Vibration Analysis

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December 3, 2017

Vibration Certifications

There are two institutes that gave certification in vibration: Vibration Institute and Mobius institute. Each of them classify the vibration analyst to four categories. Category IV is the most expert person and should have 5 years of vibration as a vibration analyst.

Vibration Institute

Mobius Institute

Vibration Certifications

1. Vibration Institute

Vibration Analyst Category	Recommended Experience for Exam Eligibility	Courses Recommended as Partial Preparation for Certification Exams* **						
		IMV	BMV	BRM	MVA	AVC	RDM	AVA
Category I	≥ 6 months	1						
Category II	≥ 18 months		2					
Category III	≥ 3 years			3	4			
Category IV	≥ 5 years					5	6	7
Training Course Cost		$1150	$1350	$1450	$1550	$1750	$1750	$1750

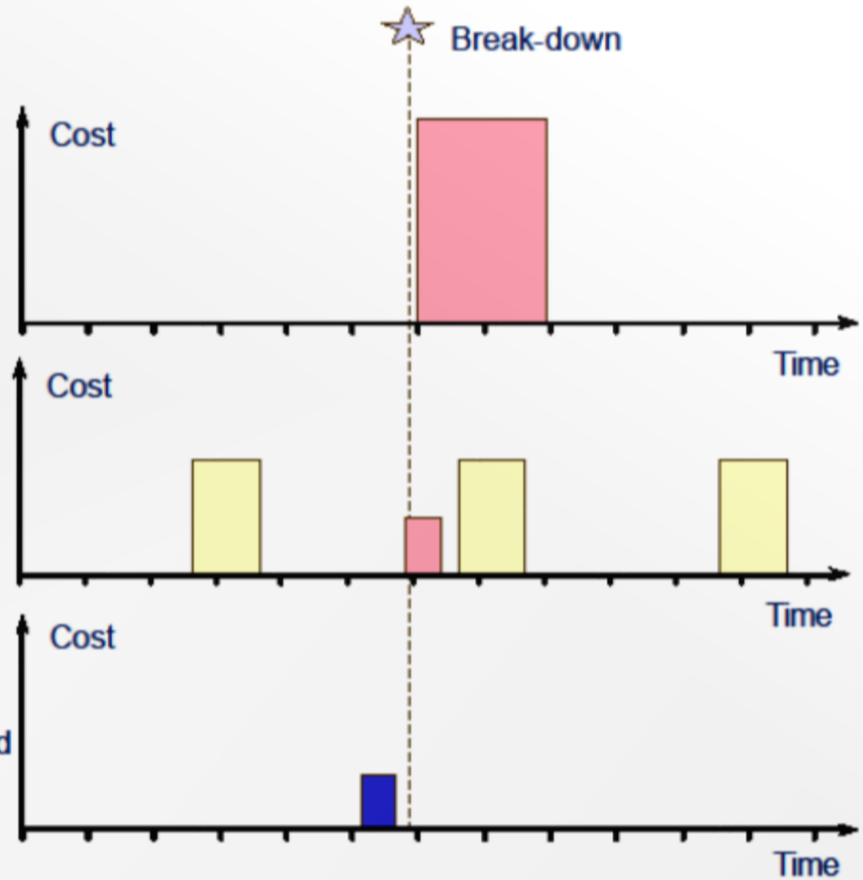
Types of Maintenance

Corrective Maintenance
(Run-to-breakdown)

Repair it when it fails

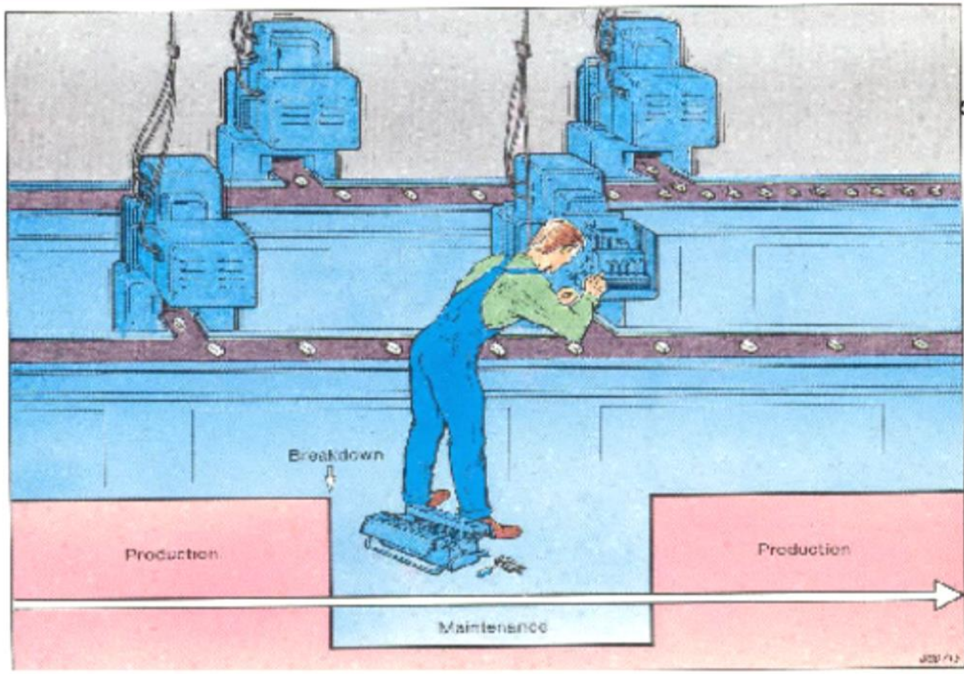
Preventive Maintenance
(Time Based Maintenance)
Maintenance at regular intervals

Predictive Maintenance
(On Condition Maintenance)
Problem detected before predicted failure.
Maintenance planned ahead



Corrective Maintenance

-Run to Breakdown-



Not recommended for critical machines

Corrective Maintenance leads to:

- **Secondary damage**
- **Safety risk**
- **Unplanned downtime**
- **Unplanned maintenance**
- **Product waste**
- **Spares inventory**

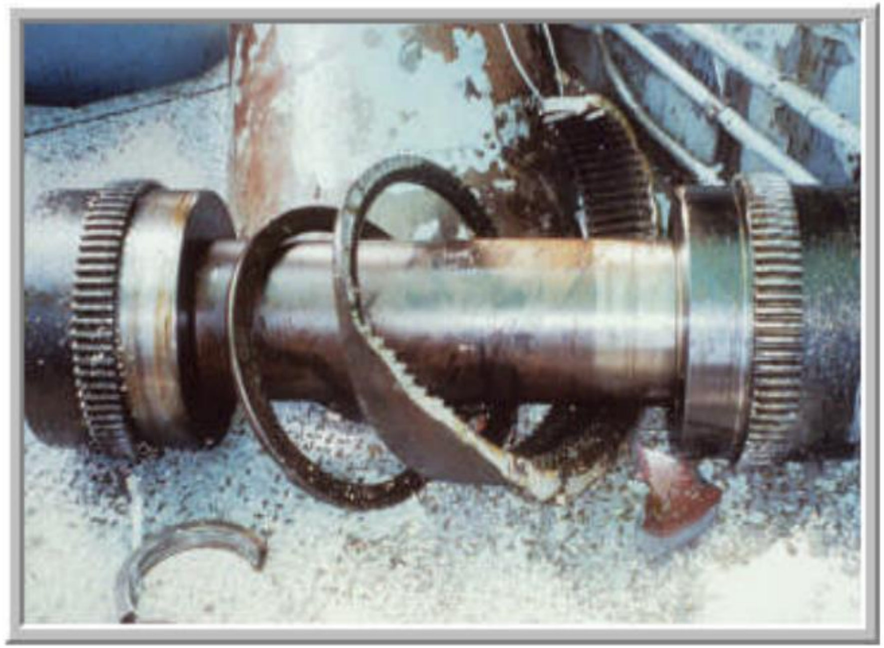
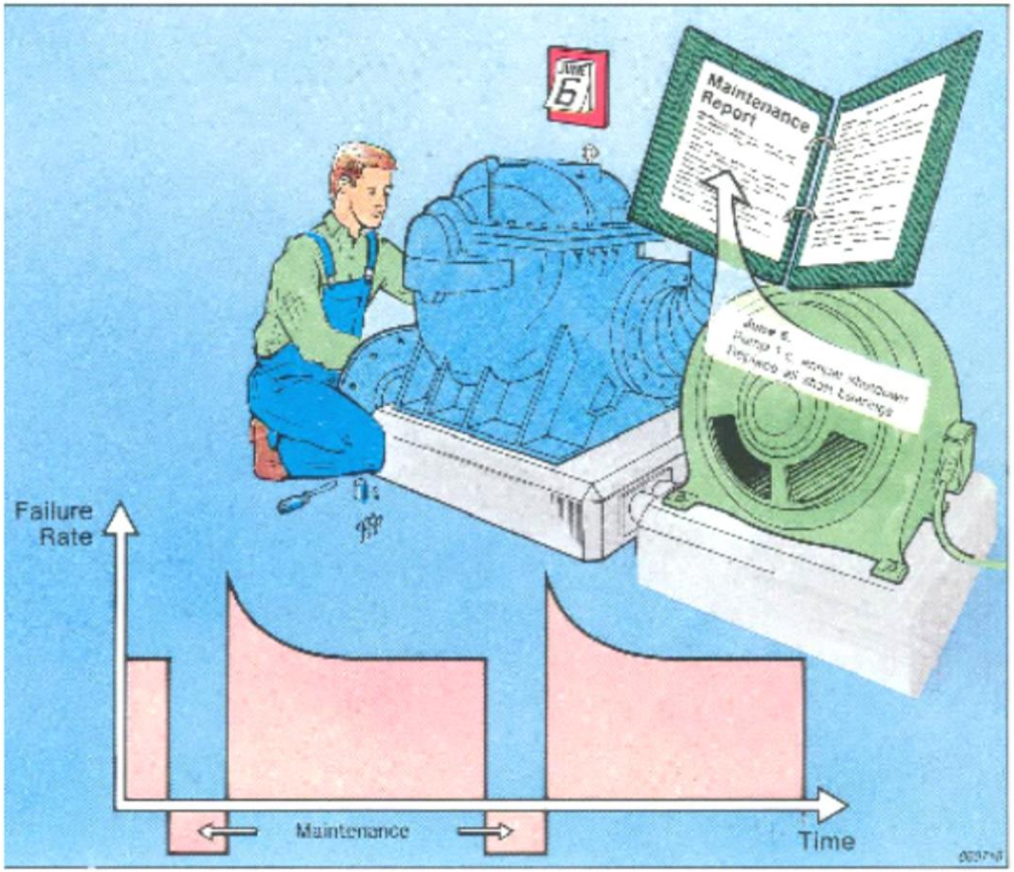


Figure 1-9 Run to Failure maintenance practices leads to very high maintenance costs.

Preventive Maintenance

Time Based Maintenance

Not recommended for critical machines

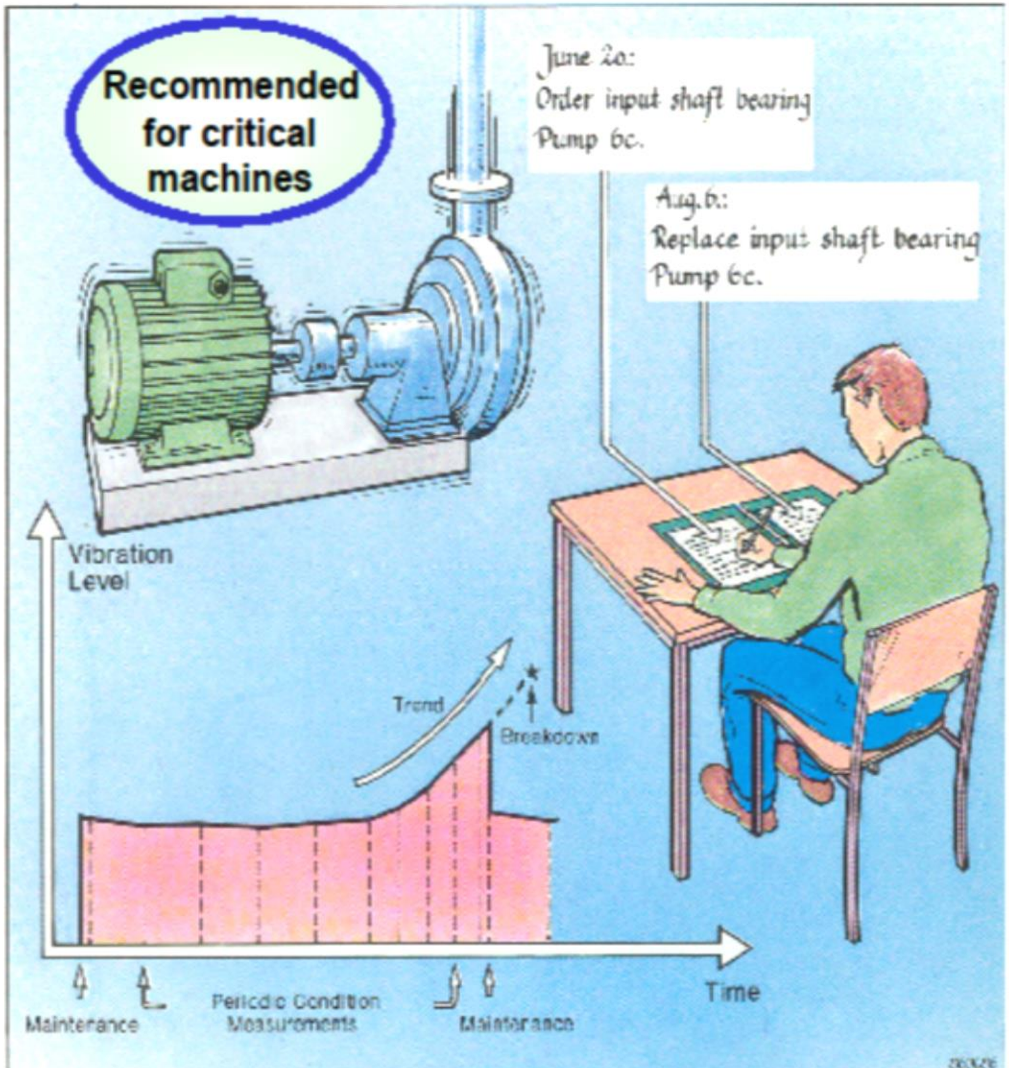


Time-based Preventive Maintenance involves:

- **More frequent overhauls**
- **Risk of early failures**
- **Tampering with good machines**
- **Time consuming overhauls**
- **Experts needed for each overhaul**

Predictive Maintenance

On-condition -Maintenance



- Monitor the condition of the machine and predict when it would fail
- Plan maintenance ahead of time and save money
- Repair the machines only when they need to
- Focus overhauls only on faulty parts

- Higher plant availability, performance and reliability
- Greater safety
- Better product quality
- Attention to environment
- Longer equipment life
- Greater cost effectiveness

Figure 1-8 New philosophy…it is worth the time

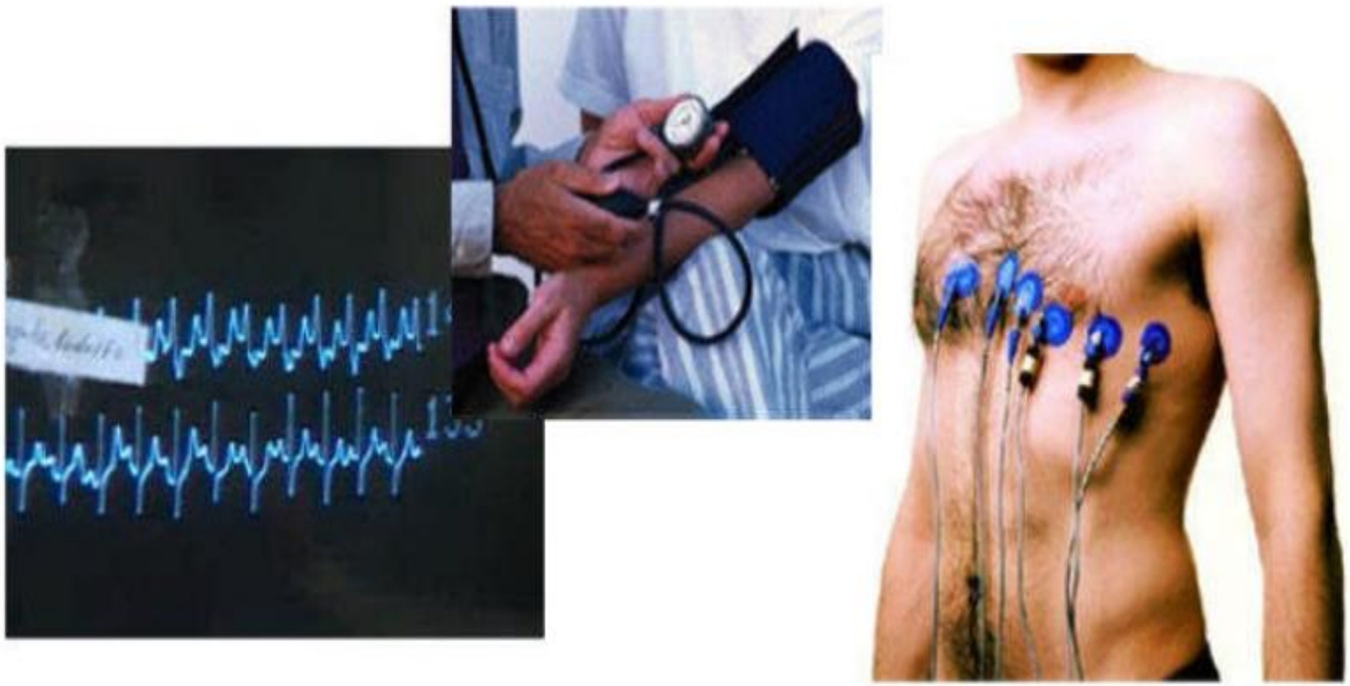


Figure 2-1 Various technologies are used to determine the state of our health

Infrared Camera

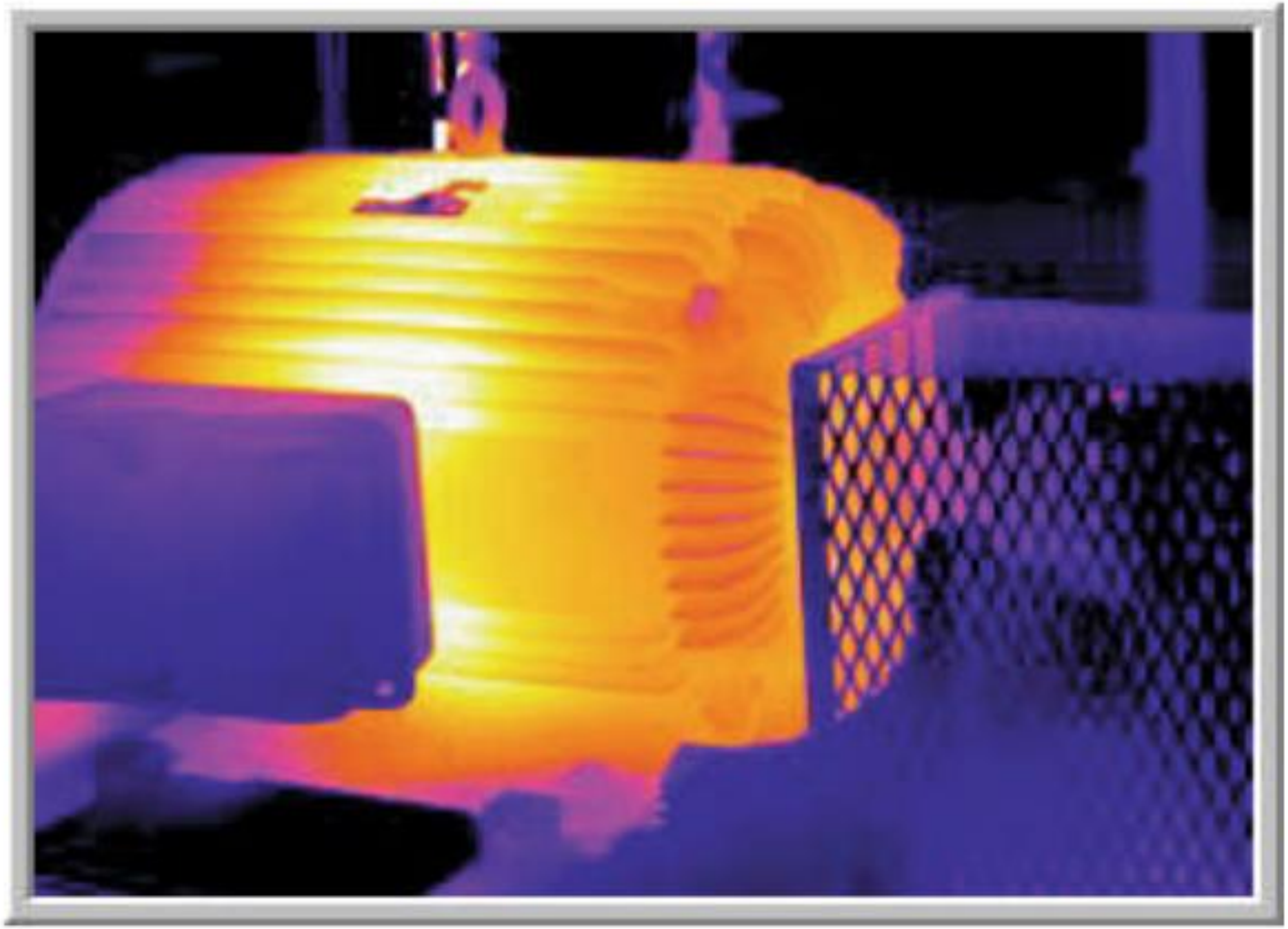


Figure 1-17 Infrared Thermography image of overheated motor

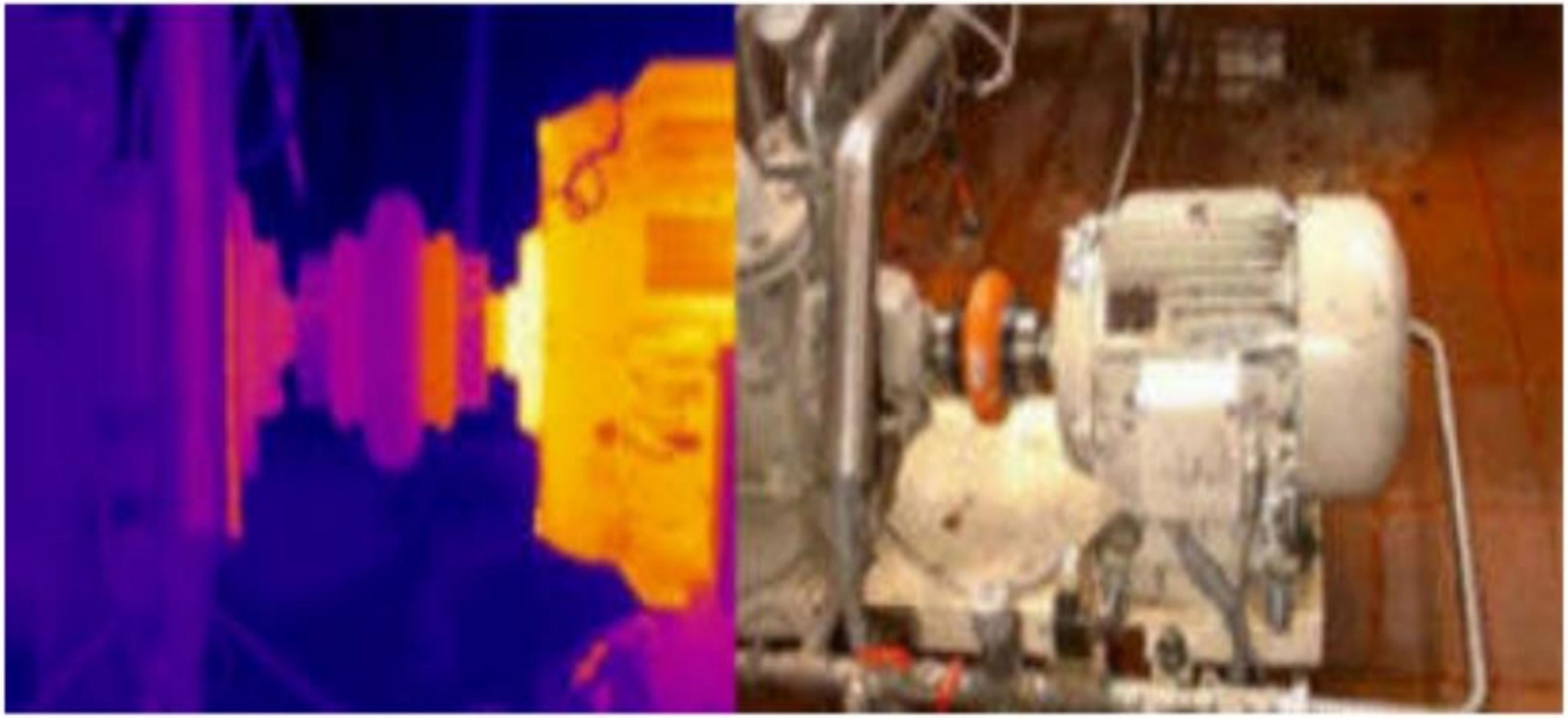


Figure 2-20 Thermographic and photographic image of overheated bearing

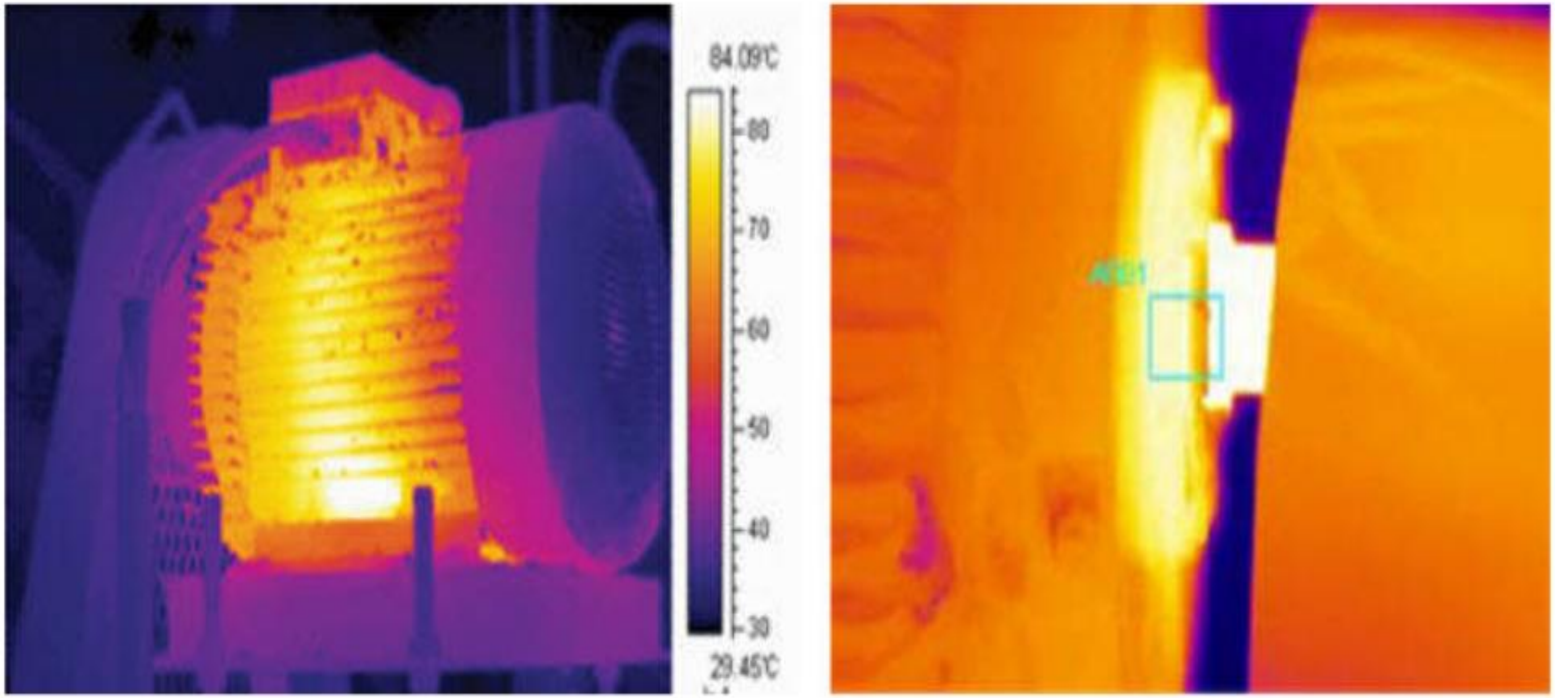


Figure 2-31 Thermographic and Photographic image of failed steam trap. Note the delta temperature of 20 degrees.

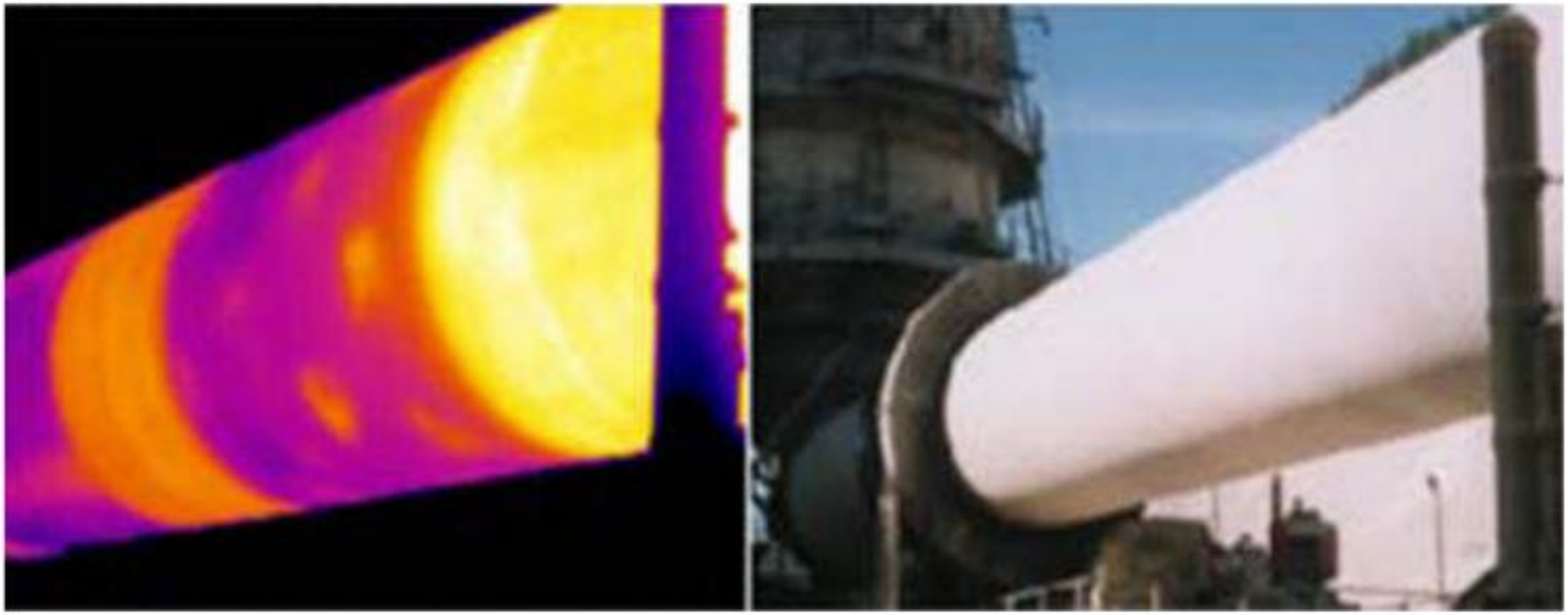


Figure 2-33 Lime kiln with hot areas indicating poor insulation.

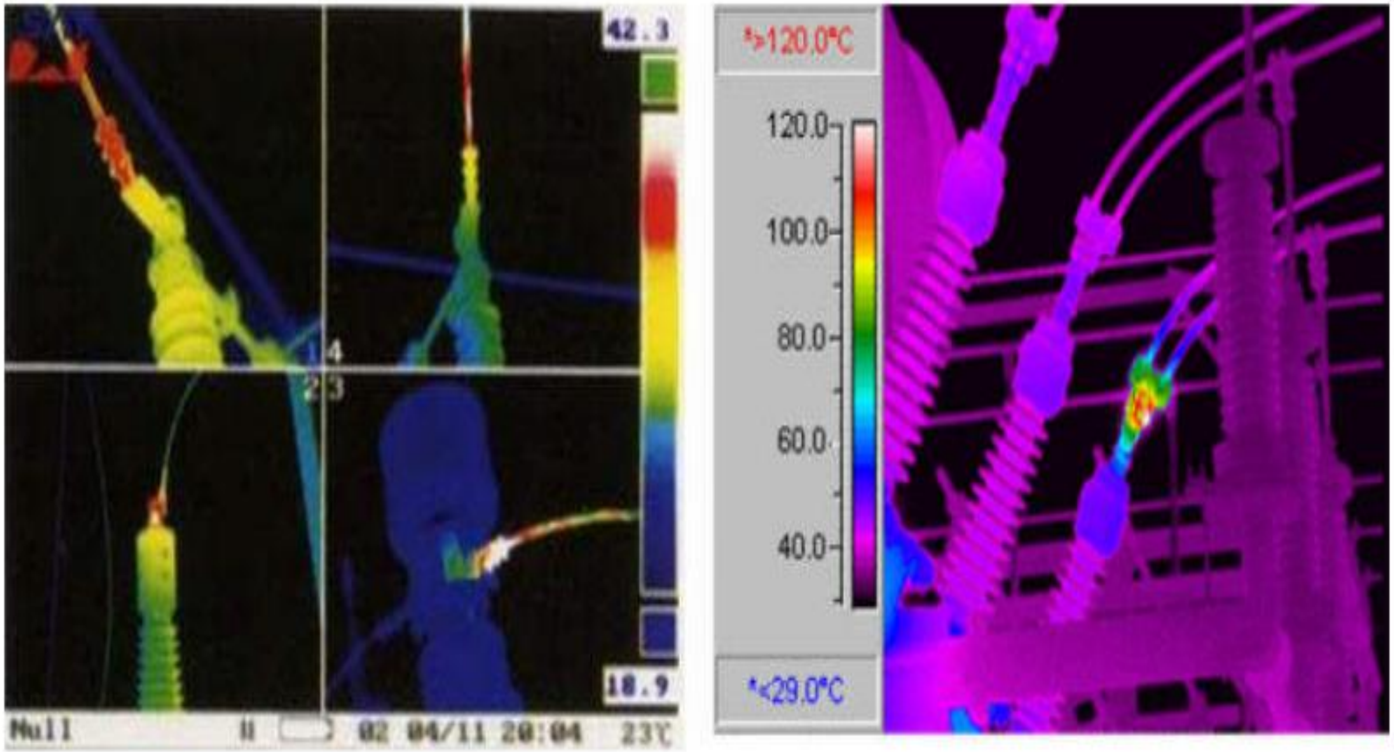
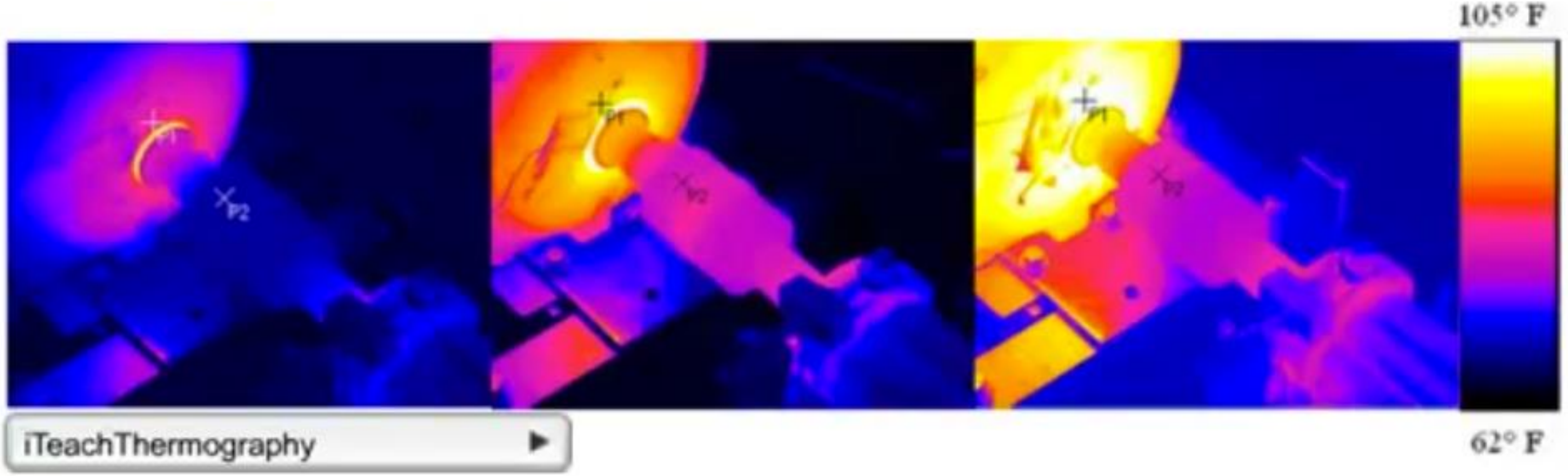


Figure 2-27 Loose connections and overloading can be detected. Overcome the effects of sun by scanning at night.


