from the source location, it experiences a time delay (phase lag). Typically, the farther from the source, the longer the phase lag. Thus, by measuring the relative phase between different axial positions in a machine and looking for the earliest signal, we can sometimes determine the location closest to the source of the problem. A healthy machine should operate and vibrate with a repeatable pattern day after day. Once the baseline vibration characteristics of a machine are known (including changes with load or other measurable factors),

changes in vibration that break the pattern indicate that something may be wrong with the machine. Changes in phase are just as important as changes in vibration amplitude or frequency, and one may change independently of the others. Changes in phase, amplitude, or frequency can warn that something is happening to the machine. For all of these reasons, phase is an important tool in machinery diagnostics.

8.4.2 The Keyphasor Event

The most common vibration in rotor systems is associated with rotor unbalance. The unbalance acts as a one-cycle-per-revolution rotating force on the rotor. This 1X forcing produces a 1X, or synchronous, vibration response in the machine. Because unbalance is so common, it is desirable to have a fixed, timing reference signal so that we can make phase measurements.

An eddy current displacement transducer looking at a keyway or key serves this purpose perfectly. Such a transducer is called a Keyphasor transducer. While a Keyphasor transducer is usually an eddy current transducer, it can be any type of transducer, as long as it provides a repeatable, once-per-turn reference signal. Figure 8.8 shows a Keyphasor transducer observing a keyway and the resulting signal. As the leading edge of the notch passes by the transducer, the observed distance will increase suddenly, and the transducer signal voltage will abruptly become more negative. When the trailing edge of the notch passes by the probe, the transducer signal voltage will return to normal. Other shaft configurations for generating the Keyphasor signal are possible. This pattern of voltage changes occurs once every revolution of the rotor and is referred to as the Keyphasor event. The Keyphasor event can be thought of as a timing signal. When the event occurs, a timing clock is set to zero and started. This event is used to measure the elapsed time between the Keyphasor event and an event on another signal. Each time the rotor completes a revolution, the Keyphasor event occurs again, resetting the imaginary timer. This once-per-turn event is the timing reference used by instrumentation to measure the absolute phase of vibration signals at 1X and integer multiples (2X, 3X, ...). It is also used to measure rotor speed and other important characteristics of the dynamic response of the rotor.

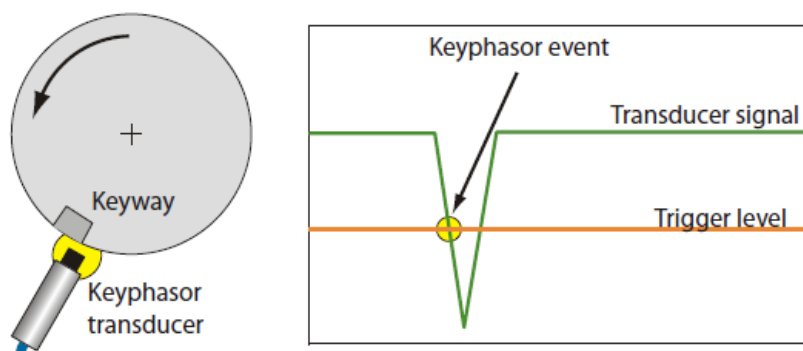


Figure 8.8. The Keyphasor event.

8.4.3 Phase Measurement

Phase is used to compare the timing of two events in different signals. If an event occurs later in time than a reference event, then it lags the reference event. Similarly, if an event occurs before the reference event, then it leads the reference event. In order to make meaningful phase measurements, the signals being used must consist of a single primary frequency component or, in the case of the Keyphasor signal, one clearly identifiable reference event. For this reason, signals are usually filtered to the frequency of interest before making the measurement, although unfiltered signals can be used if they are dominated by one frequency.

The convention used in most vibration measurement instrumentation is to measure phase lag with a positive number, sometimes called positive phase lag. For example, take the vibration signal shown in Figure 8.9. The spike in the Keyphasor signal provides the timing reference. The event of interest is the first positive peak of the vibration signal that occurs after (to the right of) the Keyphasor event. Because the first positive peak occurs after the Keyphasor event, the time delay between the two events is referred to as the phase lag.

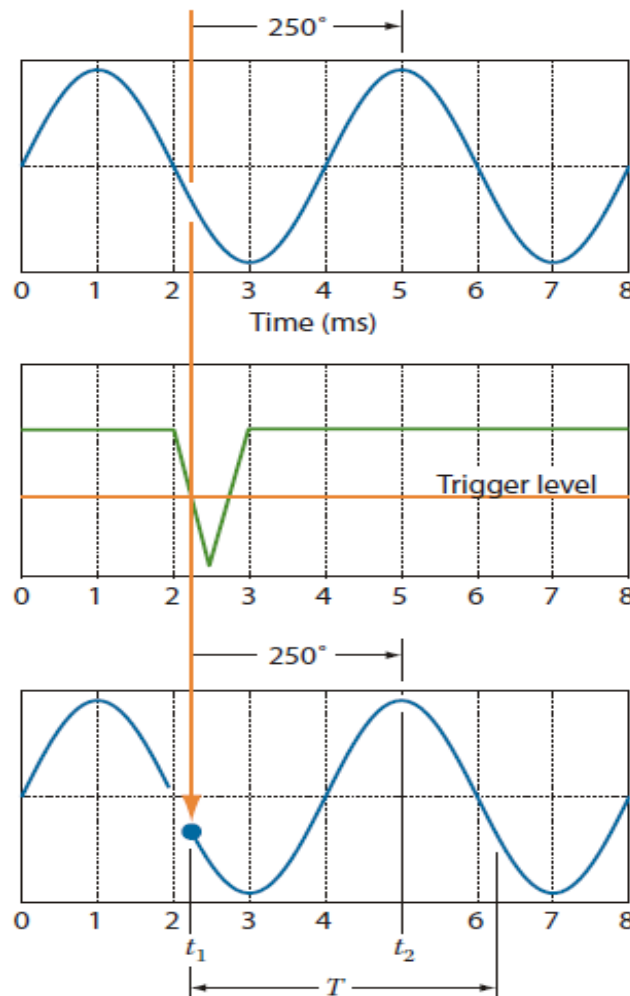


Figure 8.9. The Keyphasor event and absolute phase.

The numeric value of the phase lag is found by taking the measured time delay and comparing it to the time for one cycle of vibration:

$$\Phi = \left[\frac{t_2 - t_1}{T} \frac{ms}{ms/cycle} \right] \left[360 \frac{deg}{cycle} \right] \quad (3.2)$$

where Φ is the phase lag in degrees, t_1 is the time in milliseconds at which the reference event takes place, t_2 is the time at which the event of interest takes place, and T is the time in milliseconds for one cycle of vibration. Phase measurements are usually reduced to numbers between 0° and 360° . This is automatic in many instrumentation systems. For example, a calculated phase measurement of 395° would be reduced to $395^\circ - 360^\circ = 35^\circ$. The measurement of phase using this instrumentation convention (positive phase lag) differs from the measurement of phase using a standard mathematical convention. In the mathematical system, phase lag would be a negative number, and phase lead would be positive.

8.4.4 Absolute Phase

Absolute phase is the phase angle measured from the Keyphasor event to the *first* positive peak of the waveform. For 1X vibration in a circular orbit, this peak occurs when the rotor *high spot* is nearest the vibration transducer (the high spot and the orbit will be discussed in more detail in a later chapter). Thus, for 1X vibration, the absolute phase is sometimes said to represent the phase of the rotor high spot. The absolute phase is found by measuring the time between the Keyphasor event and the time of the first positive peak ($t_2 - t_1$), measuring the time for one complete cycle of vibration, T , and using Equation 3.2. In Figure 8.9, t_1 is measured as about 2.25 ms, and t_2 is about 5.0 ms. T is $6.25 \text{ ms} - 2.25 \text{ ms} = 4.00 \text{ ms}$. Applying Equation 3.2, the phase, Φ , is

$$\Phi = \left[\frac{5 - 2.25}{6.25 - 2.25} \frac{ms}{ms/cycle} \right] \left[360 \frac{deg}{cycle} \right]$$

$$\Phi = 250^\circ \text{ phase lag}$$

Absolute phase can only be measured on vibration signals with a frequency that is a harmonic (integer) multiple of running speed, such as 1X, 2X, 3X, etc. Attempting to measure absolute phase for frequencies that are not integer multiples, or that are sub-synchronous, can lead to ambiguity. For example, in Figure 8.10, a vibration signal is shown that has a frequency slightly higher than 1X. Measuring absolute phase from successive Keyphasor marks results in different phase values. For signals with frequencies that are not integer multiples of running speed, absolute phase measurements are meaningless. Note that, if the instrument cannot combine the Keyphasor signal with the vibration signal, it is still possible to measure absolute phase by displaying both

waveforms. The top two plots in Figure 8.9 can be used to measure the absolute phase; the basic definition of absolute phase is unchanged.

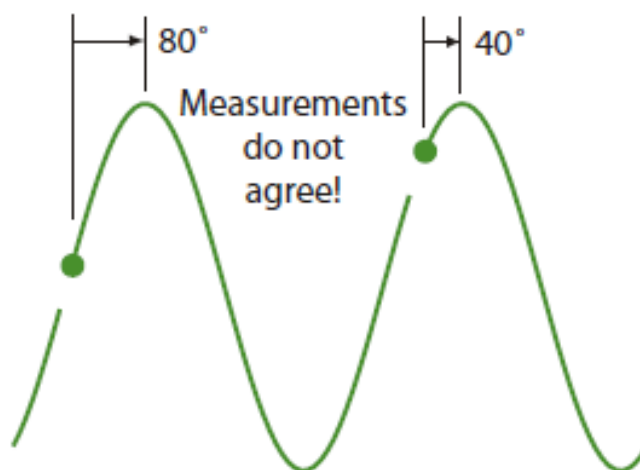


Figure 8.10. Absolute phase and nonharmonic signals.

8.4.5 Relative Phase

Relative phase is the time delay between equivalent events (peaks, zero crossings, etc.) on two separate signals, and doesn't use the Keyphasor event. Figure 8.11 shows an example of a relative phase measurement. The two vibration signals have been filtered to the same frequency and represent the displacement vibration at different axial positions on a machine (but the same transducer orientation). Select one of the signals as a reference, and then select a convenient measurement point on it. Here, the positive peak is used. The equivalent point is located on the second signal, and the relative phase is calculated using Equation 3.2. The result is, "Signal B lags signal A by 130° ." It is important to note *which* signal leads (or lags) the other and by *how much*, or the description is not correct or complete. Normally, the reference signal is stated last. In this case, signal A was the reference signal, and signal B was measured relative to the reference. For another example, suppose a relative phase measurement is made on the signals in Figure 8.12. Initially the "A" signal is selected as a reference. The relative phase is measured as "B lags A by 270° ." The result can be restated as "B leads A by 90° ." Either is correct. In order to make a relative phase measurement, three conditions should be met. First, the two signals *must* have the same frequency, but it can be any frequency, not just integer multiples of running speed. If we attempt to measure the relative phase between signals with different frequencies, we will obtain different results depending on where we make the measurement, and the result will be meaningless. Second, relative phase measurements are most often applied to vibration signals with the same units of measurement. If we want to compare the relative phase of the rotor vibration at different axial

positions, we would typically compare two displacement, two velocity, or two acceleration signals. (Since the phase relationship of acceleration and velocity, and velocity and displacement are always 90° , we could compare these signals by applying the right correction.) Third, vibration transducers should have the same radial orientation if they are in different axial planes. Relative phase measurements *can* be made between transducers with different orientations, as long as they are in the same plane, to determine the direction of precession of a rotor.

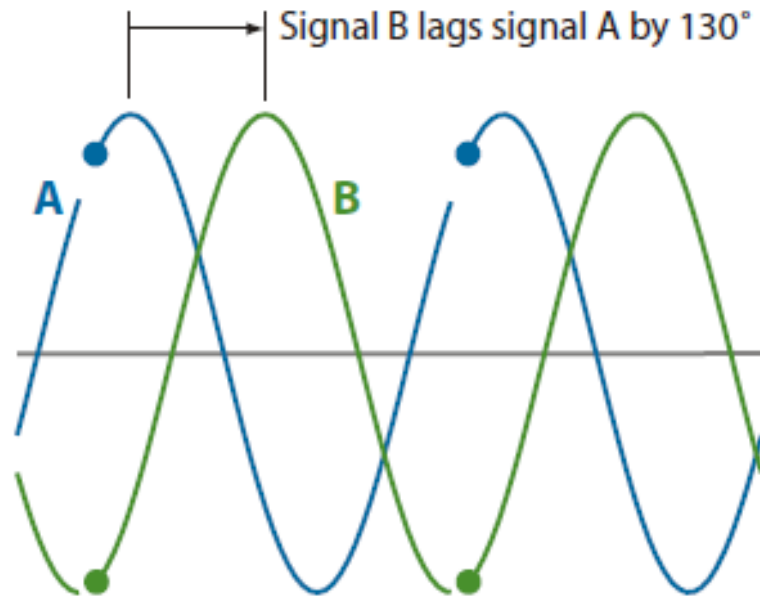


Figure 8.11. Relative phase between equivalent points on signals.

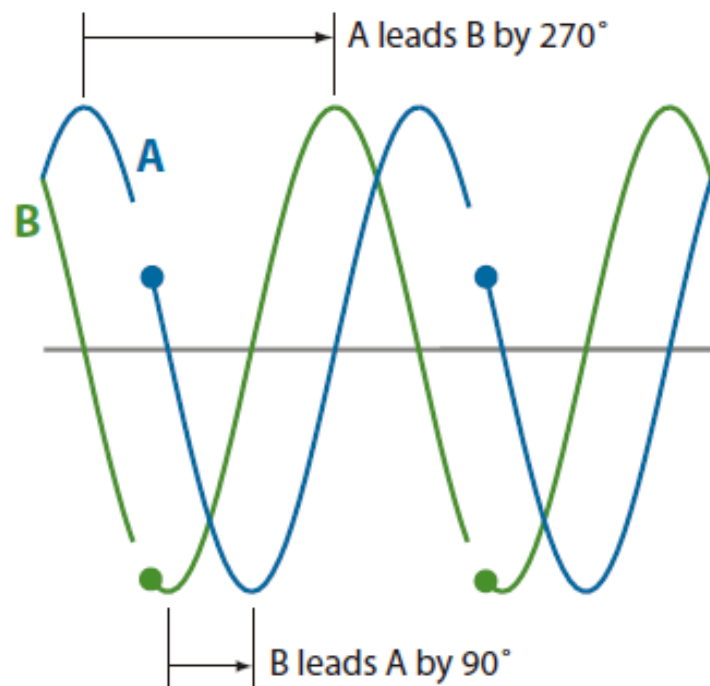


Figure 8.12. Equivalent measurements of relative phase.

8.5 Cepstrum Analysis

In Fourier analysis, the cepstrum is the result of computing the inverse Fourier transform (IFT) of the logarithm of the estimated signal spectrum (see). The method is a tool for investigating periodic structures in frequency spectra. The term cepstrum looks like a spelling error and is often corrected by proof readers. It is not an error; the term has been chosen to give the function a scent of “spectrum” because it is advantageous to consider the cepstrum as the spectrum of (the logarithm of) the autospectrum. To support this perception, all terms related to the cepstrum are warped versions of the corresponding spectrum terms:

- Spectrum: cepstrum
- Frequency: quefrequency
- Harmonics: rahmonics
- Low-pass filter: short-pass lifter
- High-pass filter: long-pass lifter

The cepstrum is useful for all periodic manifests in a response spectrum from a rotating machine like amplitude modulation of gear-tooth meshing frequency due to load variation, gear eccentricity or general wear of gears and, of course, for identifying local faults.

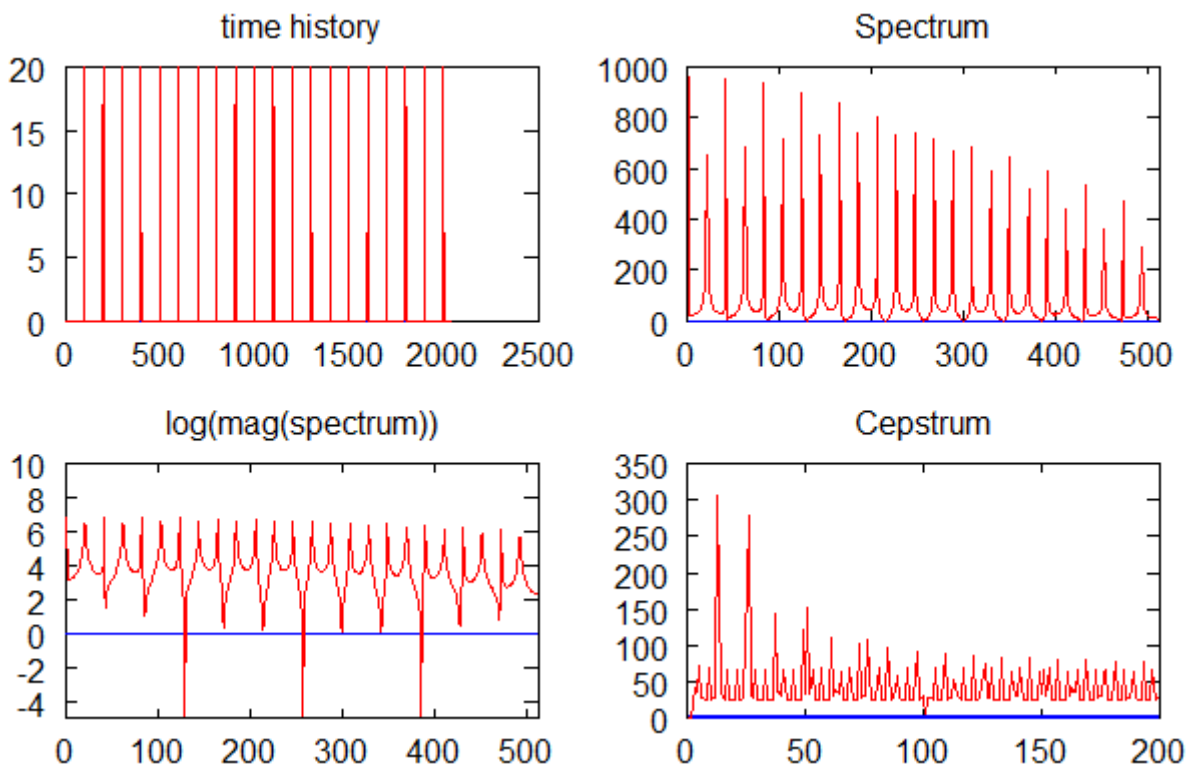


Figure 8.13. Different between spectrum and cepstrum.

Example: Local Fault in a 5:3 Gear: - The input gear spins with 3000 RPM = 50 Hz, which means that a fault on a tooth will generate a repeated impact with the repetition time $t = 1/50 \text{ Hz} = 20 \text{ ms}$. The output gear spins with $3000 \times 3/5 \text{ RPM} = 1800 \text{ RPM} = 30 \text{ Hz}$. A faulty tooth on this gear will give a repeated impact every $t = 1/30 \text{ Hz} = 33.3 \text{ ms}$. The noise from the gear suggests a local fault, and the vibration was measured on the housing of the gear. The cepstrum of the vibration spectrum will reveal a possible periodicity in the spectrum. The green curve in Figure 8.14 is the cepstrum and it reveals two rahmonic families indicating two local faults with a different fault rate:

- A rahmonic family with the fundamental rahmonic = 20 ms (50 Hz) which means a local fault on the input gear (a faulty tooth) and a rahmonic family with the fundamental rahmonic = 30 ms (33.3 Hz), indicating a faulty tooth on the output gear. The reason for the rahmonic component at quefreny 60 ms being higher than the other components is that both local faults contribute to this component: The 3rd rahmonic of the fault on the input gear and the 2nd rahmonic of the fault on the output gear.

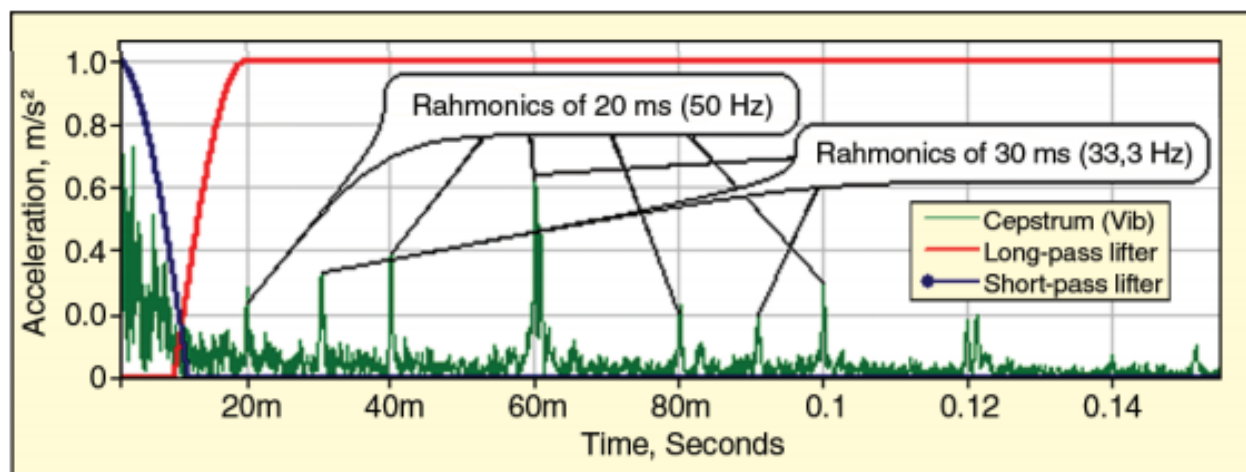


Figure 8.14. Vibration Cepstrum.