105° F
P1
P2
62° F
iTeachThermography

Oil Analysis

Figure 2-40

Oil Particle Analysis

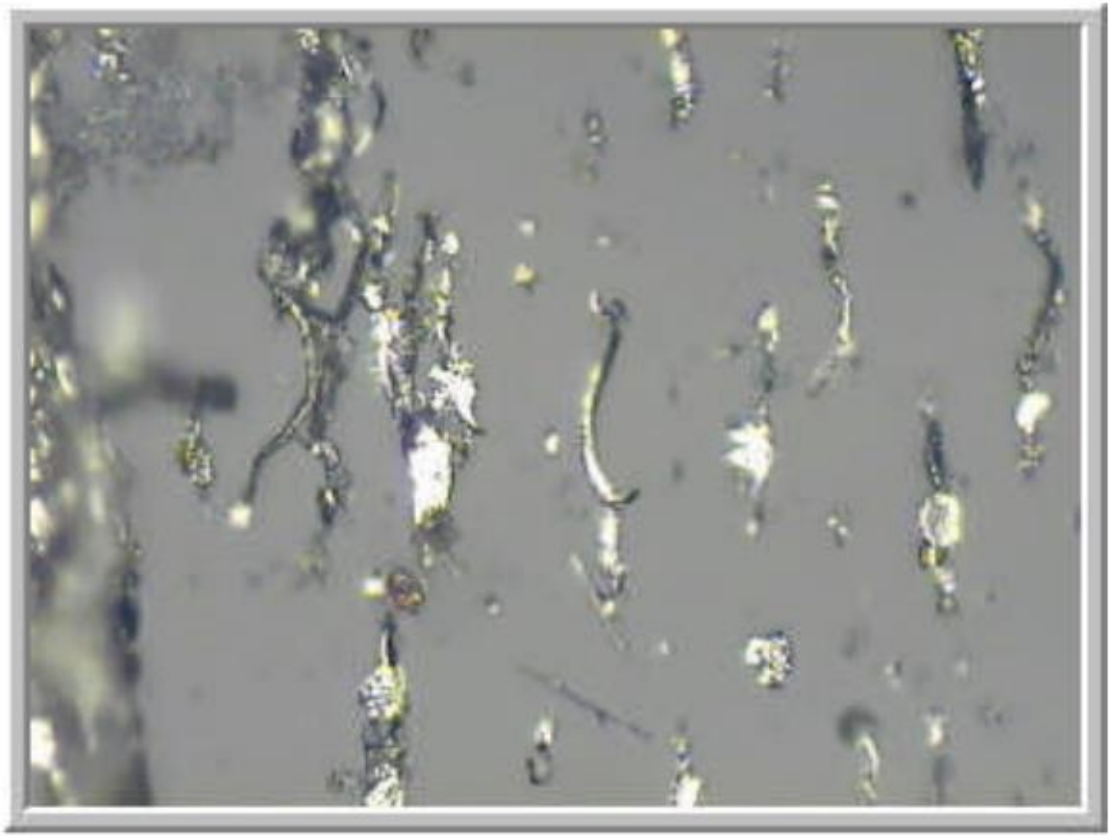


Figure 1-18 Wear Particles in Oil Sample

Vibration Analysis

Figure 1-24

Vibration Analysis

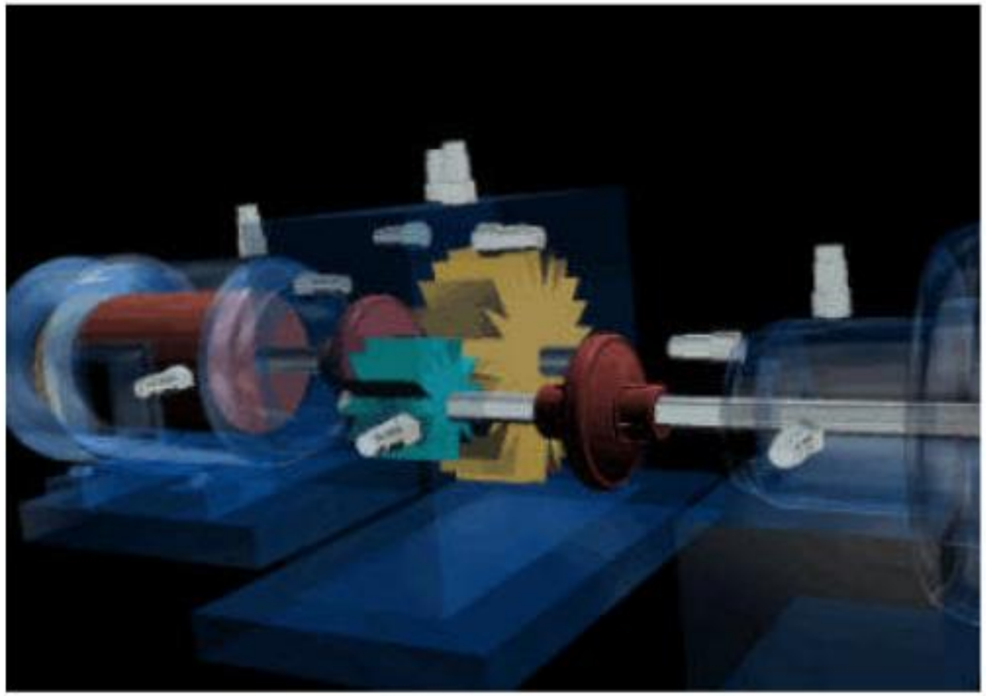
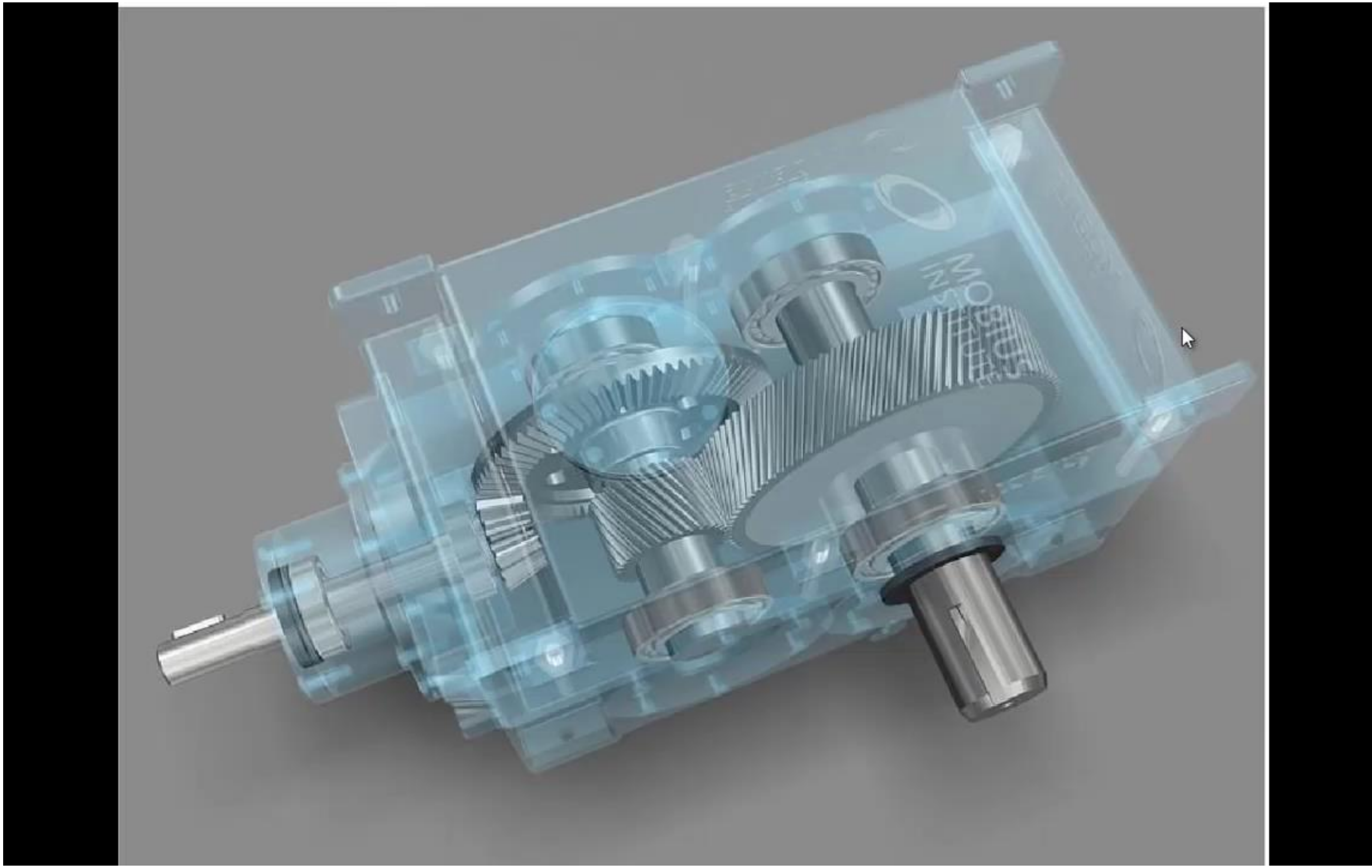


Figure 2-2 Various technologies let us see the condition inside a machine

Equipment failure happens – how do we avoid it?

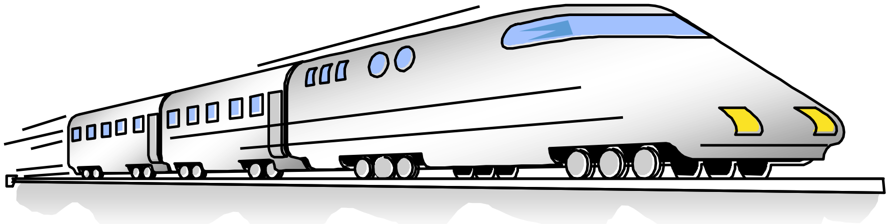


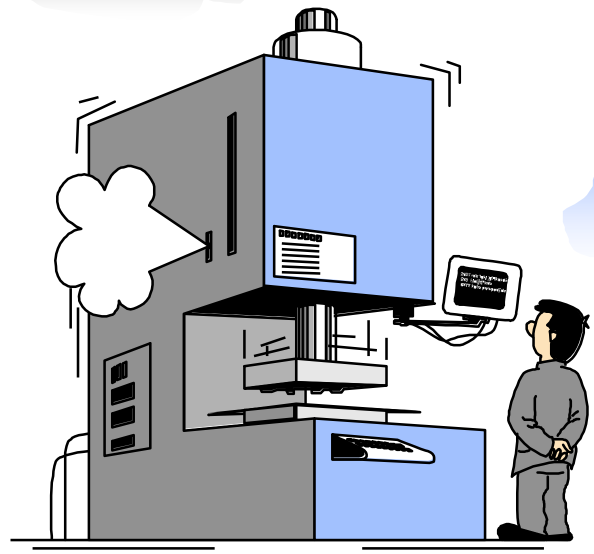
Vibration Analyzer

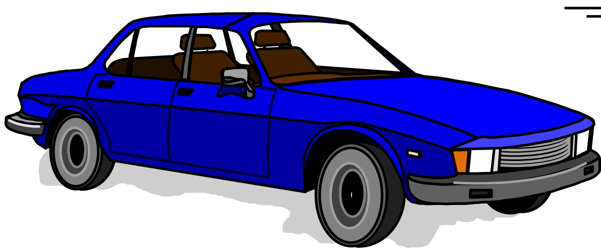
Vibration Analyzer

Vibration Analyzer

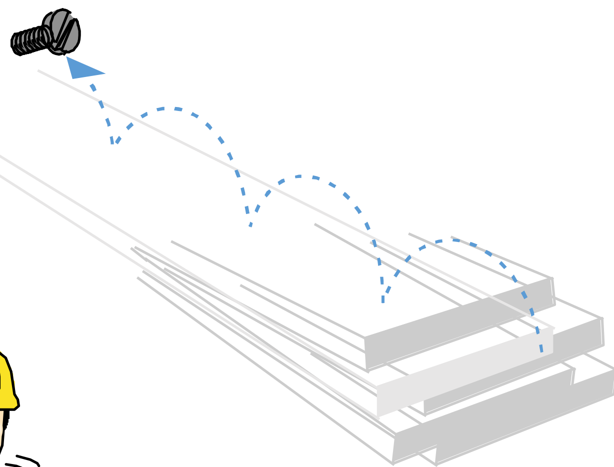
Vibration In Everyday Life

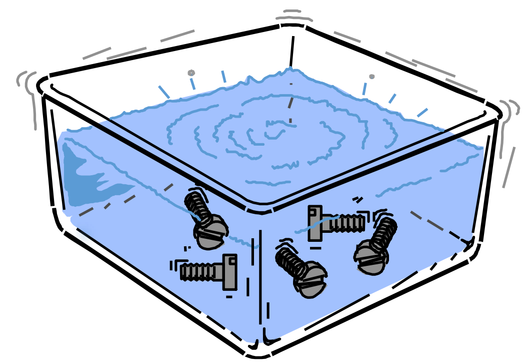


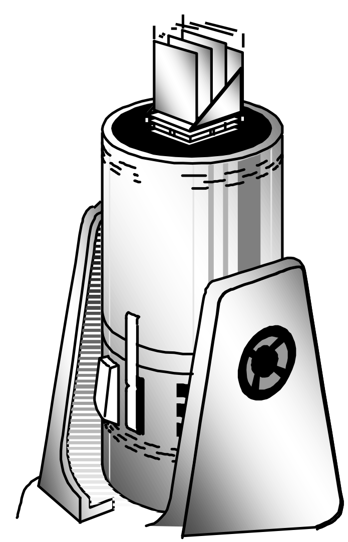




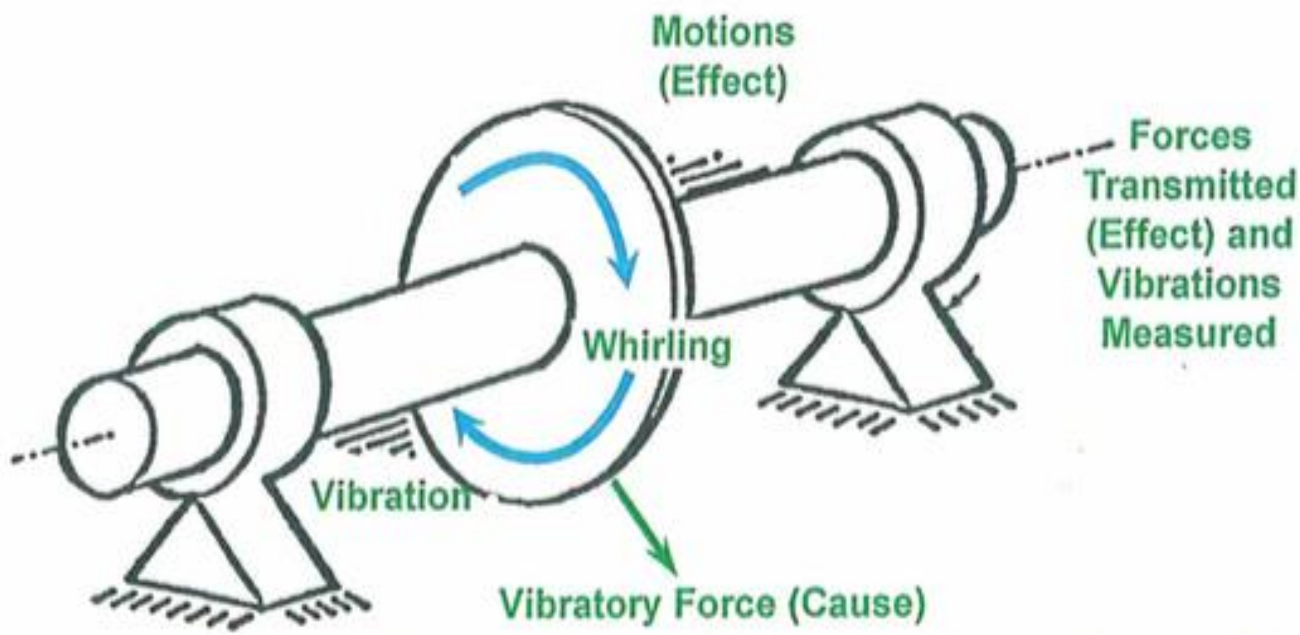
Useful Vibration







Cause and Effect Nature of Vibration



Acceptable limit of vibration

Overall levels of vibration are typically judged in terms of limits; e,g,, acceptance of new and repaired equipment, normal, surveillance and shut down.

The measured values are compared to the values in vibration standards such as

1) IRD 10816 Charts (Casing Measurements)
2) Bernhard Chart (Casing Measurements)
3) Vibration Institute Standards (Casing Measurements)
4) Update International Standards (Casing Measurements)
5) Blake Chart (Casing Measurements)
6) ISO 10816-1 Standard (Casing Measurements)
7) API 612 Standard (Shaft Displacement)
8) Dresser-Clark-Jackson Chart (Shaft Displacement)

Acceptable limit of vibration

1. ISO Standard

ISO 7919 Series	**Mechanical vibration of non-reciprocating machines - Measurement on rotating shafts and evaluation criteria**
7919-1:1996	Part 1: General Guidelines
7919-2: 2001	Part 2: Land-based steam turbines and generators in excess of 50 MW with normal operating speeds of 1500 r/min, 1800 r/min, 3000 r/min and 3600 r/min
7919-3: 1996	Part 3: Coupled industrial machines
7919-4: 1996	Part 4: Gas turbine sets
7919-5: 1997	Part 5: Machines set in hydraulic power generating and pumping plants
ISO 10816 Series	**Mechanical vibration - Evaluation of machine vibration by measurements on non-rotating parts**
10816-1: 1995	Part 1: General Guidelines
10816-2: 2001	Part 2: Land-based steam turbines and generators in excess of 50 MW with normal operating speeds of 1500 r/min, 1800 r/min, 3000 r/min and 3600 r/min
10816-3: 1998	Part 3: Industrial machines with normal power above 15kW and nominal speeds between 120 r/min and 15000 r/min when measured in situ
10816-4: 1998	Part 4: Gas turbine sets excluding aircraft derivatives
10816-5: 2000	Part 5: Machines set in hydraulic power generating and pumping plants
10816-6: 1995	Part 6: Reciprocating machines with power ratings above 100 kW
10816-7[2]	Part 7: Rotodynamic pumps for industrial application

Table 1 • ISO Standards for Evaluation of Vibration Severity[1]

Acceptable limit of vibration

2. API Standard

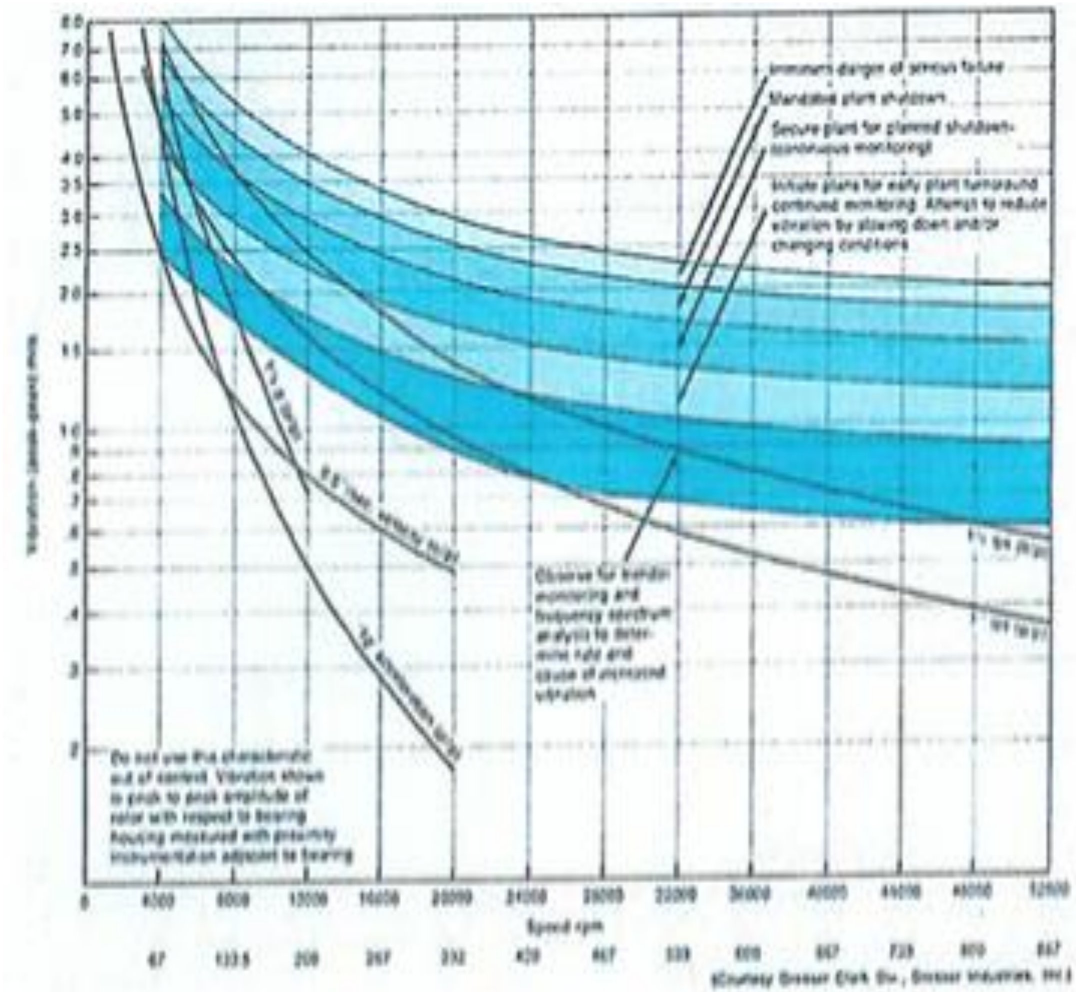
API produces a wide range of documents, including reference standards which are well suited for shop testing of new and rebuilt machinery. Note that these standards generally apply to equipment for use in the petrochemical industries. Table 4, below, shows a selection of API standards.

Equipment Type	API Standard	Acceptance Test	Other Requirements
Pumps	610 (9^{TH} edition March '03)	Shaft Relative + Casing	Vertical Pump (0.20 ips pk)
Fans	673 (2^{ND} edition November '01)	Casing (0.1 ips pk)	
Steam Turbines	612 (4^{TH} edition June '95)	Shaft Relative (mil pk-pk)	4 hour run in test required
Gears	613 (5^{TH} edition March '03)	Casing (0.15 ips pk)	Unbalance 4 W/N oz-in
Centrifugal Compressors	617 (7^{TH} edition July '02)	Shaft Relative (mil pk-pk)	4 hour run in test required
Screw Compressors	619 (3^{RD} edition June '97)	Shaft Relative (mil pk-pk)	Unbalance 4 W/N oz-in
Induction Motors (≥ 250 hp)	541 (4^{TH} edition March '03)		Unbalance 4 W/N oz-in

Table 4 • Sample API Standards for Acceptance Testing

Acceptable limit of vibration

3. Shaft vibration for turbomachinery (measuring displacement) compare to Dresser Clark chart.



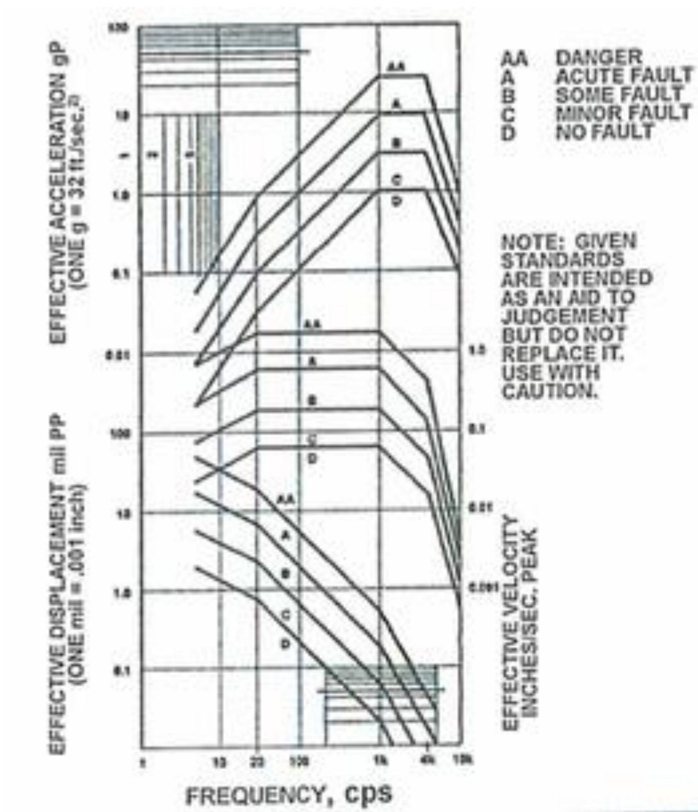
Acceptable limit of vibration

4. FLUID-FILM Bearing vibration (compare relative vibration of the rotor to the clearance in the bearing), compare to Vibration Institute tables.

MAINTENANCE ACTION	ALLOWABLE R/C	
	3,600 RPM	10,000 RPM
Normal	0.3	0.2
Surveillance	0.3 - 0.5	0.2 - 0.4
Shut down at next convenient time	0.5	0.4
Shut down immediately	0.7	0.6

Acceptable limit of vibration

5. Casing Vibration (accelerometers or velocity transducers placed as close to the bearing as possible). Compare to tables or Blake chart.



ISO Standard

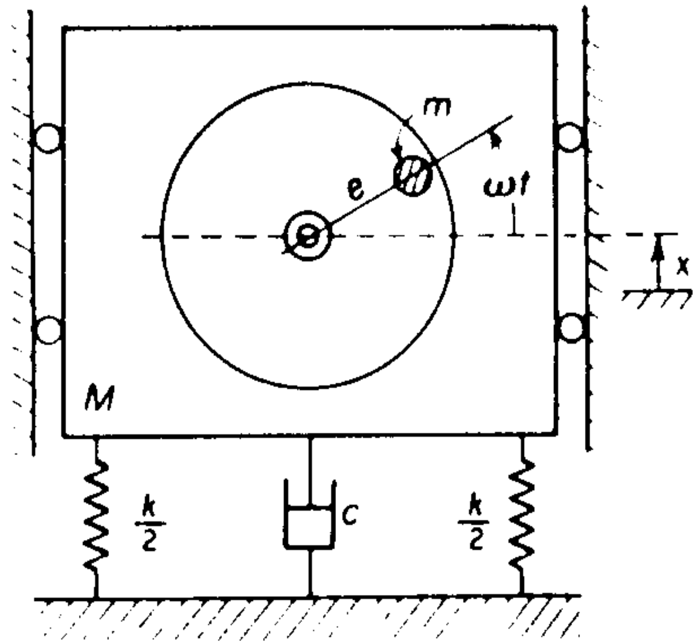
VIBRATION SEVERITY PER ISO 10816

Machine		Class I small machines	Class II medium machines	Class III large rigid foundation	Class IV large soft foundation
in/s	mm/s				
0.01	0.28	good	good	good	good
0.02	0.45	good	good	good	good
0.03	0.71	good	good	good	good
0.04	1.12	satisfactory	good	good	good
0.07	1.80	satisfactory	satisfactory	good	good
0.11	2.80	unsatisfactory	satisfactory	satisfactory	good
0.18	4.50	unsatisfactory	unsatisfactory	satisfactory	satisfactory
0.28	7.10	unacceptable	unsatisfactory	unsatisfactory	satisfactory
0.44	11.2	unacceptable	unacceptable	unsatisfactory	unsatisfactory
0.70	18.0	unacceptable	unacceptable	unacceptable	unsatisfactory
0.71	28.0	unacceptable	unacceptable	unacceptable	unacceptable
1.10	45.0	unacceptable	unacceptable	unacceptable	unacceptable