8.6 Envelope Analysis

Envelope Detection or Amplitude Demodulation is the technique of extracting the modulating signal from an amplitude-modulated signal. The result is the time history of the modulating signal (see Figure 8.15). This signal may be studied/interpreted as it is in the time domain or it may be subjected to a subsequent frequency analysis. Envelope Analysis is the FFT frequency spectrum of the modulating signal.

Envelope Analysis can be used for diagnostics/investigation of machinery where faults have an amplitude modulating effect on the characteristic frequencies of the machinery. Examples include faults in gearboxes, turbines and induction motors. Envelope Analysis is also an excellent tool for diagnostics of local faults like cracks in rolling element bearings.

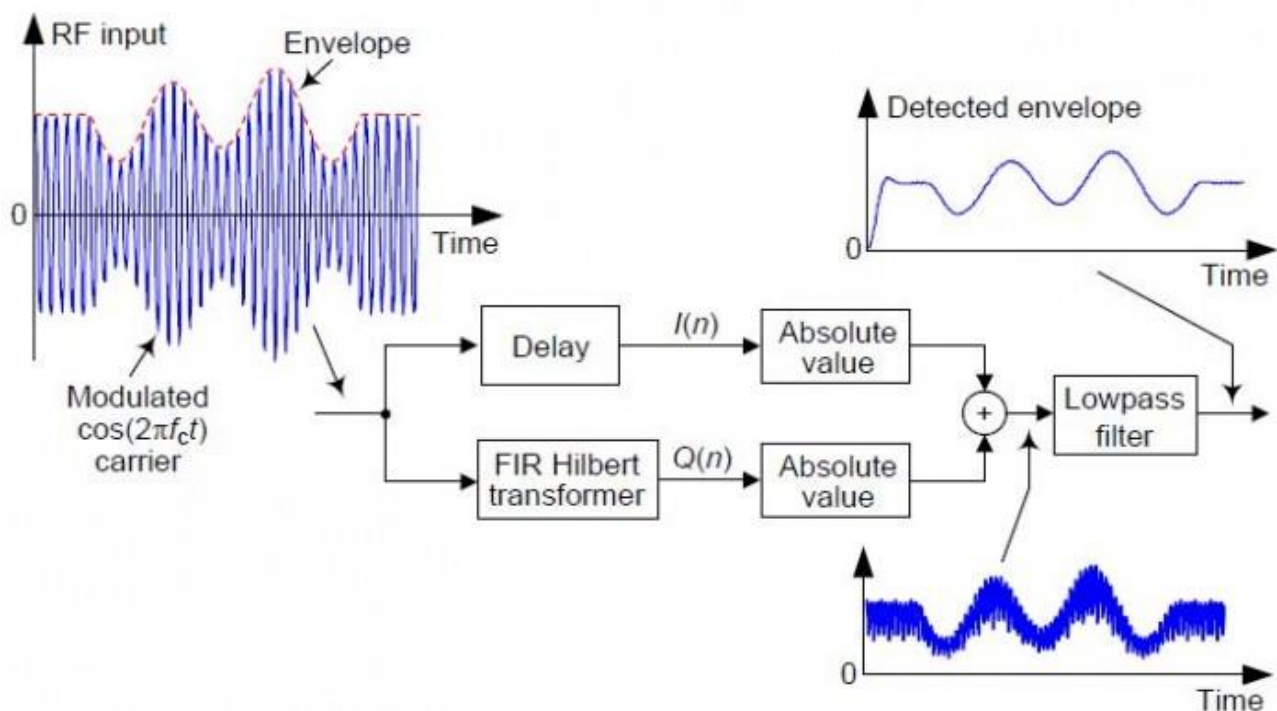


Figure 8.15. Envelope analysis concept.

8.6.1 Load Variation Modulation

The radial load in the bearing determines the strength of the impact from rolling over a fault. A fault in a stationary bearing race will be subjected to the same force at each roll and consequently all the pulses in the pulse train will be of equal strength/height. On the other hand, a fault in a rotating race will be subjected to a varying force, the variation repeating itself with the RPM of the race. This means that the pulse train will be amplitude-modulated with the RPM of the race, and in turn all the harmonics in the line spectrum, ball passing frequency outer race (BPFO), or ball passing frequency inner race (BPFI) will appear amplitude-modulated by the RPM of the race. Likewise, the ball fault frequency (BFF) caused by a ball/roller fault, will be amplitude-modulated by the fundamental train frequency (FTF). If there is more than one fault of a kind, the line spectrum will still contain the harmonics of the bearing frequency. Only the shape of the spectrum will change depending on the relative positions of the faults.

8.6.2 Envelope Detection

The bearing frequencies are present throughout the spectrum (the 1/T line spectrum), but obscured at lower frequencies by other vibrations. However, there is a technique that makes it possible to extract the bearing frequencies from the part of the vibration spectrum where the 1/T line spectrum is dominant, that is, amplitude demodulation: A band-pass filter, with center

frequency f_c , filters out the selected part of the spectrum, the output is shifted (heterodyned) to low frequency ($f_c \rightarrow \text{DC}$) and subjected to envelope detection.

If the band-pass filter encompasses a range where the 1/T line spectrum is dominant, the resulting time history will be dominated by the envelope of the original pulse train. This envelope time history can now be subjected to FFT analysis for easy identification of Bearing Frequencies. The figures below illustrate these properties. A synthesized time signal is composed as follows: A pulse with a repetition time of 25 ms (~ 40 Hz) is subjected to a certain amplitude modulation with a repetition time of 250 ms (~ 4 Hz = 240 RPM) plus random noise, 0 – 1 kHz, with substantially higher power. This resembles a BPFI = 40 Hz fault, load variation modulated by the Shaft-speed = 240 RPM, the bearing situated in a ‘noisy’ machine. The modulation of the BPFI may be perceived as contaminating the clean 40 Hz, BPFI, spectrum, but it is, on the other hand, very informative. In a real situation where bearing frequencies would not be known in advance, this spectrum is the signature of a rotating race fault, most often the inner race. In the example, the BPFI is the 10th harmonic of the shaft speed. This fact can be used to identify a load modulated race fault.

8.7 Crest Factor

The Crest Factor (CF) is used to characterize the quality of a waveform, and in other cases, CF is used to show if a waveform contains a great deal of peaks. CF is defined as the ratio of the peak level of the input signal to the RMS level.

$$\text{Crest Factor} = \frac{\text{Peak level}}{\text{RMS}} \quad (3.3)$$

Therefore, peaks in the time series signal will result in an increase in the CF value. This feature is used to detect changes in the signal pattern due to impulsive vibration sources such as tooth breakage on a gear or defect in a bearing.

When the discrete bearing frequencies cannot be identified because of high background noise and/or widespread damage, the kurtosis and CF are the best techniques for the condition monitoring of bearings. The curve in Figure 8.16 shows a typical trend for CF as bearing condition deteriorates. Initially, there is a relatively constant ratio of peak to RMS value. As a localized fault develops, the resulting short bursts increase the peak level substantially, but have little influence on the RMS level. The peak level will typically grow to a certain limit. As the bearing deteriorates, more spikes will be generated per ball-pass, finally influencing the RMS level, even though the individual peak levels are not greater. Towards the end of bearing life, the CF may have fallen to its original value, even though both peak and RMS levels have increased considerably. The best way to trend the data is as illustrated; RMS and peak levels on the same graph, with CF inferred as the difference between the two curves.

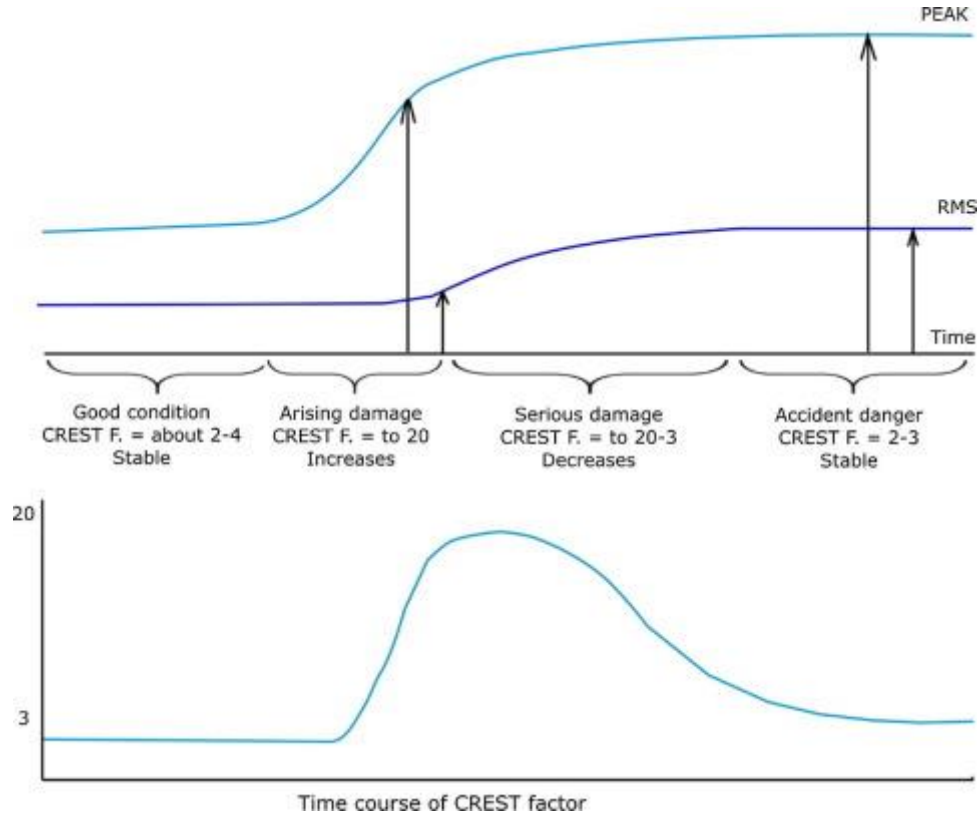


Figure 8.16. Crest Factor concept.

8.8 Kurtosis Analysis

Kurtosis is defined as the 4th moment of a distribution and measures the relative peakedness or flatness of a distribution compared to a normal distribution. Kurtosis provides a measure of the size of the tails of distribution and is used as an indicator of major peaks in a set of data. The equation for kurtosis is given by:

$$k = \frac{\sum_{n=1}^N [Y(n) - \mu]^4}{N \times (\sigma^2)^2} \quad (3.4)$$

Where $Y(n)$ is the raw time series at point n , μ is the mean of the data, σ^2 is the variance of the data, and N is the total number of data points.

Typical values of the signal kurtosis range from 3 to 45 depending upon the condition of the bearing. As a general rule, variations in kurtosis closely follow variations in the CF and the values higher than about 4 are indicative of damage. Kurtosis provides a wider dynamic range and is unaffected by changes in speed and loading because the kurtosis is based upon detecting impulsiveness.

9 Chapter (9) Fault Diagnostics

9.1 Introduction

Some of the machinery defects detected using vibration analysis are listed below:

- ☐ Unbalance
- ☐ Bent shaft
- ☐ Eccentricity
- ☐ Misalignment
- ☐ Looseness
- ☐ Belt drive problems
- ☐ Gear defects
- ☐ Bearing defects
- ☐ Oil whip/whirl
- ☐ Electrical faults
- ☐ Cavitation
- ☐ Shaft cracks
- ☐ Rotor rubs
- ☐ Resonance
- ☐ Hydraulic and aerodynamic forces.

We will now look at each one of the above cases in detail and see how they manifest in vibration analysis.

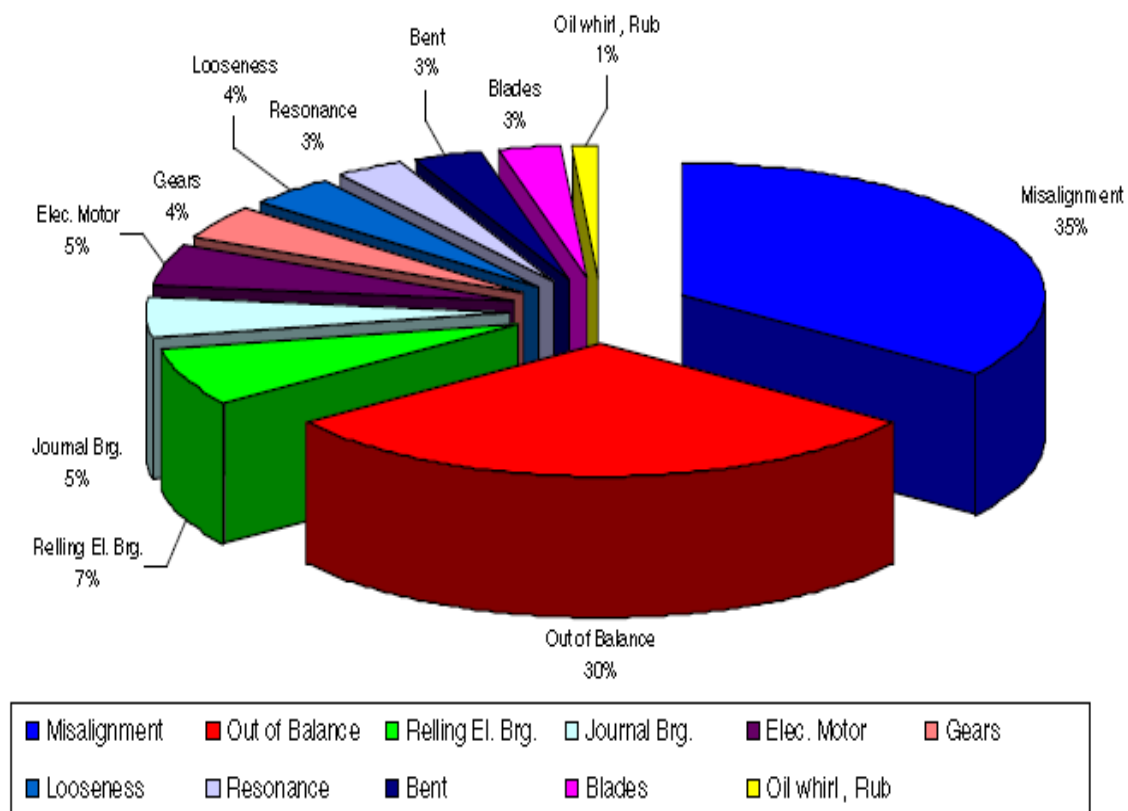


Figure 9.1. Common faults in rotary machines.

9.2 Unbalance

Vibration due to unbalance of a rotor is probably the most common machinery defect. It is luckily also very easy to detect and rectify. The International Standards Organization (ISO) define unbalance as: That condition, which exists in a rotor when vibratory, force or motion is imparted to its bearings as a result of centrifugal forces. It may also be defined as: The uneven distribution of mass about a rotor's rotating centerline. There are two new terminologies used: one is rotating centerline and the other is geometric centerline. The rotating centerline is defined as the axis about which the rotor would rotate if not constrained by its bearings (also called the principle inertia axis or PIA). The geometric centerline (GCL) is the physical centerline of the rotor. When the two centerlines are coincident, then the rotor will be in a state of balance. When they are apart, the rotor will be unbalanced. There are three types of unbalance that can be encountered on machines, and these are:

1. Static unbalance (PIA and GCL are parallel)
2. Couple unbalance (PIA and GCL intersect in the center)
3. Dynamic unbalance (PIA and GCL do not touch or coincide)
4. Unbalance – overhung rotor

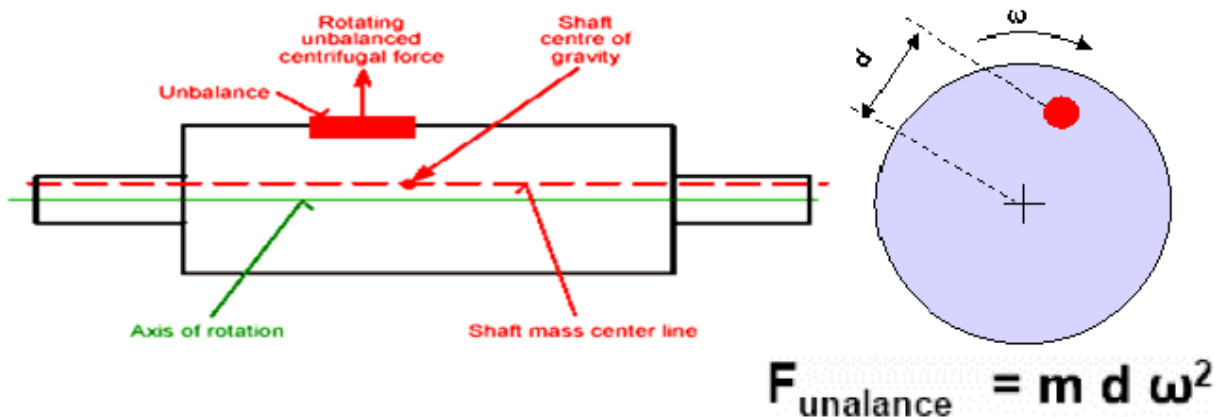


Figure 9.2. Unbalance criteria.

9.2.1 Static Unbalance

For all types of unbalance, the FFT spectrum will show a predominant 1X frequency of rotational speed in the radial direction. Vibration amplitude at 1X will vary proportional to the square of the rotational speed. It is always present and normally dominates the vibration spectrum (Figure 9.3).

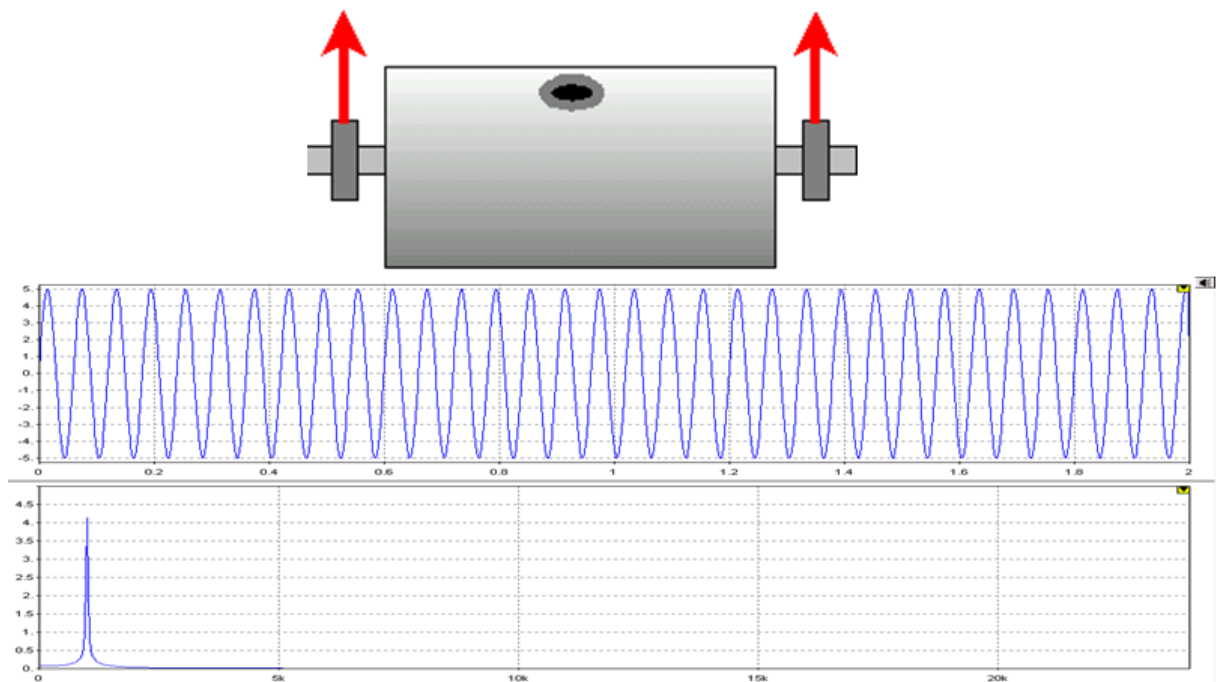


Figure 9.3. Static unbalance.

9.2.2 Couple Unbalance

In a couple unbalance (Figure 9.4), the FFT spectrum displays a single 1X frequency of rotational speed. The amplitude at 1X varies proportional to the square of speed. This defect may cause high axial and radial vibrations. Couple unbalance tends to be 180° out of phase on the same shaft. Note that almost a 180° phase difference exists between two bearings in the radial directions.

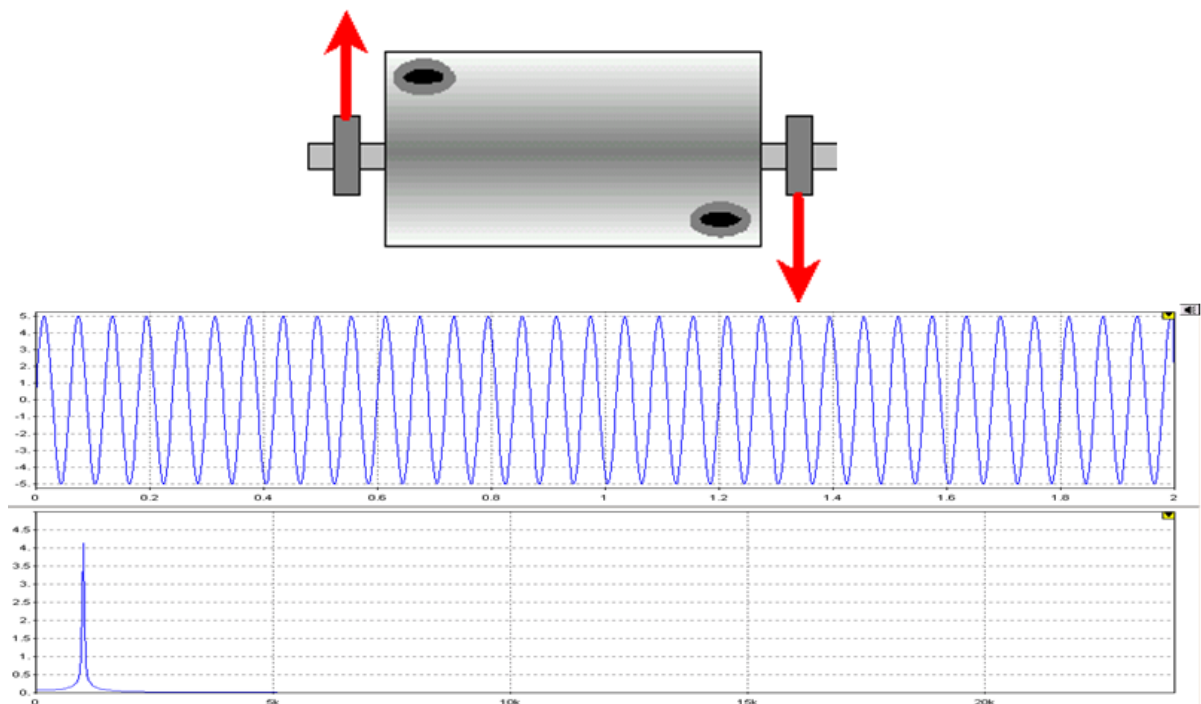


Figure 9.4. Couple unbalance.