Vibration Velocity Vrms

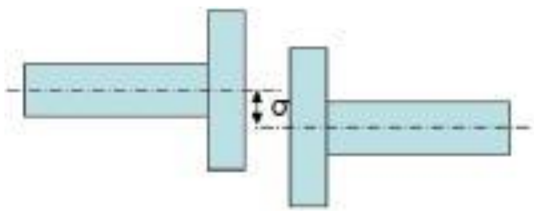
Causes of Vibrations

1. Mass unbalance

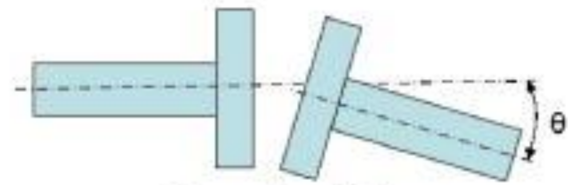


Causes of Vibrations

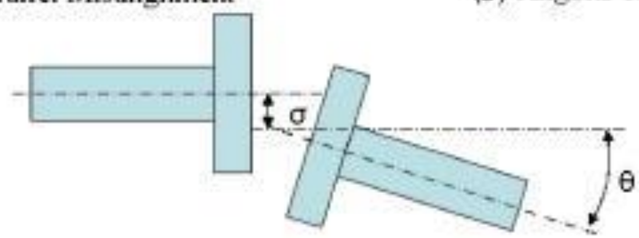
2. Misalignment



(a) Parallel Misalignment



(b) Angular Misalignment

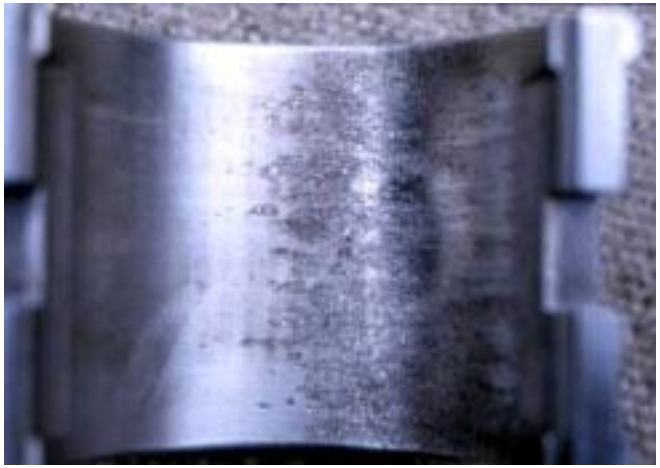


(c) Combined Parallel-Angular Misalignment

Causes of Vibrations

3. Fluid-film bearing wear

JOURNAL BEARING



WEAR IN PADS

Causes of Vibrations

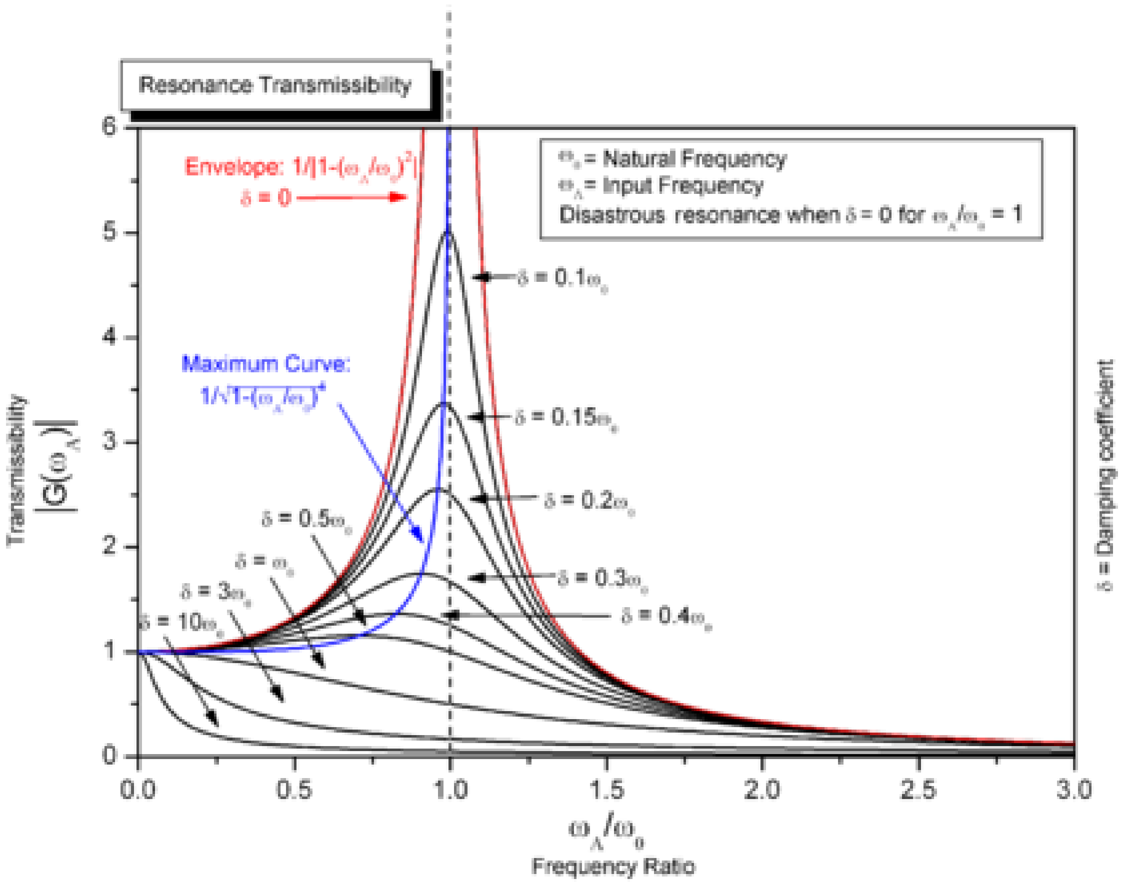
3. Fluid-film bearing wear

Thrust Bearing

WEAR IN PADS

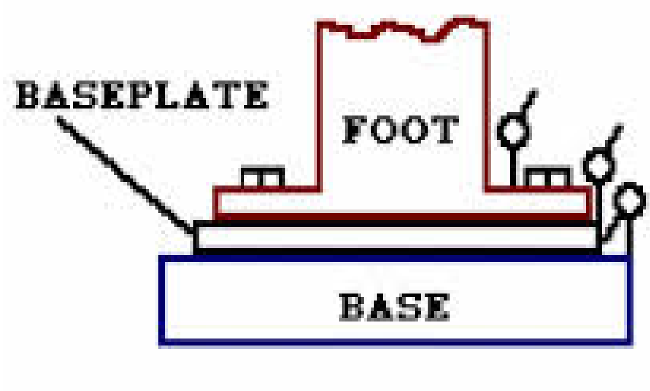
Causes of Vibrations

4. Resonance



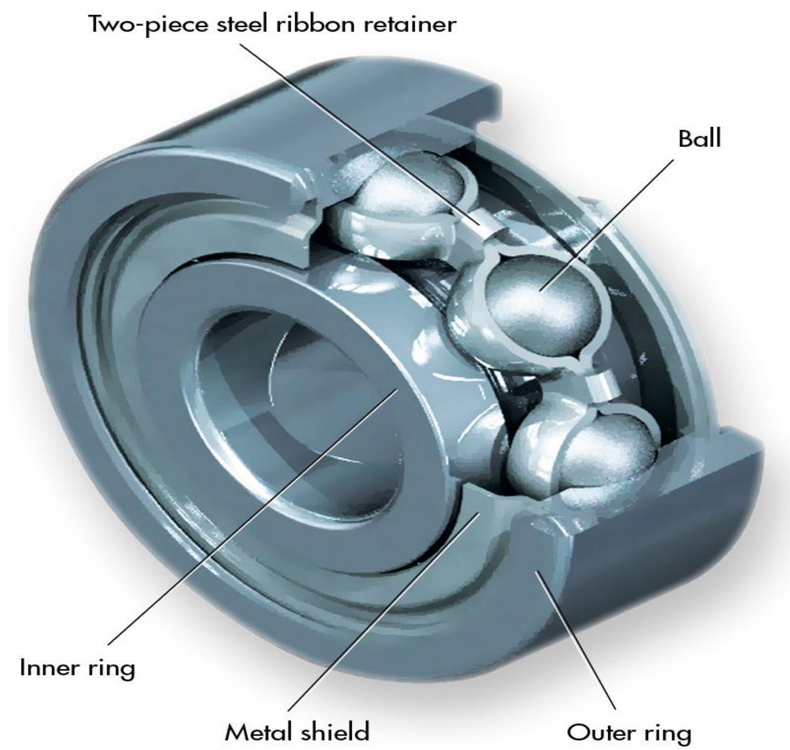
Causes of Vibrations

5. looseness



Causes of Vibrations

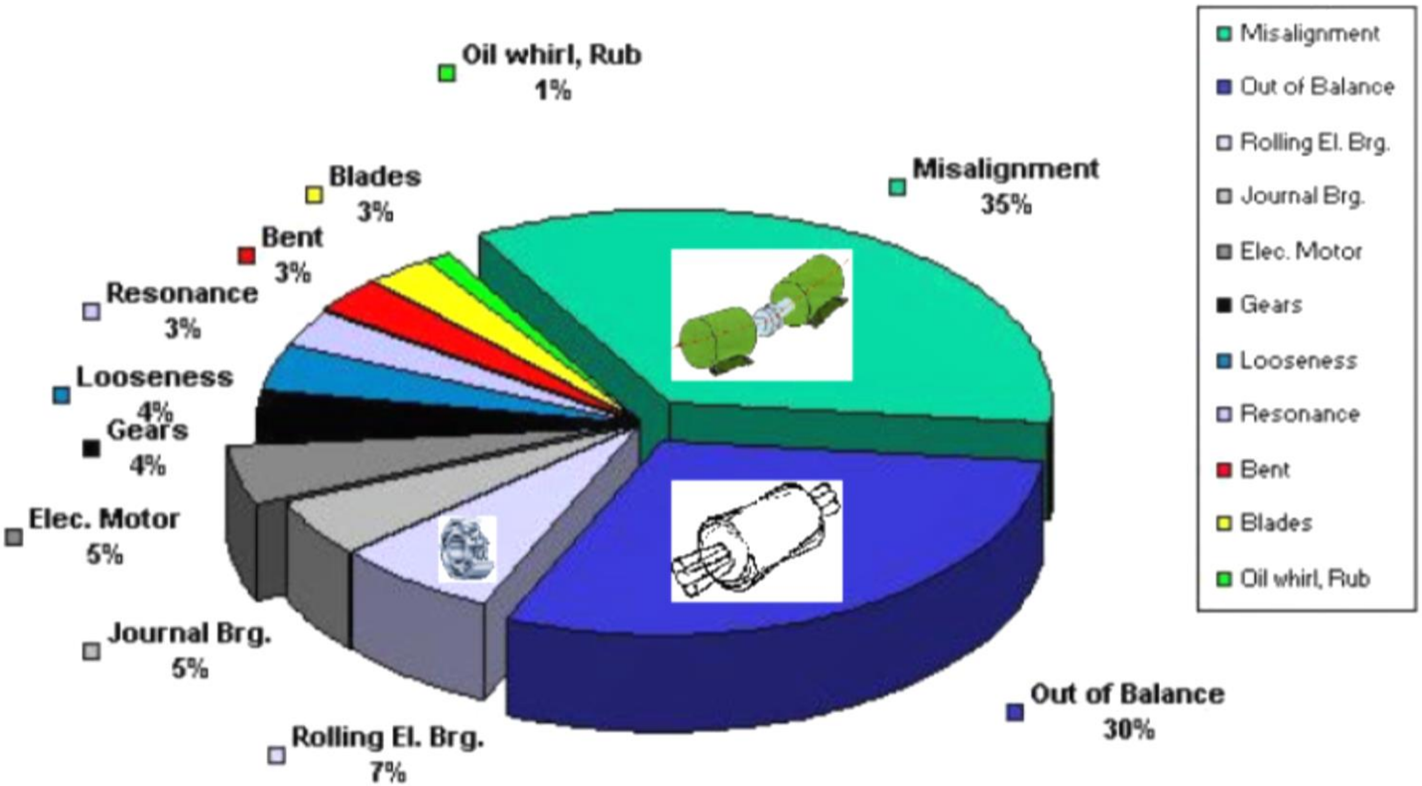
6. Rolling element bearing (outer race defect – inner race defect – ball defect – excessive internal clearance)



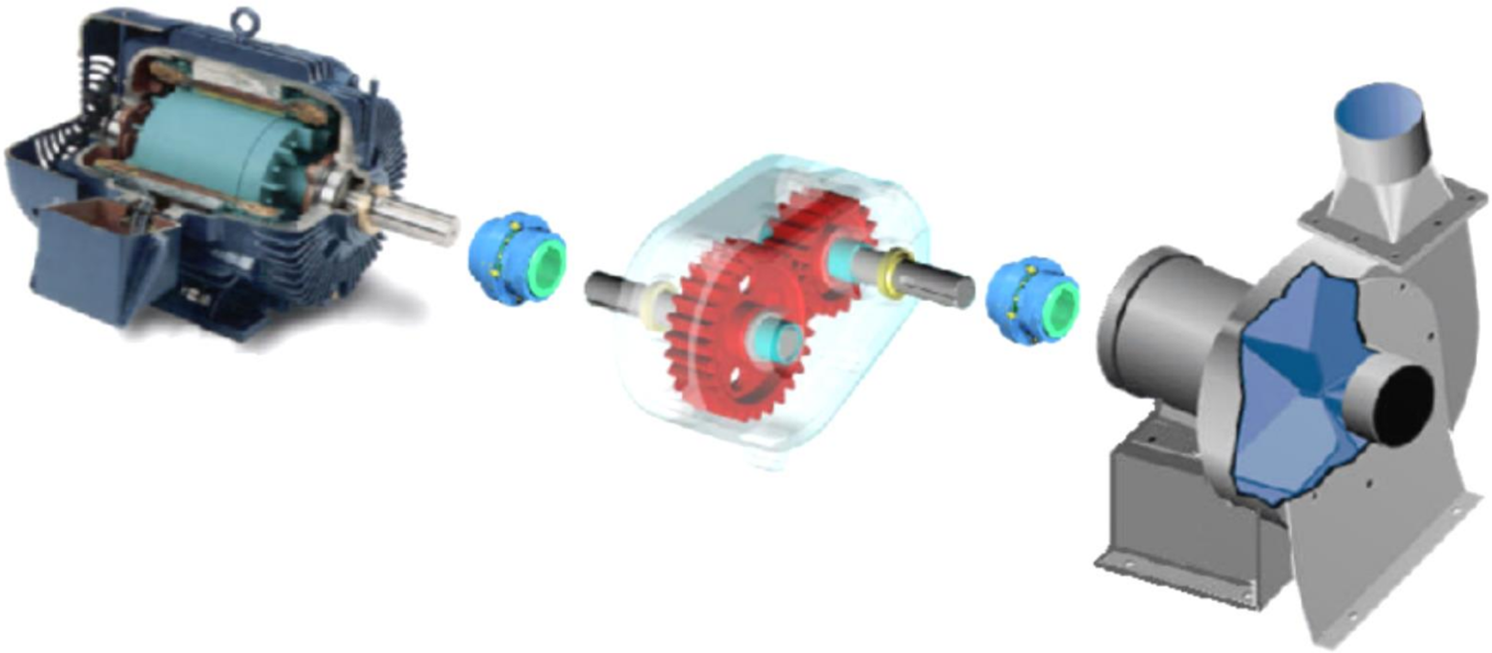
Causes of Vibrations

7. Gear box (eccentric gears – gear wear- broken teeth)

Machine Potential Failures Analysis

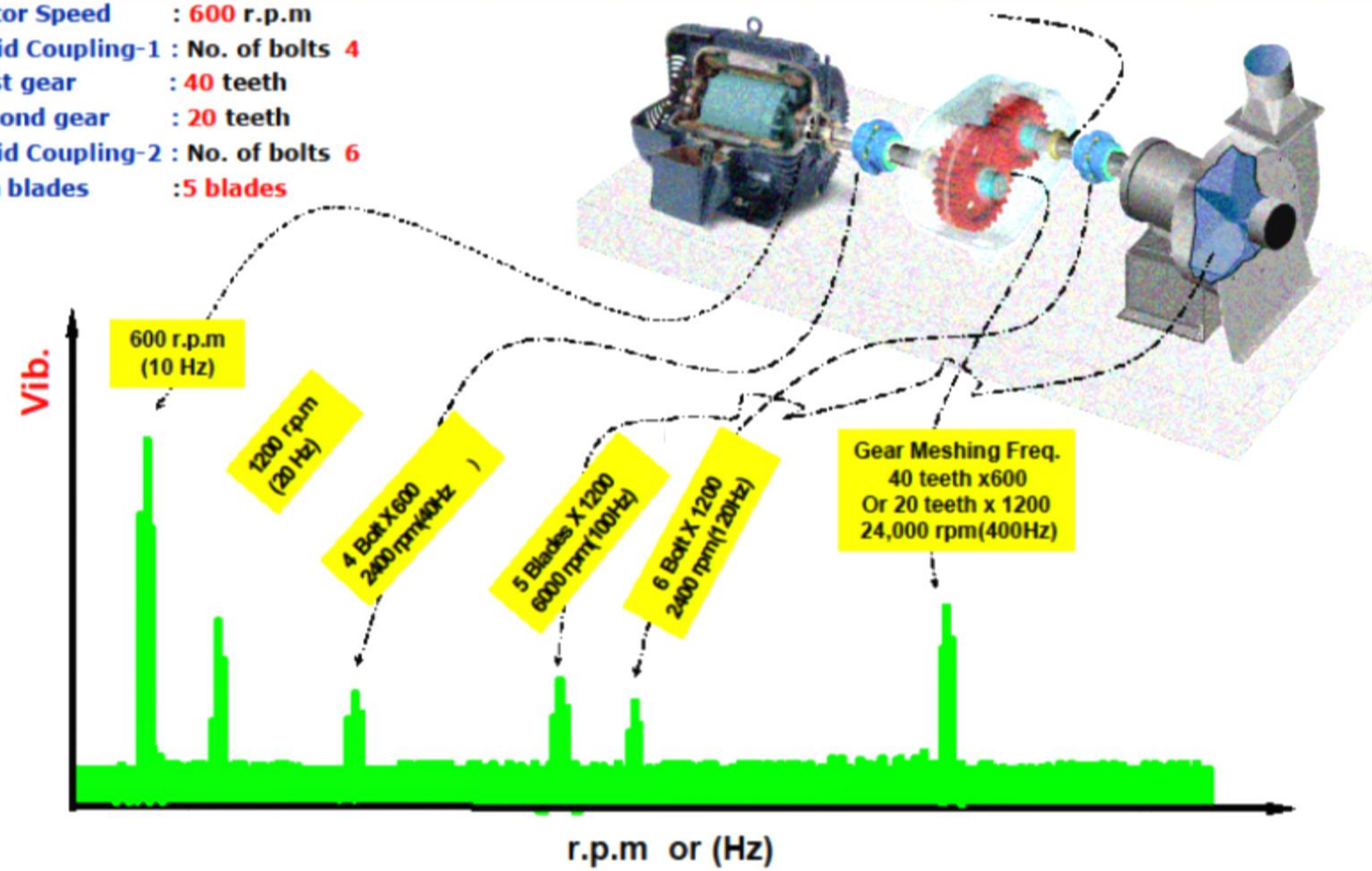


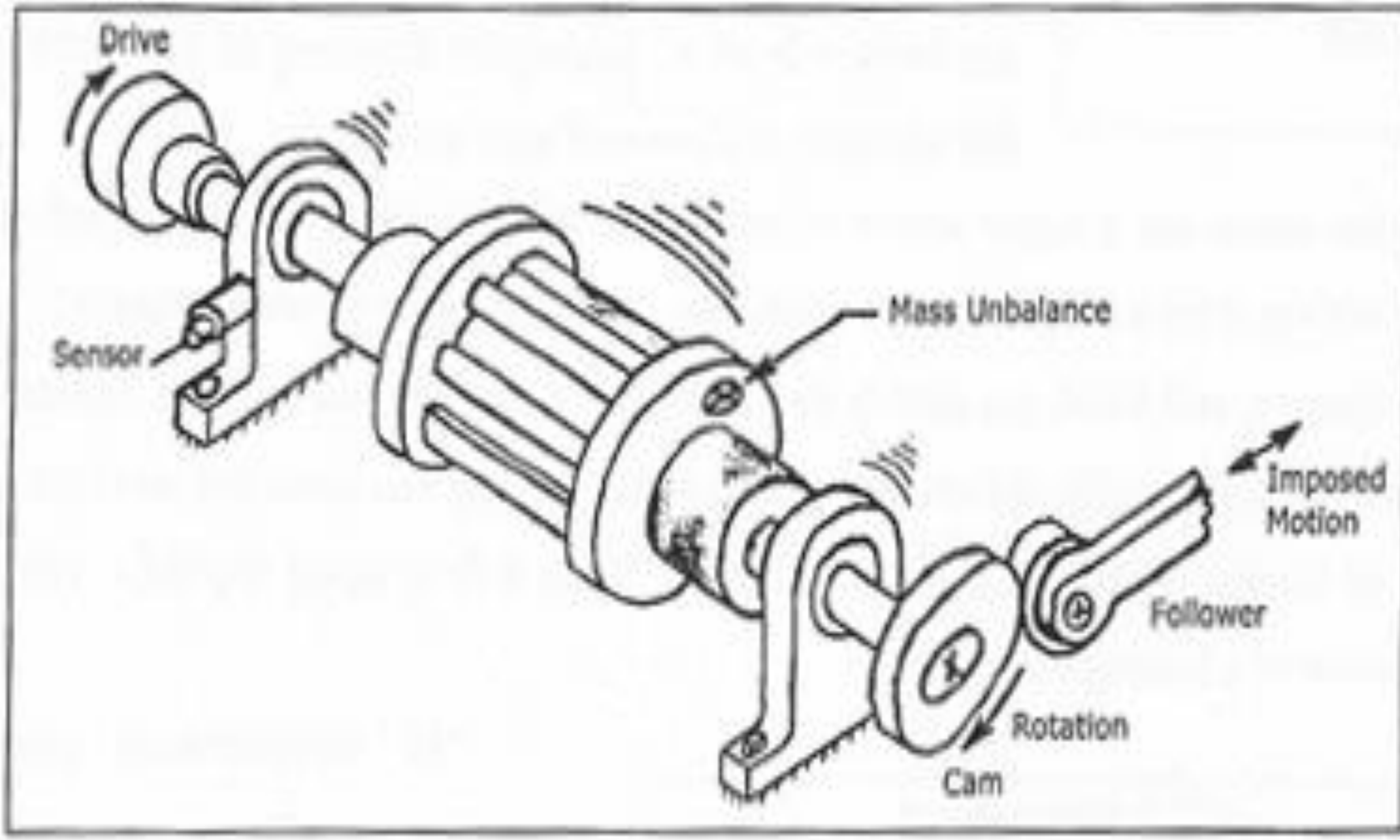
Example of Machine Component



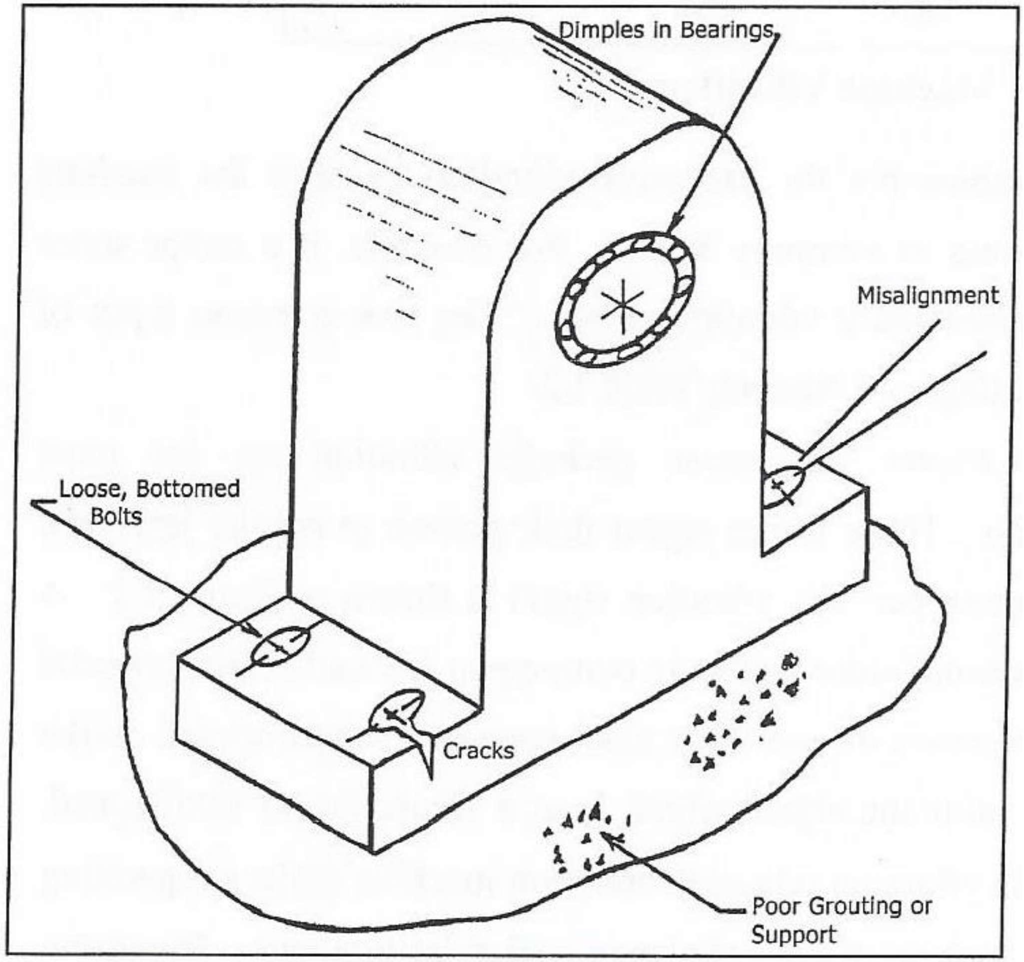
Frequency Spectrum interpretation

Motor Speed : 600 r.p.m
Rigid Coupling-1 : No. of bolts 4
First gear : 40 teeth
Second gear : 20 teeth
Rigid Coupling-2 : No. of bolts 6
Fan blades :5 blades





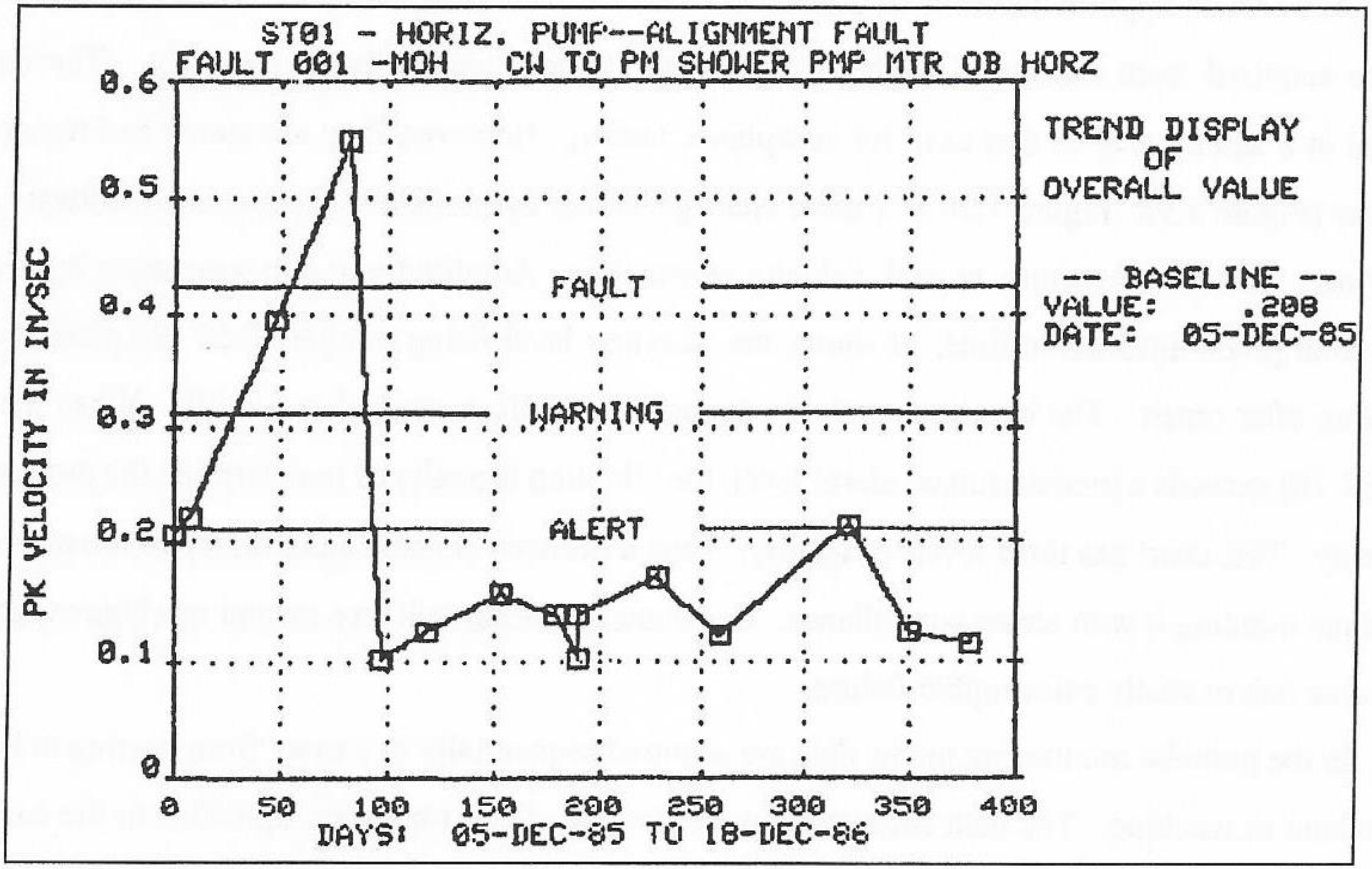
Drive
Sensor
Mass Unbalance
Imposed
Motion
Follower
Rotation
Cam


Dimples in Bearings
Misalignment
Loose, Bottomed Bolts
Cracks
Poor Grouting or Support

LOCATION	UNITS	DATE	PREV. VALUE	LAST VALUE	% CHANGE	ALARM STATUS
MOTOR OBD HORIZONTAL	IPS-Pk	1/23/03	0.237	0.183	-23	
MOTOR OBD VERTICAL	IPS-Pk	1/23/03	0.251	0.230	-8	
MOTOR OBD AXIAL	IPS-Pk	1/23/03	0.402	0.304	-24	A1
MOTOR IBD HORIZONTAL	IPS-Pk	1/23/03	0.223	0.168	-25	
MOTOR IBD VERTICAL	IPS-Pk	1/23/03	0.251	0.350	39	A1
MOTOR IBD AXIAL	IPS-Pk	1/23/03	0.430	0.354	-18	A1
DRIVE INPUT HORIZONTAL	IPS-Pk	1/23/03	0.320	0.206	-36	
DRIVE INPUT VERTICAL	IPS-Pk	1/23/03	0.259	0.174	-33	
DRIVE INPUT AXIAL	IPS-Pk	1/23/03	0.183	0.179	-2	
DRIVE OUTPUT HORIZONTAL	IPS-Pk	1/23/03	0.295	0.152	-14	
DRIVE OUTPUT VERTICAL	IPS-Pk	1/23/03	0.284	0.154	-46	
DRIVE OUTPUT AXIAL	IPS-Pk	1/23/03	0.212	0.136	-36	
PUMP IBD HORIZONTAL	IPS-Pk	1/23/03	0.190	0.168	-12	
PUMP IBD VERTICAL	IPS-Pk	1/23/03	0.174	0.175	1	
PUMP IBD AXIAL	IPS-Pk	1/23/03	0.110	0.093	-15	
PUMP OBD HORIZONTAL	IPS-Pk	1/23/03	0.230	0.202	-12	
PUMP OBD VERTICAL	IPS-Pk	1/23/03	0.124	0.140	13	
PUMP OBD AXIAL	IPS-Pk	1/23/03	0.113	0.060	-47	
MOTOR OBD VERTICAL	Mils-Pk-Pk	1/23/03	1.03	0.80	-22	
MOTOR OBD HORIZONTAL	Mils-Pk-Pk	1/23/03	1.43	1.42	-1	
MOTOR IBD VERTICAL	Mils-Pk-Pk	1/23/03	2.00	1.54	-23	
MOTOR IBD HORIZONTAL	Mils-Pk-Pk	1/23/03	2.50	1.32	-47	
DRIVE INPUT VERTICAL	Mils-Pk-Pk	1/23/03	2.97	3.10	4	A1
DRIVE INPUT HORIZONTAL	Mils-Pk-Pk	1/23/03	2.80	3.66	31	A1
DRIVE OUTPUT VERTICAL	Mils-Pk-Pk	1/23/03	2.57	1.70	-34	
DRIVE OUTPUT HORIZONTAL	Mils-Pk-Pk	1/23/03	2.57	2.10	-18	
PUMP IBD VERTICAL	Mils-Pk-Pk	1/23/03	1.09	0.85	-22	
PUMP IBD HORIZONTAL	Mils-Pk-Pk	1/23/03	1.42	1.14	-20	
PUMP OBD VERTICAL	Mils-Pk-Pk	1/23/03	0.23	0.23	0	
PUMP OBD HORIZONTAL	Mils-Pk-Pk	1/23/03	0.49	0.63	29	

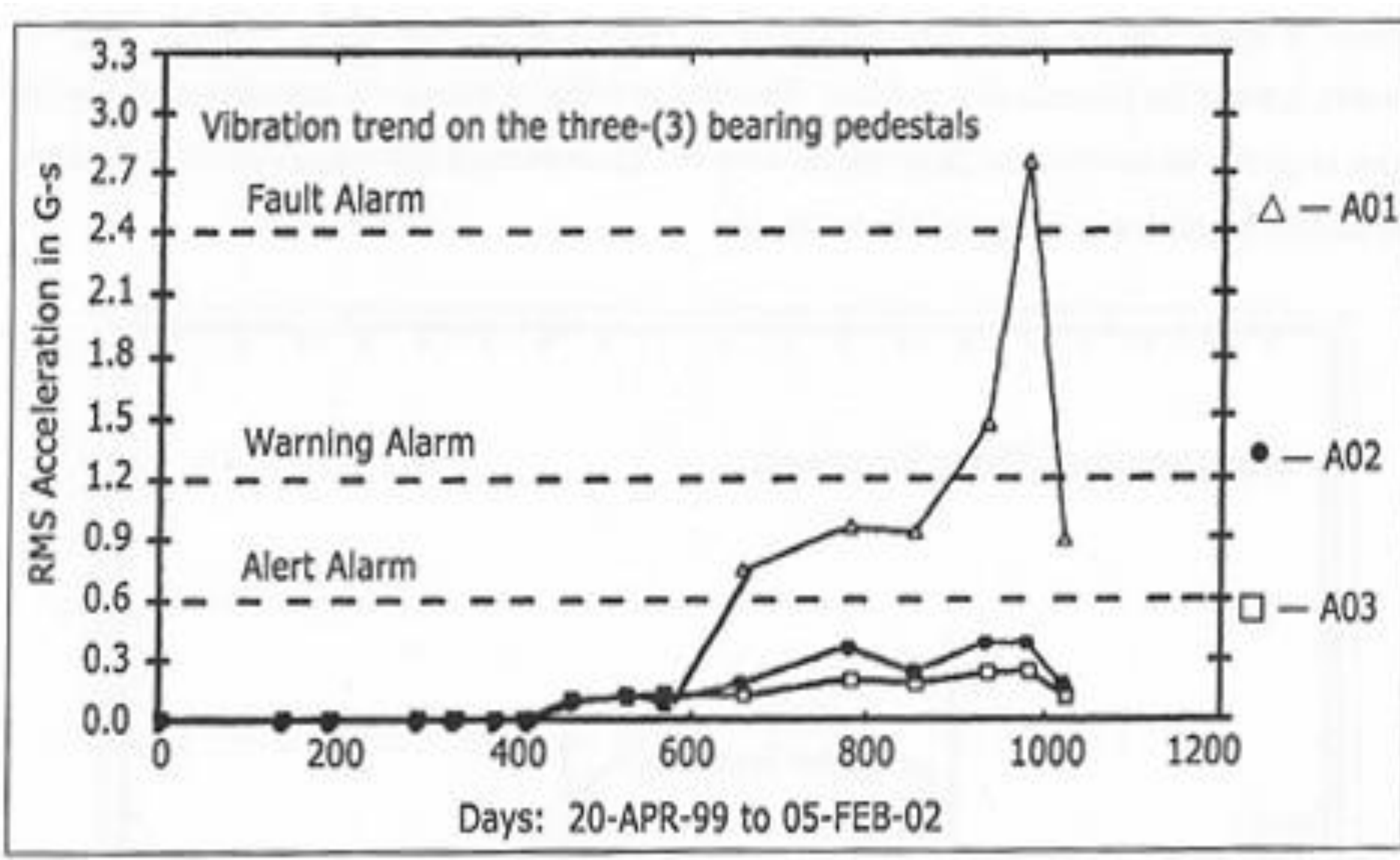
Example of a Route for Motor Driven Boiler Feed Pump

Vibration



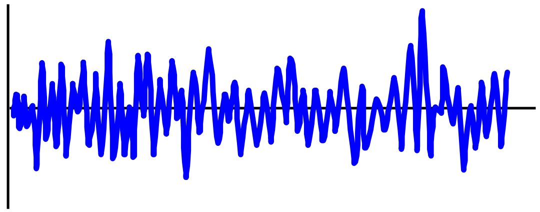
Monthly Trend Plot of a Pump Motor for Peak Velocity.