9.2.3 Dynamic Unbalance

Dynamic unbalance is the most common type of unbalance and is a combination of static and couple unbalances. It occurs when the principal axis of inertia and the axis of rotation are skew lines. This unbalance can be understood as two different masses placed arbitrary on the rotor. FFT spectrum displays a single 1X frequency of rotational speed. The amplitude at 1X varies proportional to the square of speed. This defect may cause high axial and radial vibrations. Dynamic unbalance gives different phase on the same shaft.

9.2.4 Unbalance - Overhung Rotors

The axial phase on the two bearings will seem to be in phase whereas the radial phase tends to be unsteady. A special case of dynamic unbalance can be found in overhung rotors (see Figure 9.5). Dynamic unbalance in overhung rotors causes high 1X levels in radial and axial direction due to shaft bending which causes axial movement.

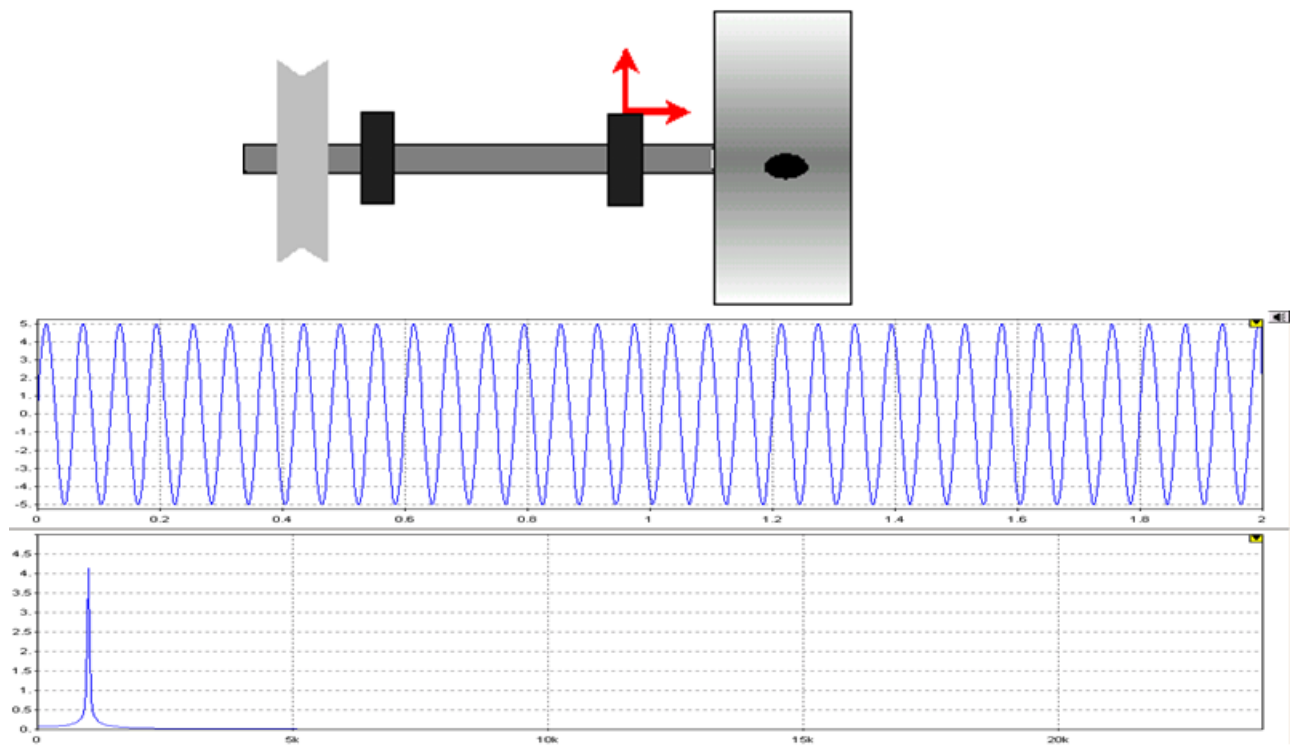


Figure 9.5. Unbalance-overhung rotor.

9.3 Bent Shaft

A bent shaft and bowed rotor are actually the same phenomena. The bent shaft is measurable outside the machine housing while the bowed rotor is the same condition inside. These defects sometimes develop on a motor that has been allowed to sit stationary for a long time. When a bent shaft is encountered, the vibrations in the radial as well as in the axial direction will be high. Axial vibrations may be higher than the radial vibrations. The FFT will normally have 1X and 2X

components. If the amplitude of 1X is dominant, then the bend is near the shaft center as shown in Figure 9.6. If the amplitude of 2X is dominant, then the bend is near the shaft end.

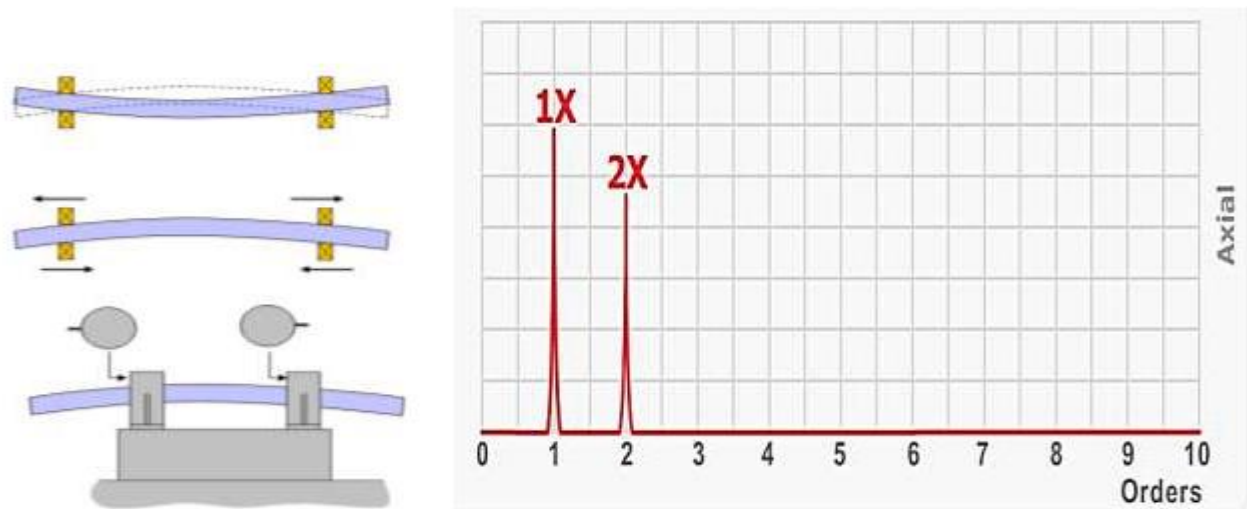


Figure 9.6. Bent shaft near the shaft center.

9.4 Eccentric rotor

Eccentricity occurs when the center of rotation is at an offset from the geometric centerline of a sheave, gear, bearing, motor armature or any other rotor. The maximum amplitude occurs at 1X of the eccentric component in a direction through the centers of the two rotors (see Figure 9.7)

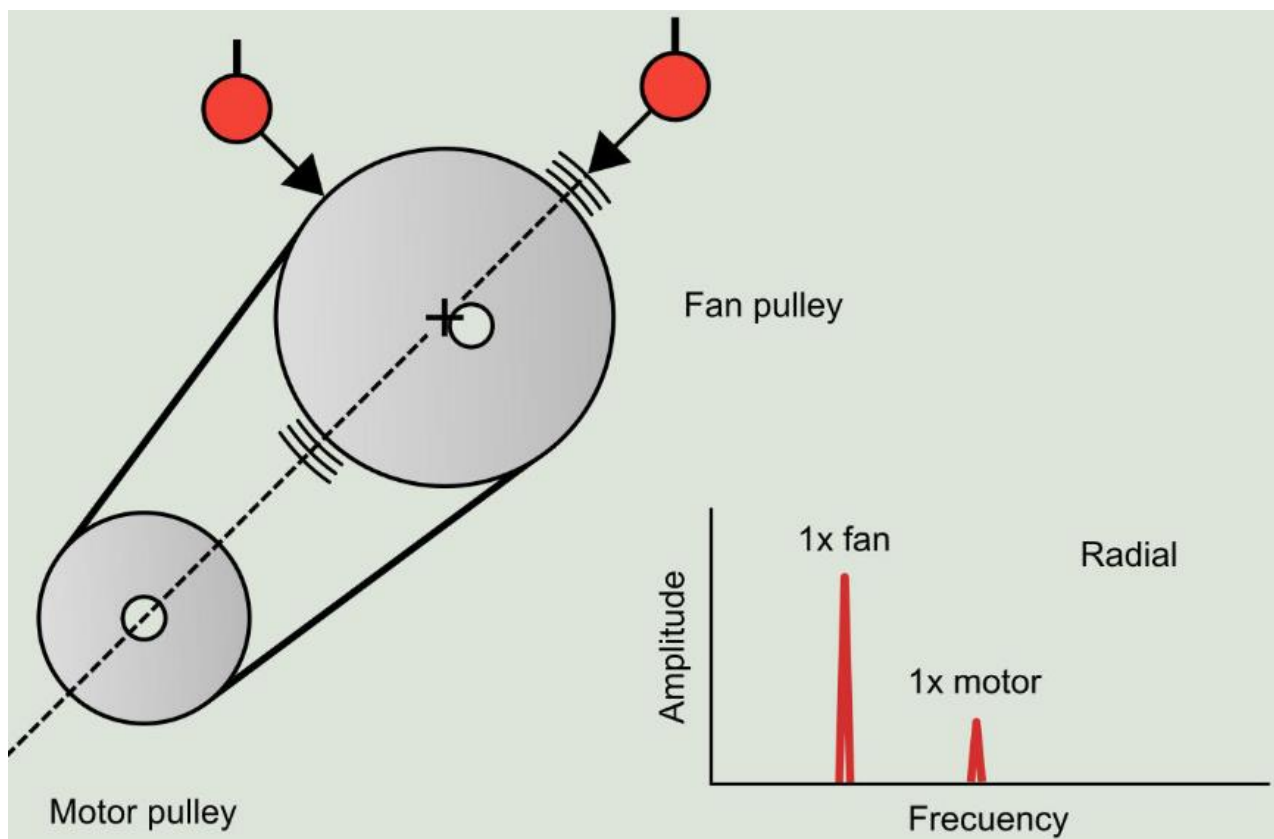


Figure 9.7. Eccentric rotor.

9.5 Misalignment

Misalignment, just like unbalance, is a major cause of machinery vibration. Some machines have been incorporated with self-aligning bearings and flexible couplings that can take quite a bit of misalignment. However, it is not uncommon to come across high vibrations due to misalignment. There are basically two types of misalignment:

1. Parallel misalignment: the shaft centerline of the two machines is parallel to each other and have an offset.
2. Angular misalignment: the shaft centerline of the two shafts meets at angle with each other.
3. Combination of parallel and angular misalignments

9.5.1 Parallel Misalignment

Parallel misalignment results in two *hits* per cycle and therefore, 2X vibration in the radial direction is dominant. Parallel misalignment shows high radial vibration that approaches a 180° phase difference across the coupling in the radial direction. Thus, we will see both the 1X and 2X peaks. Coupling construction will often significantly influence the shape of the spectrum if misalignment is severe (see Figure 9.8).

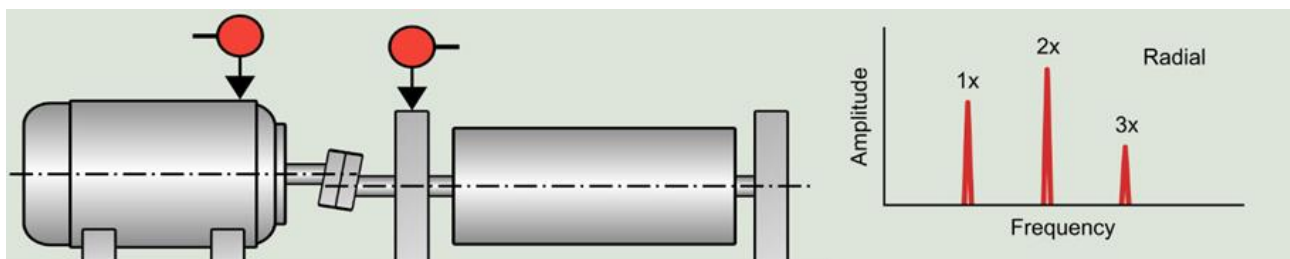


Figure 9.8. Parallel (offset) misalignment.

9.5.2 Angular misalignment

Angular misalignment primarily subjects the driver and driven machine shafts to axial vibrations at the 1X, high 2X, and 3X in addition to 180° phase difference across the coupling in the axial direction.

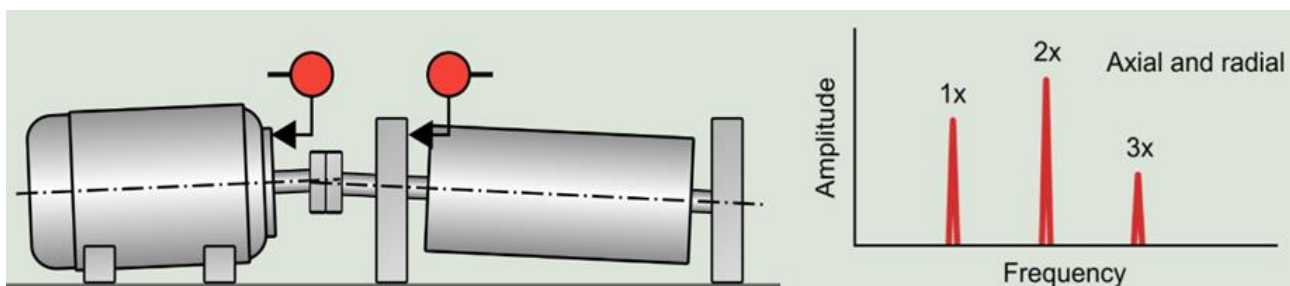


Figure 9.9. Angular misalignment.

9.6 Soft Foot

Soft foot is one of the most prevalent conditions found in rotating machinery. This condition, if not corrected, makes an alignment job much more difficult and sometimes impossible. Soft foot can cause high levels of vibration. Soft foot can be seen as looseness and exhibit a high 1X vibration signature and 180o phase difference between base plate and concrete base. Soft foot can be occurred due to base plate problem or bad concrete base leveling as shown in Figure 9.10. There are four types of soft foot as discussed in the following sections.

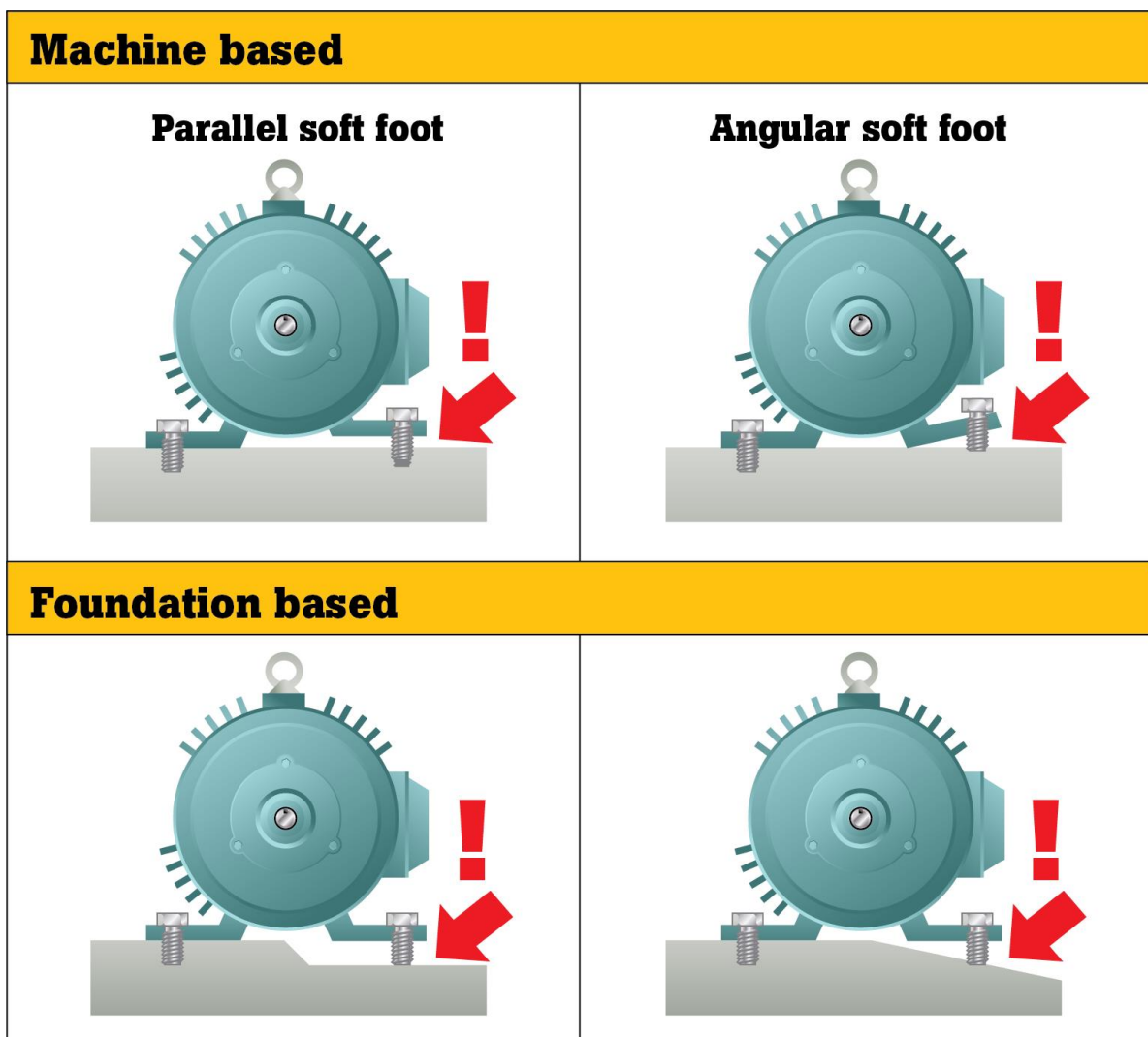


Figure 9.10. Causes of soft foot.

9.6.1 Parallel Soft Foot

Parallel soft foot or (short foot) exists when the foot simply does not reach the base and creates a gap between the foot and base. The bottom of the foot is parallel to the base plate. Tightening the hold-down bolts will result in a distortion of the machine's frame as the soft foot is

drawn down to the base. This condition is easiest to detect using either a feeler gauge or dial indicator.

9.6.2 Angular Soft Foot

Angular soft foot can occur when the foot is touching the base on either the outside or inside portion of the foot, but the other side of the foot is bent away creating an angle between the base and the bottom of the foot. This type can be corrected by using plastic shims to fill the angular gap.

9.6.3 Squishy Foot

Squishy foot, sometimes called spring foot, exists when the gap between the foot and base has already been filled with corroded / soft shims. The machine will appear to be fixed of soft foot problems until the hold-down bolts are tightened. This type can be corrected by replacing the corroded / soft shims by integrated / hardened once.

9.6.4 Stress-Induced Soft Foot

Perhaps the most difficult soft foot condition to detect is caused by forces that are external to the machine. This is referred to as stress-induced soft foot. It can be the result of pipe strain or stresses induced by the electrical connections as well as drastic misalignment. Stress-induced forces can be created during any stage of the alignment process therefore, eliminating this kind of soft foot may require more than one check such as piping strain check.

9.7 Looseness

Mechanical looseness or the improper clearance between component parts is generally characterized by a long string of harmonics of running frequency with abnormally high amplitudes. In some machines, vibration levels may be excessive as a consequence of components being assembled too loosely, for example in the case of a bearing, which is not properly secured. Mechanical looseness can be classified to three categories:

- 1) Structure looseness.
- 2) Machine / base plate looseness.
- 3) Internal assembly looseness.

9.7.1 Structure Looseness (Non-Rotating Looseness - Type A)

This type of looseness is caused by structural looseness or weakness in the machine's feet, base plate, or foundation. It can also be caused by deteriorated grouting, loose hold down bolts at the base, and distortion of the frame or base (soft foot). Phase analysis may reveal approximately 180° phase shift between vertical measurements of machine's foot / base plate and foundation as shown in Figure 9.11. When the soft foot is suspected, an easy test to confirm for it is to loosen

each bolt, one at a time, and see if this brings about significant changes in the vibration. In this case, it might be necessary to re-machine the base or install shims to eliminate the distortion when the mounting bolts are tightened again.