PERIODIC MONITORING



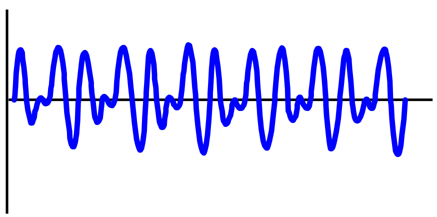
Types of Signals

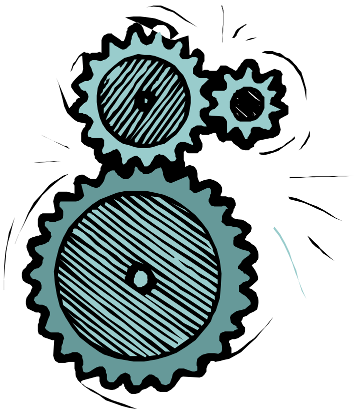
Stationary signals



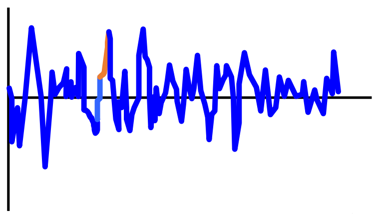
Non-stationary signals

Deterministic

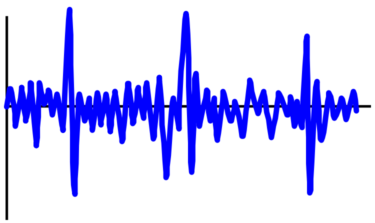




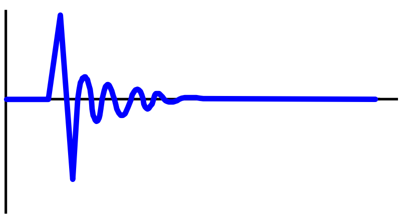
Random



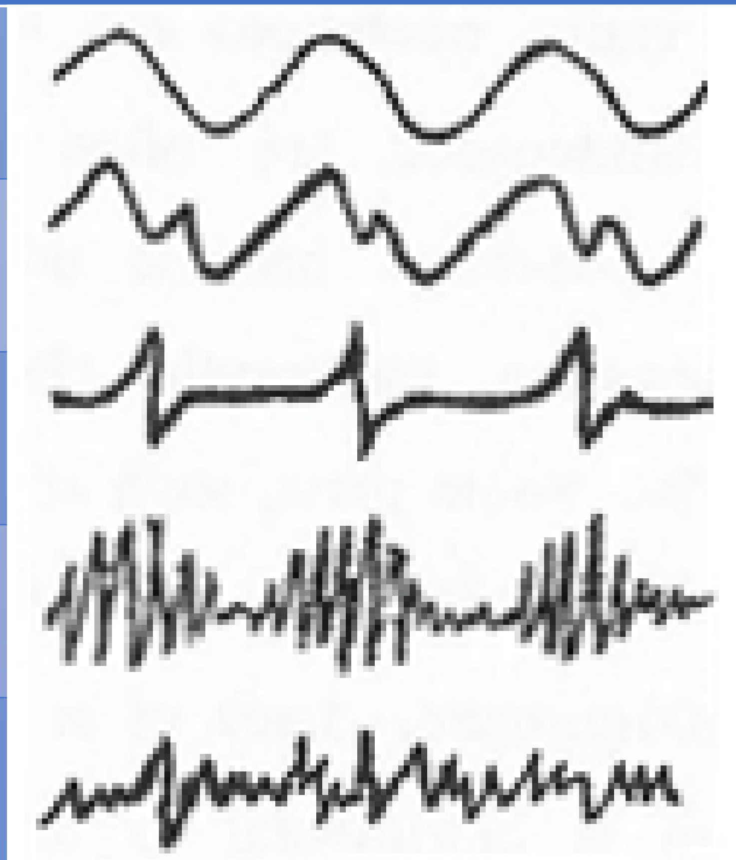
Continuous



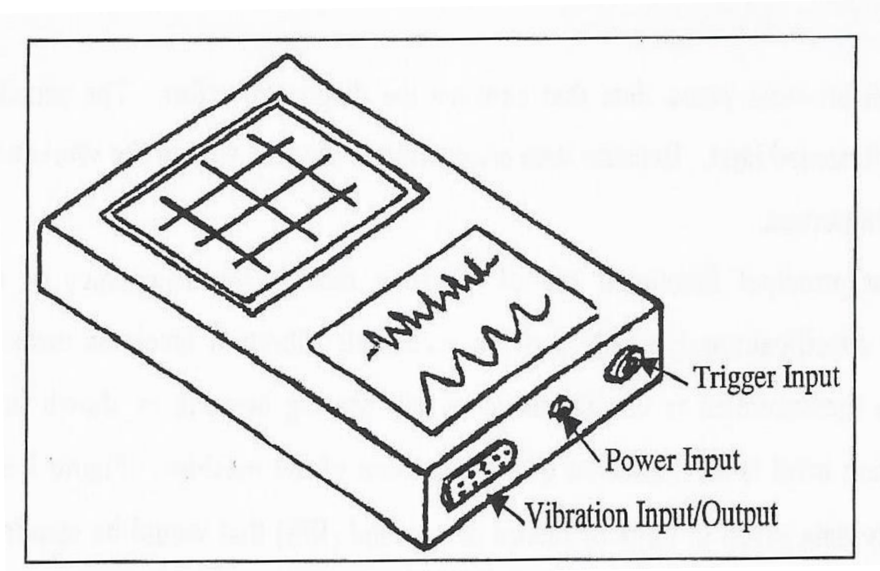
Transient



Types of Vibrations

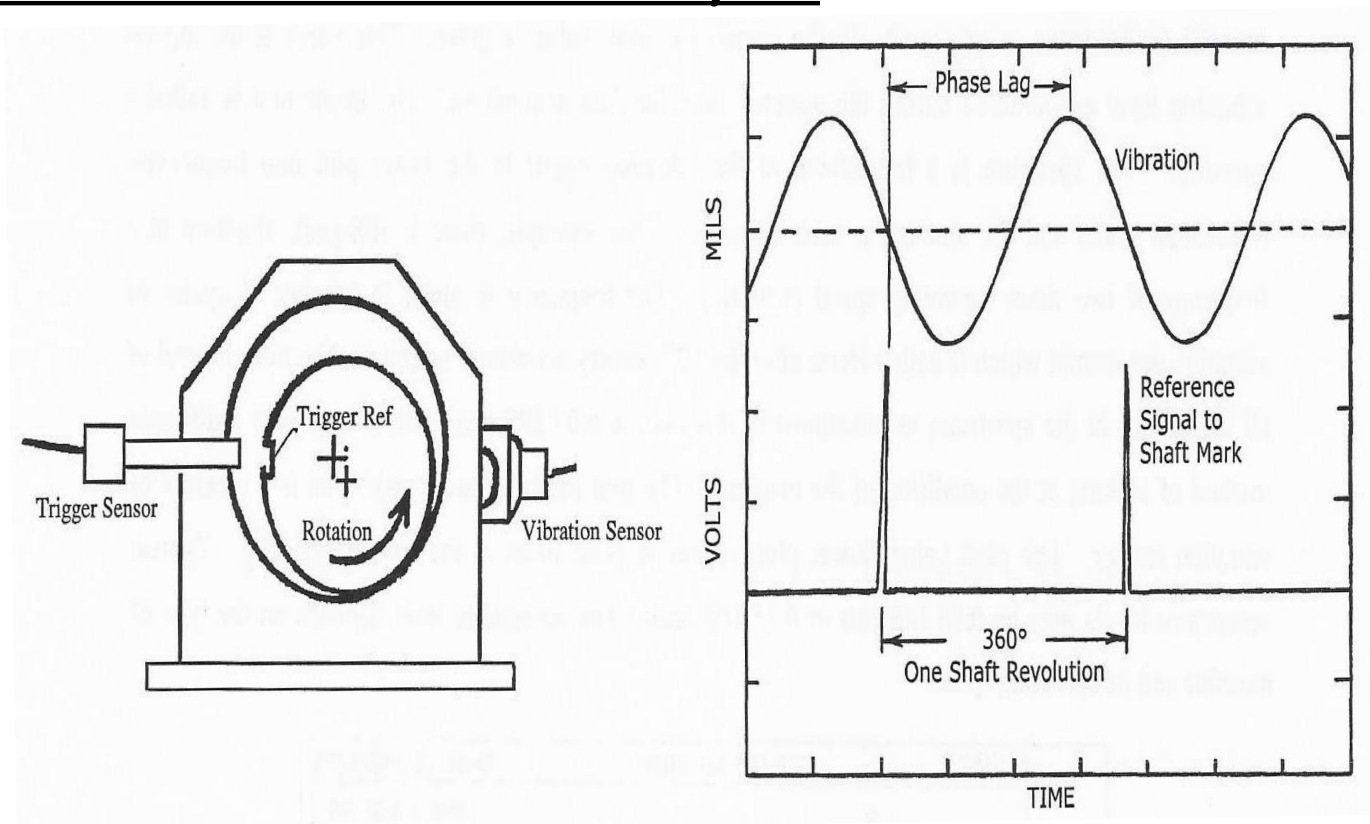
Harmonic		Mass Unbalance
Periodic		Misalignment
Impulsive		Rolling Element Bearing, Gears tooth
Pulsating		
Random		Cavitation in Pumps

Measurement and Analysis



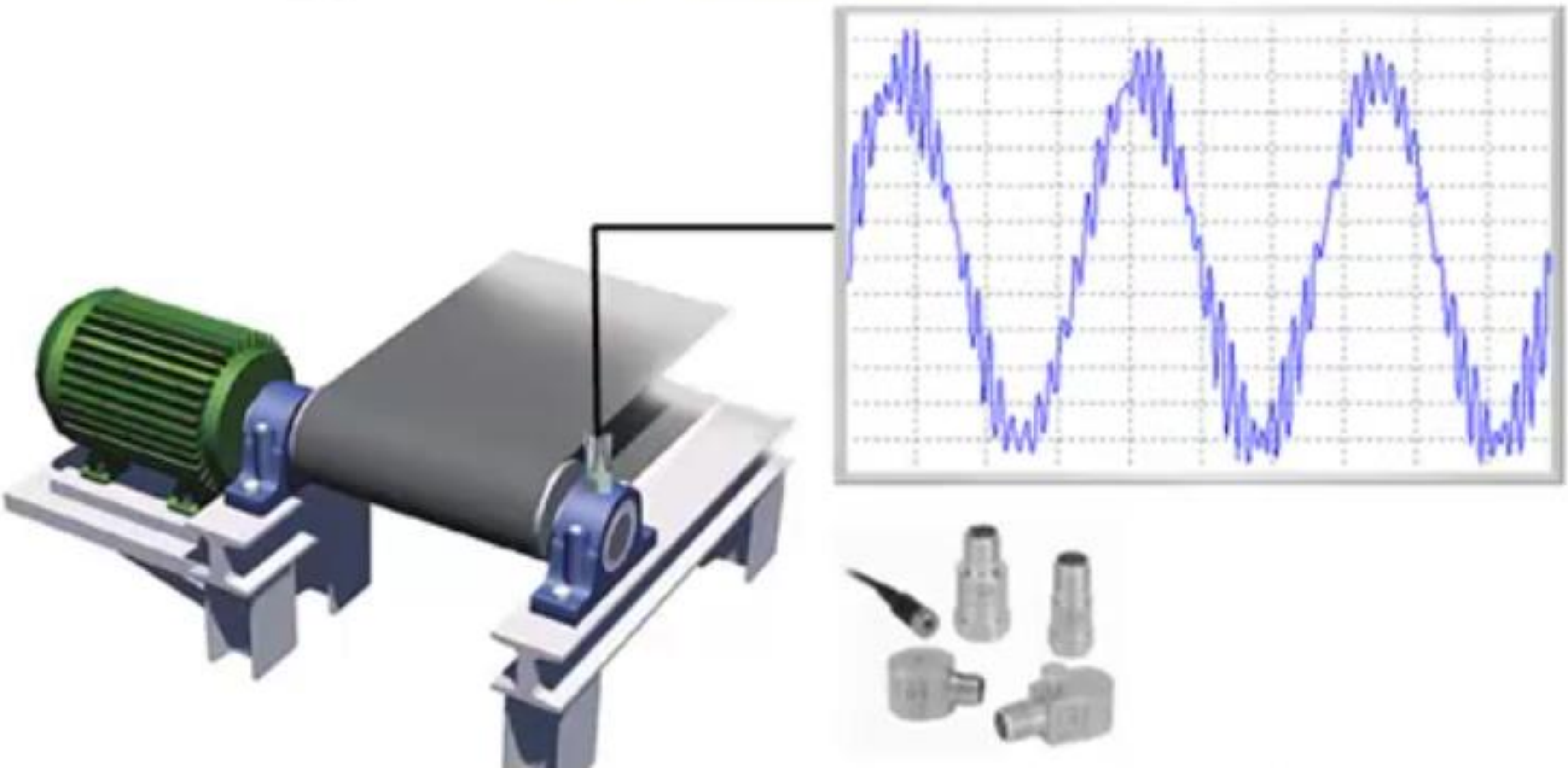
Schematic of Data Collection Instrument.

Measurement and Analysis

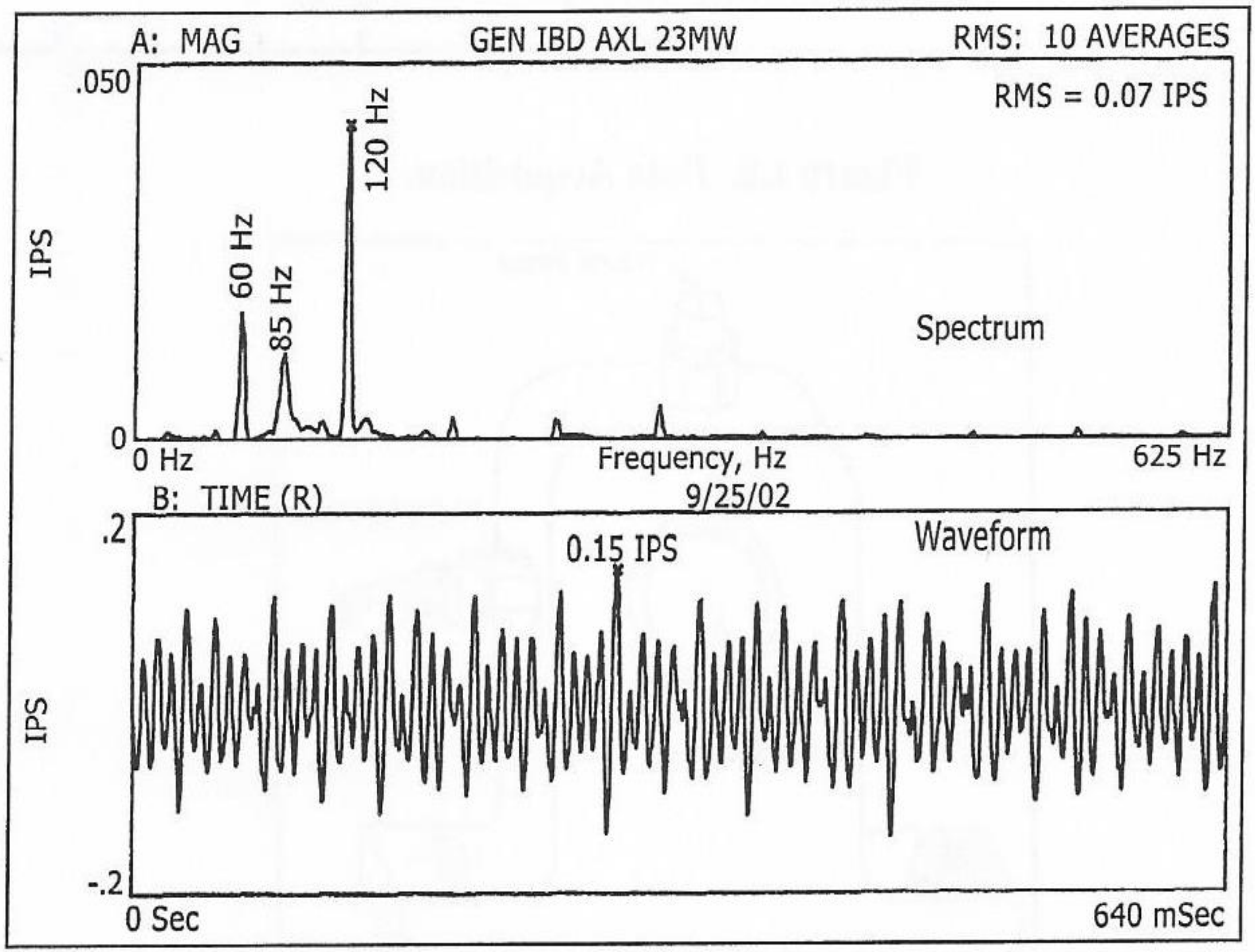


Data Acquisition

Measurement and Analysis



Measurement and Analysis



PHYSICAL OBSERVATIONS by Human Senses

While there are several types of recorded data that form the basis for machine fault and condition analysis, among the most basic data are direct observations by the person doing the data collection based on human senses - hearing, sight, touch, smell, and taste. Human sensory capabilities, although not analytical, cannot be underestimated in the machine analysis process.

Noise

Unusual noises can indicate rubs, bearing defects, looseness, improper assembly, lack of lubrication, and any other metal to metal contact problems. A listening rod or screw driver can be used to detect a bearing defect or rubbing in a low speed machine. In pumps, a sign of flow problems is a noise that sounds like gravel in the piping. Motors and generators may emit high frequency whining noises when they are subject to excessive vibration due to casing distortion, misalignment, or coupling unbalance.

Noise

High pitched noise from new gears indicates bad construction and machinery quality or design (low contact ratio). Rubbing of guards by pulleys and belts will cause impacting and noise. Lack of lubrication in oil starved bearings or bearings with excessive clearance means that the bearing needs attention. Excessive noise is almost always an indicator of trouble. The experienced data collector will be able to enhance their analytical capability by learning to identify noise sources and associate the physical problem with them.

Sight

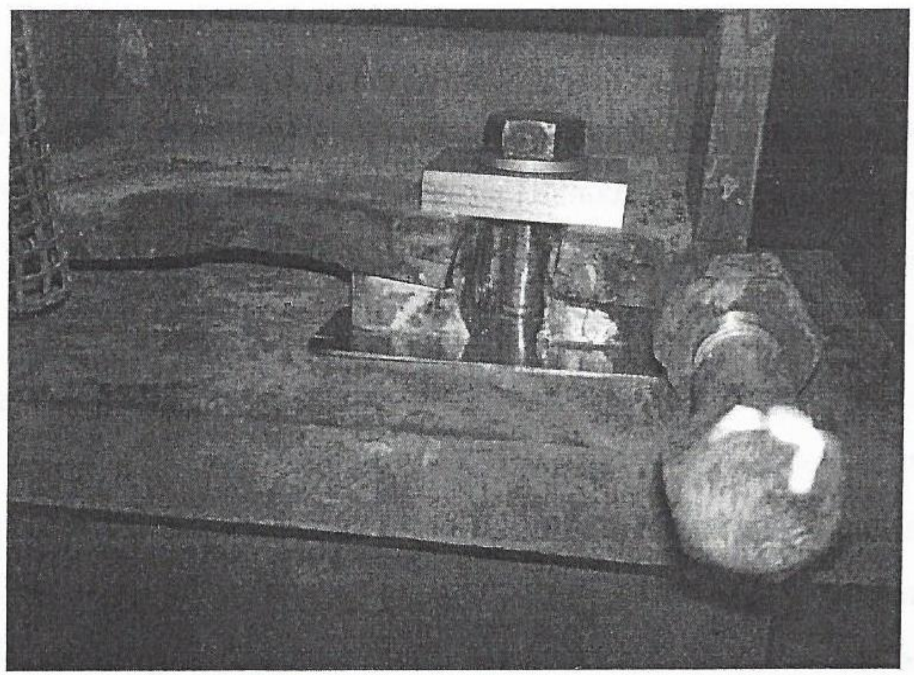
The use of sight is an even more powerful tool for data collectors. Smoke, fire, and catastrophic failures need and get immediate attention. However, other mundane faults may go unnoticed for months. Foundation and bearing pedestal faults are the source of many cases of excessive vibration.

A flashlight and feeler gage or knife help to root out these type problems. Squishing oil between joints is a certain clue of looseness.

Cracks in ducting and piping and other machine components provide clues to the presence of excessive vibration. Vibration analysis will confirm these faults

Sight

Vibration analysis will confirm these faults. The data collector may have to go off route to measure these cases.



Hammered and Torched to Fit.

Smell and Touch

The senses of smell and touch are less important but should not be neglected. Unusual, abnormal odors are easily detected by the human sense of smell. Oil smoke can be smelled long before an oil fire. Ammonia and other chemical and gas leaks are best detected by the nose. Even small quantities can be detected. Hot bearings or other machine parts that are not normally operating above ambient temperature can be identified by touch. However, the data collector needs to exercise extreme caution. A steaming or red hot machine should not be touched. The water can confirm the temperatures are above 100^{o} C.

The use of taste is not recommended in this work.

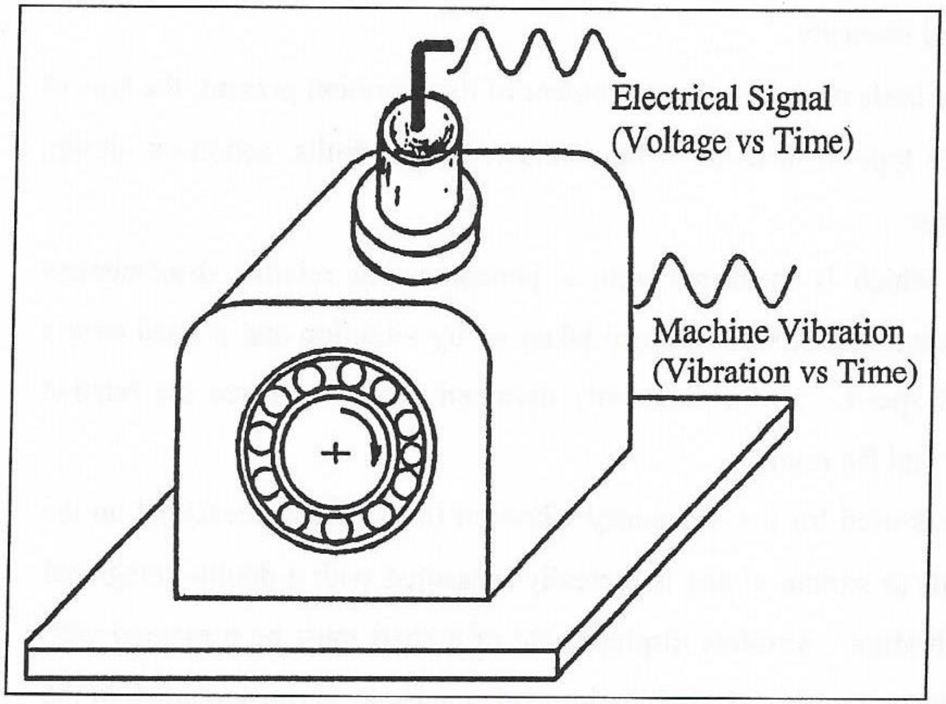
PERIODIC AND CONTINUOUS DATA COLLECTION

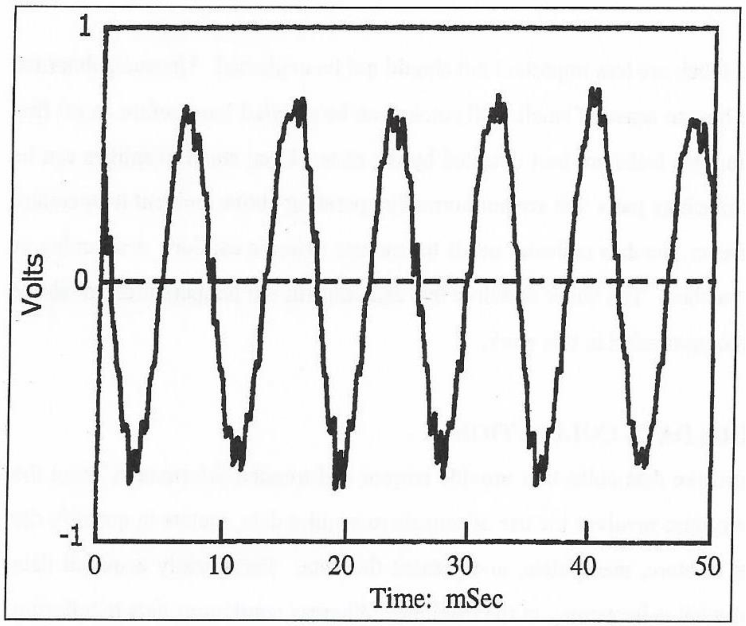
Periodic and continuous non-intrusive data collection provide current and trended information about the condition of a machine. The procedure involves the use of sensors to acquire data, meters to quantify the measured data, and instruments to store, manipulate, and present the data. Periodically acquired data provide an intermittent record of what is happening in the machine. Whereas continuous data monitoring and collection provides continuous surveillance along with the ability to protect the machine through data based automatic shutdown.

PERIODIC AND CONTINUOUS DATA COLLECTION

Measurement of vibration for analytical use is performed by a sensor, sometimes called a transducer or pickup, and is nonintrusive to the machine or process. The sensor transforms the vibration (mechanical motion) of the mounting location to an electrical voltage which varies with time.

PERIODIC AND CONTINUOUS DATA COLLECTION





Selecting a Measure

A measure is a unit or measures of vibration are standard of measurement that provides a means for physical evaluation. Examples of measures are pounds for weight and feet for height. Three basic available displacement, velocity, and acceleration. Ideally the sensor would directly provide the selected measure. Unfortunately, sensor limitations do not always allow direct measurement of vibration in the proper measure. Other predictive maintenance based measures are temperature, pressure, and viscosity.

Selecting a Measure

The measure is selected on the basis of the frequency content of the vibration present, the type of sensor, the design of the machine, the type of analysis to be conducted (e.g., faults, condition, design information), and the information sought.

Selecting a Measure

Relative shaft displacement

which is measured with a non-contacting relative displacement sensor, proximity probe, shows the extent of bearing clearance taken up by vibration and is used over a frequency range as wide as the shaft speed. This permanently mounted probe measures the relative motion between the point of mounting and the rotor.

Selecting a Measure

Absolute displacement

which is used for low-frequency vibration (0 to 10Hz) measured on the bearing pedestal, relates to stress (shaft or structure) and is typically measured with a double integrated accelerometer. It is called seismic vibration. Absolute displacement of a shaft must be measured with either a contacting sensor or a noncontacting sensor in combination with a seismic sensor mounted on the bearing pedestal.

Selecting a Measure

Velocity

For general machinery monitoring and analysis in the span from 10 Hz to 1,000 Hz, velocity is the default measure. Velocity as a time rate of change of displacement is dependent upon both frequency and displacement and related to fatigue. It has been shown to be a good measure in the span for 10Hz to 1,000 Hz because a single value for rms or peak velocity can be used in rough assessments of condition without the need to consider frequency. Most modem data collectors use accelerometers but the signal must be integrated to obtain velocity.

Selecting a Measure

Acceleration

is the measure used above 1,000 Hz; it relates to force and is used for such high- frequency vibrations as gearmesh and rolling element bearing defects. Acceleration and velocity are absolute measures taken on the bearing housing or as close to the bearing as possible.

Selecting a Measure

• Measure	Useful Frequency Span	Physical Parameter	Application
Relative displacement (Proximity probe)	0 – 1000 HZ	stress/motion	relative motions in bearings/casings.
Absolute displacement (seismic)	0 – 10 Hz	stress/motion	machine condition
Velocity (seismic)	10 – 1000 Hz	energy/fatigue	general machine, medium-frequency vibrations
Acceleration (seismic)	>1000 Hz	force	general machine, medium-high-frequency vibrations

Selecting a Measure

The rule of thumb for measure selection is that velocity is used for bearing pedestal measurement up to 1,000 RPM and acceleration is used above that machine speed. If the machine has permanent non-contacting displacement sensors, then displacement is acquired.

Selecting a Measure

FREQUENCIES
Bearing Frequencies

$$\mathrm{FTF} = \left(\frac{\Omega}{2}\right)\left[1 - \left(\frac{\mathrm{B}}{\mathrm{P}}\right)\cos \mathrm{CA}\right]$$

$$\mathrm{BPFI} = \left(\frac{\mathrm{N}}{2}\right)\Omega\left[1 + \left(\frac{\mathrm{B}}{\mathrm{P}}\right)\cos \mathrm{CA}\right]$$

$$\mathrm{BPFO} = \left(\frac{\mathrm{N}}{2}\right)\Omega\left[1 - \left(\frac{\mathrm{B}}{\mathrm{P}}\right)\cos \mathrm{CA}\right]$$

$$\mathrm{BSF} = \left(\frac{\mathrm{P}}{2\mathrm{B}}\right)\Omega\left[1 - \left(\frac{\mathrm{B}}{\mathrm{P}}\right)^2 \cos^2 \mathrm{CA}\right]$$

FTF = fundamental train frequency
BPFI = ball pass frequency, inner race
BPFO = ball pass frequency, outer race
BSF = ball spin frequency
RPM = shaft speed
B = ball or roller diameter, in
Bearing defect frequencies are same units as machine speed

CA = contact angle
Ω = machine speed
N = number of rolling elements
P = pitch diameter, in

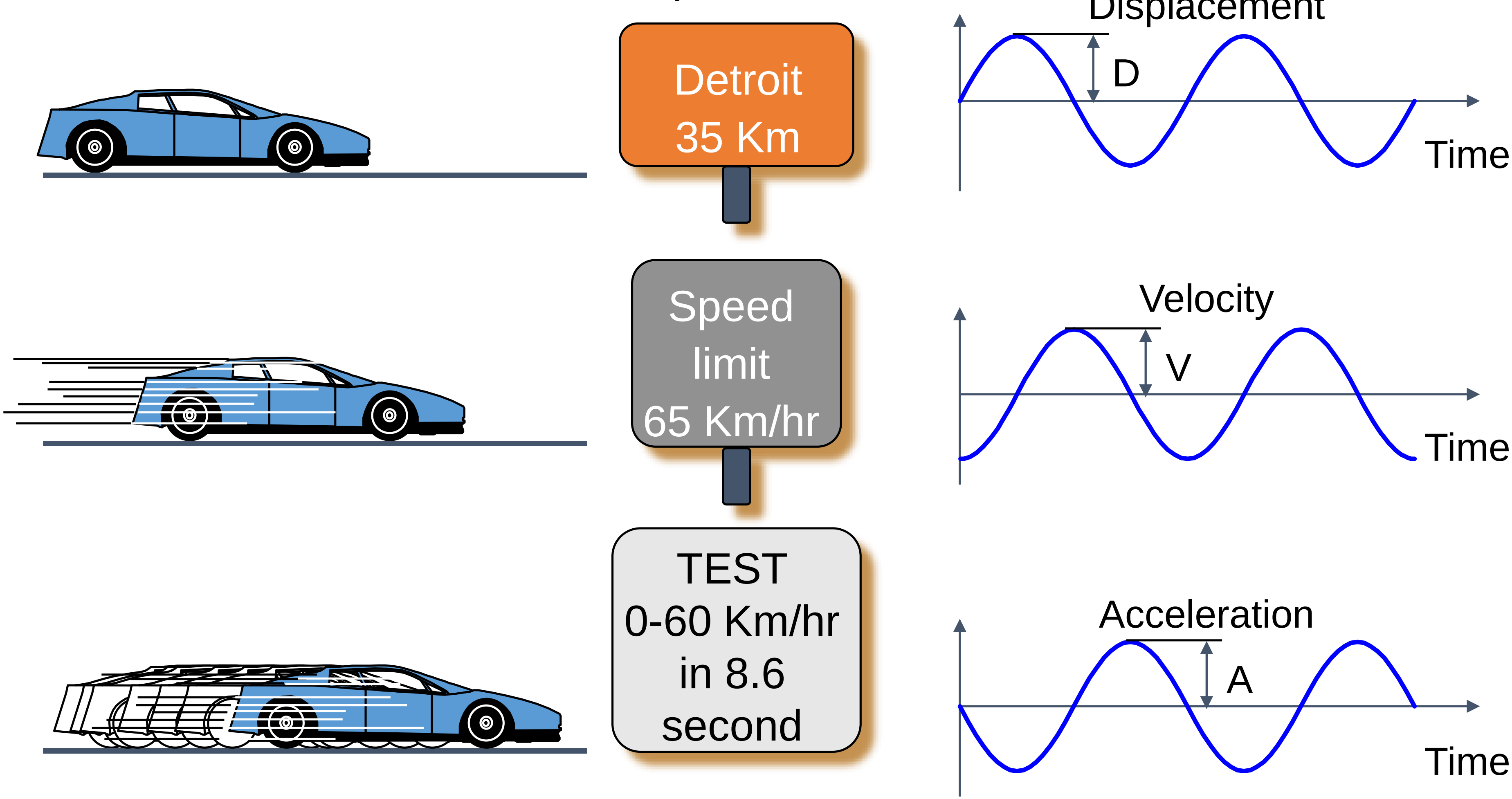
Selecting a Measure

FREQUENCIES

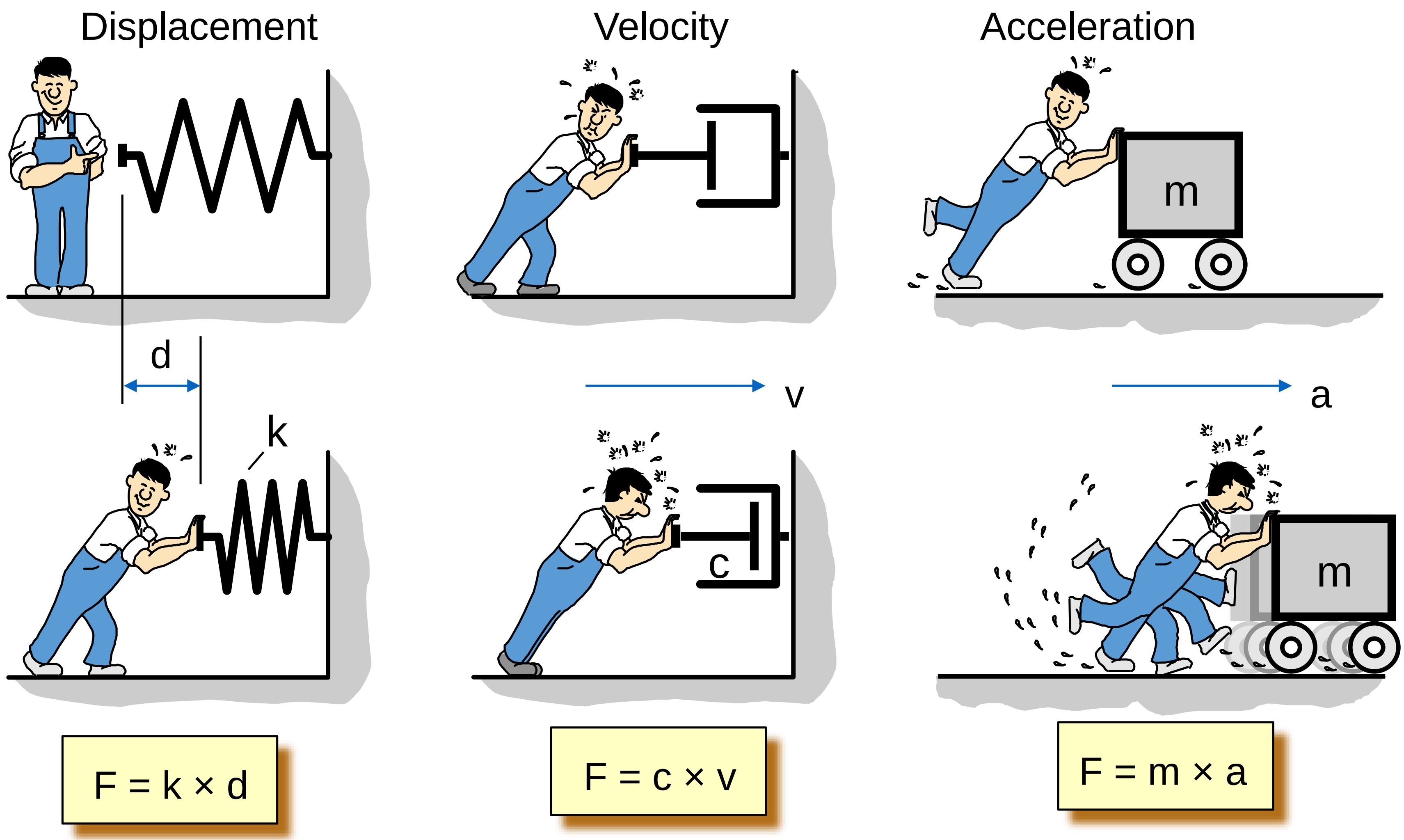
FAN

blade pass frequency = no blades x RPM

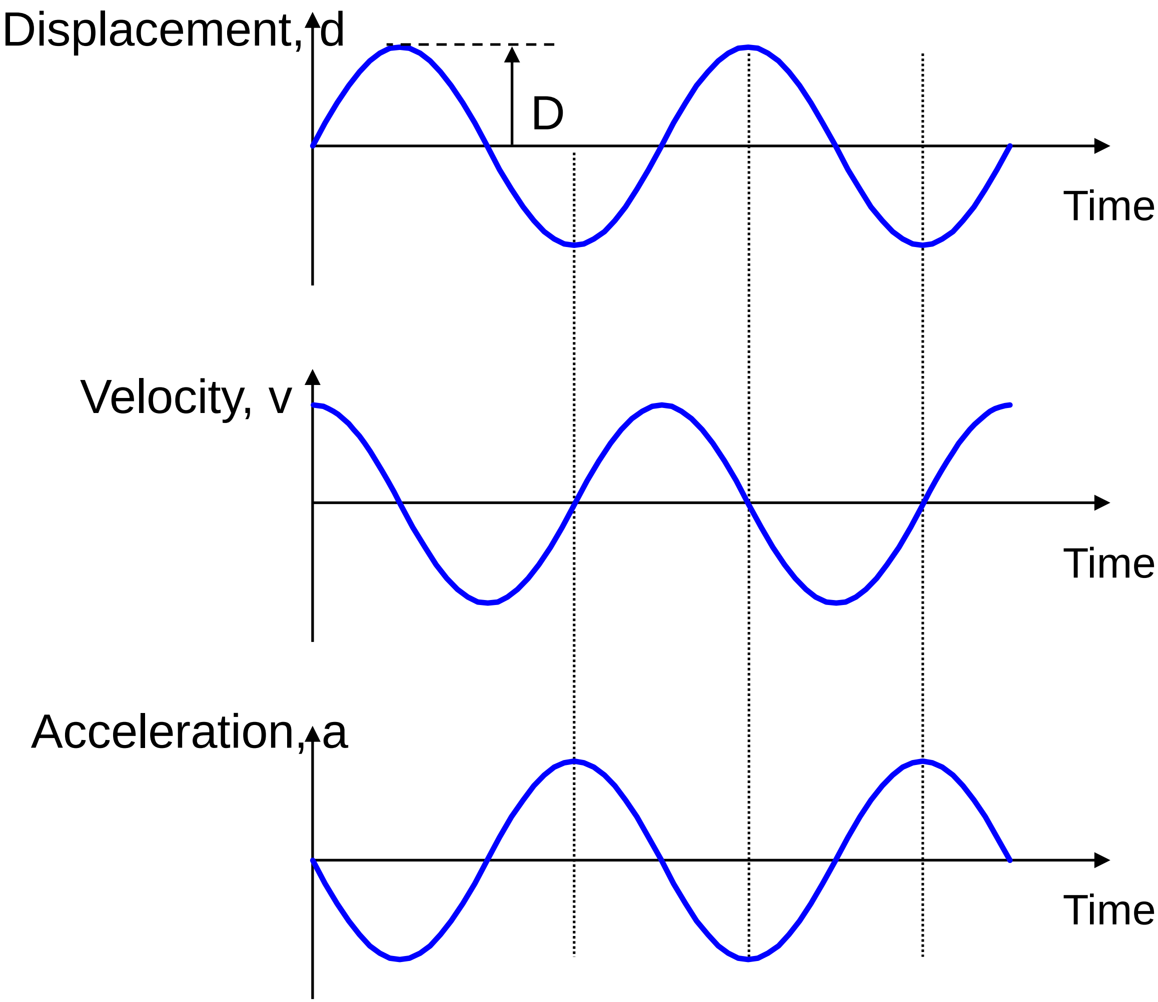
Linear vs. Oscillatory Motion



Mechanical Parameters and Components



Conversion from Displacement to Acceleration



$$d = D \sin \omega t$$

$$\boxed{d = D}$$

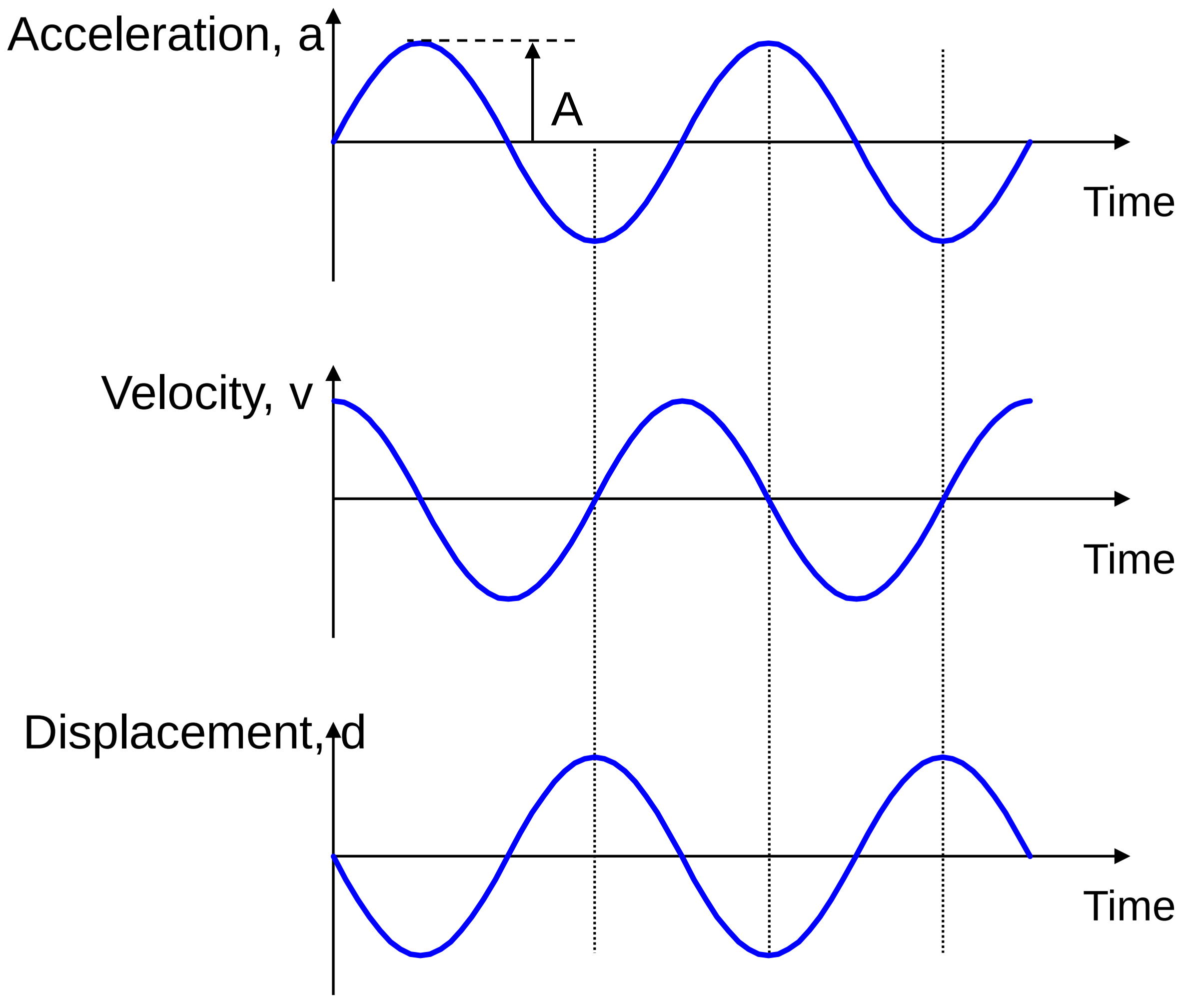
$$v = \frac{dd}{dt} = D\omega \cos \omega t$$

$$\boxed{v = D\omega = D2\pi f}$$

$$a = \frac{d^2d}{dt^2} = D\omega^2 \sin \omega t$$

$$\boxed{a = D\omega^2 = D4\pi^2 f^{\,2}}$$

Conversion from Acceleration to Displacement



$$a = A \sin \omega t$$

$$\boxed{a = A}$$

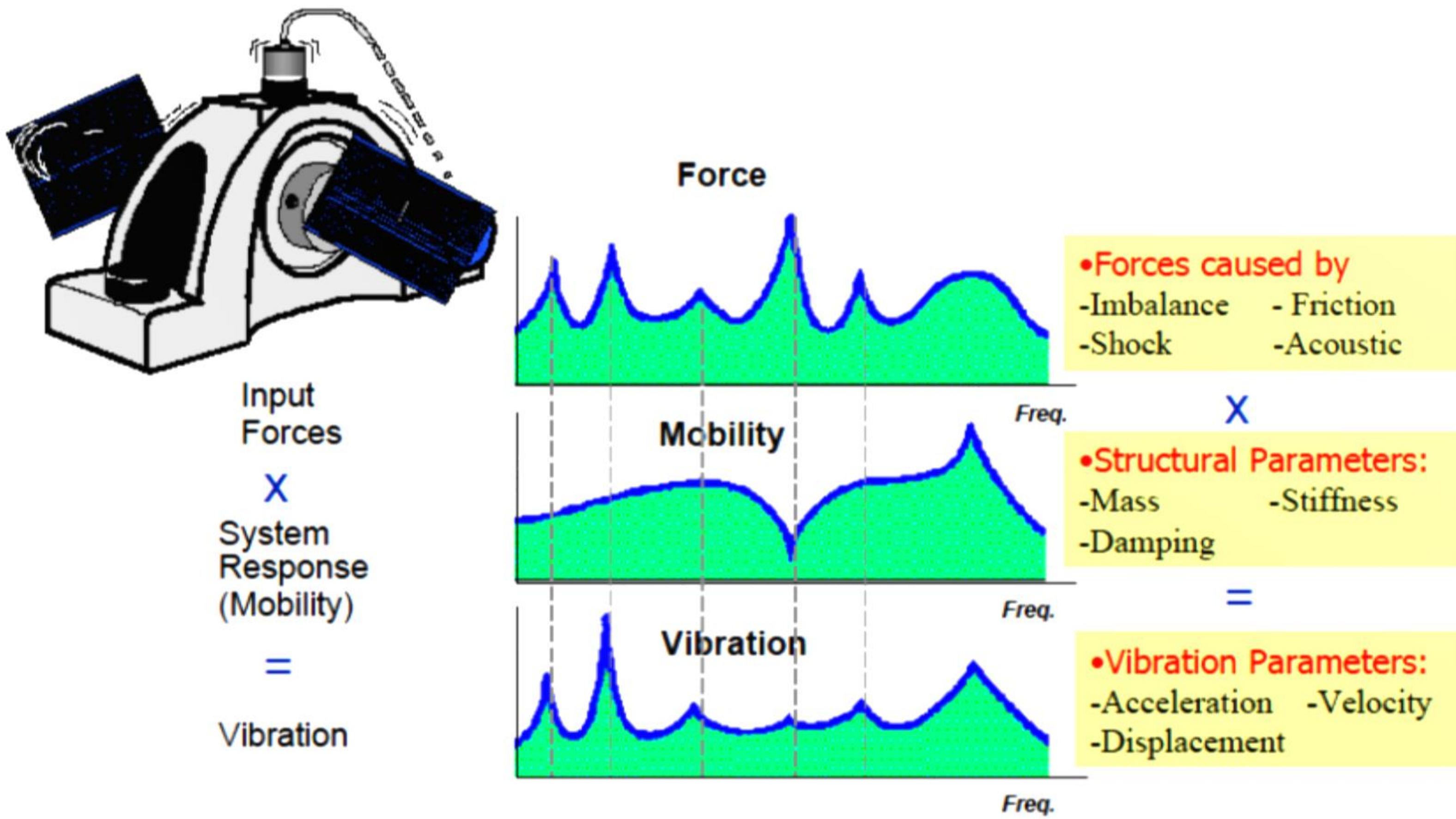
$$v = \int a \, dt = -\frac{A}{\omega} \cos \omega t$$

$$\boxed{v = \frac{A}{\omega} = \frac{A}{2\pi f}}$$

$$d = \iint a \, dt \, dt = -\frac{A}{\omega^2} \sin \omega t$$

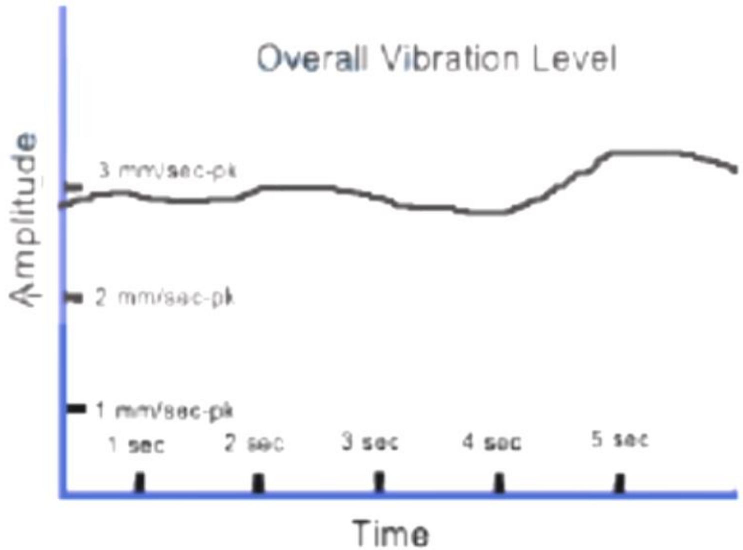
$$\boxed{d = \frac{A}{\omega^2} = \frac{A}{4\pi^2 f^{\,2}}}$$

What is Vibration



Overall Amplitude

- It is the total vibration amplitude over a wide range of frequencies.
- Acceleration, Velocity, or Displacement.



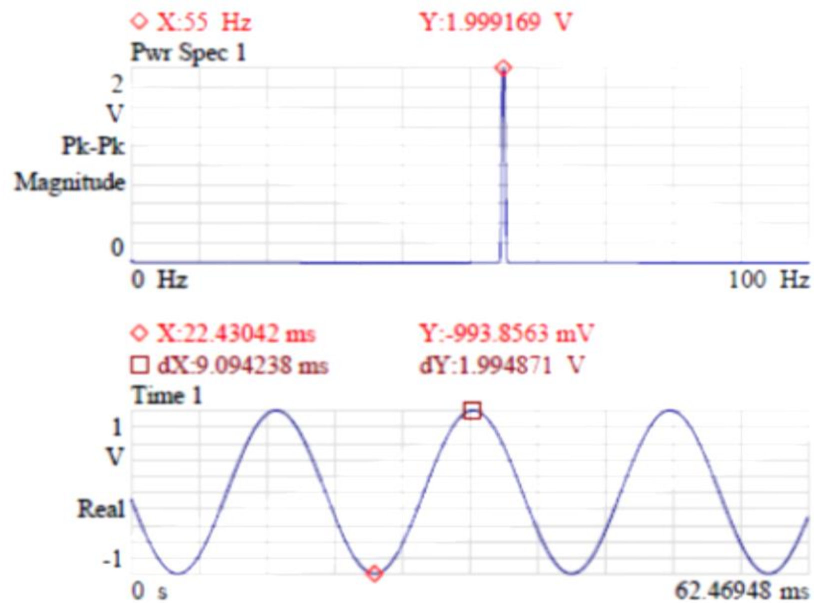
Vibration Terminology

- Displacement [peak-peak]
- Velocity [peak]
- Velocity [rms]
  - Velocity rms tends to provide the energy content in the vibration, whereas the Velocity peak depicts more of the intensity of vibration.
- Acceleration - peak

Pk-Pk (Peak - Peak)

The Peak - Peak value is expressed from the peak to peak amplitude.

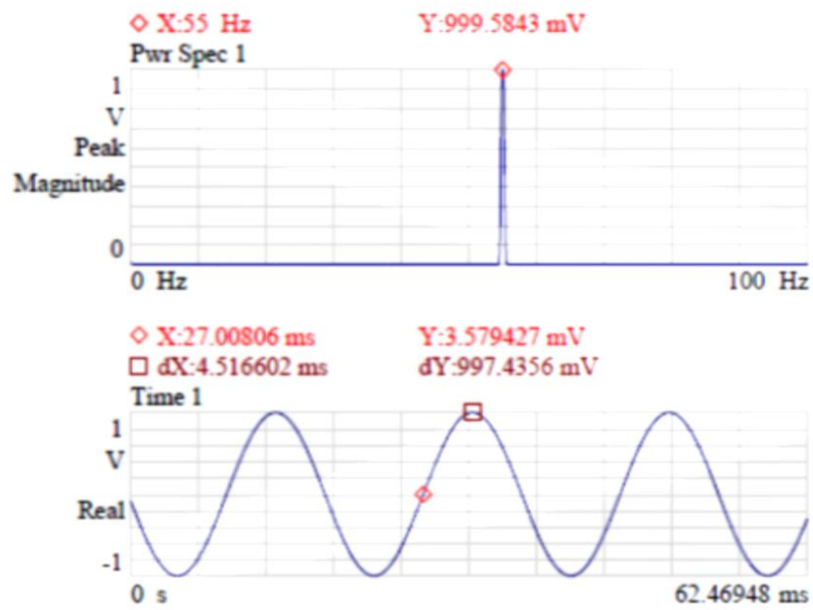
The spectrum value uses the suffix “Pk-Pk” to denote this.



Pk (Peak)

The time wave has not changed. The Peak value is expressed from zero to the peak amplitude.

The spectrum value uses the suffix “Peak” to denote this.



Vibration Measurement

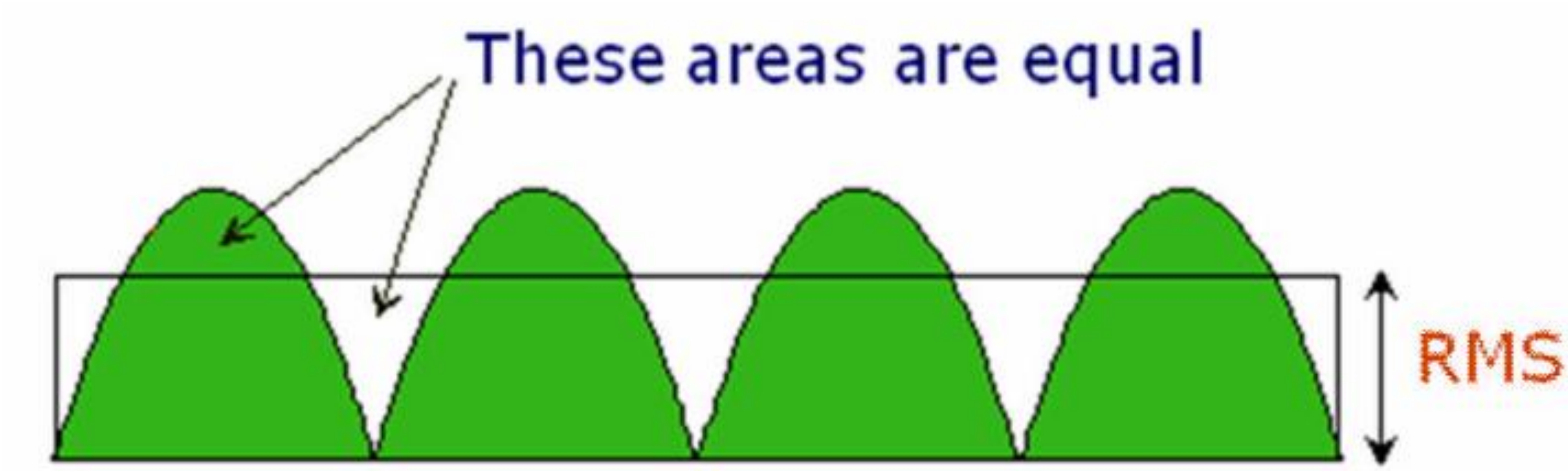
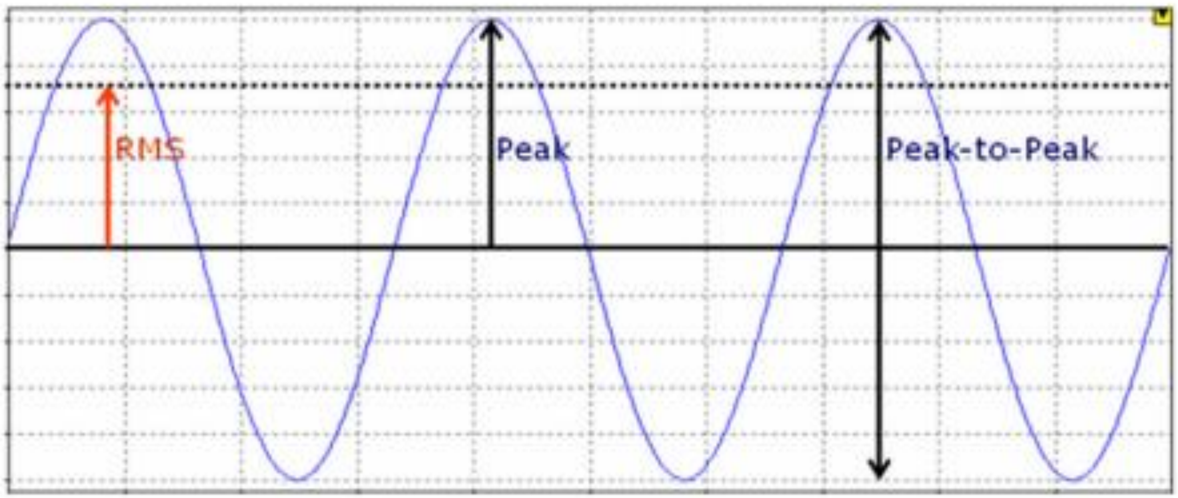


Figure 3-41 The RMS is calculated after rectifying (making the negative parts positive) the waveform

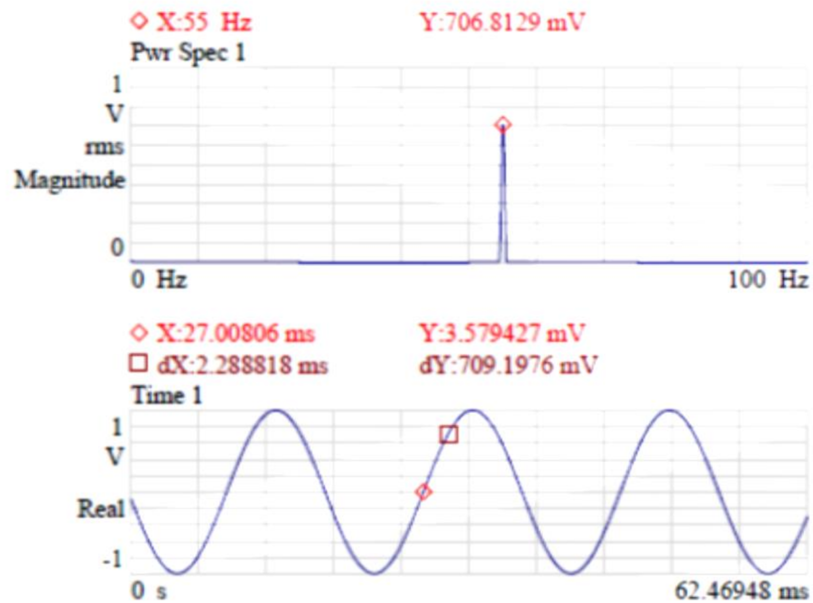
Vibration Measurement



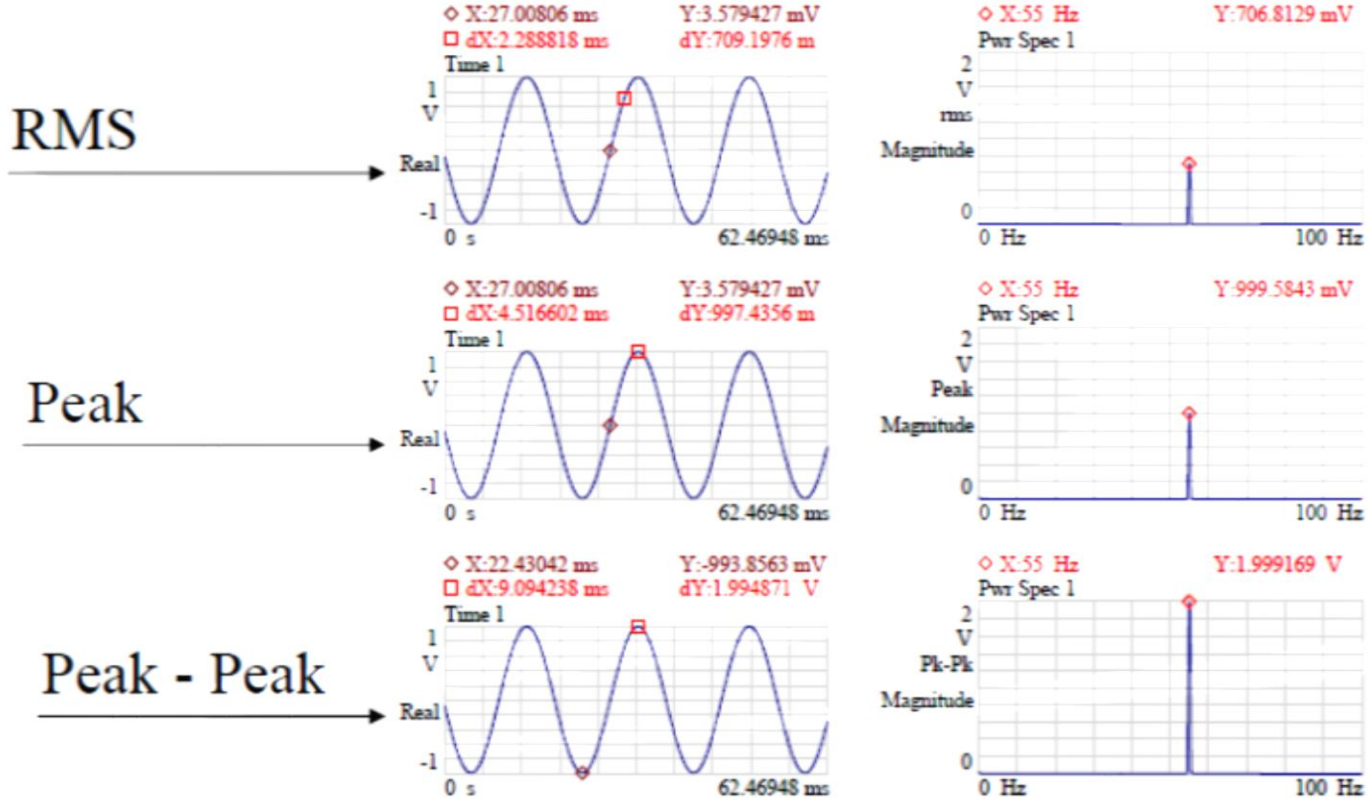
RMS (Root Mean Square)

The time wave has not changed. The RMS value is expressed from zero to 70.7% of the peak amplitude.

The spectrum value uses the suffix “RMS” to denote this.

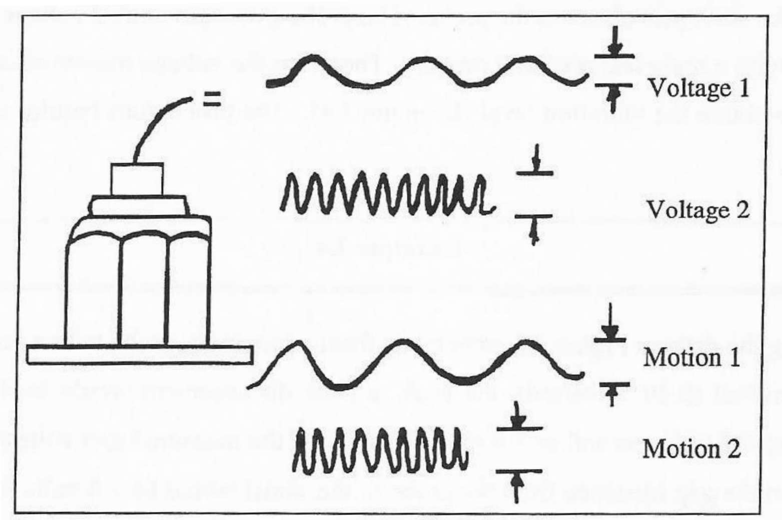


Suffix Comparison



Vibration Sensors

Magnitude, frequency, and phase between two signals are used for evaluation. Sensor selection is based on sensitivity, size required, selected measure, frequency response, and machine design and speed. The sensor should be mounted as close to the source of vibration as possible.



Vibration Sensors

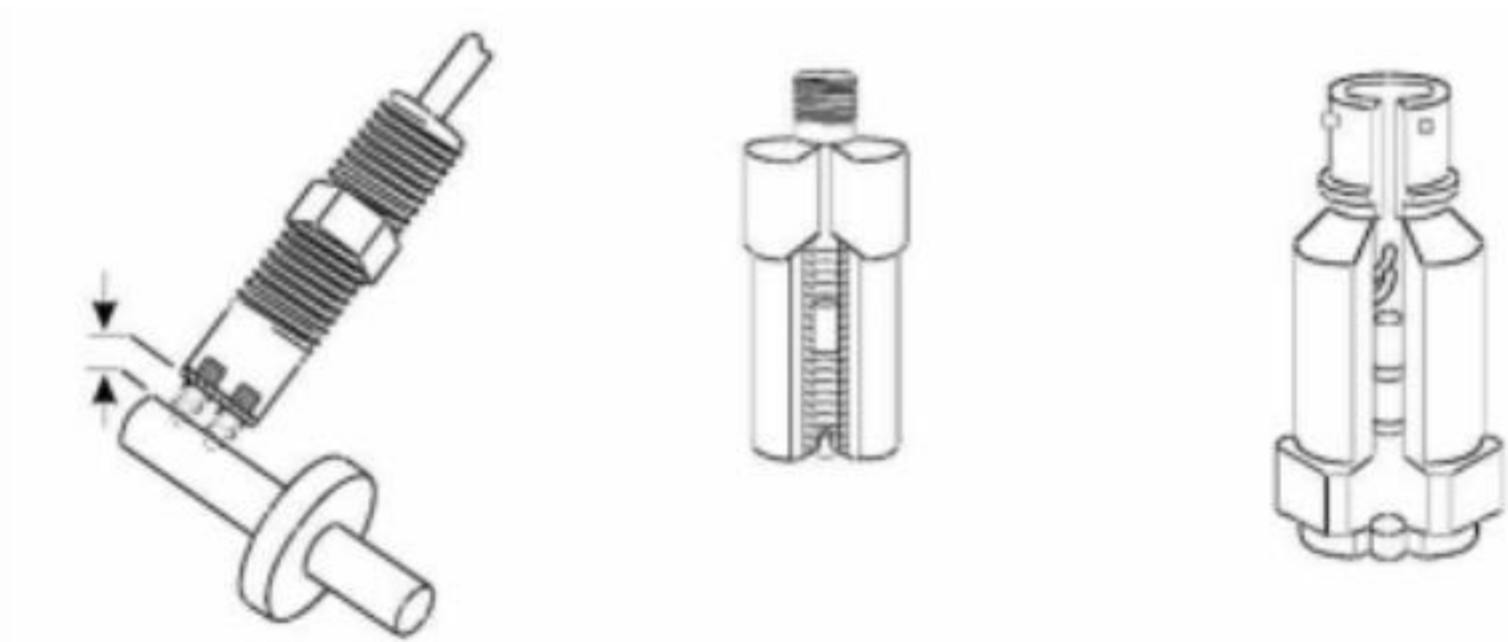
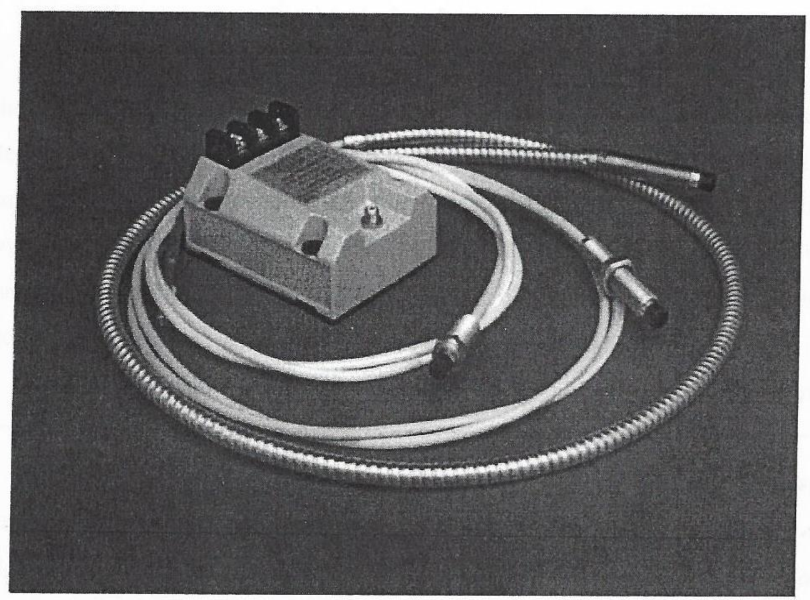


Figure 4-4 A proximity probe, a velocimeter, and an accelerometer.

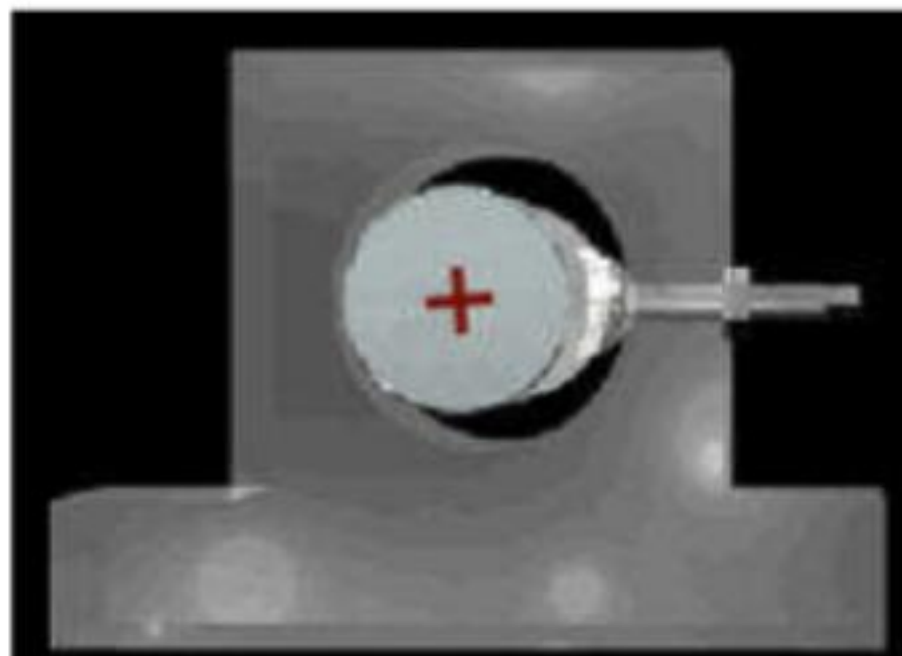
Proximity probes

The proximity probe (non-contacting eddy current displacement transducer) shown in Figure 3.5 measures static and dynamic displacement of a shaft relative to the bearing housing. It is permanently mounted on many large (greater than 1,000 HP) machines for monitoring (protection and trending) and analysis.



Proximity probes

The probe generates a negative DC voltage proportional to the distance of the shaft from the sensor (gap). The typical gap is 40 mils or at 200 mv/mil, 8 volts. The negative voltage decreases as the shaft gets closer to the probe. The probe generates an AC voltage proportional to the vibration with a scale factor of 200 mv/mil. Therefore, the voltage measured is divided by the scale factor to obtain the vibration level (Example 3.4). The probe doe power supply.

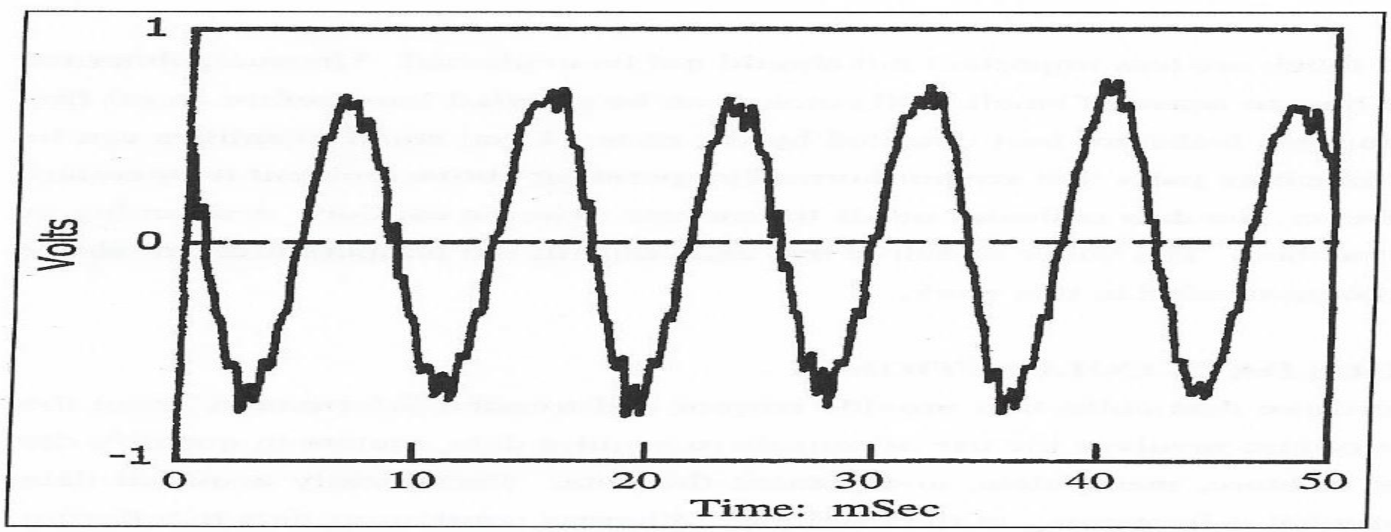


Proximity Probe

Figure 4-7 Eddy current probes have 3 components: the probe, a driver, and the cable between them.

Example

Assuming the data on Figure were taken from a proximity probe with a scale factor of 200 mv/mm (0.20 Volts/mm), the peak to peak displacement would be 1.58 volts divided by 0.2 volts per mm or 7.9 mm-pk to pk. If the measured gap voltage was 7.6 volts, then the gap (distance from the probe to the shaft) would be 7.6 volts divided by 0.2 Volts/mm or 38 mm.