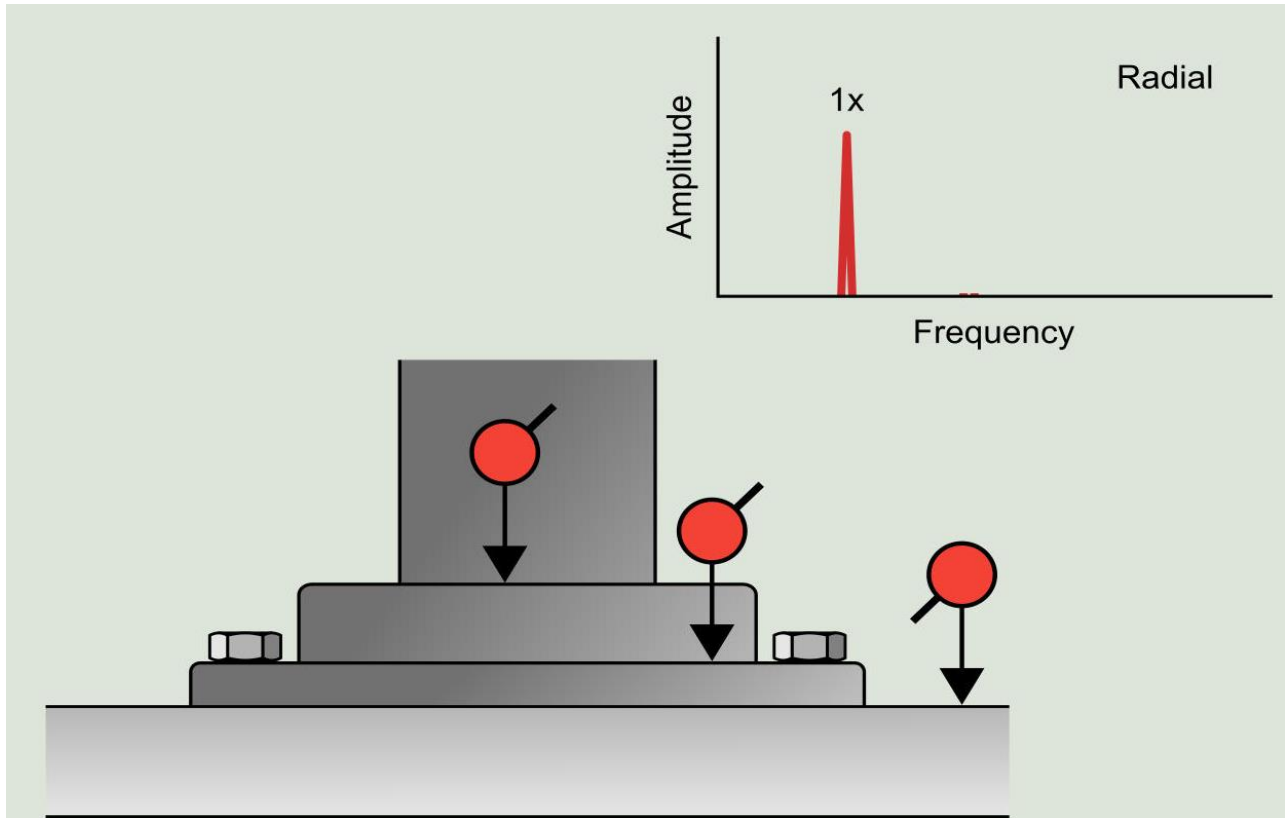


Figure 9.11. Structural looseness.

9.7.2 Machine Components / Base Plate Looseness (Non-Rotating Looseness - Type B)

This problem is associated with loose pillow-block bolts, cracks in the frame structure, or the bearing pedestal as displayed in Figure 9.12. This generates higher harmonics due to the rocking motion of the pillow block with loose bolts. Since the movement is restricted in one direction, we can see a truncated time wave form (This is the reason why harmonics appear in spectrum).

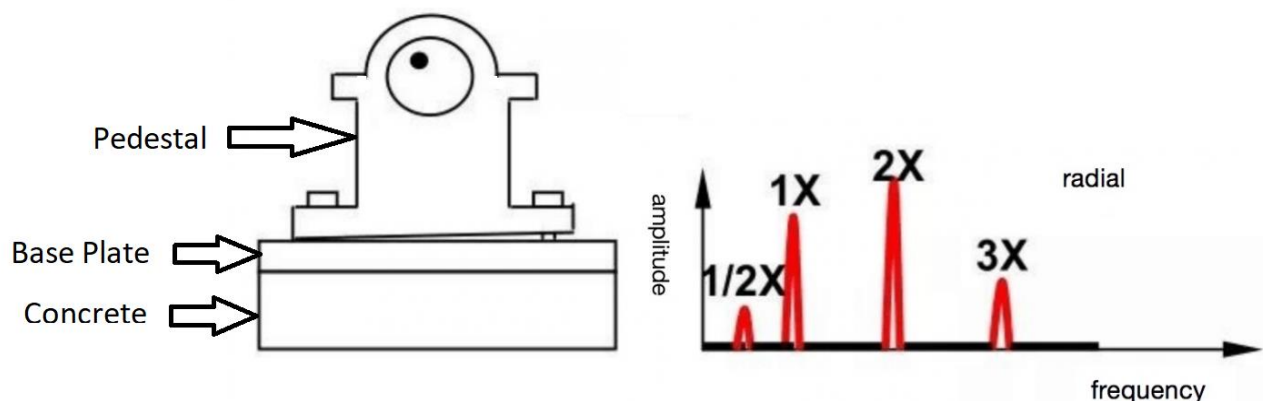


Figure 9.12. Non- rotating looseness – Type B.

9.7.3 Internal Assembly Looseness (Rotating Looseness - Type C)

Internal assembly looseness is normally caused by an improper fit between bearing component parts which produce many harmonics in the FFT due to the nonlinear response of the loosed parts to the exciting forces from the rotor. The phase is often unstable and can vary broadly from one measurement to another. Looseness will often cause sub-harmonic multiples at 0.5X and 1.5X as shown in Figure 9.13.

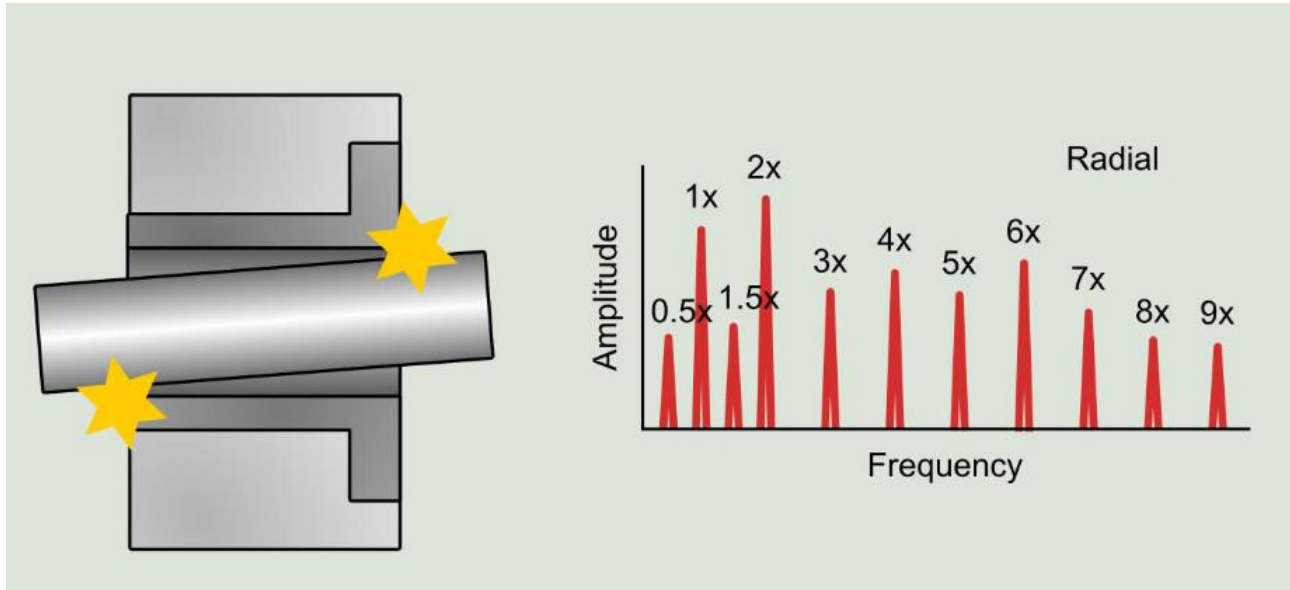


Figure 9.13. Internal (rotating) looseness.

9.8Belt Drive Defects

Worn, loose and mismatched belts are considered the main problem in the rotating machines. Belt defect frequencies are of the sub-harmonic type. Belt-drive problems, which include shaft misalignment, pulley misalignment, belt wear, belt resonance, belts too tight, belts too loose, pulley eccentricity and bent shafts, can be relatively straight forward to detect but can be far more difficult to specifically diagnose and correct. That is mainly due to the wide variety of problems that can occur in the installation and assembling of the belt drive, the difficulty of doing field testing on belts, and the possibility of other influences (i.e. the base) having some effect. With timing belt drives, it is useful to know that high amplitudes at the timing belt frequency indicate wear or pulley misalignment. The belt frequency (Figure 9.14) is given by the formula:

$$\text{Belt frequency} = \frac{\text{Pulley rpm} \times \text{Pulley Pitch Diameter}}{\text{Belt Length}} \quad (4.1)$$



Figure 9.14. Belt frequency with sub-harmonic belt frequencies.

9.8.1 Pulley Misalignment

There are two types of misalignment as shown in Figure 9.15. These conditions not only result in destructive vibration but also cause accelerated wear of both the belt and the sheaves. Misalignment of sheaves produces high vibration at 1X, predominantly in the axial direction. The ratio of amplitudes of driver to driven rpm depends on the measurement position, relative mass, and the frame stiffness. With sheave misalignment in fans, the highest axial vibration will be at the fan rpm. When the belt drives an overhung rotor, which is in an unbalanced condition, it will have to be resolved with phase analysis.

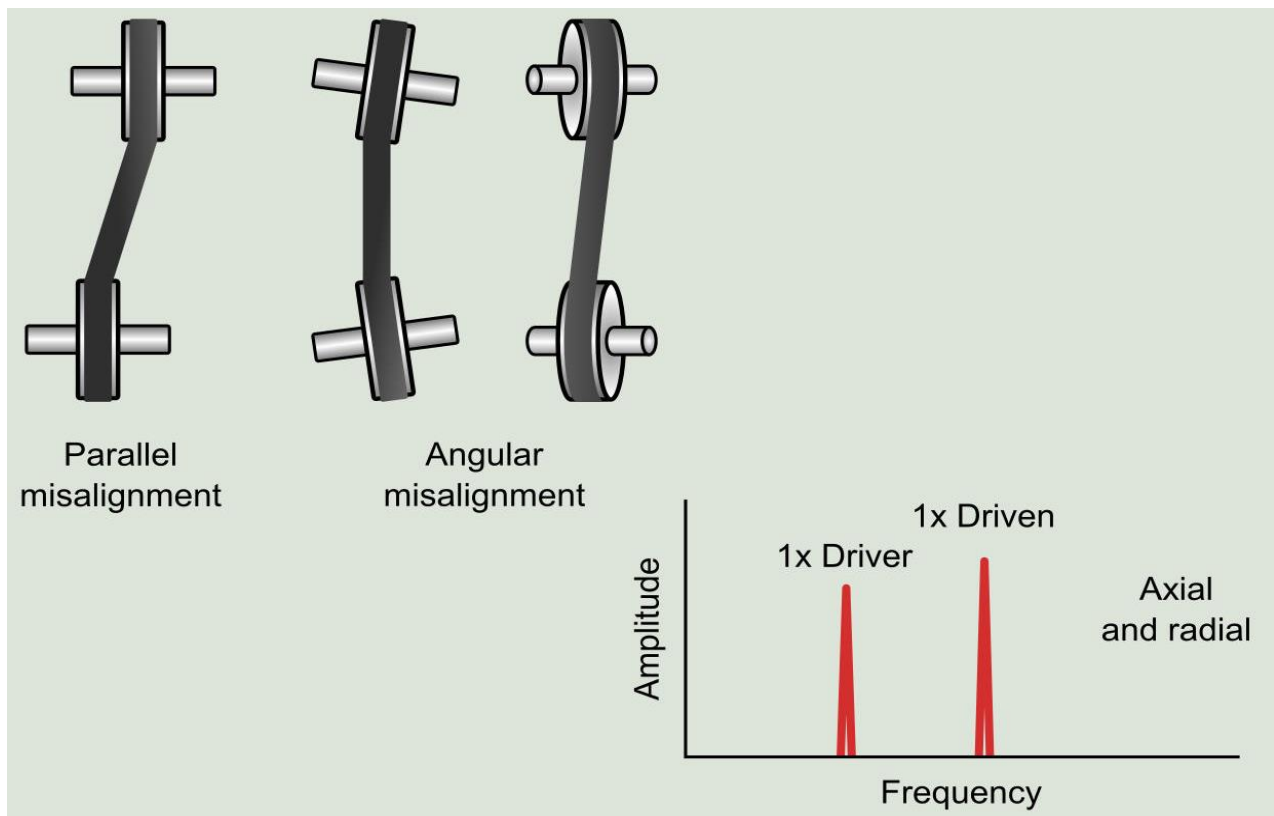


Figure 9.15. Pulley misalignment.

9.8.2 Belt Wear

Worn sheaves or belts may also increase vibration at certain rotational frequencies; however, belt wear is more often indicated by increased slippage and drive wear. Belt defects appear in the vibration signature as sub-synchronous peaks, often with harmonics. Figure 9.16 shows a typical spectral plot (i.e., vibration profile) for a defective belt. If a belt is worn or loose, you will witness a peak at the belt rate, and harmonics. The highest amplitude peak in the series will often be the twice the belt rate frequency. The key forcing frequency is known as the "belt rate" or "fundamental belt pass frequency". It is the rate at which a point on the belt passes a fixed reference point. It is always less than the speed of either sheave.

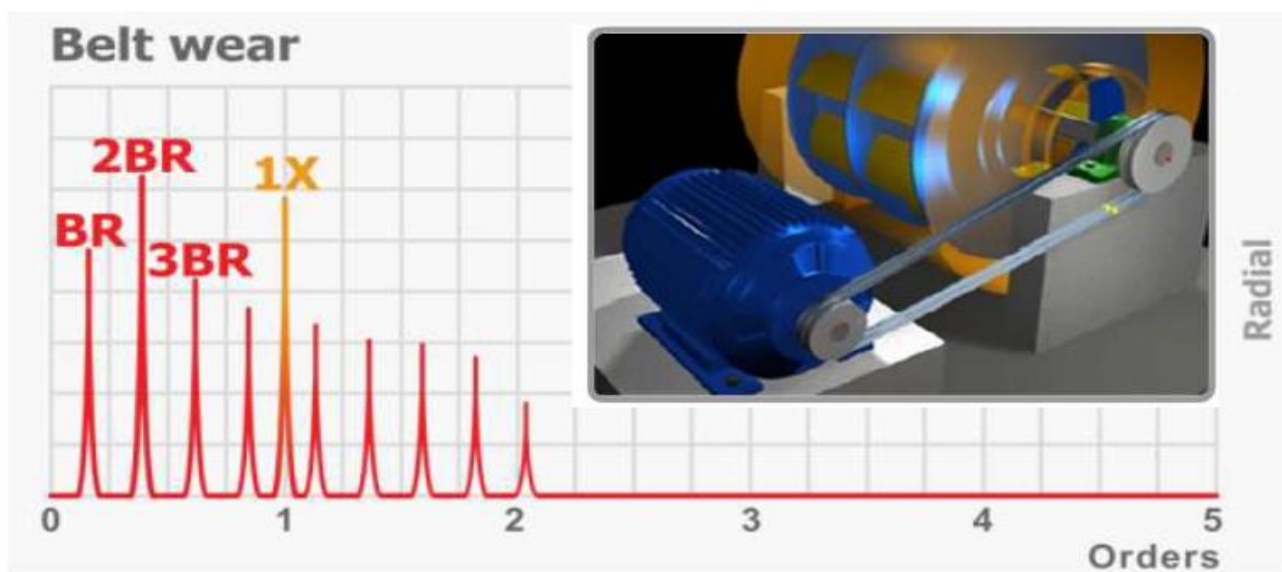


Figure 9.16. Belt wear rate with harmonics.

9.8.3 Eccentric Pulleys

Eccentric or unbalanced sheaves cause maximum vibration at 1X of the sheave rotating speed (see Figure 9.7). To resolve this condition, it may sometimes be possible to balance eccentric sheaves by attaching washers to taper lock bolts. However, even if balanced, the eccentricity will still induce vibration and cause fatigue stresses in the belt.

9.8.4 Belt Resonance

Resonance in the belt can cause high amplitudes if the belt's natural frequency should happen to coincide with either the driving or driven sheave RPM. The natural frequency can be altered by either changing the length or tension of the belt.

Spectrum: If the belt's natural frequency coincides with the RPM of either driven component, then you expect that the 1X peak to increase in amplitude as shown in Figure 9.17. The base of the peak may be more broad than normal.

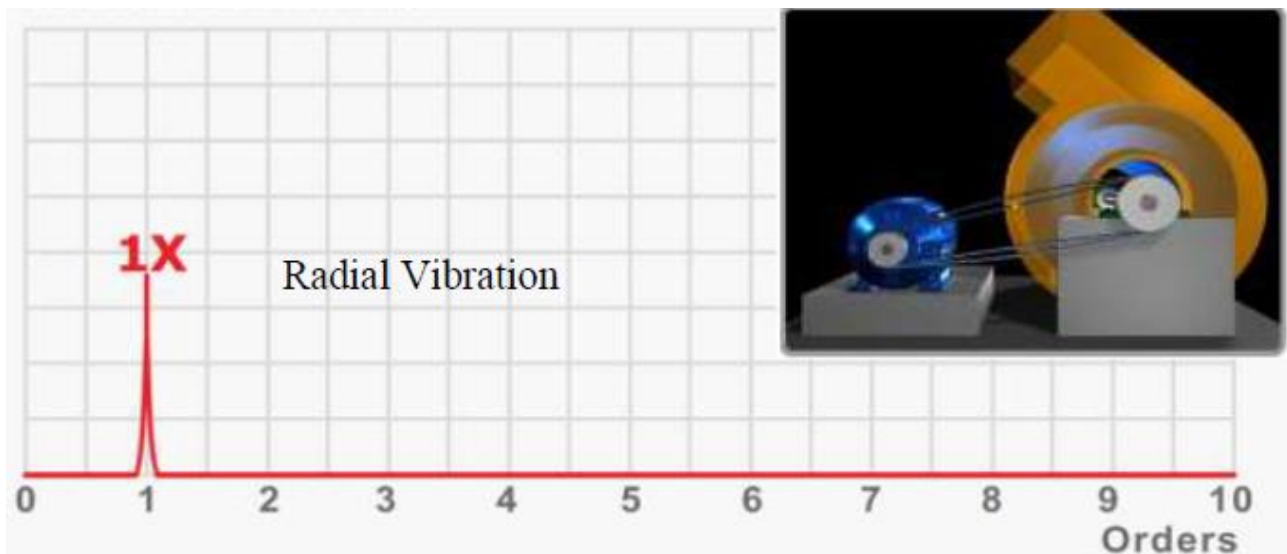


Figure 9.17. Belt resonance.

9.9 Gear Defects

All gear sets create a frequency component referred to as gear mesh. The fundamental gear mesh frequency is equal to the number of gear teeth times the running speed of the shaft. In addition, all gear sets create a series of sidebands or modulations that are visible on both sides of the primary gear mesh frequency.

9.9.1 Normal Profile

In a normal gear set, each of the sidebands is spaced by exactly the 1X running speed of the input shaft, and the entire gear mesh is symmetrical as shown in Figure 9.18. In addition, the sidebands always occur in pairs, one below and one above the gear mesh frequency and the amplitude of each pair is identical. If we split the gear-mesh profile for a normal gear by drawing a vertical line through the actual mesh (i.e., number of teeth times the input shaft speed), the two halves would be identical. Therefore, any deviation from a symmetrical profile indicates a gear problem; however, care must be exercised to ensure that the problem is internal to the gears and not induced by outside influences. External misalignment, abnormal induced loads, and a variety of other outside influences destroy the symmetry of a gear-mesh profile. For example, a single-reduction gearbox used to transmit power to a mold-oscillator system on a continuous caster drives two eccentric cams. The eccentric rotation of these two cams is transmitted directly into the gearbox, creating the appearance of eccentric meshing of the gears; however, this abnormal induced load actually destroys the spacing and amplitude of the gearmesh profile. A gearbox is a piece of rotating equipment that can cause the normal low-frequency harmonics in the vibration spectrum, but also show a lot of activity in the high frequency region due to gear teeth and bearing impacts.

The spectrum of any gearbox shows 1X and 2X, along with the gear mesh frequency (GMF). The GMF will have running speed sidebands relative to the shaft speed to which the gear is attached. Gearbox spectrums contain a range of frequencies due to the different GMFs and their harmonics. All peaks have low amplitudes and no natural gear frequencies are excited if the gearbox is still in a good condition. Sidebands around the GMF and its harmonics are quite common.