Example

Select a measure and sensor for a fan operating at 950 RPM. The fan has seven (7) blades and fifteen (15) rolling elements in its bearings.
The frequencies of interest are operating speed and orders, blade pass frequency and multiples, and rolling element fault frequencies, and multiples.

$$\text{operating speed frequency} = \frac{950 \text{ RPM}}{60} = 15.83 \text{ Hz and orders}$$

$$\text{blade pass frequency} = \text{no blades x RPM}$$

$$\text{blade pass frequency} = \frac{950 \text{ RPM}}{60} \times 7 = 110.8 \text{ Hz and multiples}$$

$$\text{ball pass frequency of inner race} = 0.6 \text{ x no. balls x RPM}$$

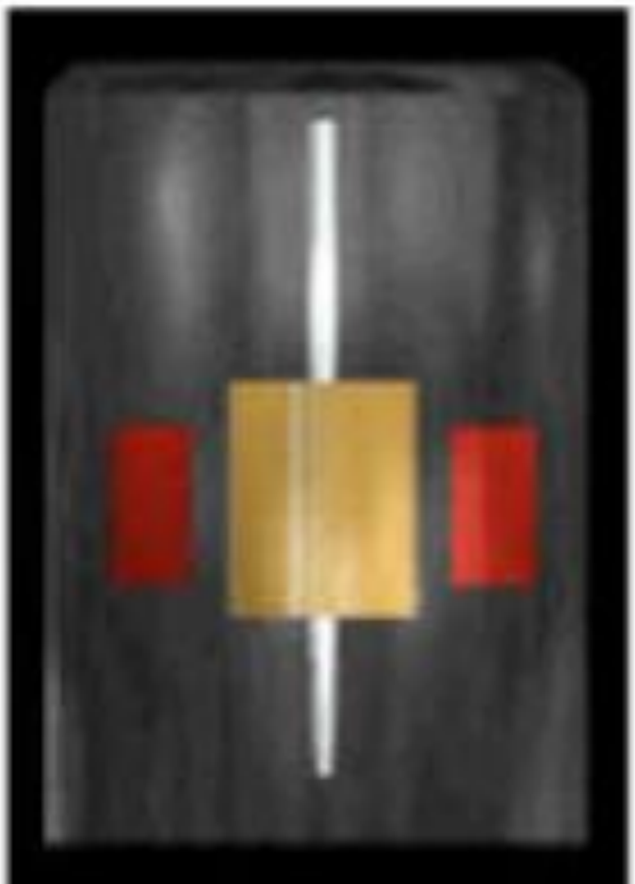
$$\text{bearing fault frequency} = \frac{950 \text{ RPM}}{60} \times 0.6 \times 15 = 142.5 \text{ Hz and multiples}$$

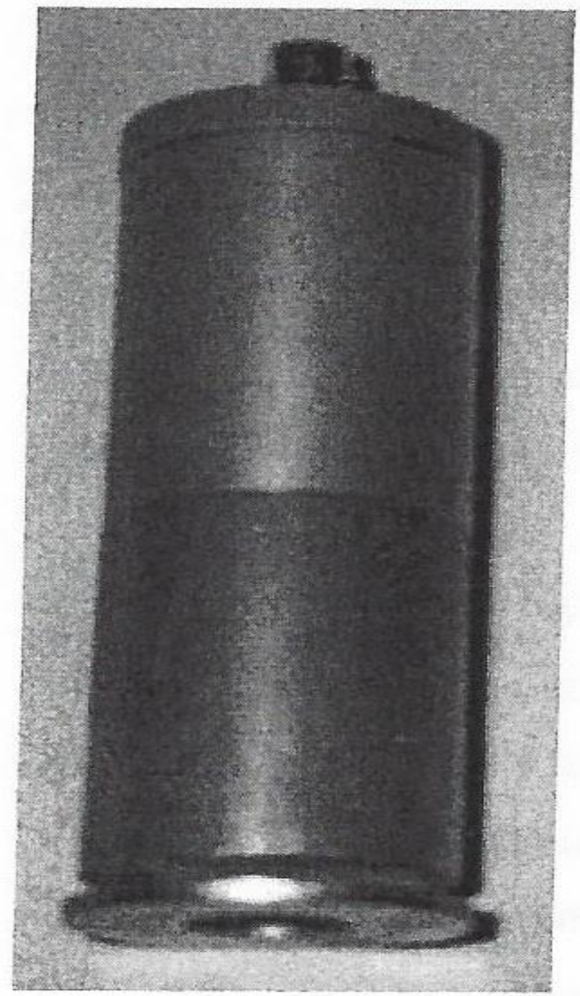
The majority of the frequency activity is between 10 and 1,000 Hz. Therefore, velocity measure will provide the best information. An integrated accelerometer or velocity sensor can be used to acquire the data.

Velocity transducers

Velocity transducers. The velocity transducer is a seismic transducer (i.e., it measures absolute vibration) that is used to measure vibration levels on casings or bearing housings in the range from 10 Hz to 2,000 Hz. The transducer is self-excited - that is, it requires no power supply. The self-generated signal can be directly passed to an oscilloscope, meter, or analyzer for evaluation. A typical velocity transducer generates 500 mv/(in./sec).

Velocity transducers





Accelerometers

Accelerometers are used to measure vibration levels on casings and bearing housings; they are the transducers typically supplied with electronic data collectors. An accelerometer (Figure 3.7) consists of a small mass mounted on a piezoelectric crystal that produces an electrical output proportional to acceleration when a force is applied from the vibrating mass.

Accelerometers

Figure 4-12 Accelerometers are typically mounted on a bearing housing via magnet

Accelerometers

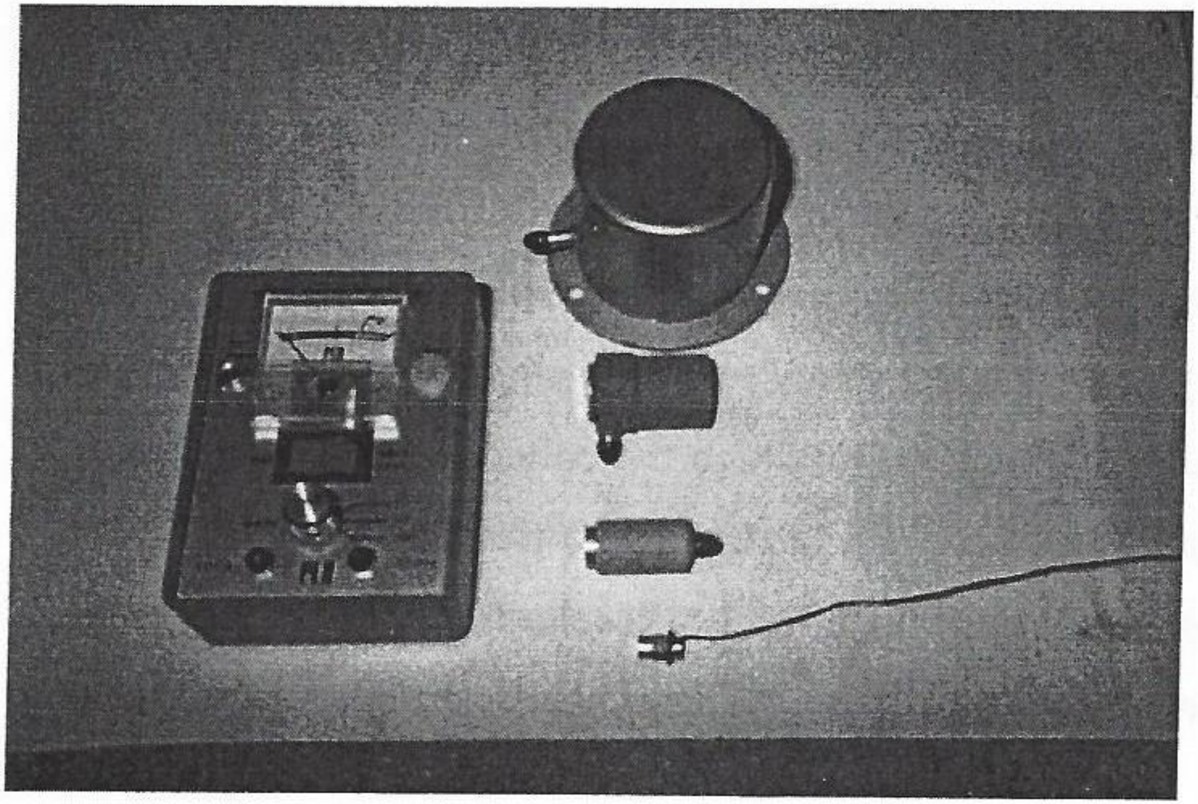
Accelerometers

Figure 4-15 Triaxial accelerometers have 3 accelerometers in one unit

Accelerometers

The size of an accelerometer is proportional to its sensitivity. Small accelerometers (the size of a pencil eraser) have a sensitivity of 5 mv/g (1 g = 386.1 in./sec2) and a flat frequency response to 25 kHz. A 1,000 mv/g accelerometer, which is used for low-frequency measurement, may be as large as a velocity sensor; however, the limit of its usable frequency span may be to 1,000 Hz. The analyst should be aware of the properties of each accelerometer being used.

Accelerometers



Accelerometers

If vibration velocity is desired, the signal is usually integrated, which electronically converts acceleration to velocity, before it is recorded or analyzed; an analog integrator/power supply is shown in Figure 3.8.

Analog Integrator and Power

Accelerometers

Accelerometers are recommended for permanent seismic monitoring because of their extended life and because their cross sensitivity is low. (Cross sensitivity means that the transducer generates a signal in horizontal direction from vibration in the vertical direction.)
However, cable noise, transmission distance, and temperature sensitivity of the accelerometer must be carefully evaluated. Excellent guidelines are available from vendors for accelerometer use.

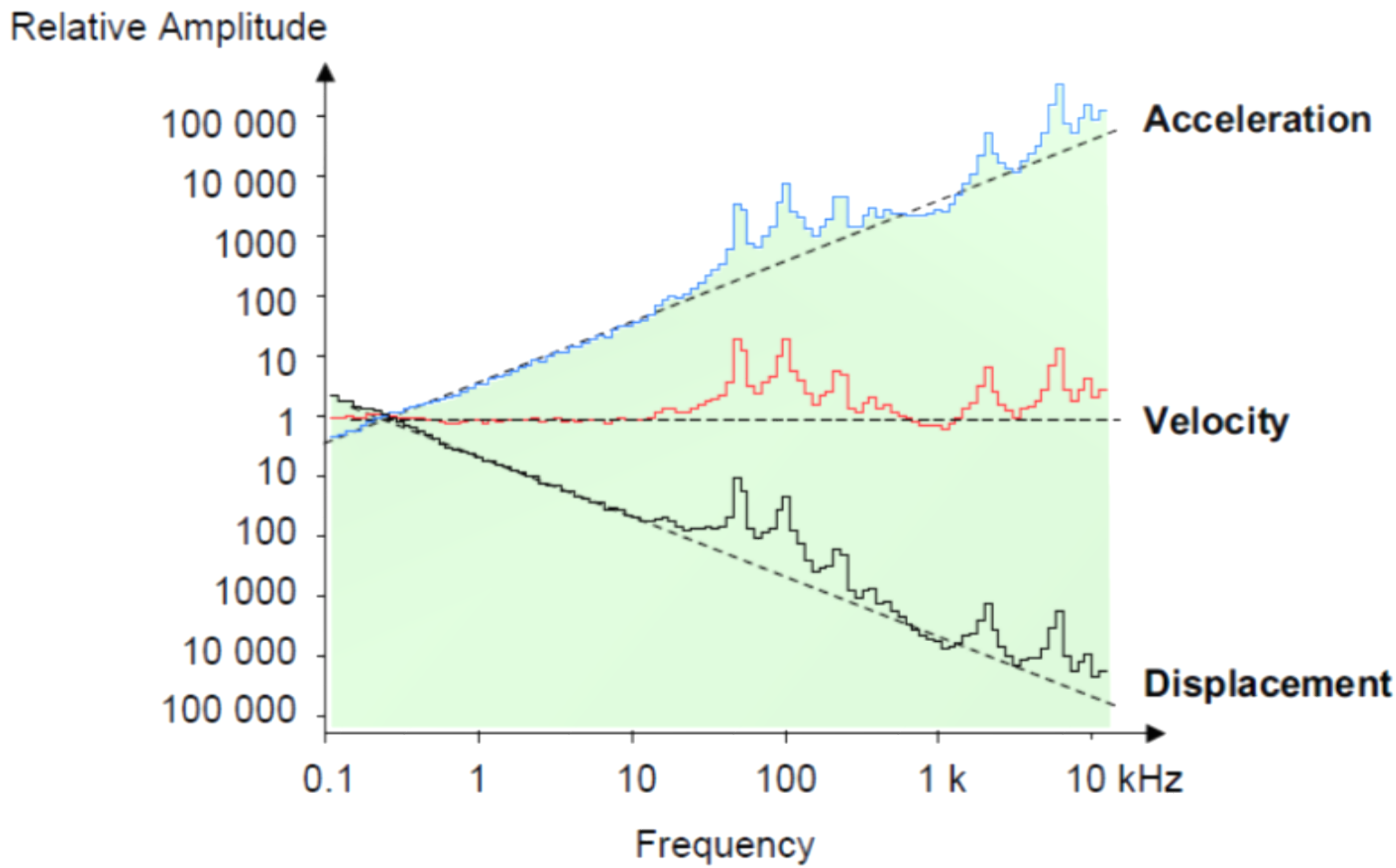
Sensor Selection

Important considerations in sensor selection include frequency response, signal-to-noise ratio, size, thermal and amplitude sensitivity of the sensor, and the strength of the signal being measured. The frequency range of the sensor must be compatible with the frequencies generated by the mechanical components of the machine. Otherwise, another transducer must be selected and the signal converted to the proper measure. For example, if the velocity measure is desired at frequencies above 2,000 Hz, an accelerometer integrated to velocity should be selected to obtain the signal. If the time waveform of the velocity measure is desired, the signal must be acquired from a velocity pickup or analog integrated signal from an accelerometer, either within or external to the data collector.

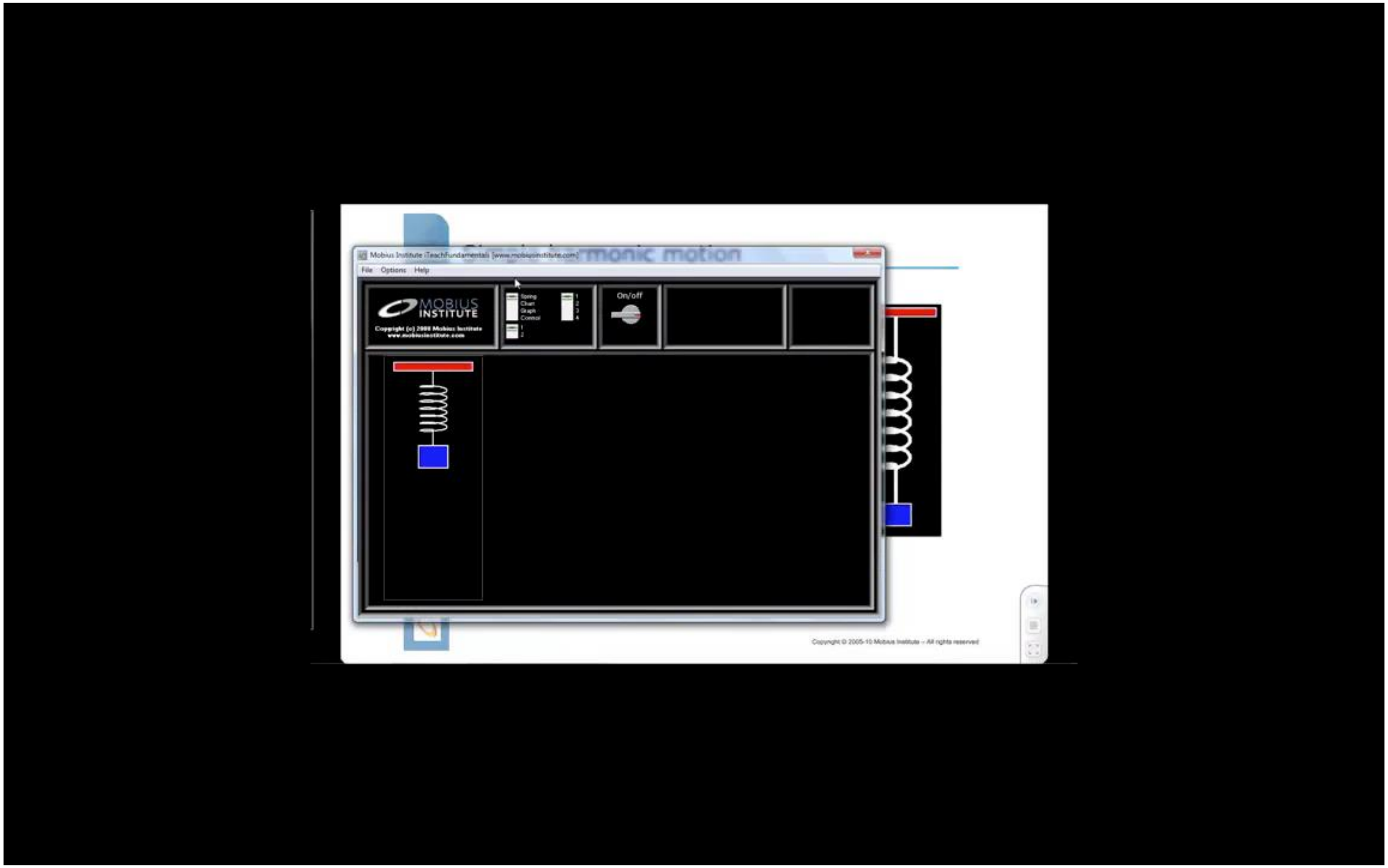
Sensor Selection

The cable that transmits the signal to the data collector can cause erroneous readings. Many standard cables are specially wound cords that are more convenient than the standard coaxial construction. But, because many conductors are flexible at the core, individual strands may fail at stress points as a result of handling or packing in a carrying case. In addition, the terminals must be handled carefully.

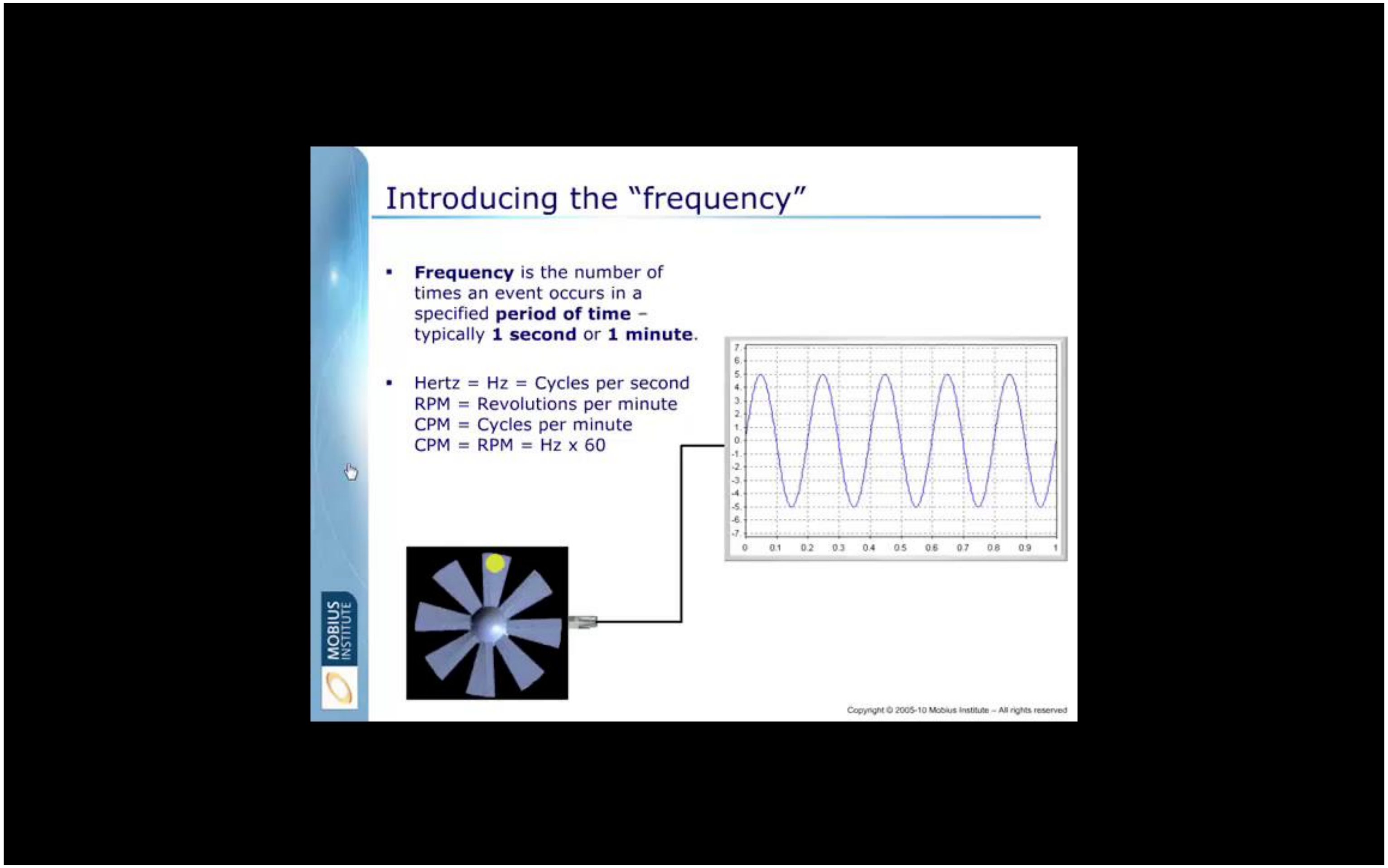
Vibration Parameters



Simple Harmonic Motion



Time period


Introducing the "frequency"
Frequency is the number of times an event occurs in a specified period of time – typically 1 second or 1 minute.
Hertz = Hz = Cycles per second
RPM = Revolutions per minute
CPM = Cycles per minute
CPM = RPM = Hz x 60
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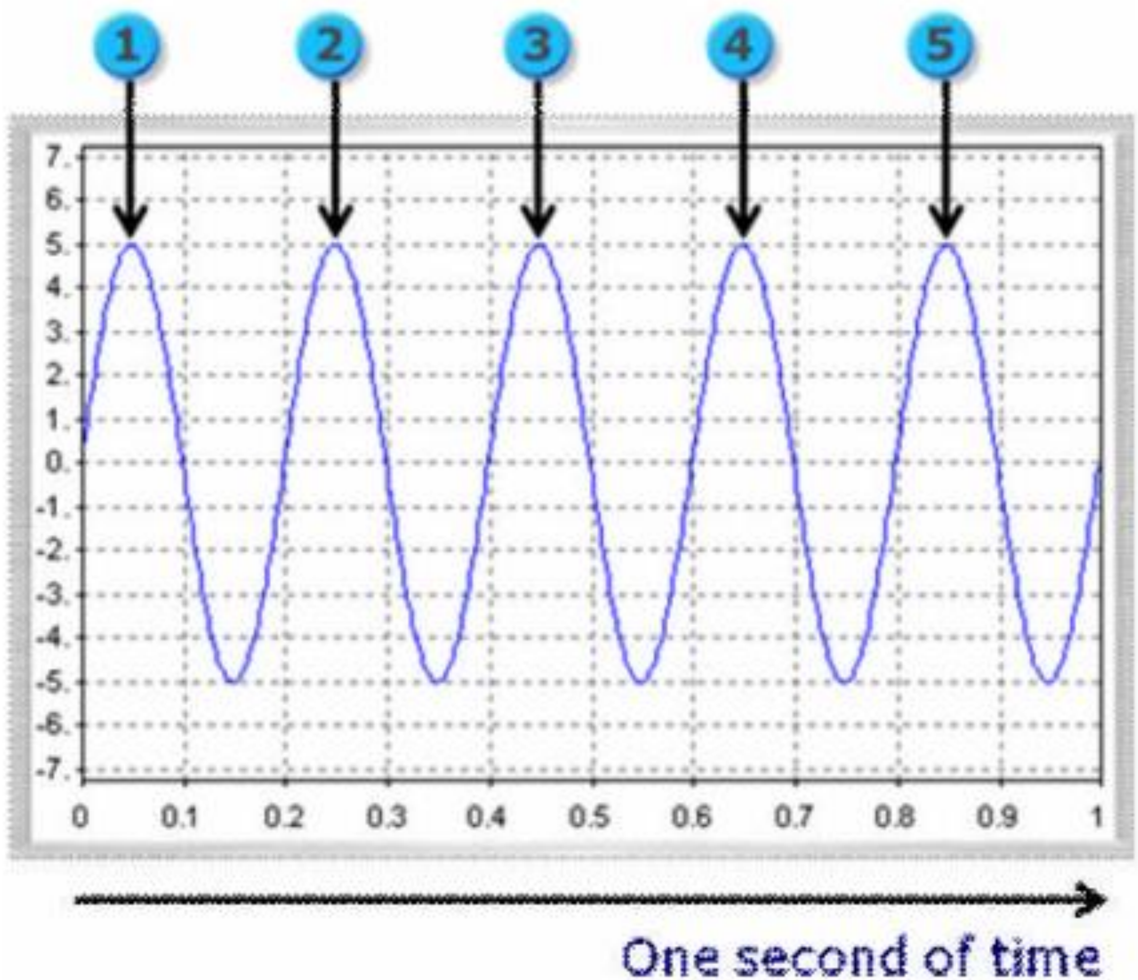
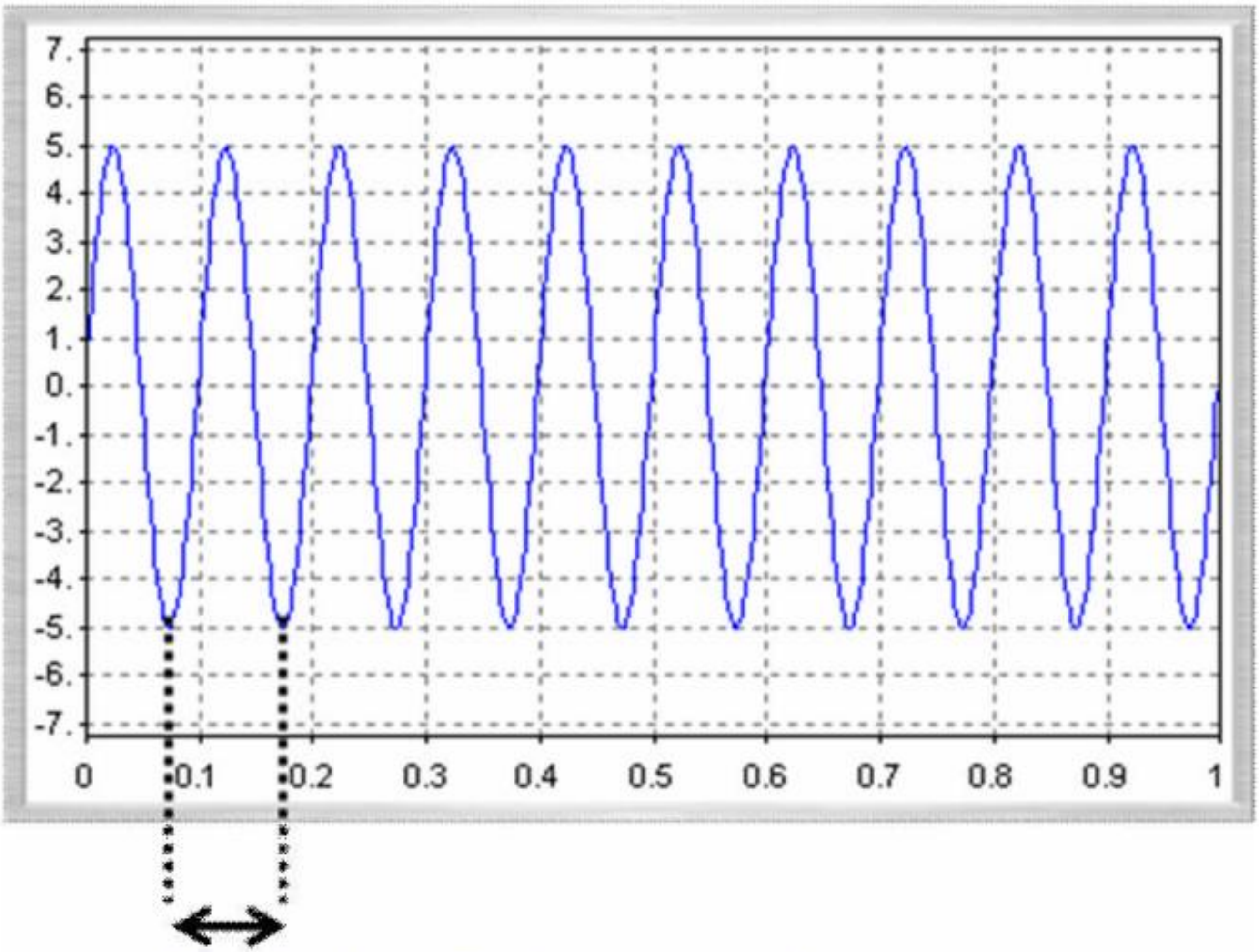
Vibration Measurement
Frequency



Vibration Measurement
Period



Period = 0.1 seconds

Vibration Measurement Amplitude

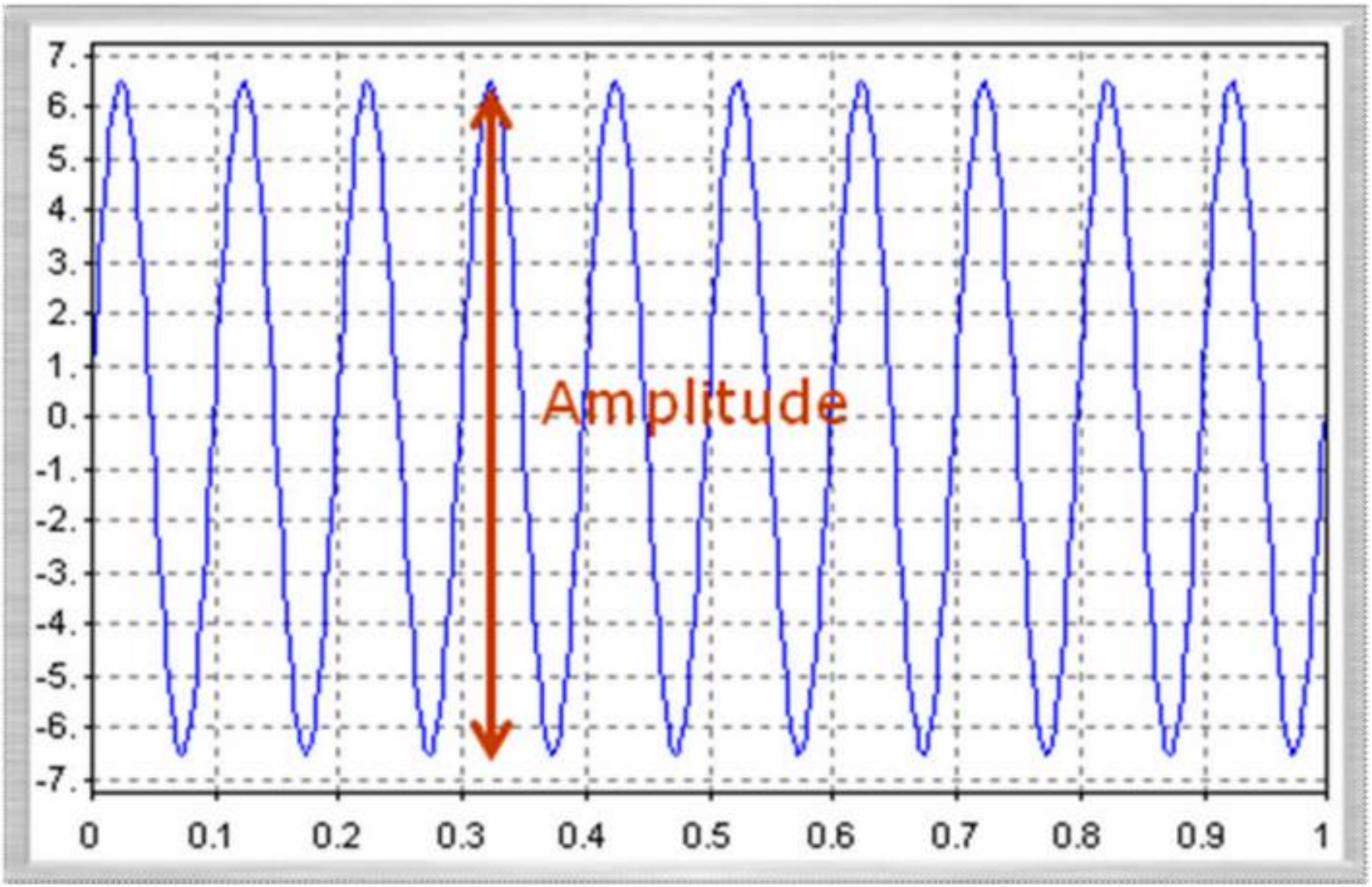


Figure 3-15 As speed increases, the amount of vibration, or amplitude, increases.

Vibration Measurement

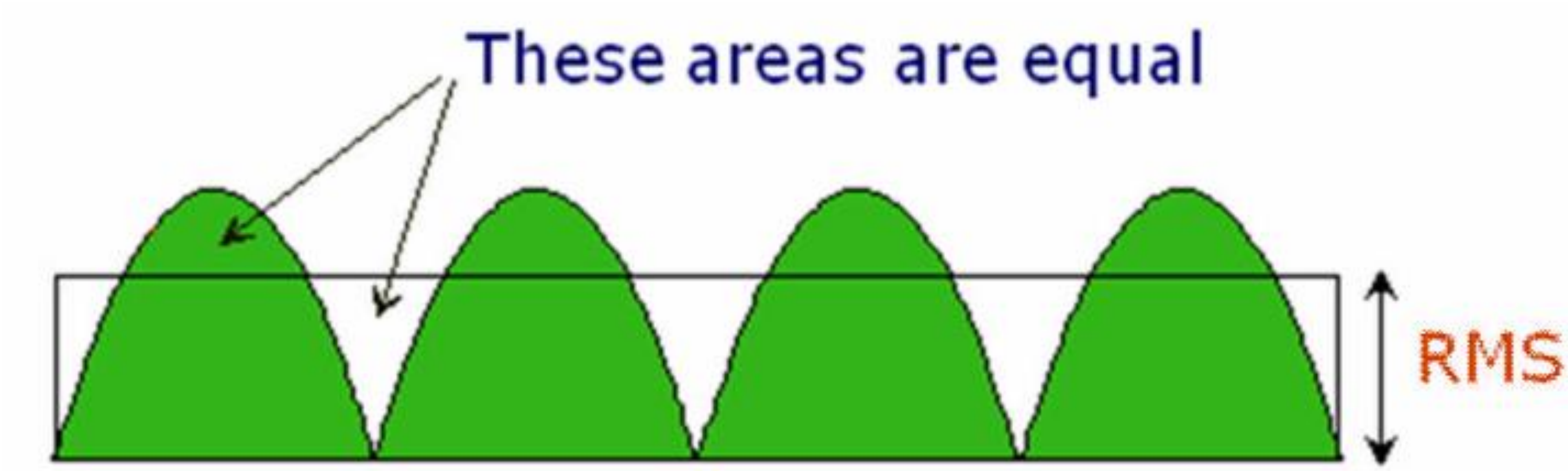
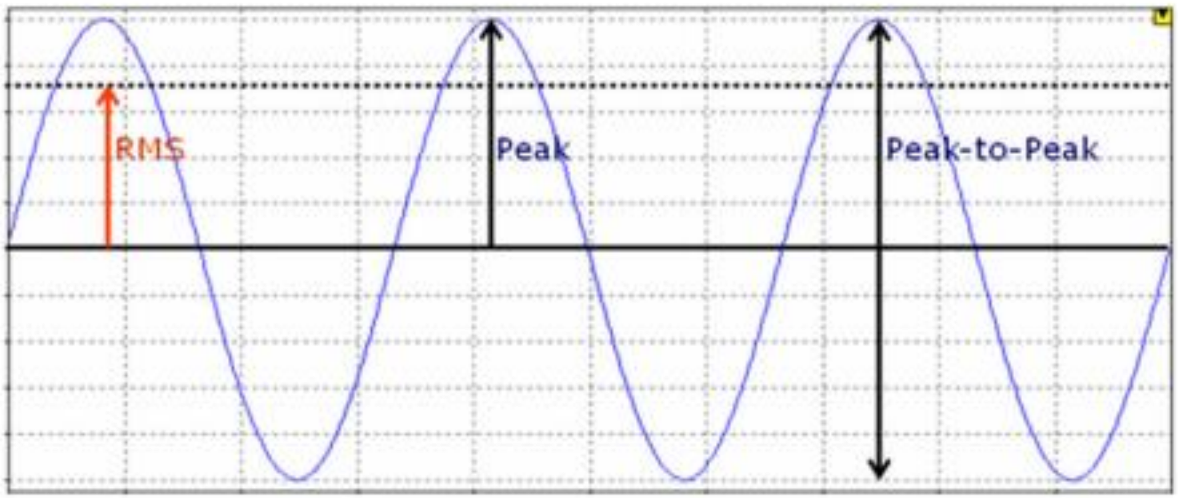
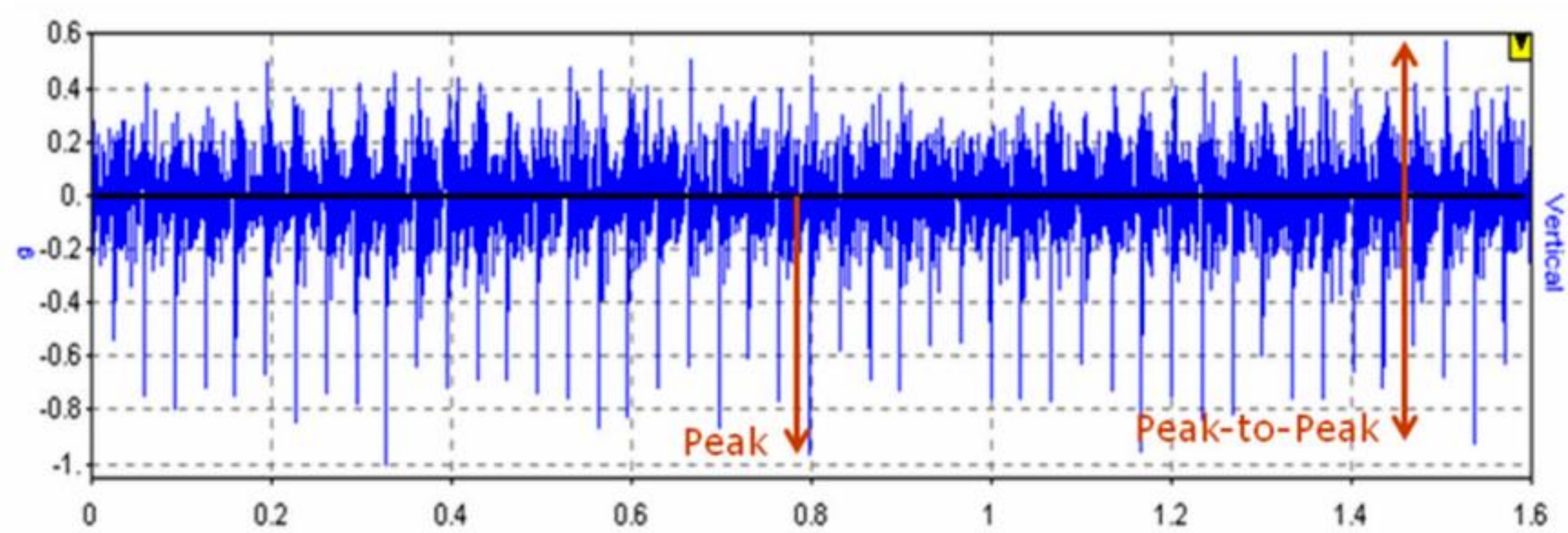


Figure 3-41 The RMS is calculated after rectifying (making the negative parts positive) the waveform

Vibration Measurement



Vibration Measurement



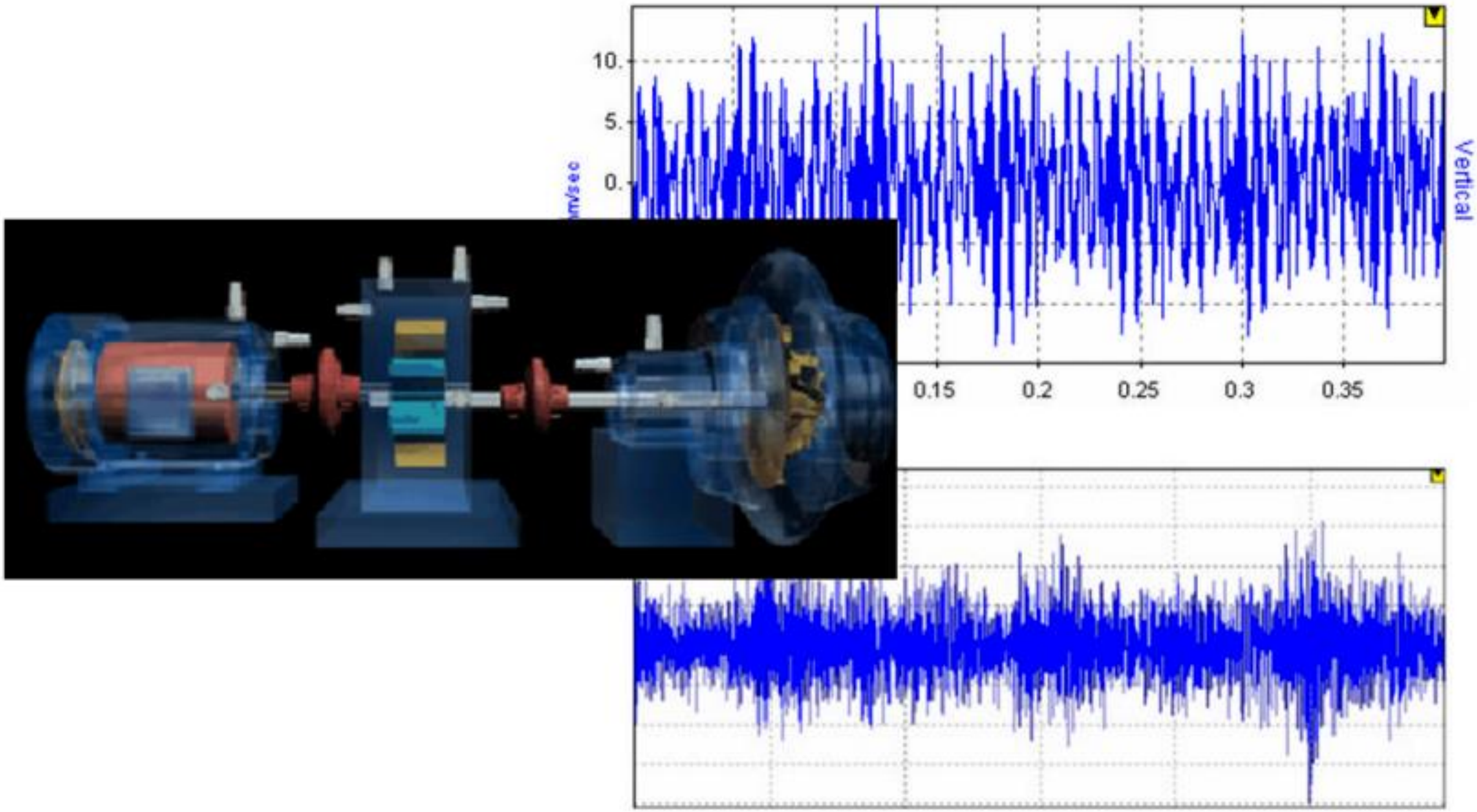
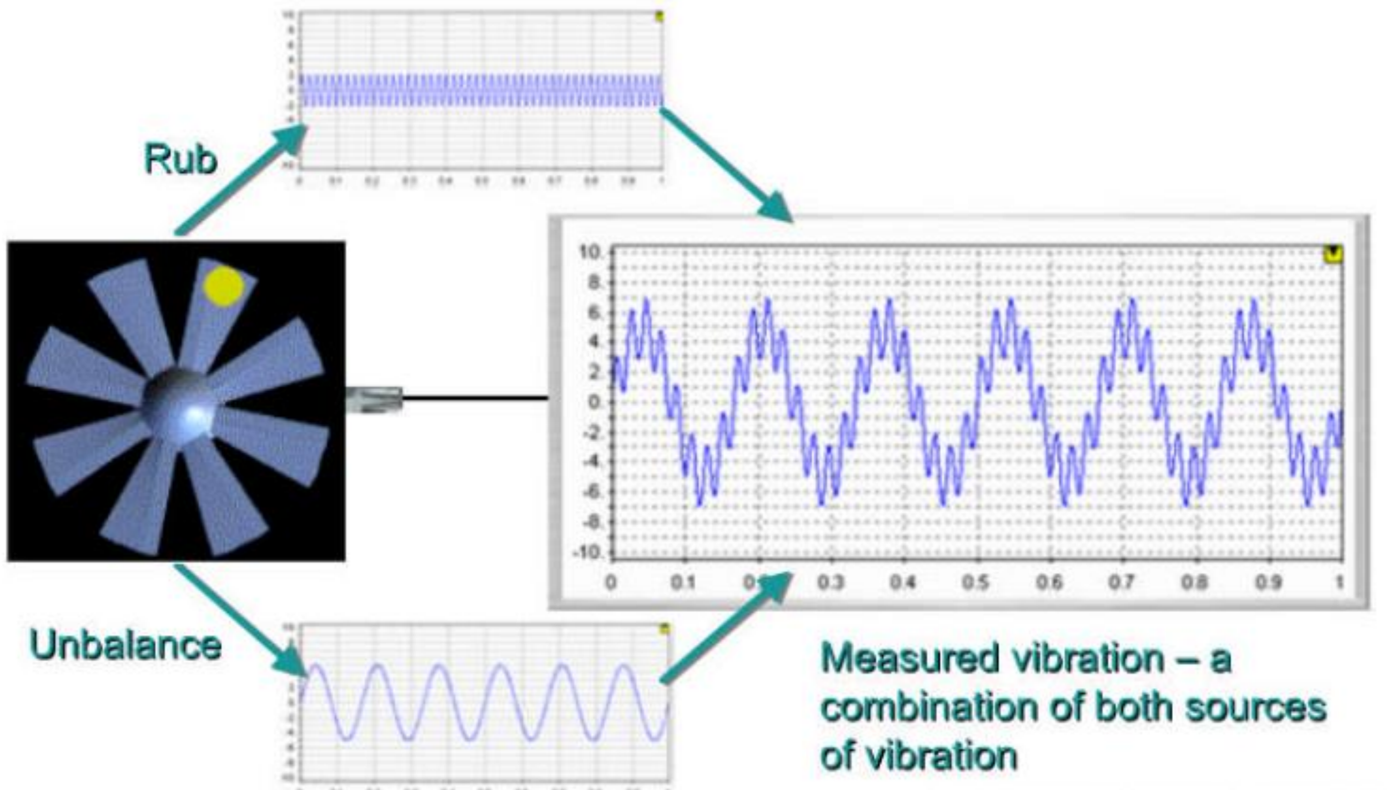


Figure 3-20 The goal is look inside the machine to see what is happening. Vibration can do that. The waveform can be too complex to sort out the sources.

Vibration Measurement



Spectrum

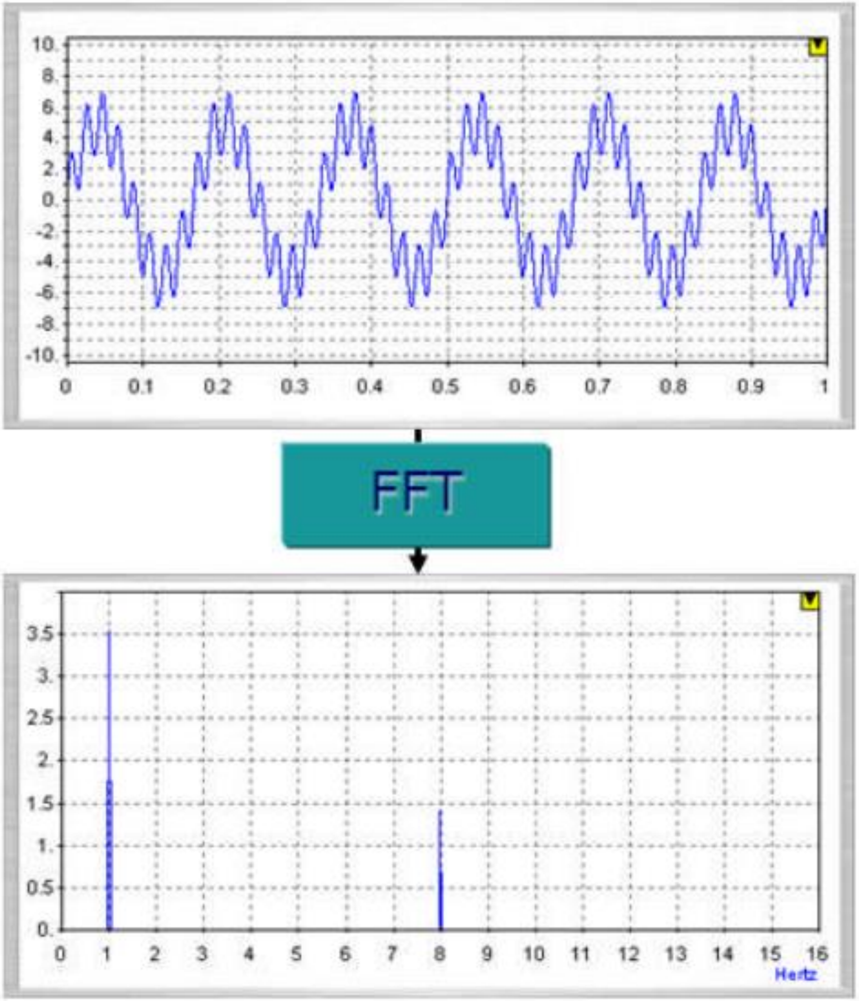


Figure 3-21 The FFT process separates the individual waveforms and displays them according to frequency.

Spectrum

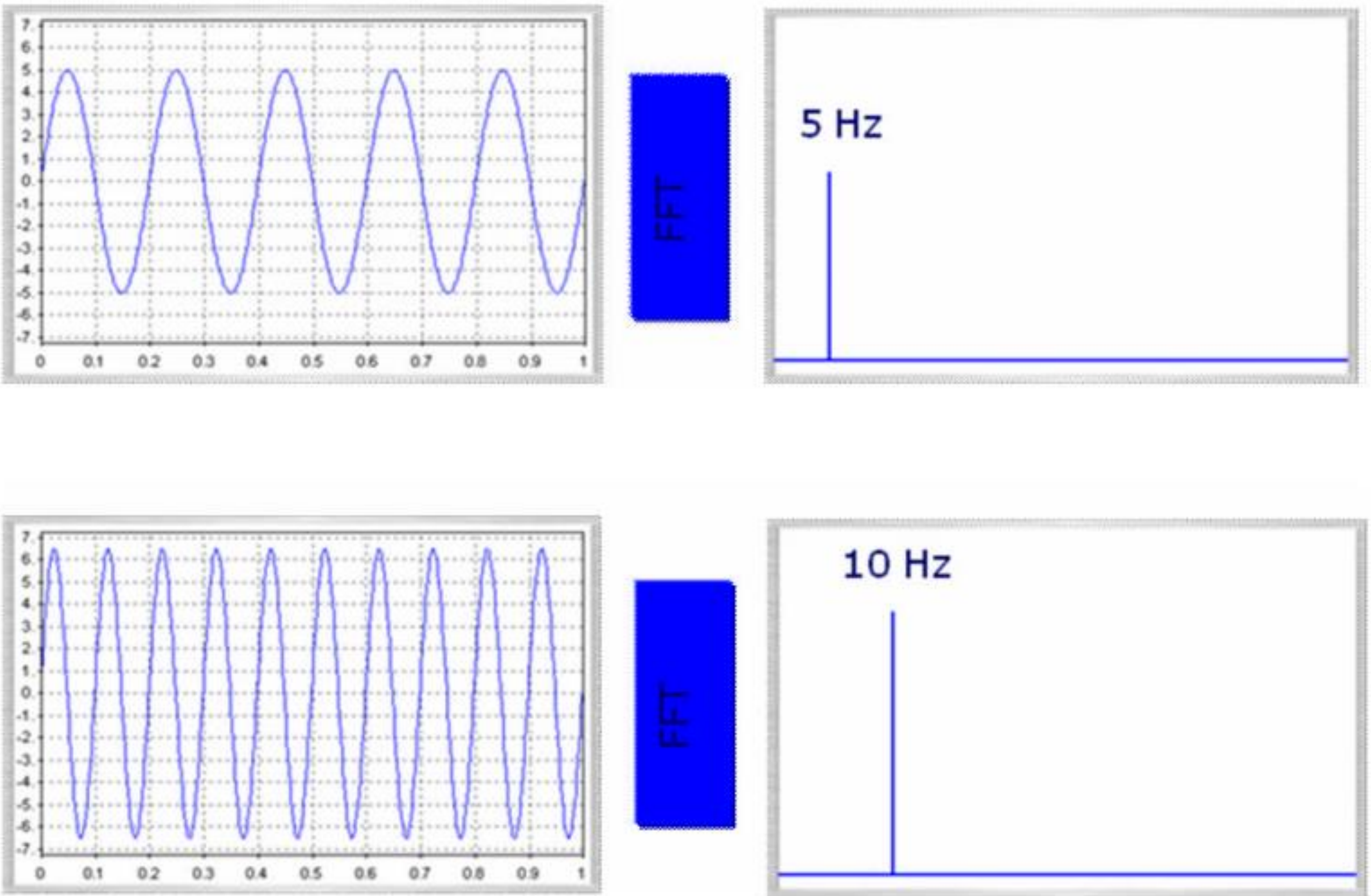


Figure 3-22 The top waveform of a 5 Hz cycle produces a spectrum with the 5 Hz peak. The bottom waveform is a result of doubling the speed to 10Hz. It produces a spectrum with a peak at 10 Hz. Notice the height of the peaks reflects the amplitude of the waveform.

Spectrum

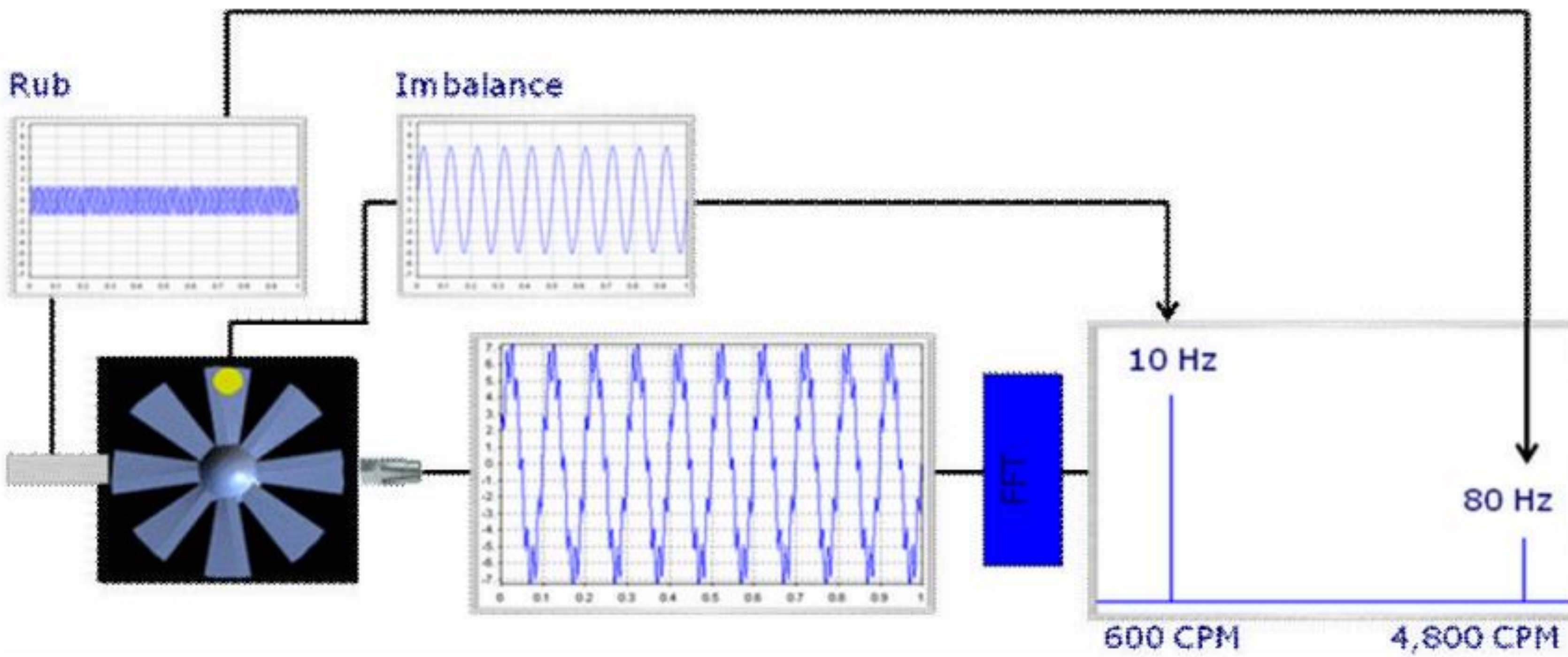
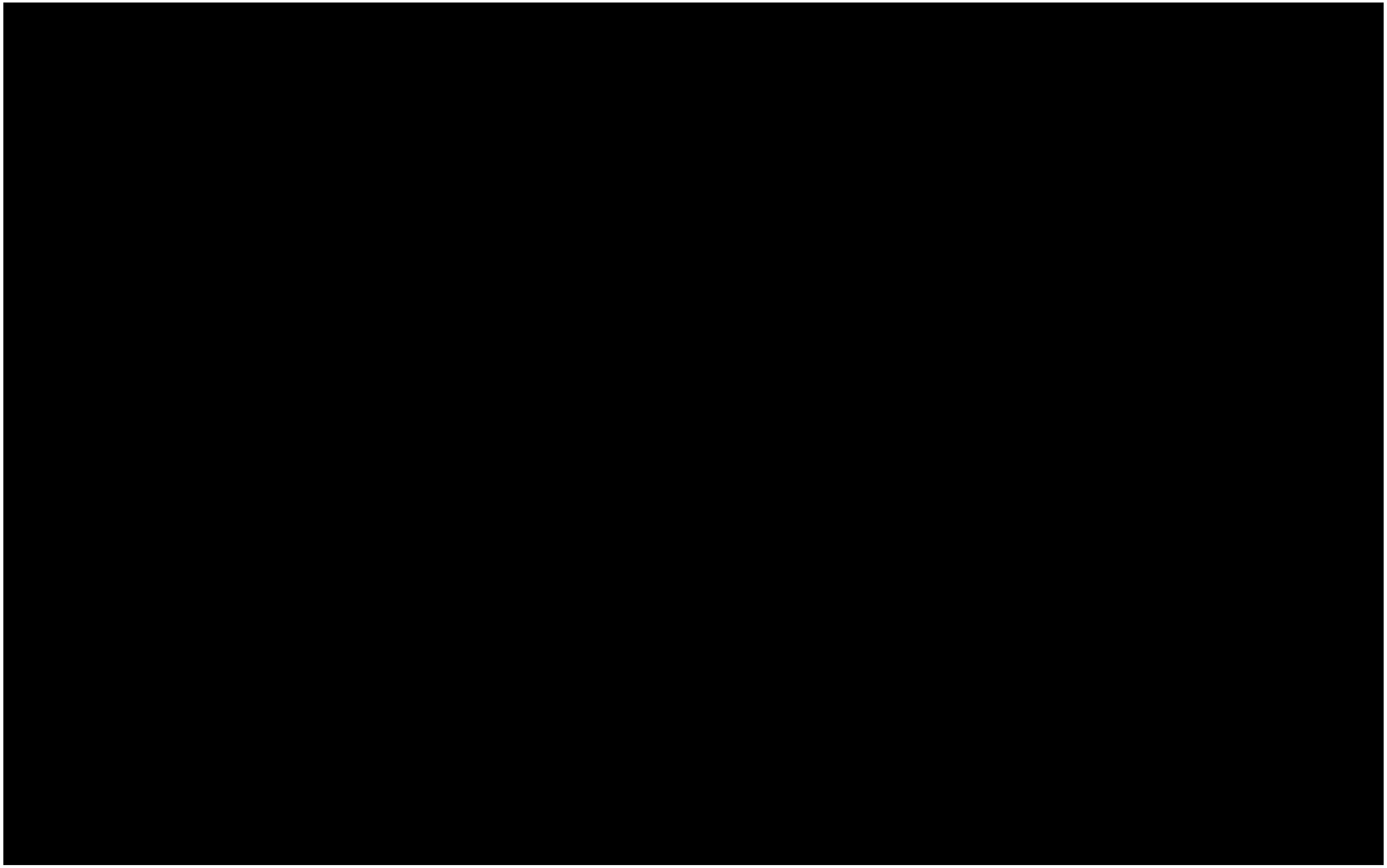
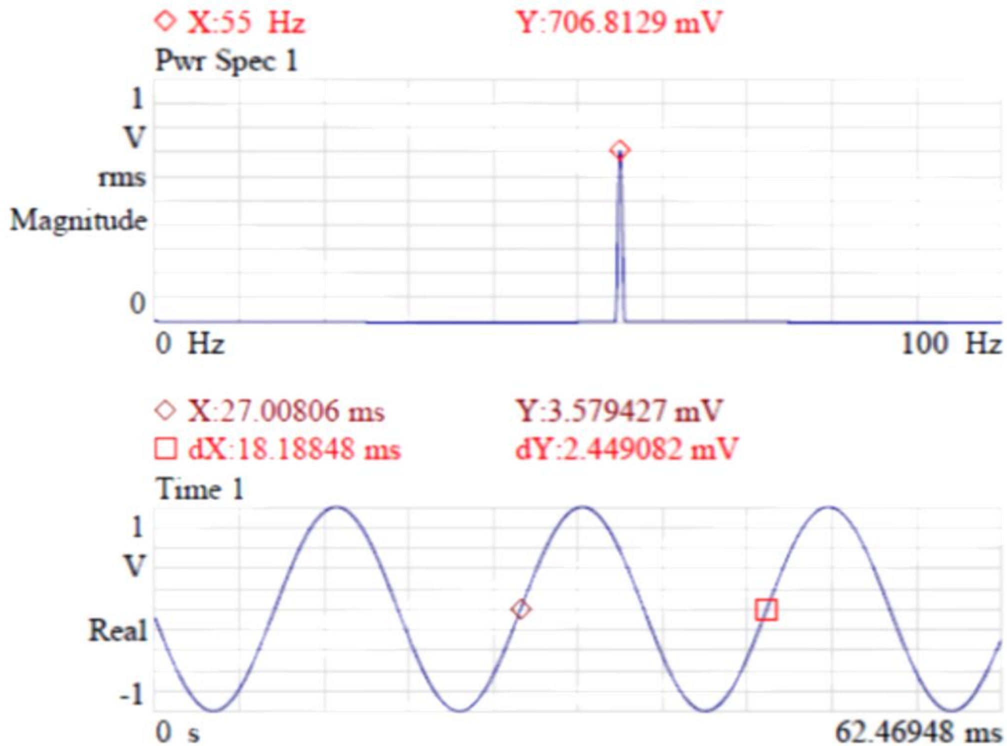


Figure 3-23 The FFT process separates the complex waveform into the individual waveforms and displays them in the spectrum.

Spectrum

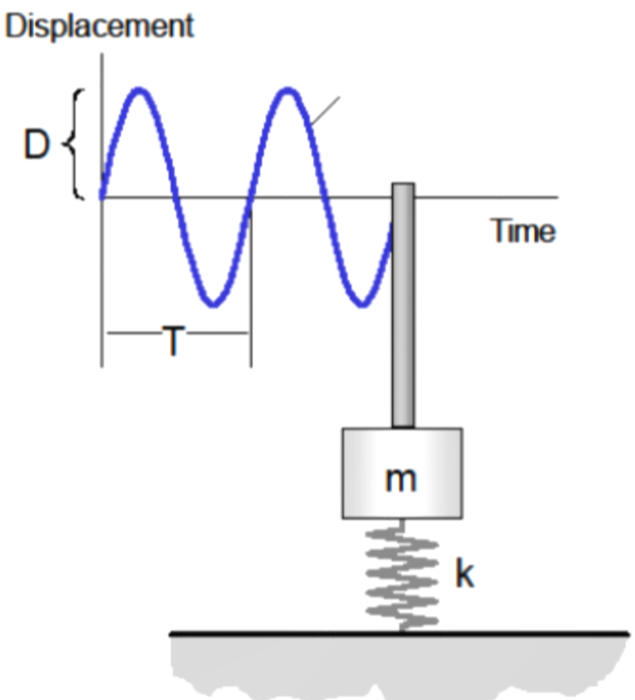


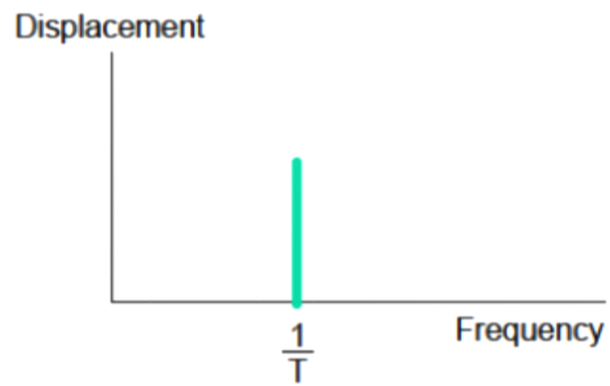
Single Frequency



FFT transformation

$d = D \sin ?_n t$





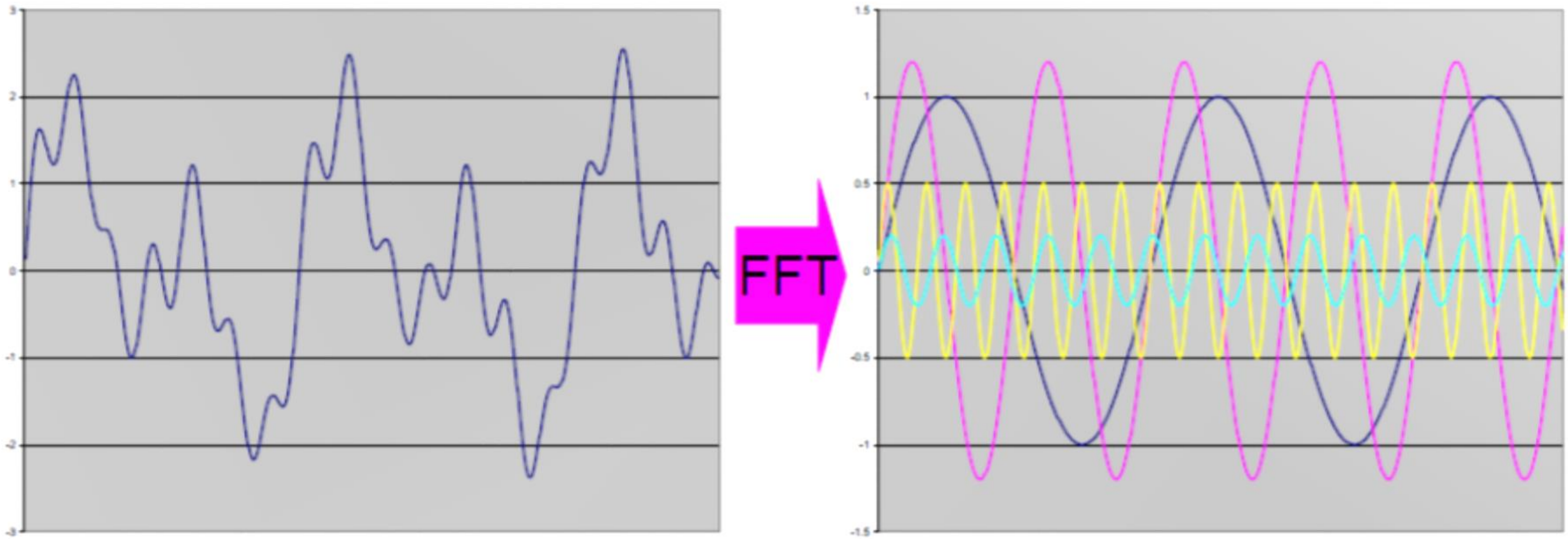
Period, T_n in [sec]

Frequency, $f_n = \frac{1}{T_n}$ in [Hz = 1/sec]

$? _n = 2 ? f_n = \sqrt{\frac{k}{m}}$

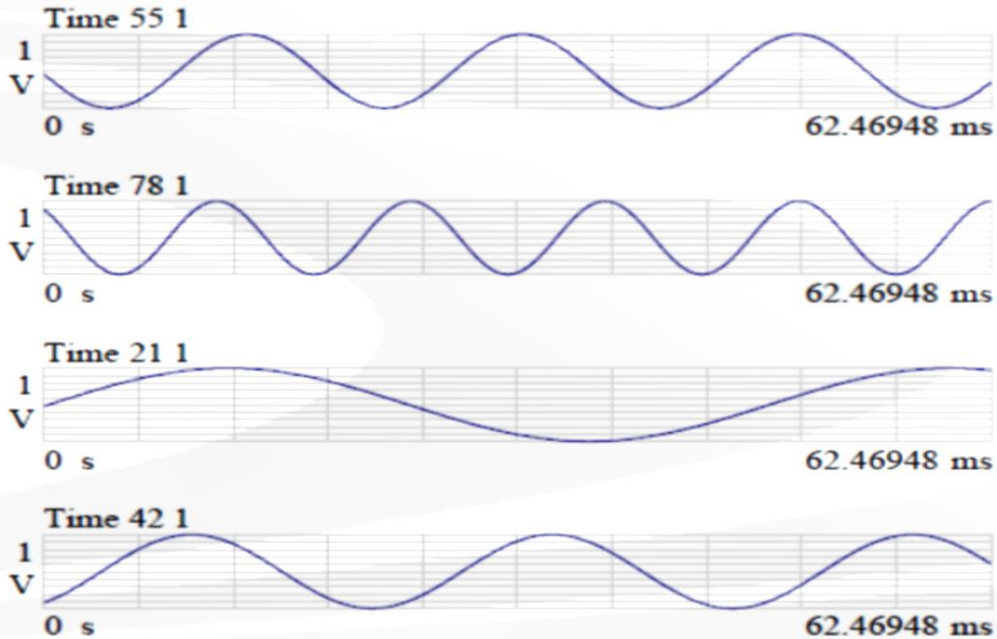
Signal Processing – break down complex waveform in to waveform components

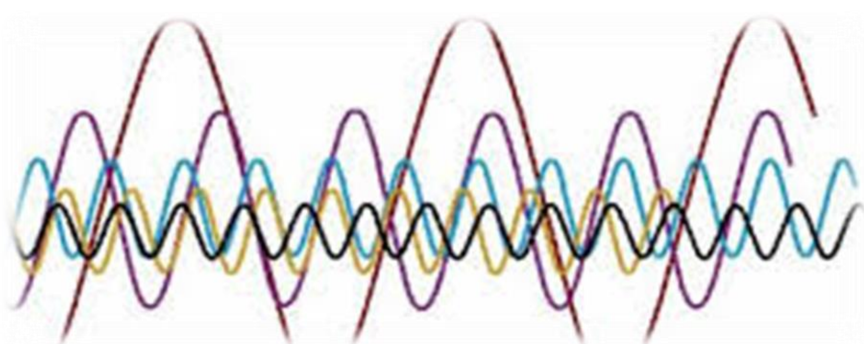
The Fast Fourier Transform (FFT) takes the complex waveform and breaks it down into the component sine waves