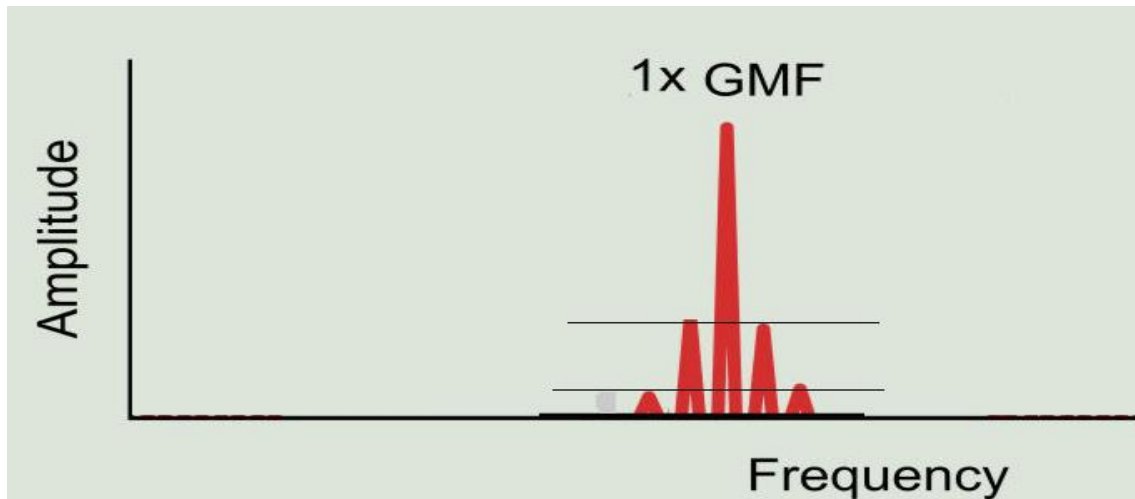


Figure 9.18. Sidebands (paired and equal).

9.9.2 Gear Tooth Wear

Since the spacing between the sidebands is erratic and is no longer evenly spaced by the input shaft speed frequency, the sidebands for a worn gear set tend to occur between the input and output speeds and are not evenly spaced. An important characteristic of gear tooth wear is that gear natural frequencies are excited with sidebands around them. These are spaced with the running speed of the bad gear. The GMF may or may not change in amplitude, although high-amplitude sidebands surrounding the GMF usually occur when wear is present. Sidebands are a better wear indicator than the GMF itself (see Figure 9.19).

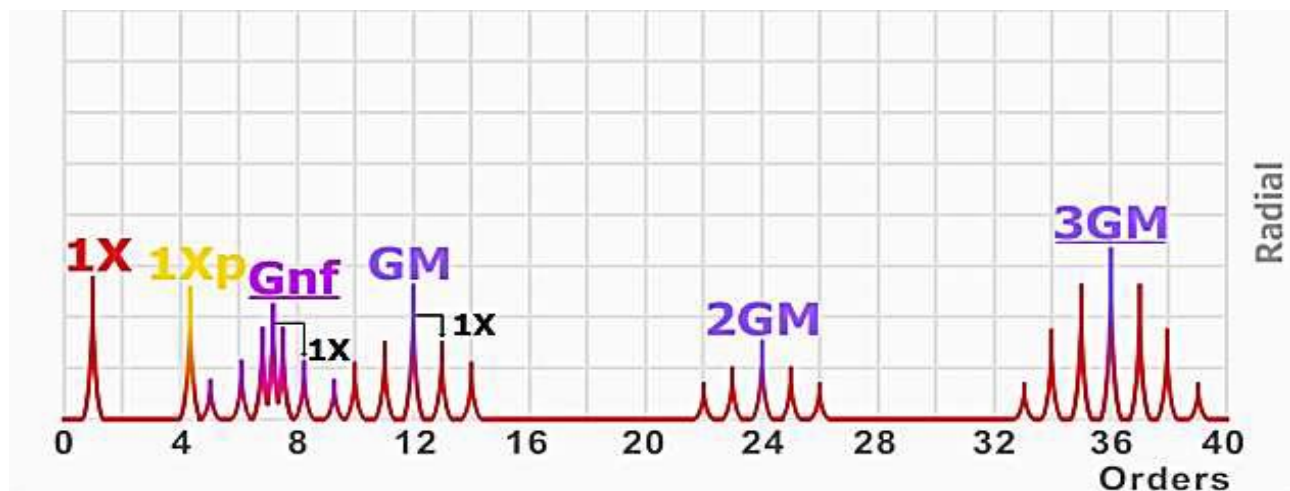


Figure 9.19. FFT spectrum analysis for gear tooth wear.

9.9.3 Gear Tooth Load

As the load on a gearbox increases, the GMF amplitude may also increase. High GMF amplitudes do not necessarily indicate a problem, particularly if sideband frequencies remain low and no gear natural frequencies are excited as shown in Figure 9.20. It is advised that vibration analysis on a gearbox be conducted when the gearbox is transmitting maximum power.

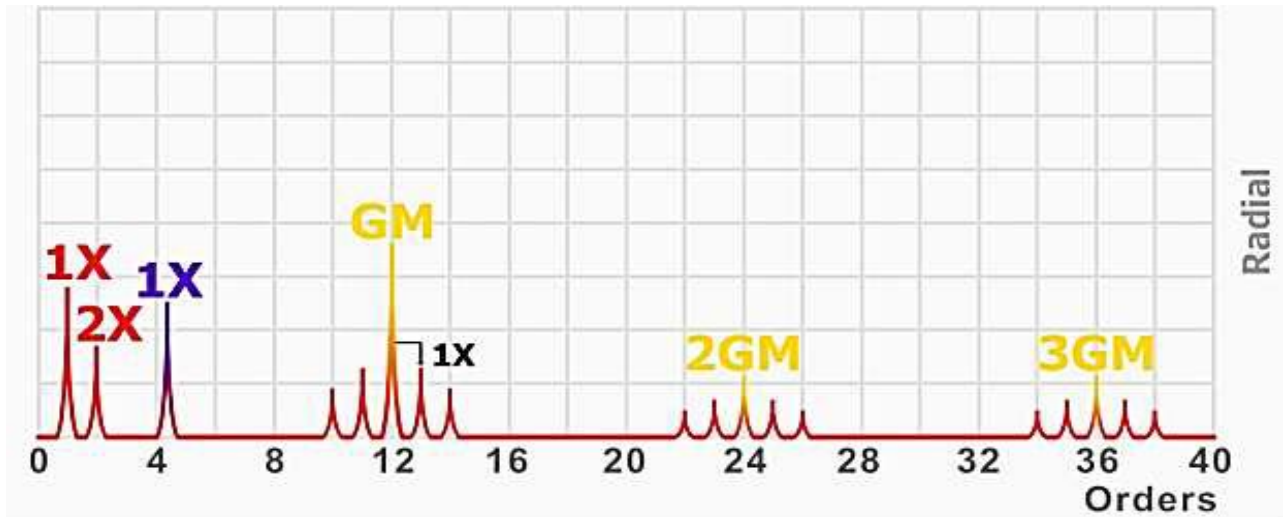


Figure 9.20. FFT spectrum for gear tooth load.

9.9.4 Eccentric Gears and Backlash

If a gear is eccentric (or the shaft is bent), there will be an increase in the amplitude of the sidebands around the gear mesh frequency. The key is that rather than witnessing a family of sidebands, there may only be one dominant sideband on either side of the gear mesh peak. In addition, it is not uncommon to see an increase in the 1XGMF and 3XGMF peaks (see Figure 9.21). Eccentricity can result in backlash therefore; you may also witness a peak at the natural frequency of the gear. This peaks (whose frequency is unknown) will also have 1X sidebands.

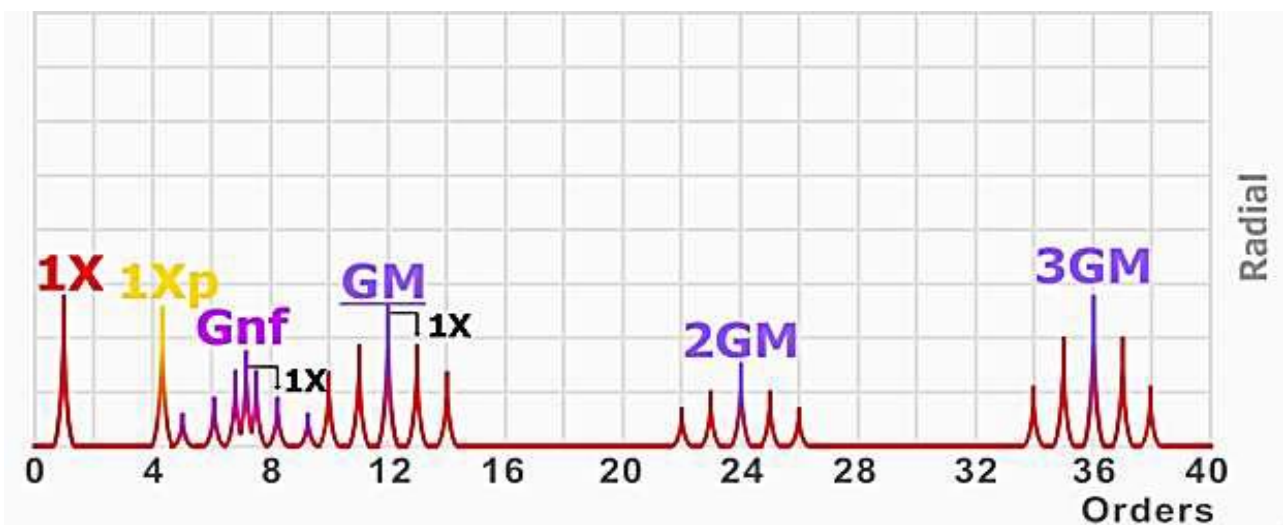


Figure 9.21. FFT spectrum for eccentric gears and backlash.

9.9.5 Misaligned Gears

Gear misalignment always excites second order or higher GMF harmonics, which will have sidebands spaced with the running speed. It will often show only small amplitudes at 1XGMF, but much higher levels at 2X (see Figure 9.22). It is important to set the frequency range of the FFT spectrum more than 3XGMF.

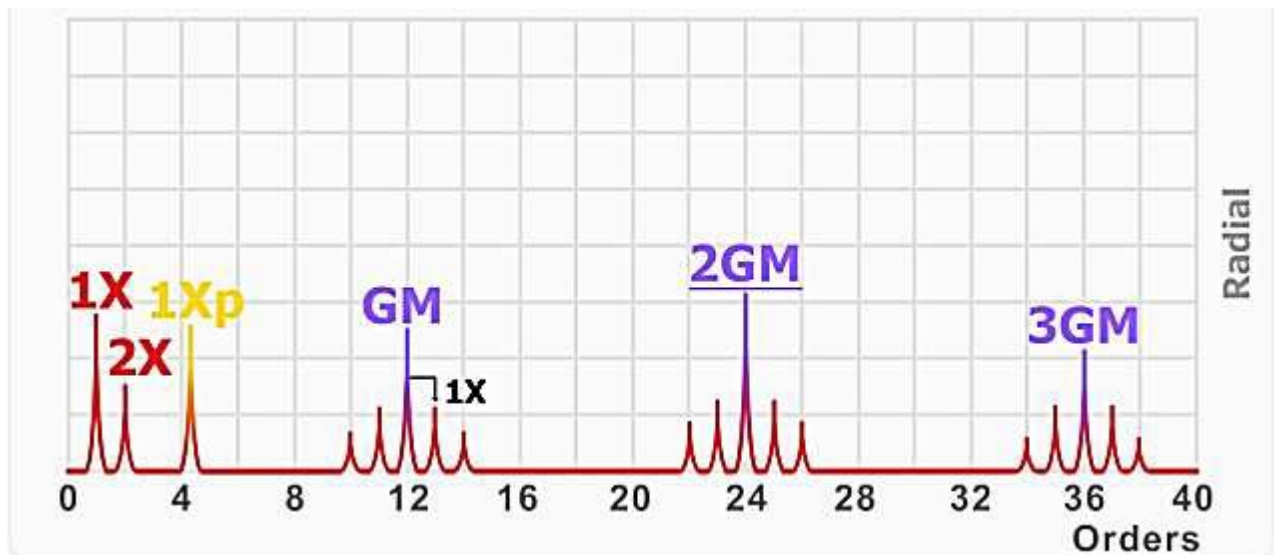


Figure 9.22. FFT spectrum for Gear misalignment

9.9.6 Cracked or Broken Teeth

When the gear has a cracked/broken tooth and when this tooth meshes the mating gear, the clearance between two meshed teeth increases leading to low amplitude sidebands in the left side of gear mesh frequency. When the next undamaged teeth are meshed, higher energy impact is generated leading to higher amplitude sidebands to the right side of gear mesh frequency as shown in Figure 9.23. As a result, the paired sidebands have non symmetrical amplitude due to disproportional clearance and impact energy.

The cracked/broken gear tooth generates high amplitude at 1X rpm of this gear and excites the gear natural frequency with sidebands spaced with its running speed. The time wave form is the best indicator for this fault since the impact shocks appear clearly and periodically repeated every one revolution of defected gear as shown in Figure 9.24.

It is best

detected in the time domain, which will show a pronounced spike every time the problematic tooth tries to mesh with teeth on the mating gear. The time between impacts will correspond to 1/speed of the gear with the broken tooth. The amplitude the impact spike in the time waveform will often be much higher than that of the 1X gear rpm in the FFT spectrum (Figure 7.36).

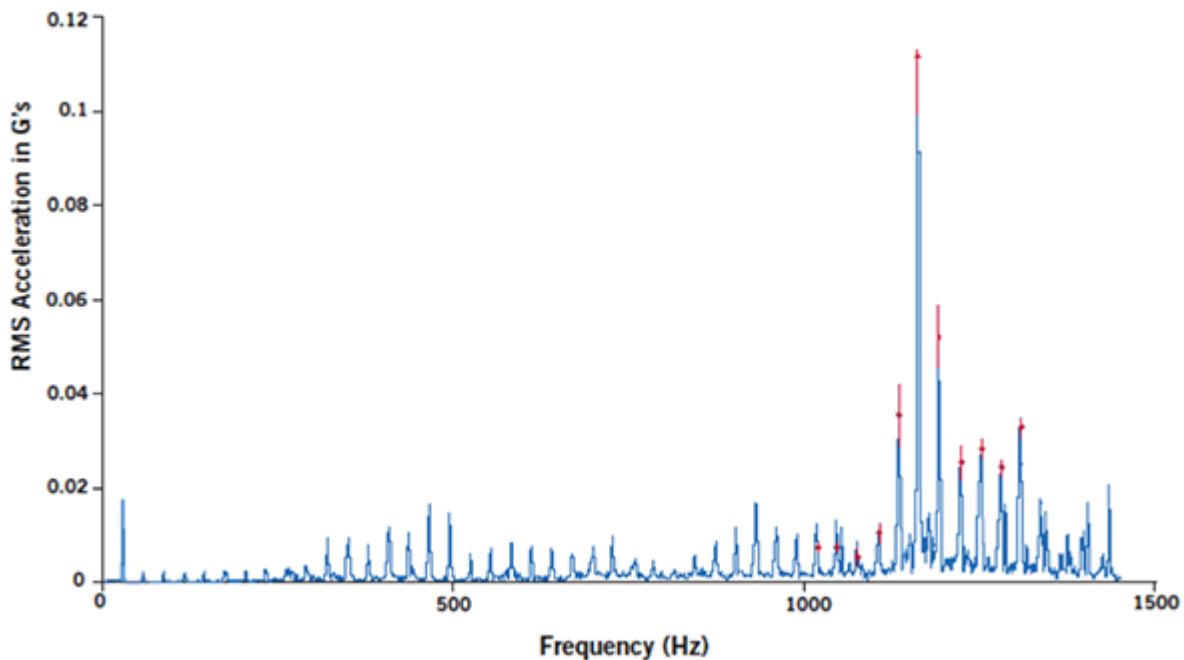


Figure 9.23. Non symmetrical sidebands profile due to broken tooth.

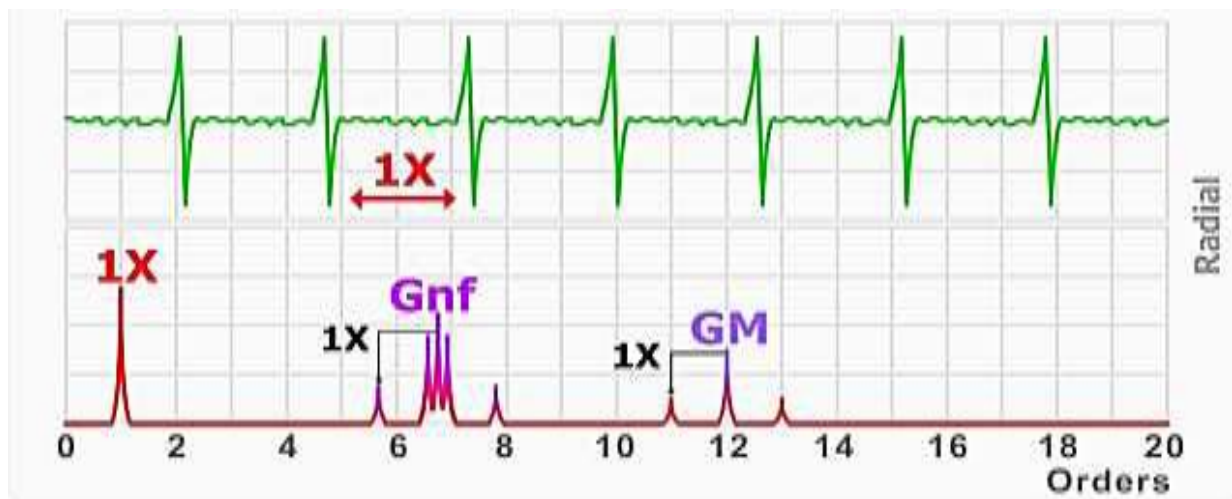


Figure 9.24. The pattern of cracked/broken tooth

9.9.7 Hunting Tooth Frequency

If each one of two mating gears has a defected tooth, the hunting tooth frequency (HTF) is generated which is the rate of a tooth in one gear mates with a particular tooth in the other gear. HTF is located in the low frequency as shown in Figure 9.25. A small defect in one tooth will repeatedly contact the same teeth in the other gear causing localized wear on those teeth. In practice, the HTF is used to detect faults on both the gear and pinion that may have occurred during manufacturing or as a result of mishandling.

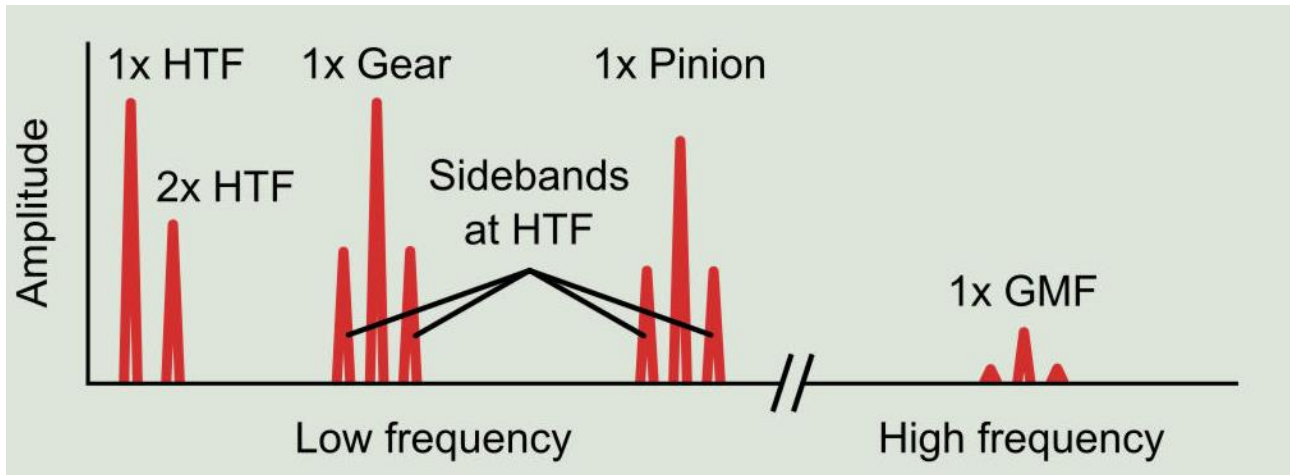


Figure 9.25. Hunting tooth frequency

9.9.8 Load Changes

The energy and vibration profiles of gear set change with load. When the gear is fully loaded, the profiles exhibit the amplitudes discussed previously. When the gear is unloaded, the same profiles are present, but the amplitude increases dramatically (see Figure 9.26). The reason for this change is the gear-tooth roughness. In normal practice, the backside of the gear tooth is not finished to the same smoothness as the power or drive side. Therefore, more looseness is presented on the non-power or back side of the gear.

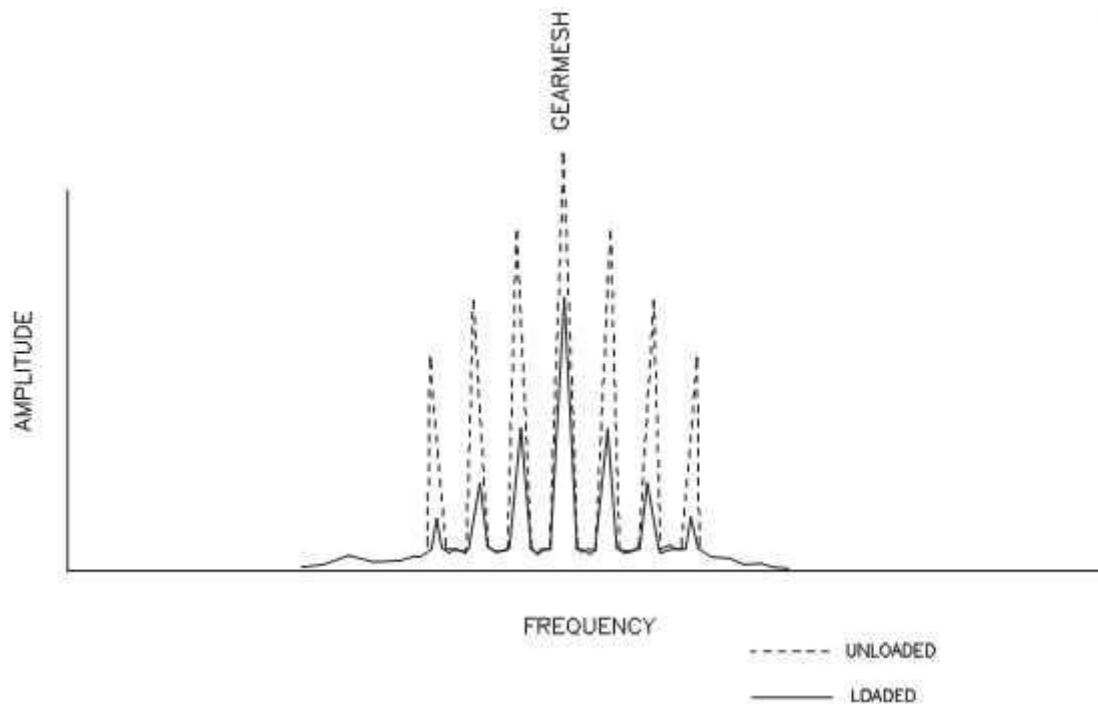


Figure 9.26. Unloaded gear has much higher vibration levels

9.10 Bearing Defects

Bearings are an important machine element that supports the rotor system. They are designed to provide less friction at the supports and carry the loads. In machineries, the two most common types of bearings are used—the antifriction bearing or the rolling element bearing, and the journal bearing working on the principle of hydrodynamic lubrication. We will focus on fault diagnosis in rolling element bearing.

Bearing defects are one of the most common faults identified by vibration-monitoring programs less than 10% run for design lifetime, ~40% fail due to improper lubrication, and 30% due to misalignment. Although bearings wear out and fail, defects are normally symptoms of other problems within the machine train or process system. The real problem of bearing elements can be identified as follow:

- **Lubrication:** Insufficient lubrication.
- **Wear/cracks/spalls:** Damage on the inner or outer race and damage on the rolling elements. Damaged cages are difficult to detect with vibration analysis.
- **Cocked bearing:** Inner race not true on shaft or outer race not true in housing.
- **Excessive clearance:** Due to excessive wear or poor bearing selection.
- **Skidding:** Rolling elements skidding over inner race when out of the load zone.
- **Loose on shaft:** Inner race sliding on shaft.
- **Loose in housing:** Outer race turning in housing.

Four rotational frequencies are associated with rolling element bearings as shown in Figure 9.27. These components frequencies are fundamental train frequency (FTF), ball spin frequency (BSF), ball pass frequency outer race (BPFO), and ball pass frequency inner race (BPFI). There is a large number of fault conditions that can be associated with rolling element bearings such as bearing wear, lubrication problems, cocked bearing, fluting/EDM, skidding, and looseness. These faults have a significant effect and the effectiveness level was classified into four stages which will be discussed in the following sections.

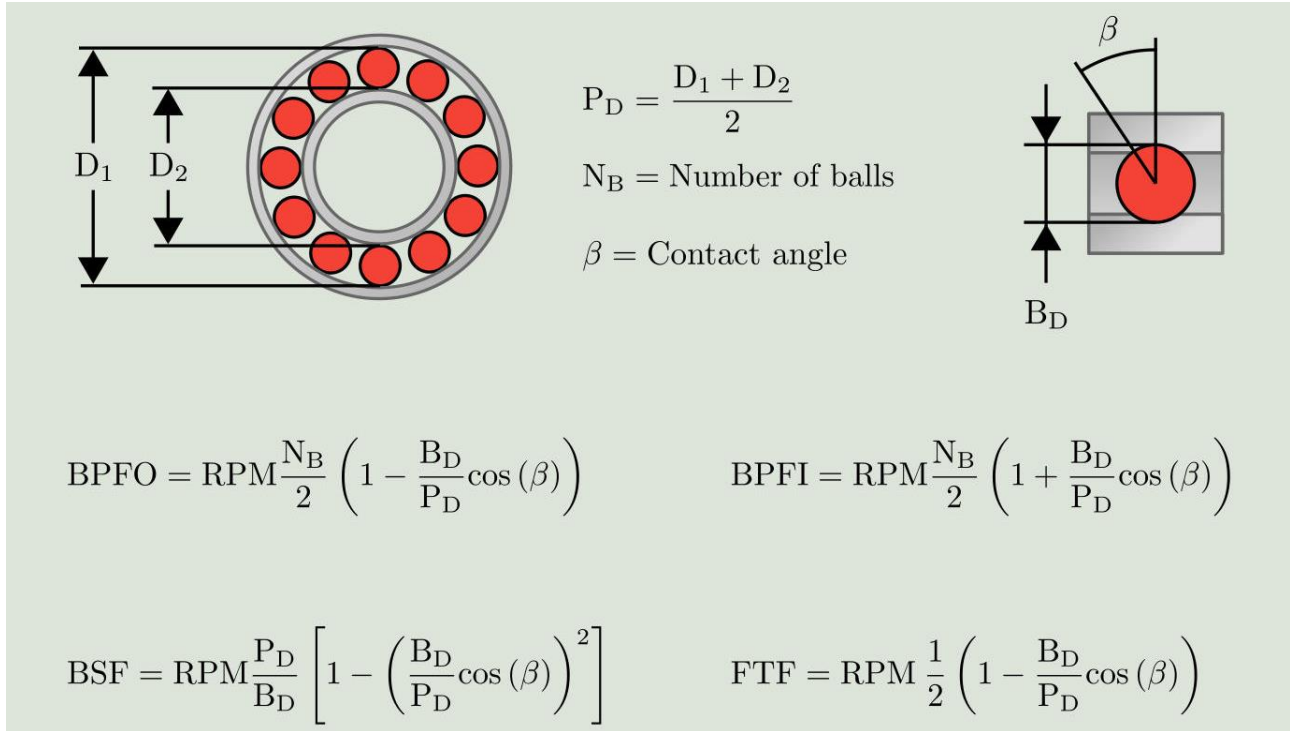


Figure 9.27. Bearing components frequencies

9.10.1 Stage One

The earliest indications of bearing problems including poor lubrication occur in the ultrasonic frequency (greater than 20 kHz). This is the stage of sub-surface damage which produces friction and minor impacts. It has a significant pattern in the frequency range 5 to 40 kHz as shown in Figure 9.28. This requires to use ultrasound measurement devices. Traditional spectrum analysis and time waveform techniques will not detect the fault in this stage. High frequency techniques such as enveloping, demodulation, PeakVue™, Shock Pulse™, and Spike Energy™ can detect the fault in Stage One.

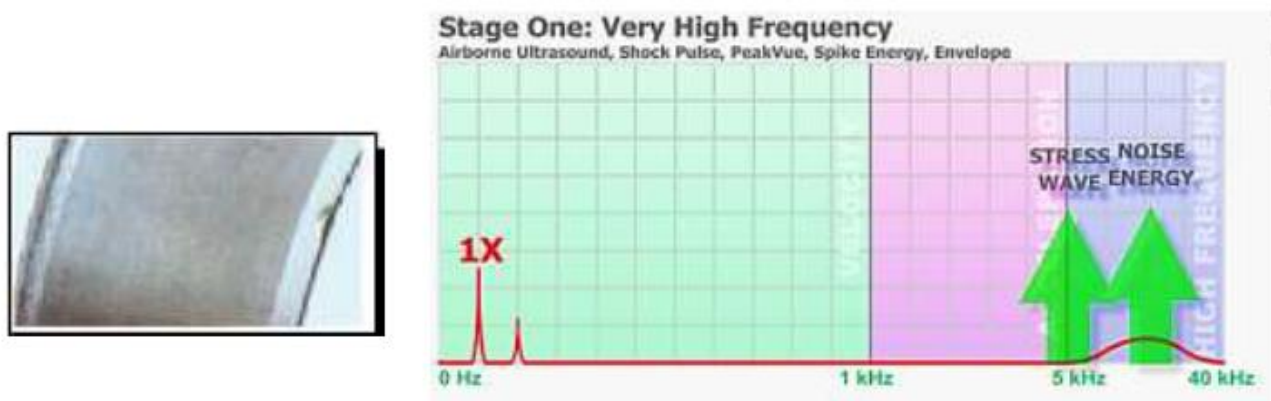


Figure 9.28. Stage one: very low amplitude-very high frequency vibration.

9.10.2 Stage Two

As the bearing fault develops, techniques such as enveloping, demodulation, and acceleration spectrum are more successful to use in this stage. The velocity spectrum still ineffective indicator to detect the faults in this stage. Small defects in the bearing begin to "ring" – they excite the natural frequencies of the bearing components, which are mainly in the range from 500 Hz to 2 kHz (see Figure 9.29). These may also be resonances of supporting parts of the bearing. At the end of the second stage, sidebands around the resonance peaks appear. This stage can be detected using demodulated high frequency envelope spectra.

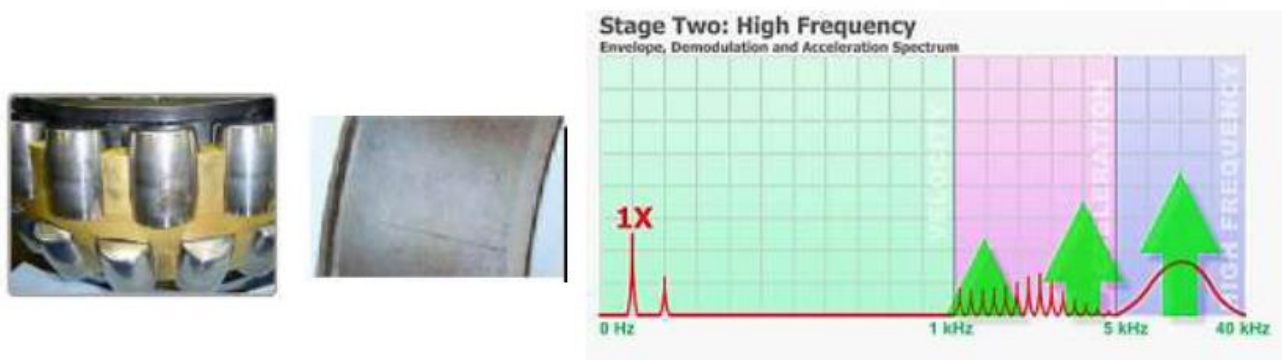


Figure 9.29. Stage two: Very low amplitude-high frequency vibration

9.10.3 Stage Three

As the wear increases, bearing defect frequencies and their harmonics occur in addition to sidebands around these harmonics. This stage can be determined from the vibration velocity spectra.

9.10.3.1 Stage Three: Outer Race

When the bearing fault reaches stage three, the damage is more severe and will be visible if the bearing is removed. The velocity spectrum can be used to detect the fault in addition to the time waveform and high frequency techniques (demodulation, enveloping, and acceleration spectra). If the damage is on the outer race, there will be an impact each time the ball or roller comes into contact with the damaged area and 1XBPFO and its harmonics appear as shown in Figure 9.30. The amplitude should be constant, therefore no modulation (or sidebands). If the outer race is rotating, 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 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 as shown in Figure 9.31.

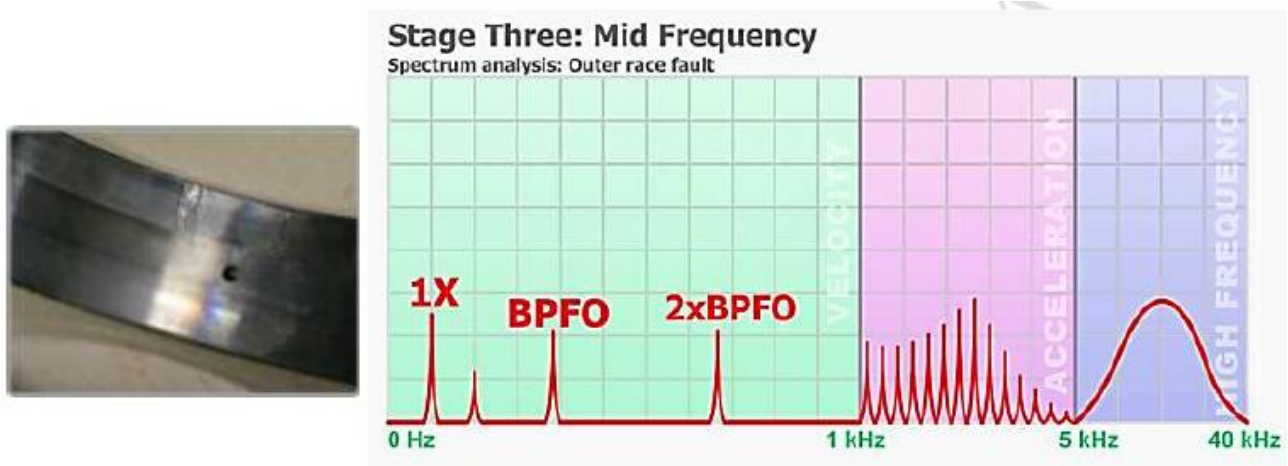


Figure 9.30. The pattern of outer race defect in FFT spectrum (case of inner race is rotating).

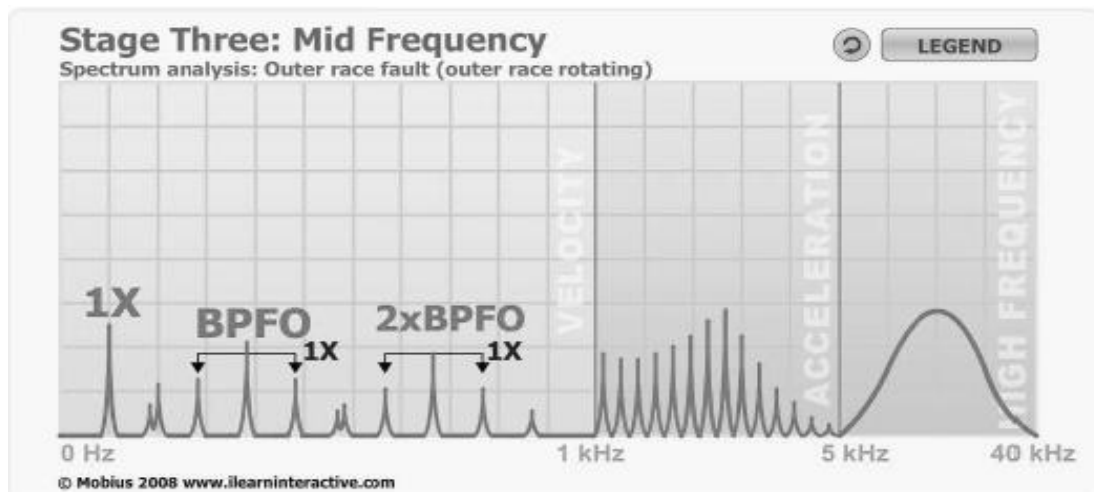


Figure 9.31. The pattern of outer race defect in FFT spectrum (case of outer race is rotating).

9.10.3.2 Stage Three: Inner Race

If the damage is on the inner race, there will be an impact each time the ball or roller comes into contact with the damaged area and 1XBPFI and its harmonics appear as shown in Figure 9.32. The amplitude will be highest when the damaged area is in the load zone, therefore there will be 1X sidebands.

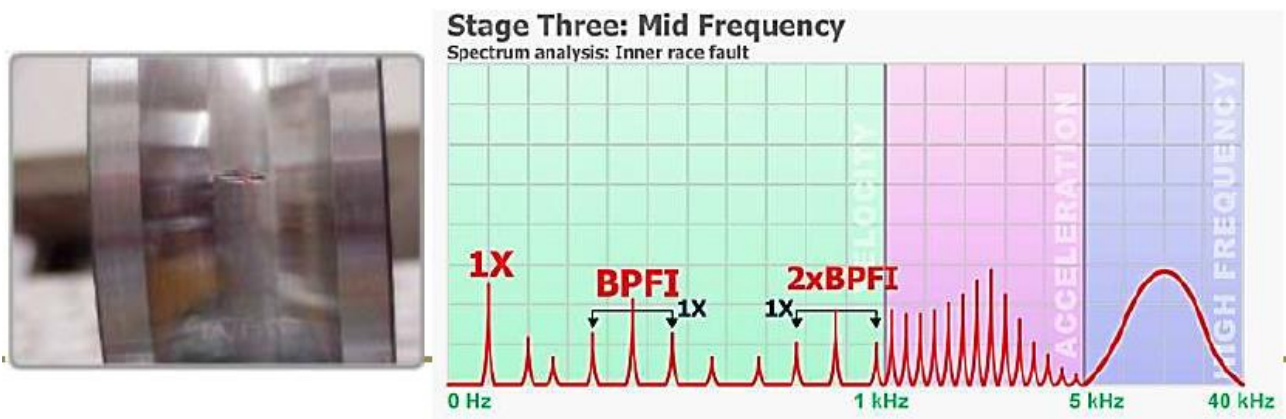


Figure 9.32. The pattern of inner race defect in FFT spectrum

9.10.3.3 Stage Three: Ball Damage

If the damage is on the balls or rollers, there will be an impact each time the damaged area comes into contact with the inner or outer race, so the defected ball appears at $1XBSF$ and its harmonics as shown in Figure 9.33. The amplitude will be highest when the damaged ball is in the load zone, therefore there will be FTF (cage) sidebands.

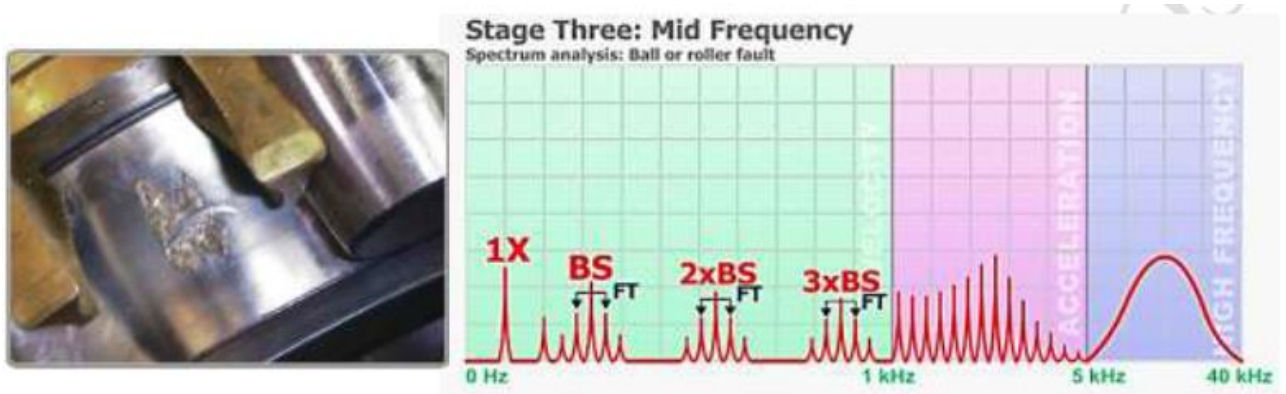


Figure 9.33. The pattern of roller defect in FFT spectrum

9.10.4 Stage Four

When the bearing fault reaches stage four, the bearing has significant damage and should be replaced. With time the tell-tale harmonics and sidebands may disappear. As the clearance in the bearing increases due to wear, you will see signs of looseness ($1X$ and its harmonics) as shown in Figure 9.34. High frequency techniques become less effective as the condition worsens. Overall levels will increase and the velocity spectrum will show the fault clearly. Spectrum will become very noisy and the noise floor will lift up ("haystacks" will appear in certain areas).

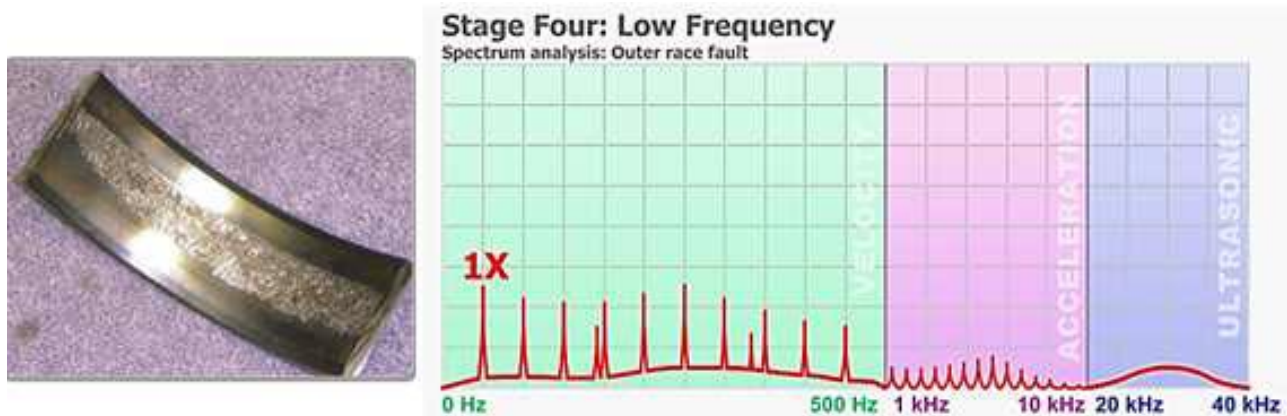


Figure 9.34. Stage four: outer race fault

9.10.5 Cocked Bearing: on Shaft

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 as shown in Figure 9.35. The bearing can be cocked on the shaft or in the housing. If it is cocked on the shaft, then it will "wobble" as it rotates generating a rotating vibration. Given that there is such a strong axial vibration, it can be confused with misalignment, and with unbalance in an overhung pump or fan. The presence of peaks at 2X and 3X would indicate a cocked bearing condition over unbalance. Relative phase measurement is the key indicator to differentiate between misalignment and cocked bearing.

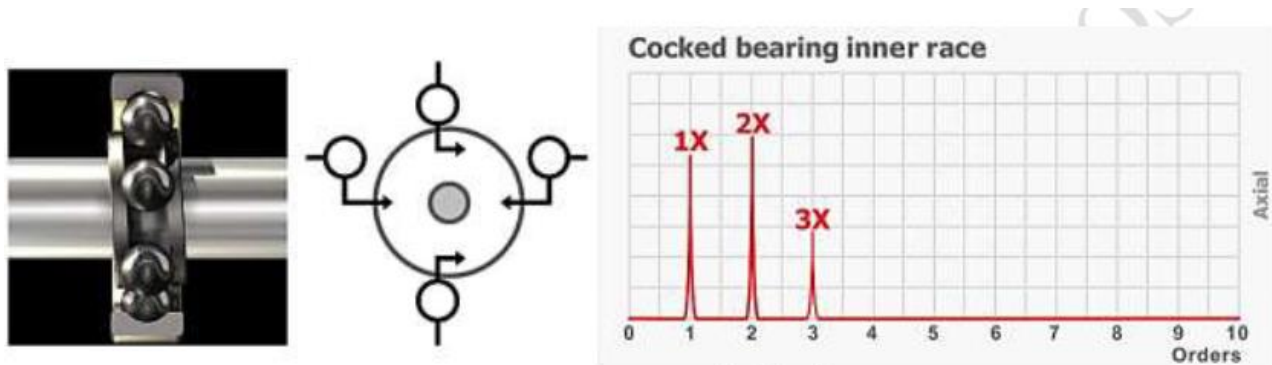


Figure 9.35. Cocked bearing on shaft at inner race

9.10.6 Cocked Bearing: in Housing

If the bearing is cocked in the housing, the vibration will more closely mimic misalignment. There will be two points on either side of the shaft on the face of the component (axial direction) where the amplitude will be the highest and the vibration will be 180° out-of-phase as displayed in Figure 9.36.

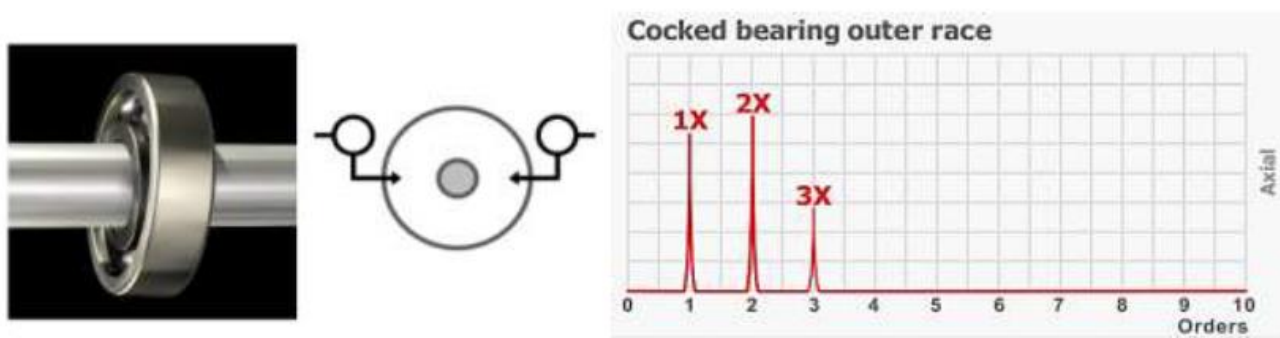


Figure 9.36. Cocked bearing in the housing

9.10.7 Inner Race Sliding on Shaft

If the bearing is not installed correctly, the inner race may slide on the shaft – i.e. it does not always turn at the same speed of the shaft. Depending upon the nature of the sliding, the vibration spectrum may have an elevated 3X and its harmonics (6X, 9X, etc.) as displayed in Figure 9.37. If the sliding motion is intermittent, i.e. it slides a little then stops, the change in the spectrum may not be observed. If it is constantly sliding, the vibration pattern should change accordingly.

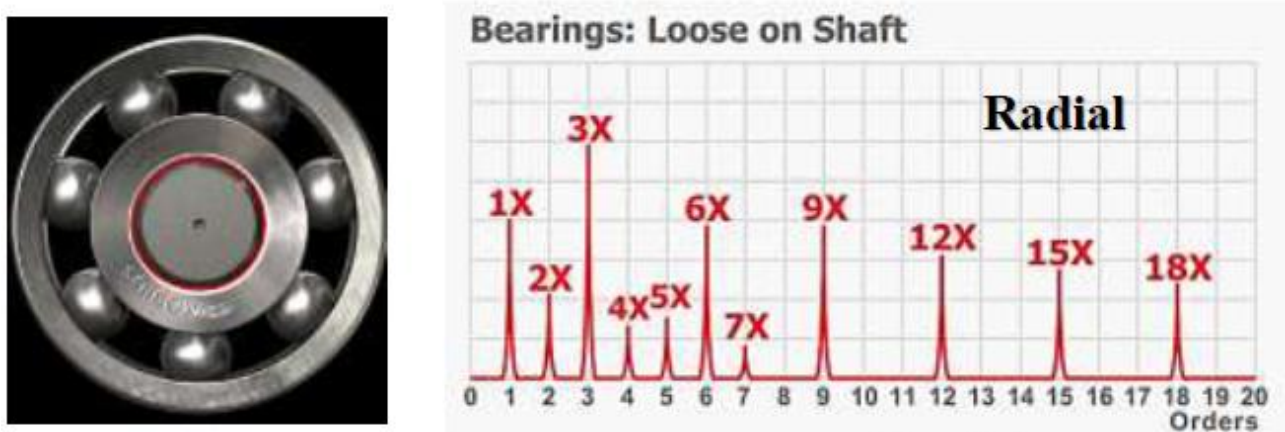


Figure 9.37. Bearing loose on shaft for inner race

9.10.8 Outer Race Loose in Housing

If the bearing is not installed correctly within the housing, the outer race may move relative to the housing; it may begin to spin and even "rattle" within the housing. The peaks at 4X running speed may be elevated as shown in Figure 9.38. If it is possible to view the bearing within the machine (by removing the cover if allowable), the outer race can be seen with a relative movement to the housing.



Bearings: Loose in Housing

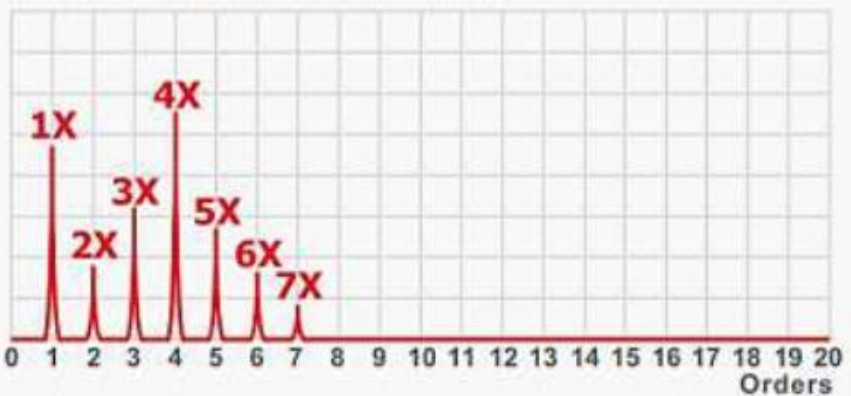


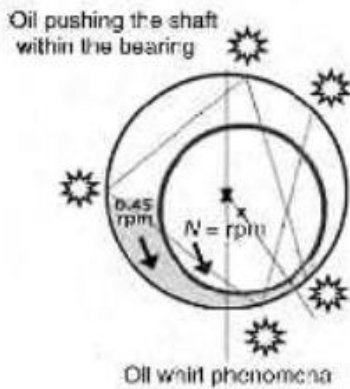
Figure 9.38. Bearing loose in housing

9.11 Oil Whirl

Oil whirl is a self-excited fluidic malfunction that typically occurs in plain sleeve bearings. It may occur for a number of reasons including:

- 1) Misalignment – bearing unloaded.
- 2) Excessive clearance
- 3) Lightly loaded, low damping

Oil whirl generates a characteristic source of vibration in the range of approximately 0.38X to 0.48X as shown in Figure 9.39.



Journal bearing oil whirl

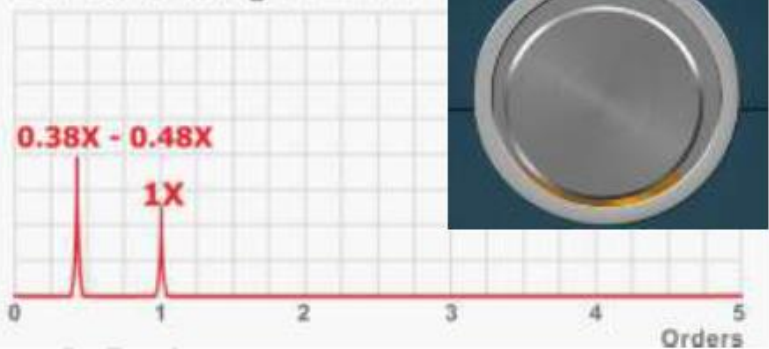


Figure 9.39. Oil whirl

9.12 Oil Whip

Oil whip is a very destructive condition. This condition can occur when the speed of the machine is over twice the critical speed (or the first balance rotor resonance); therefore, the critical speed is close to the oil whirl frequency. The oil whirl excites the resonance and a violent condition arises as shown in Figure 9.40. Vibration levels are very high and the machine must be stopped or else catastrophic failure may occur.

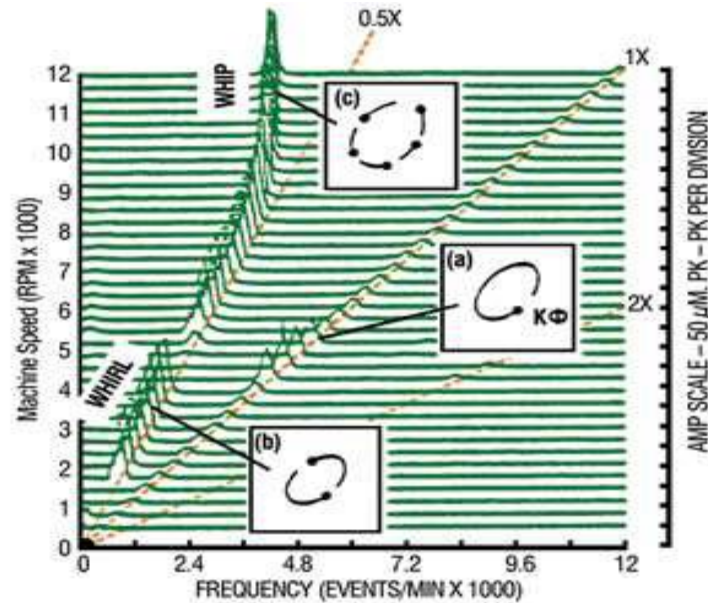


Figure 9.40. Oil whip

9.13 Electrical Problems

Vibrations of electrical machines such as motors, generators, and alternators can be either due to mechanical or electrical issues. Almost of mechanical problems have been discussed in above sections. Some of electrical defects can be investigated from the vibration spectrum such as:

- ☐ Cracked/broken rotor bar
- ☐ Eccentric rotor
- ☐ Eccentric stator / soft foot
- ☐ Phase loss
- ☐ Unequal air gap between stator and rotor

9.13.1 Cracked/Broken Rotor Bars

Cracked/broken rotor bars, shorted end rings or rotor laminations, or loose rotor bar joints develop a series of harmonics of turning speed with sidebands of the pole pass frequency (PPF) as displayed in Figure 9.41. Since the PPF has a very low frequency, it is necessary to use a high resolution spectrum. Also, audible and visible beating is appearing in the time waveform. There is another pattern can be observed with cracked/broken/loosed rotor bars which is two. Line frequency (2XLF) sidebands surrounding the rotor bar pass frequency (RBPF).

Hint:

PPF = No of poles × slip frequency

No of poles = 2 (3000 rpm) | 4 (1500 rpm) | 6 (1000 rpm)

Slip frequency = synchronized motor speed – actual speed

RBPF = Number of rotor bars × actual motor speed

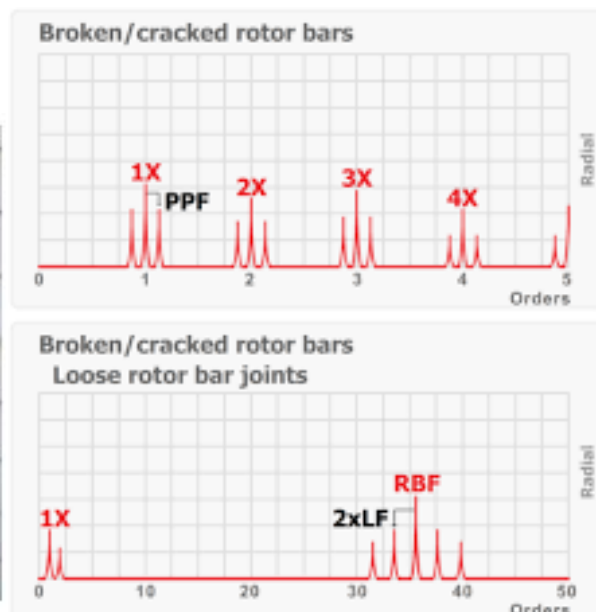


Figure 9.41. Broken/cracked/loose rotor bar

9.13.2 Eccentric Rotor

Eccentric rotors produce a rotating variable air gap between the rotor and stator which induces pulsating vibrations. The key indicator is the presence of the pole-pass sidebands around 1X and 2XLF as observed in Figure 9.42.

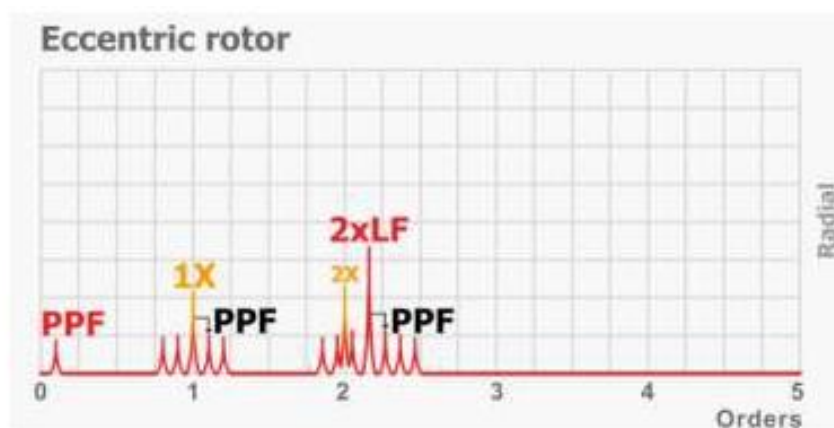
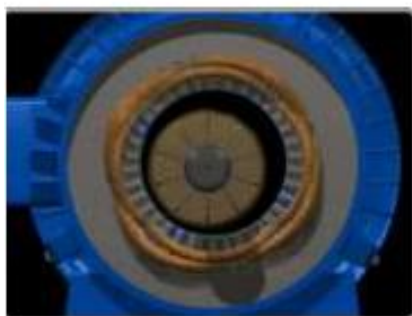


Figure 9.42. Eccentric rotor

9.13.3 Eccentric Stator / Soft Foot

Stator eccentricity produces an uneven stationary air gap between the rotor and stator and may be occurred due soft foot problem. This produces a very directional source of vibration generating from two-line frequency (100 or 120 Hz) as presented in Figure 9.43.



Figure 9.43. Eccentric stator

9.14 Cavitation

Cavitation is a phenomenon in which the static pressure of a liquid reduces to below the liquid's vapor pressure, leading to the formation of small vapor-filled cavities in the liquid. When subjected to higher pressure, these cavities, called "bubbles" or "voids", collapse and can generate shock waves that may damage machinery. These shock waves are strong when they are very close to the imploded bubble, but rapidly weaken as they propagate away from the implosion. Cavitation normally generates random and high-frequency broadband energy which is superimposed with the vane pass (VP) frequency and sometimes with its harmonics as shown in Figure 9.44.



Figure 9.44. Pump cavitation

Case Study (1): Cooling Fan - Severe fan bearing looseness



Analysis Results

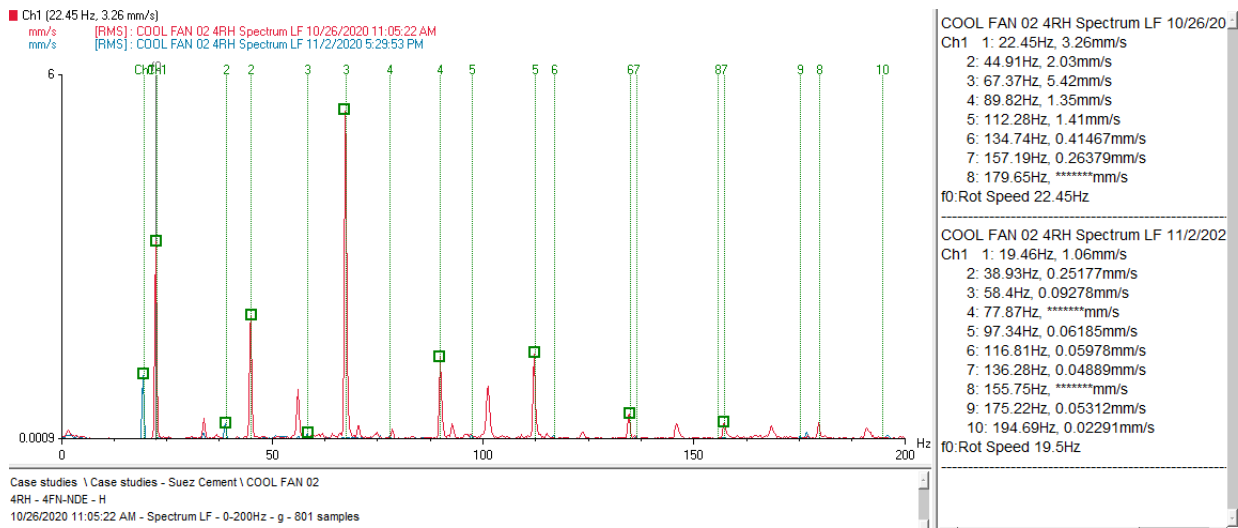
- There is high internal looseness at FNDE bearing leading to 1X harmonics in spectrum and envelope in addition to 1/2X harmonics in envelope.
- This looseness has a significant effect on the vibration levels of motor bearings and these levels were decreased after fan bearings replacement.
- Two fan bearings give the same pattern in spectrum and envelope and FNDE bearing takes the largest vibration levels

Important Diagnostic Vision

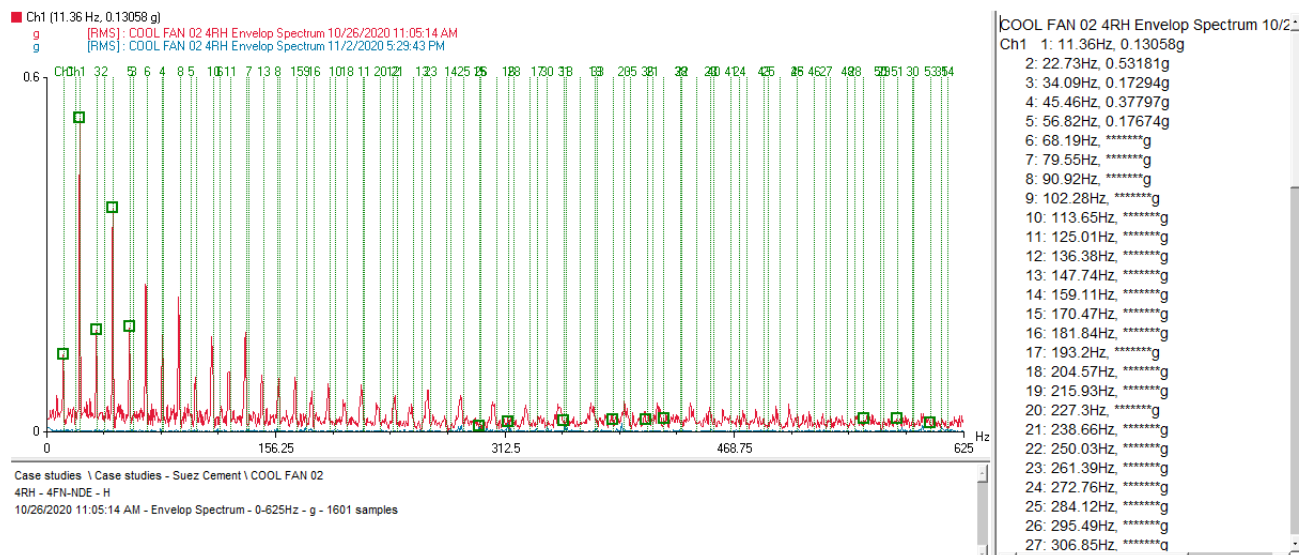
- It was found that the overall acceleration levels are low and the detected severe looseness has not effect on change the acceleration level.
- The best acceleration indicator is Pk - Pk acceleration since it reaches 49g before bearings replacement and reduced to 5g after replacement.
- The high vibration level on a healthy component (motor) is transmitted from another faulted component (fan).
- There is an observed increase in 3X compared to other harmonics.

Overall vibration level according to ISO 10816-3 and ISO 14694

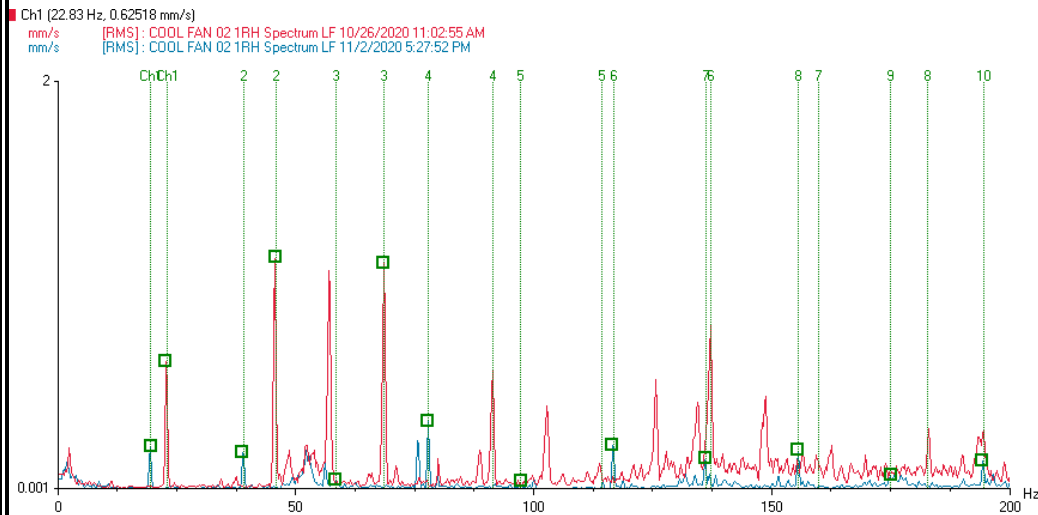
Cooling FAN	Value			Alarm threshold	
Point Param	Unit	Current 11/2/2020 RPM (19.5Hz)	Previous 10/26/2020 RPM (22.5Hz)	Alarm	Danger
1-MT-Ax					
Overall Velocity	mm/s	0.886	2.02	2.8	4.5
Overall Acceleration	g	0.463	0.704	2	4
1-MT-RH					
Overall Velocity	mm/s	1.22	3.95	2.8	4.5
Overall Acceleration	g	0.306	0.725	2	4
1-MT-RV					
Overall Velocity	mm/s	1.04	3.47	2.8	4.5
Overall Acceleration	g	0.267	0.502	2	4
2-MT-Ax					
Overall Velocity	mm/s	0.522	2.07	2.8	4.5
Overall Acceleration	g	0.351	0.354	2	4
2-MT-RH					
Overall Velocity	mm/s	1.59	3.28	2.8	4.5
Overall Acceleration	g	0.272	0.313	2	4
2-MT-RV					
Overall Velocity	mm/s	2.26	3.22	2.8	4.5
Overall Acceleration	g	0.559	0.458	2	4
3-FN-Ax					
Overall Velocity	mm/s	0.849	1.74	7.1	9
Overall Acceleration	g	0.236	0.608	4	8
3-FN-RH					
Overall Velocity	mm/s	0.789	3.11	7.1	9
Overall Acceleration	g	0.467	1.03	4	8
3-FN-RV					
Overall Velocity	mm/s	0.644	1.75	7.1	9
Overall Acceleration	g	0.293	0.683	4	8
4-FN-Ax					
Overall Velocity	mm/s	0.725	3.43	7.1	9
Overall Acceleration	g	0.367	1.12	4	8
4-FN-RH					
Overall Velocity	mm/s	1.23	8.25	7.1	9
Overall Acceleration	g	0.805	1.86	4	8
4-FN-RV					
Overall Velocity	mm/s	1.04	5.16	7.1	9
Overall Acceleration	g	0.319	0.942	4	8



Frequency Spectrum - FNDE - Horizontal Direction



Envelope Spectrum - FNDE - Horizontal Direction



COOL FAN 02 1RH Spectrum LF 10/26/2020

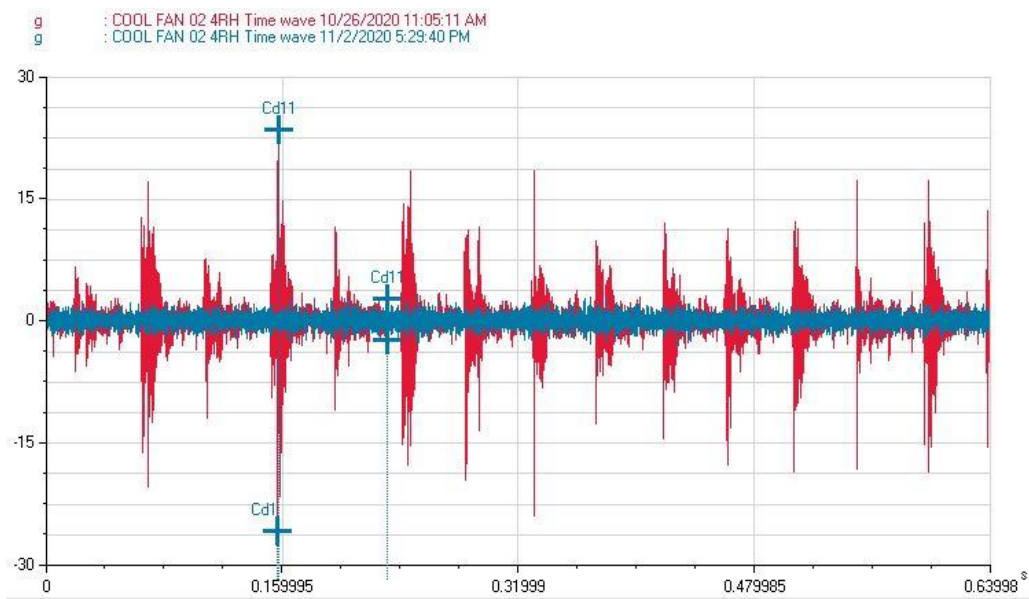
Ch1 1: 22.83Hz, 0.62518mm/s
2: 45.67Hz, 1.13mm/s
3: 68.5Hz, 1.11mm/s
4: 91.34Hz, *****mm/s
5: 114.17Hz, *****mm/s
6: 137.01Hz, *****mm/s
7: 159.84Hz, *****mm/s
8: 182.68Hz, *****mm/s

COOL FAN 02 1RH Spectrum LF 11/2/2020

Ch1 1: 19.43Hz, 0.20453mm/s
2: 38.86Hz, 0.18008mm/s
3: 58.3Hz, 0.03975mm/s
4: 77.73Hz, 0.33276mm/s
5: 97.17Hz, 0.03488mm/s
6: 116.6Hz, 0.21413mm/s
7: 136.03Hz, 0.14694mm/s
8: 155.47Hz, 0.18794mm/s
9: 174.9Hz, 0.06811mm/s
10: 194.34Hz, 0.13809mm/s

Case studies \ Case studies - Suez Cement \ COOL FAN 02
1RH - 1MT-NDE - H
10/26/2020 11:02:55 AM - Spectrum LF - 0-200Hz - g - 801 samples

Spectrum - MNDE - Horizontal Direction



COOL FAN 02 4RH Time wave 10/26/2020

Cd1 0.15703s
-25.933g
0.15822s
23.36g
dX:[0.00119s, 839.34Hz]
dY:49.3g (-190.1%)

COOL FAN 02 4RH Time wave 11/2/2020

Cd1 0.2315s
-2.5081g
0.23166s
2.66g
dX:[0.00015s, 6399.85Hz]
dY:5.16g (-206.07%)

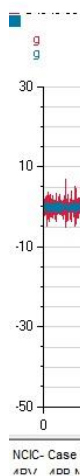
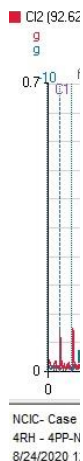
Case Study (2): Water Pump – Pump bearing defect



Analysis Results

- It was observed an increase in the acceleration on pump bearings
- High acceleration is resulting from 2X ball spin frequency and sidebands with cage frequency
- Shocks in time wave resulting from 2X ball spin frequency and cage frequency
- Another measurement was performed on 26th December, 2020 after pump bearing replacement and it was found a superior improvement

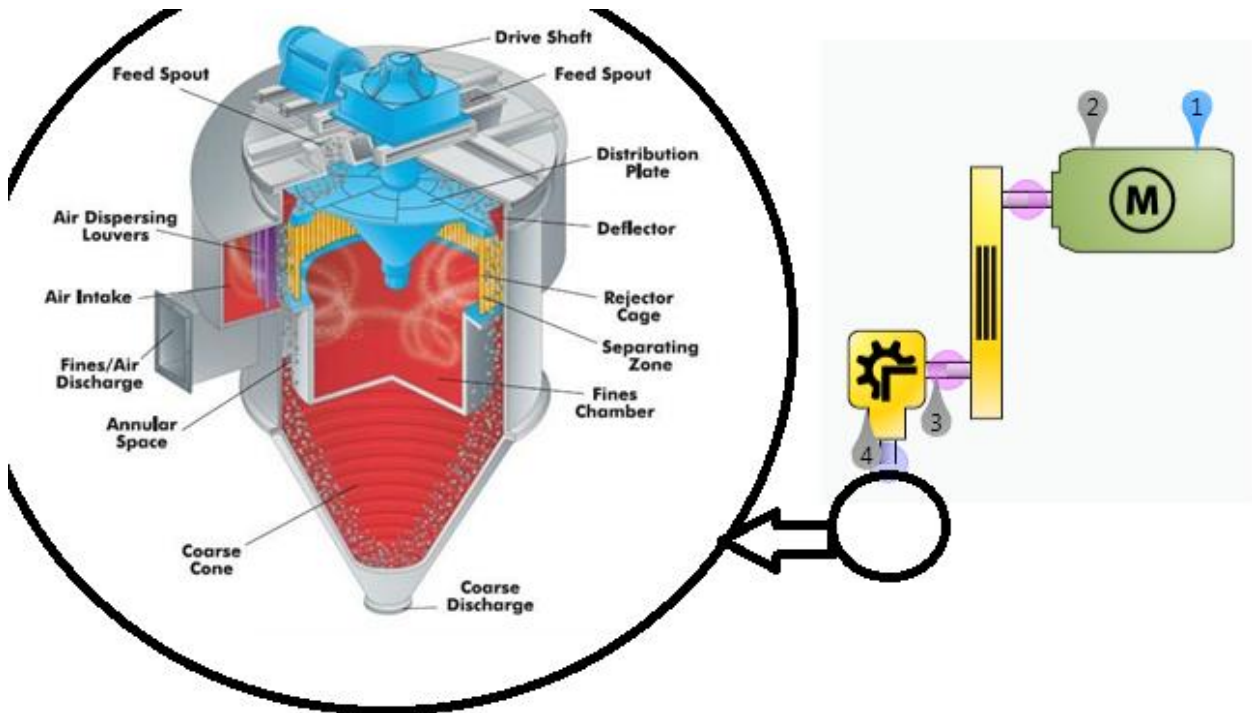
30-CTP 02			Value		Alarm threshold	
Point	Status	Unit	Current 12/26/2020 11:38 AM	Previous 8/24/2020 12:00 PM	Al	Dg
Param						
1-MT-Ax						
Overall Vib Vel	Ok	mm/s	0.984	0.962	2.8	4.5
Overall Acc	Ok	g	0.101	0.326	2	4
1-MT-RH						
Overall Vib Vel	Ok	mm/s	0.716	2.34	2.8	4.5
Overall Acc	Ok	g	0.185	0.573	2	4
1-MT-RV						
Overall Vib Vel	Ok	mm/s	0.863	0.927	2.8	4.5
Overall Acc	Ok	g	0.163	0.419	2	4
2-MT-Ax						
Overall Vib Vel	Ok	mm/s	1.27	1.18	2.8	4.5
Overall Acc	Ok	g	0.116	0.205	2	4
2-MT-RH						
Overall Vib Vel	pAl	mm/s	1.64	3.08	2.8	4.5
Overall Acc	Ok	g	0.311	0.207	2	4
2-MT-RV						
Overall Vib Vel	pAl	mm/s	2.71	1.6	2.8	4.5
Overall Acc	Ok	g	0.198	0.184	2	4
3-PP-Ax						
Overall Vib Vel	pAl	mm/s	1.62	1.72	2.8	4.5
Overall Acc	Ok	g	0.565	2.65	2	4
3-PP-RH						
Overall Vib Vel	pAl	mm/s	1.59	3.51	2.8	4.5
Overall Acc	Ok	g	0.72	2.05	2	4
3-PP-RV						
Overall Vib Vel	Ok	mm/s	0.692	1.74	2.8	4.5
Overall Acc	Ok	g	0.614	2.18	2	4
4-PP-Ax						
Overall Vib Vel	Ok	mm/s	1.17	1.79	2.8	4.5
Overall Acc	Ok	g	0.715	2.92	2	4
4-PP-RH						
Overall Vib Vel	Ok	mm/s	0.789	1.62	2.8	4.5
Overall Acc	Ok	g	0.867	3.32	2	4
4-PP-RV						
Overall Vib Vel	Ok	mm/s	0.461	1.24	2.8	4.5
Overall Acc	Ok	g	0.658	2.5	2	4



Case Study (3): Dynamic Separator - Severe wear in separator blades/fins

Definition

The main function of separator is to divide a given material path into two separated paths using air as the carrying medium. One path contains only fine particles, and the other only coarse particles. A distribution plate was used to guide the feed material into the separating space. The mechanical air separator generates the circulating air inside the separator itself which employed as the classifying force



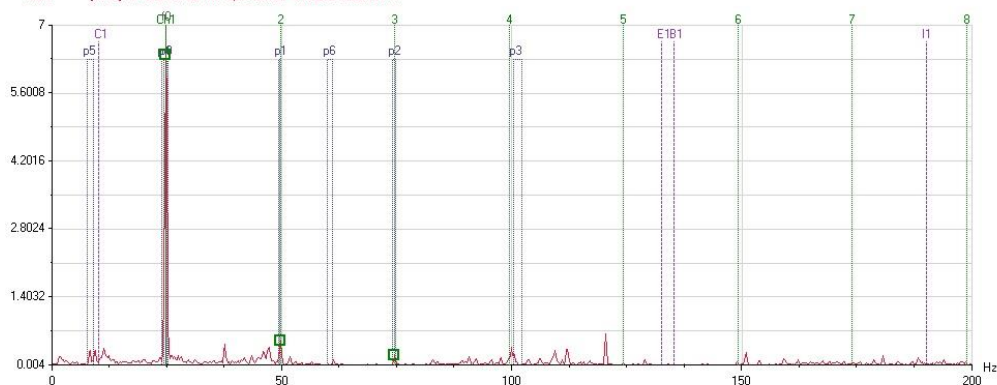
Analysis Results

- 1- The overall vibration level is not accepted on motor and gearbox according to ISO 10816-3.
- 2- Motor: - The main source of vibration is 1X motor speed.
- 3- Gearbox: Blade pass frequency (60.28Hz) with several harmonics in addition to sidebands with 1X input gear speed. Also, there are shocks with blade pass frequency
- 4- After inspection, it was found that the blades/fins of separator impeller have severe wear

Overall vibration level according to ISO 10816-3

Dynamic Separator			Value	Alarm threshold	
Point Parameter	Status	Unit	Current 4/4/2021 2:05 PM	Alarm	Danger
	1-MT-Ax				
Overall Vibration Velocity	Dg	mm/s	8.06	2.8	4.5
	1-MT-RH				
Overall Vibration Velocity	pAl	mm/s	2.07	2.8	4.5
	1-MT-RV				
Overall Vibration Velocity	Dg	mm/s	7.57	2.8	4.5
	2-MT-Ax				
Overall Vibration Velocity	Dg	mm/s	7.07	2.8	4.5
	2-MT-RH				
Overall Vibration Velocity	pAl	mm/s	1.71	2.8	4.5
	2-MT-RV				
Overall Vibration Velocity	Al	mm/s	2.91	2.8	4.5
	3-ANG-Ax				
Overall Vibration Velocity	Al	mm/s	3.95	2.8	4.5
	3-ANG-RH				
Overall Vibration Velocity	Dg	mm/s	8.17	2.8	4.5
	3-ANG-RV				
Overall Vibration Velocity	Dg	mm/s	8.31	2.8	4.5
	4-ANG-Ax				
Overall Vibration Velocity	Al	mm/s	3.98	2.8	4.5
	4-ANG-R=				
Overall Vibration Velocity	Dg	mm/s	5.02	2.8	4.5
	4-ANG-RT				
Overall Vibration Velocity	Dg	mm/s	5.4	2.8	4.5

■ Ch1 (24.84 Hz, 6.37 mm/s)
mm/s [RMS]: RM2 D. SP2x 1RV Spectrum LF 4/4/2021 2:02:15 PM

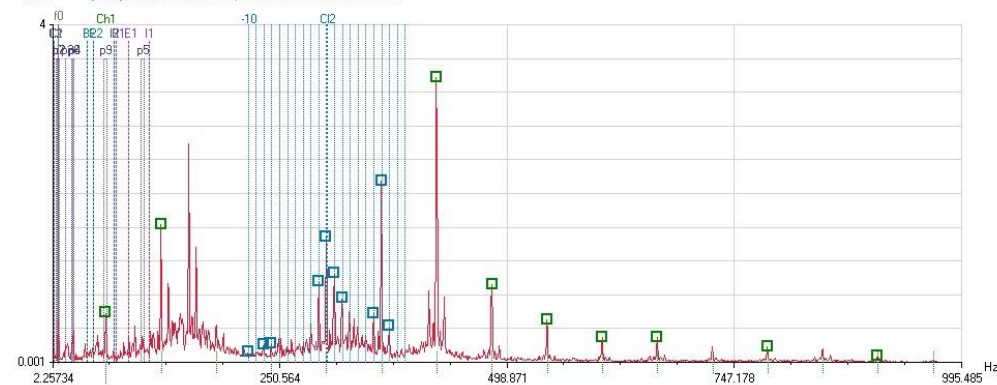


Case studies \ RM2 D. SP2x
1RV - 1MT-NDE - V
4/4/2021 2:02:15 PM - Spectrum LF - 0-200Hz - g - 801 samples

RM2 D. SP2x 1RV Spectrum LF 4/4/2021 2:02:15 PM
Ch1 1: 24.84Hz, 6.37mm/s
2: 49.69Hz, 0.49966mm/s
3: 74.53Hz, 0.19983mm/s
4: 99.38Hz, *****mm/s
5: 124.23Hz, 0.05242mm/s
6: 149.07Hz, 0.06551mm/s
7: 173.92Hz, *****mm/s
8: 198.77Hz, *****mm/s
f0: Rot Speed 24.84Hz
NU320 [SKF] I1: 190.27Hz E1: 132.64Hz
B1: 135.37Hz C1: 10.18Hz
p0: 1X 24.84Hz (+/- 0.5Hz)
p1: 2X 49.68Hz (+/- 0.25Hz)
p2: 3X 74.52Hz (+/- 0.5Hz)
p3: GMF (101.5Hz) 101.5Hz (+/- 1Hz)
p4: 1X motor 24.75Hz (+/- 0.75Hz)
p5: 1X (input gear 8.56Hz) 8.56Hz (+/- 0.75Hz)
p6: Blade PF (20blades) 60.59Hz (+/- 0.75Hz)

Motor NDE - RV

■ C12 (301.54 Hz, 1.48 mm/s) $d\lambda=8.75$ Hz
mm/s [RMS]: RM2 D. SP2x 3RH Spectrum MF 4/4/2021 2:04:09 PM

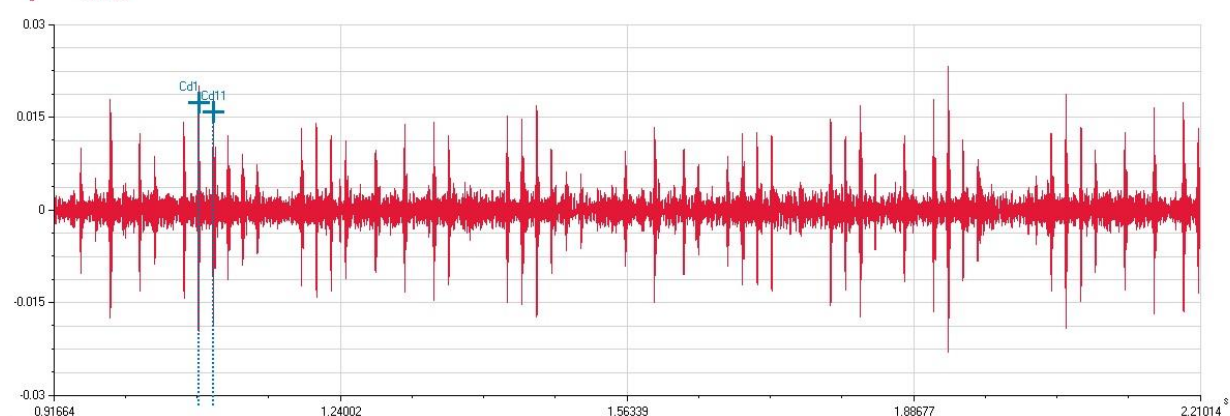


Case studies \ RM2 D. SP2x
3RH - 3ANG-InDE - H
4/4/2021 2:04:09 PM - Spectrum MF - 0-2000Hz - g - 3201 samples

RM2 D. SP2x 3RH Spectrum MF 4/4/2021 2:04:09 PM
Ch1 1: 60.28Hz, 0.58837mm/s
2: 120.57Hz, 1.63mm/s
3: 180.85Hz, *****mm/s
4: 241.14Hz, 0.21968mm/s
5: 301.42Hz, 1.48mm/s
6: 361.71Hz, 2.15mm/s
7: 421.99Hz, 3.37mm/s
8: 482.28Hz, 0.92045mm/s
9: 542.56Hz, 0.50314mm/s
10: 602.85Hz, 0.2993mm/s
11: 663.13Hz, 0.29748mm/s
12: 723.42Hz, *****mm/s
13: 783.71Hz, 0.18646mm/s
14: 843.99Hz, *****mm/s
15: 904.28Hz, 0.07893mm/s
16: 964.56Hz, *****mm/s
17: 1024.85Hz, 0.02647mm/s
18: 1085.13Hz, *****mm/s
19: 1145.42Hz, 0.0197mm/s
20: 1205.7Hz, 0.01086mm/s
21: 1265.99Hz, *****mm/s
22: 1326.27Hz, *****mm/s
23: 1386.56Hz, *****mm/s
24: 1446.85Hz, *****mm/s
25: 1507.13Hz, *****mm/s
26: 1567.42Hz, *****mm/s
27: 1627.7Hz, *****mm/s

Separator DE - RH

■ Cd1 (1.08 s, 0.01736 g) Cd11 (1.03 s, 0.01584 g) $d\lambda=0.01656$ s [60.37 Hz] $d\gamma=0.0015$ g [-8%]
g : SFI Filter



Case studies \ RM2 D. SP2x
4RT - 4ANG-OutDE - P

Separator NDE - RT

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