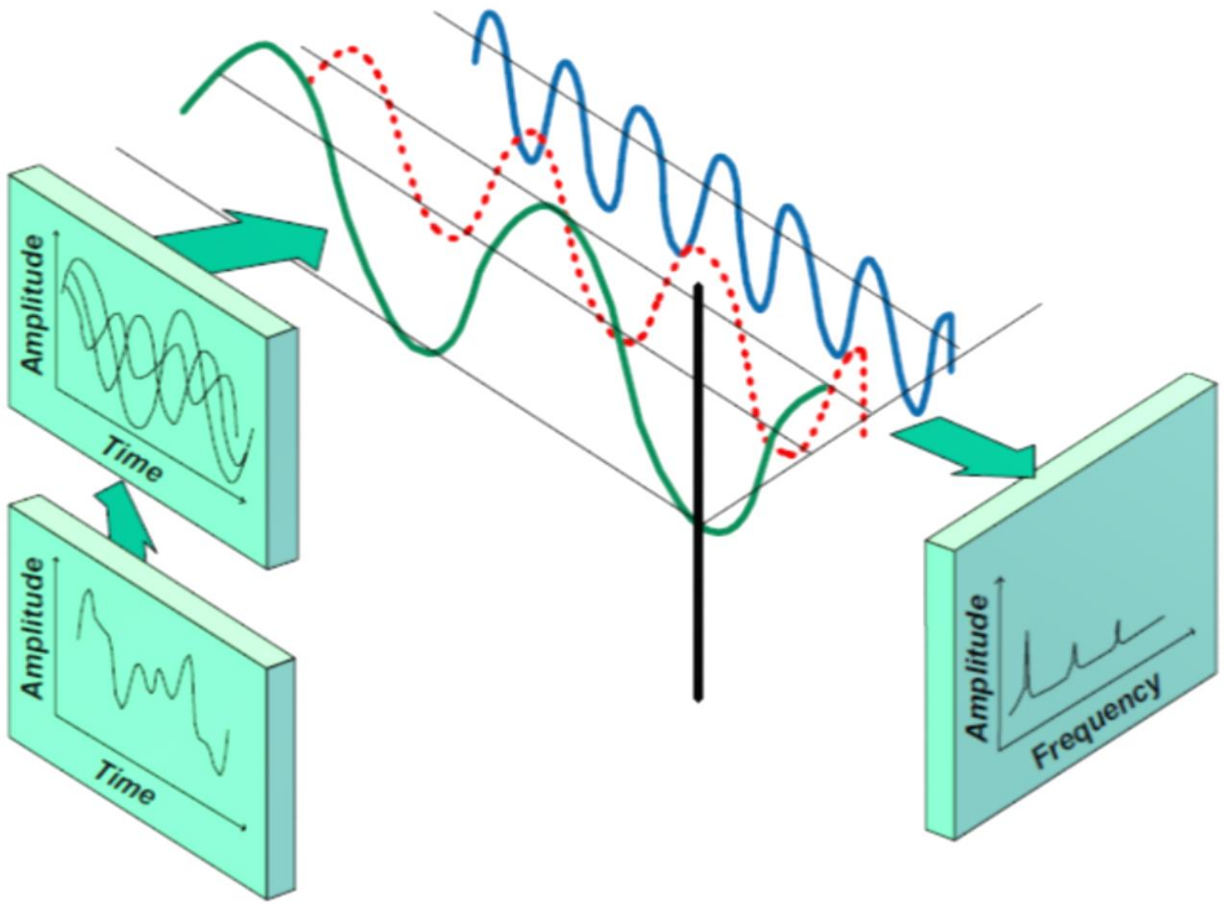
The amplitudes for each sine wave is then plotted at the frequency of the sine wave, creating the Spectrum

Multiple Time

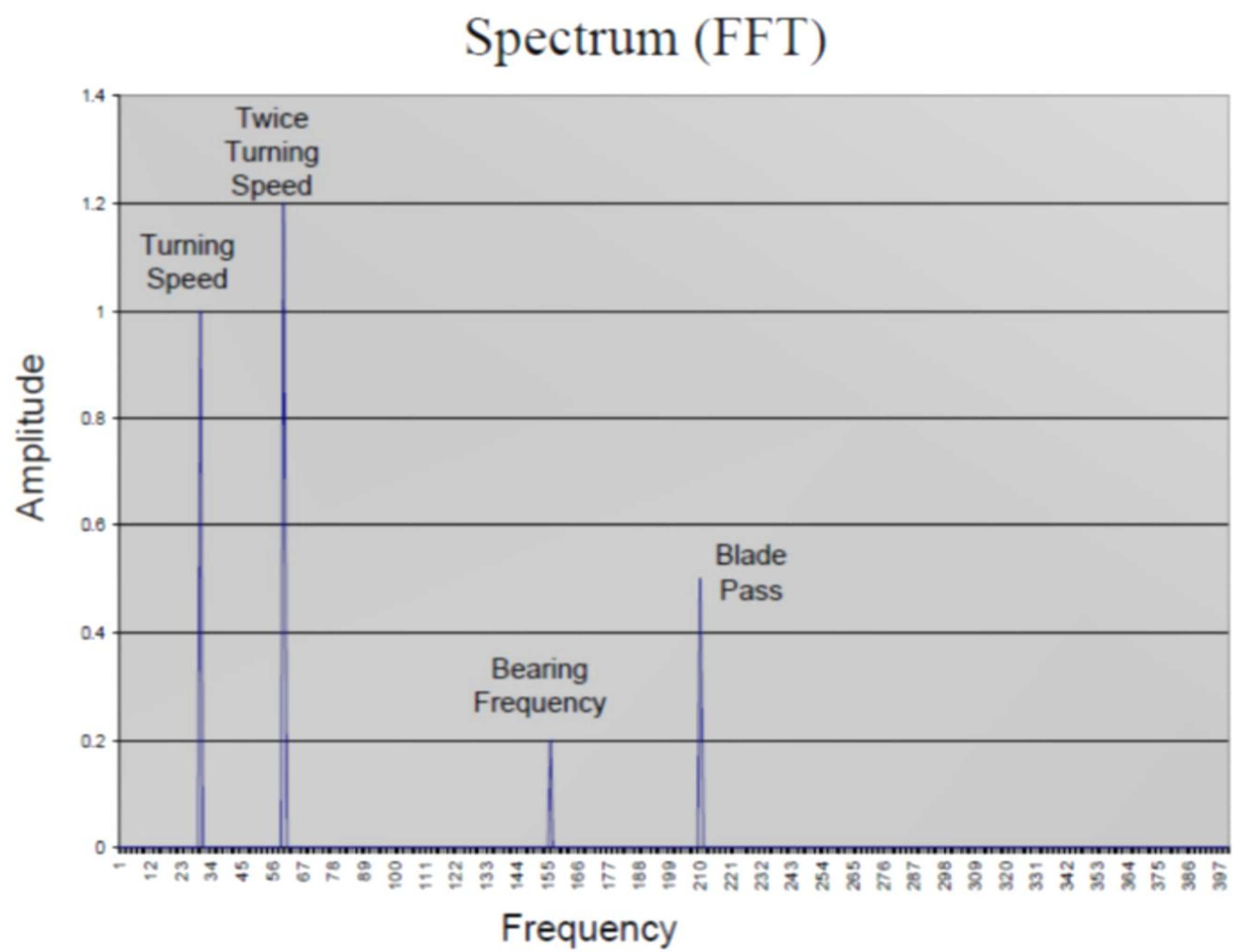




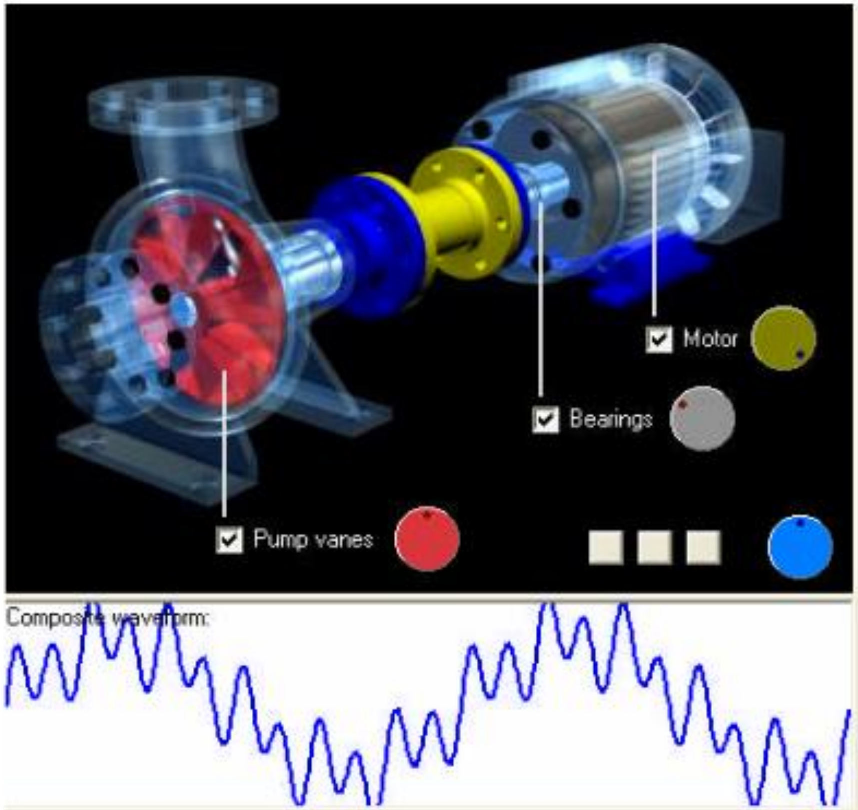
FFT Signal Processing



Signal Processing – The FFT or Spectrum



Spectrum



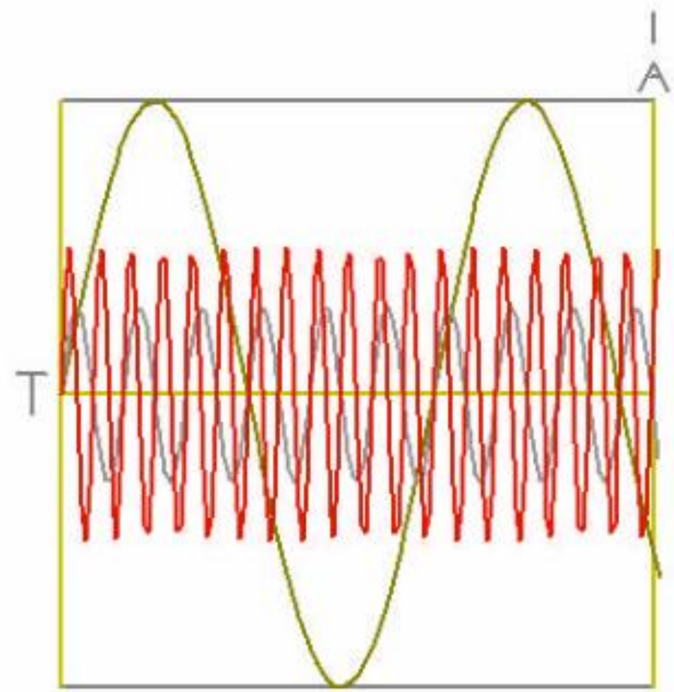


Figure 3-24 Three sources of vibration are combined in the composite waveform. They are individually overlaid in the box to the right of the machine.

Spectrum

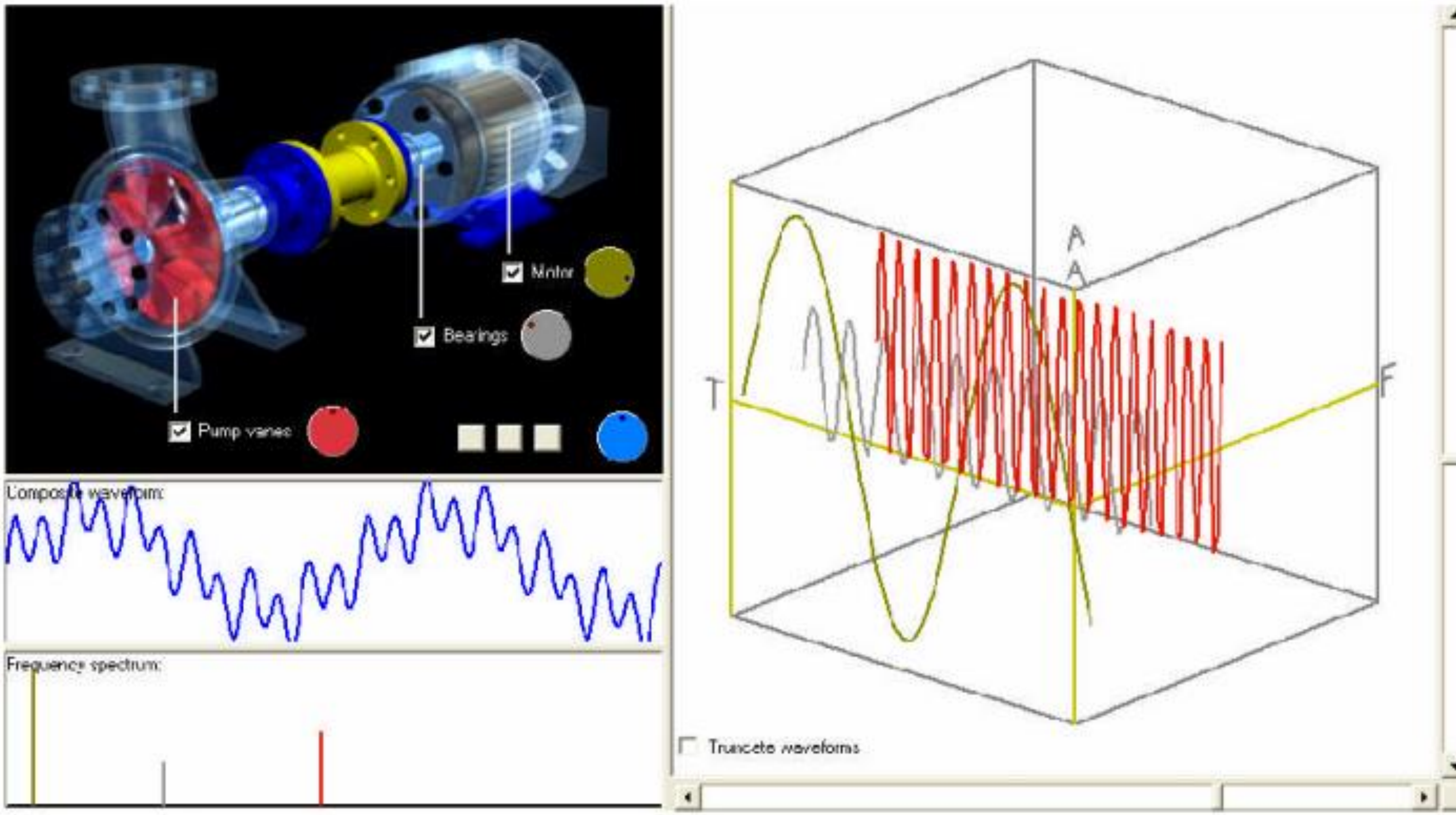


Figure 3-25 The individual waveforms are shown in a 3 dimensional box that is partially rotated.

Spectrum

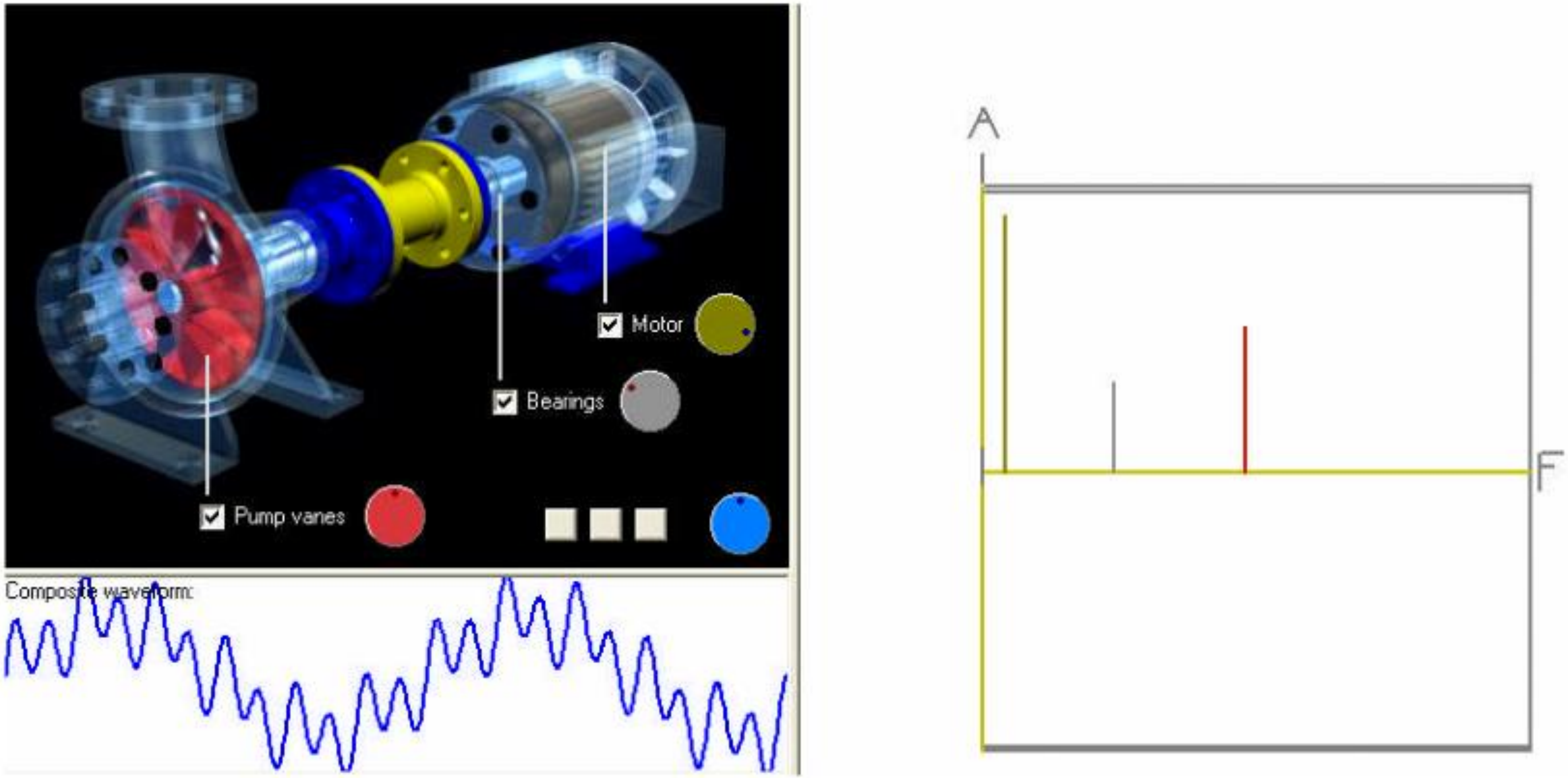
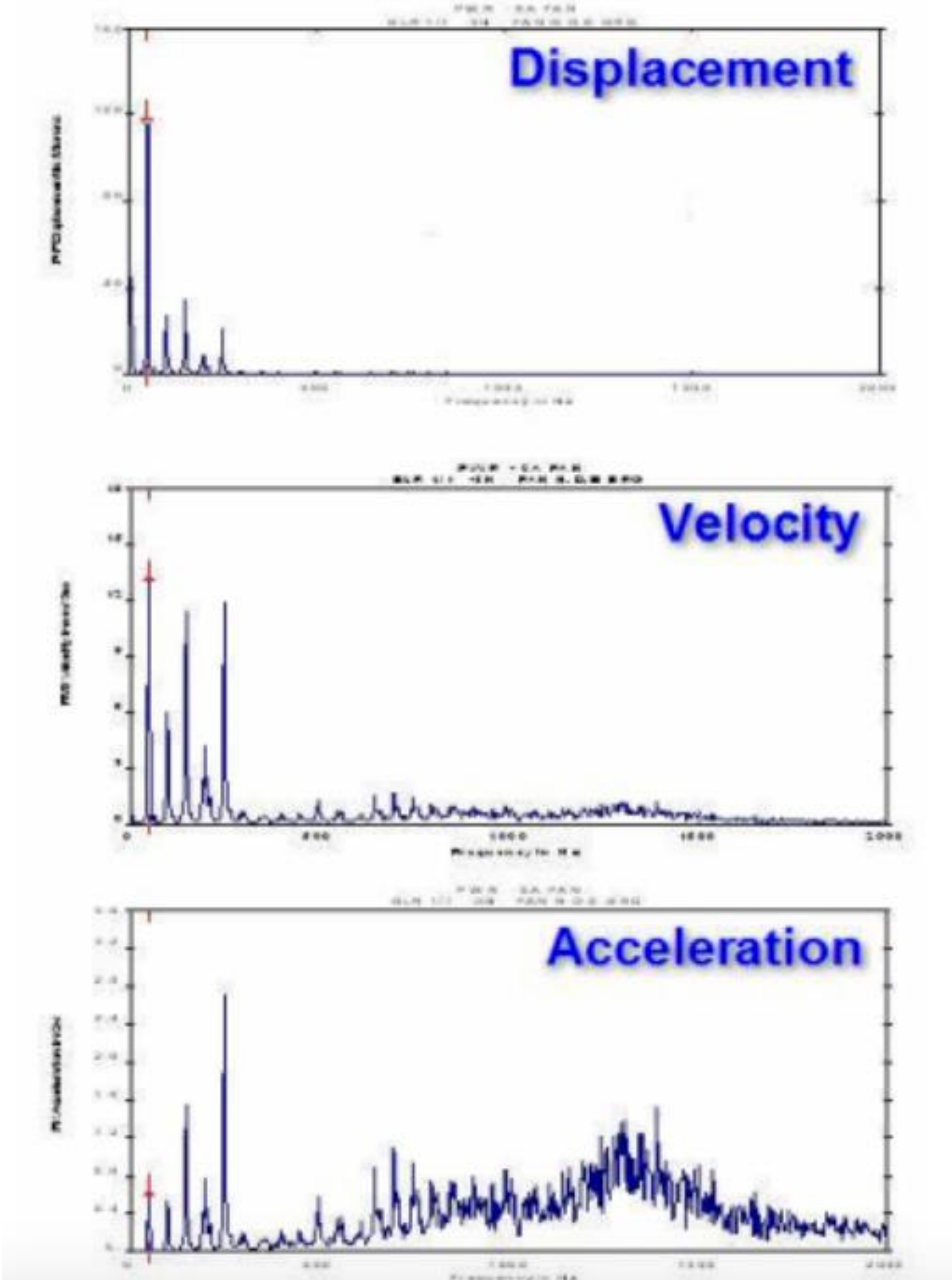
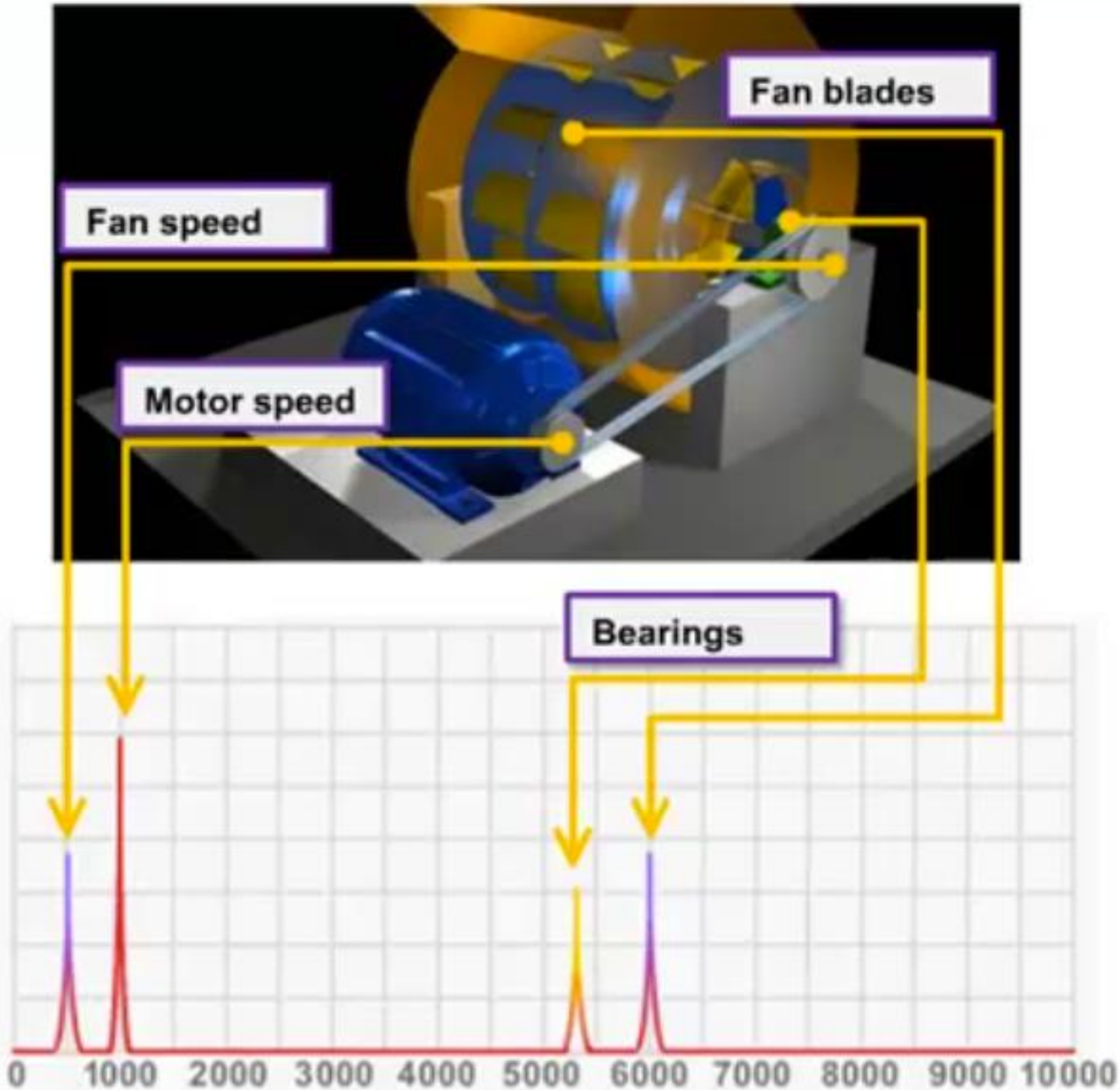


Figure 3-26 The 3 dimensional box is shown rotated 90 degrees revealing the end-view of the waveforms. They have been truncated so that nothing is shown below the zero line. The X axis is Frequency.

Spectrum

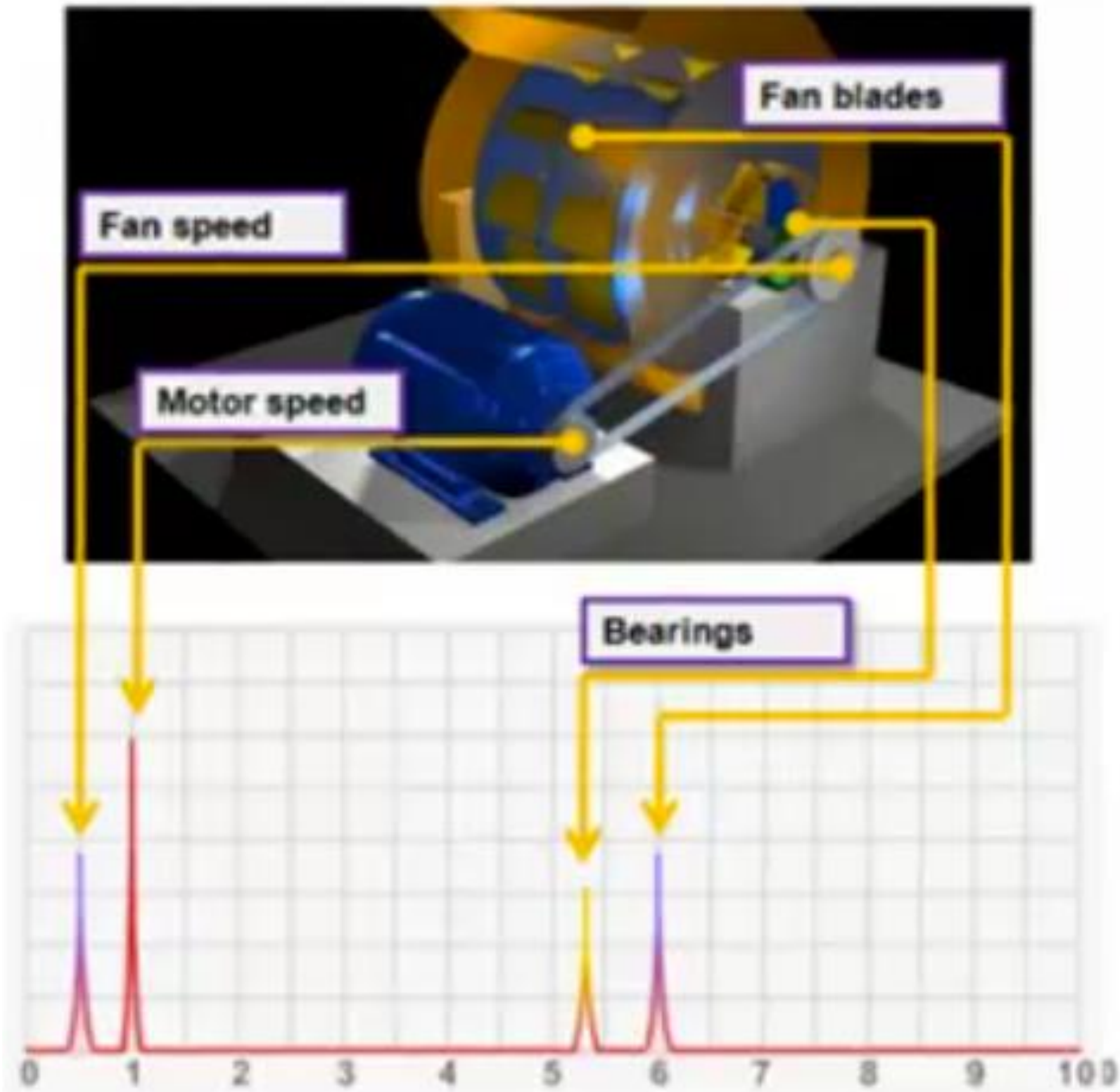


Spectrum



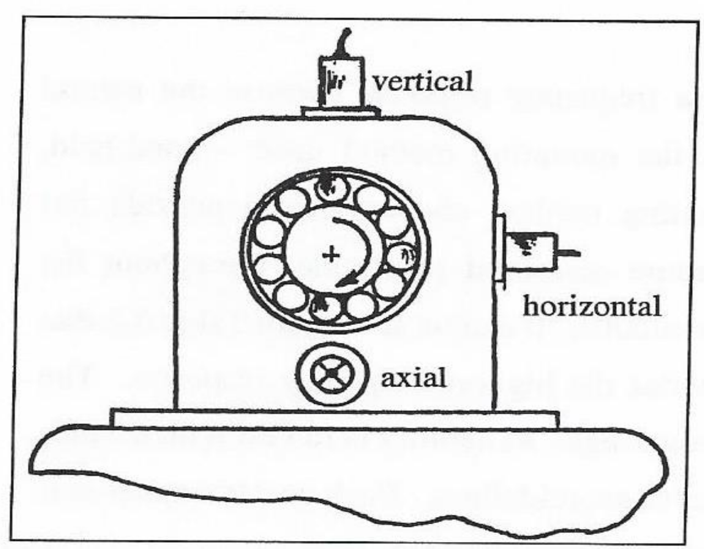
Fan blades
Fan speed
Motor speed
Bearings
0
1000
2000
3000
4000
5000
6000
7000
8000
9000
10000

Spectrum Orders



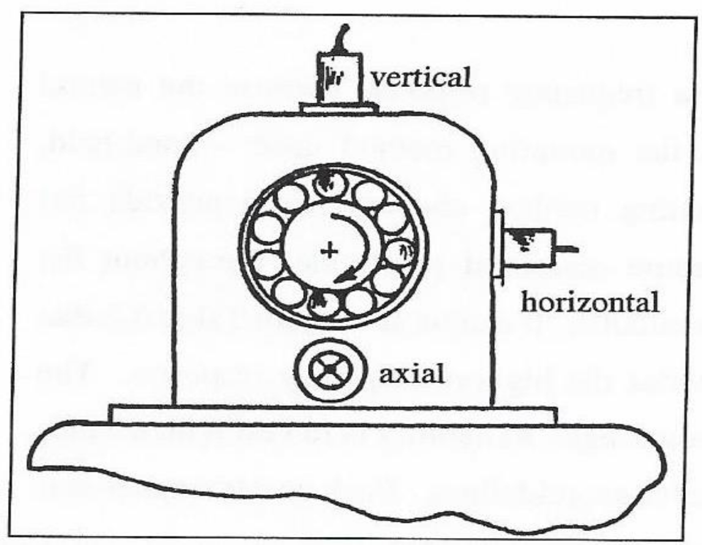
Sensor Location

The key to accurate vibration measurement is placement of the sensors at a point that is responsive to machine condition. In any event the sensor should be placed as close to the bearing as is physically possible and in the load zone. Figure 3.10 shows the optimum points for mounting sensors for data acquisition in a normal bearing mounting



Sensor Location

The horizontal and vertical locations at the bearing centerline are shown. These locations are used to sense the vibrations from radial forces such as mass unbalance. Vibrations from axially-directed forces such as gearmesh and bearing faults are measured in the axial direction in the load zone.



Sensor Location

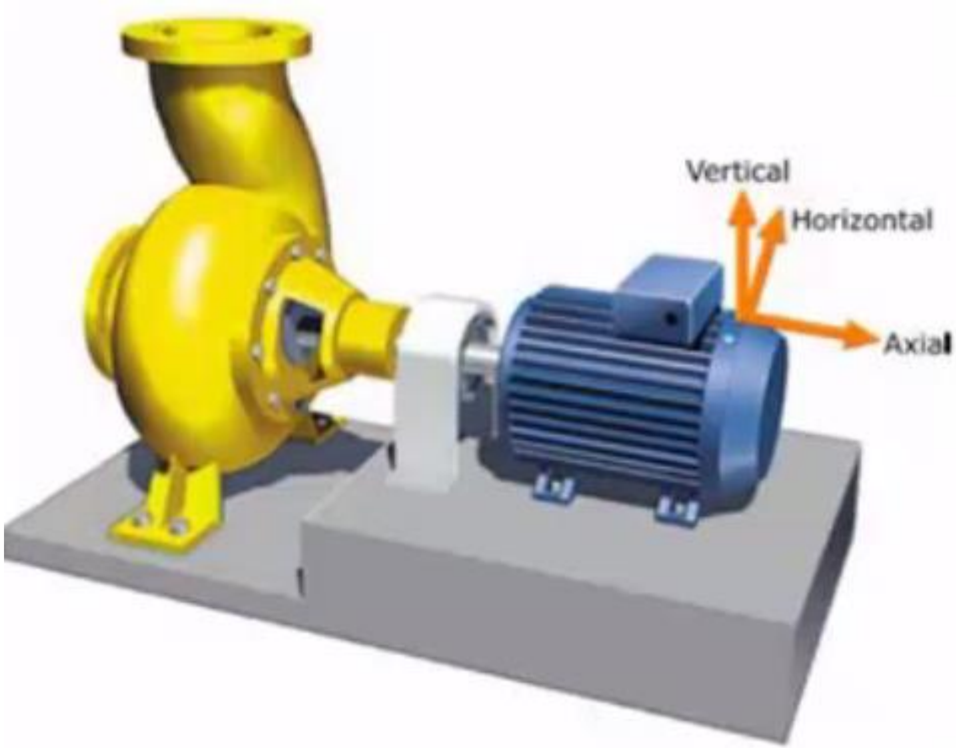
The sensor must be placed as close to the bearing as possible, even though placement is restricted by such components as housings, coupling guards, and fan covers.

In general, radial readings are taken on radial bearings; that is, any antifriction bearing with a contact angle of 0°.

Radial bearings are used in electric motors, in medium- to light-duty fans, and in power transmission units not subject to axial loading.

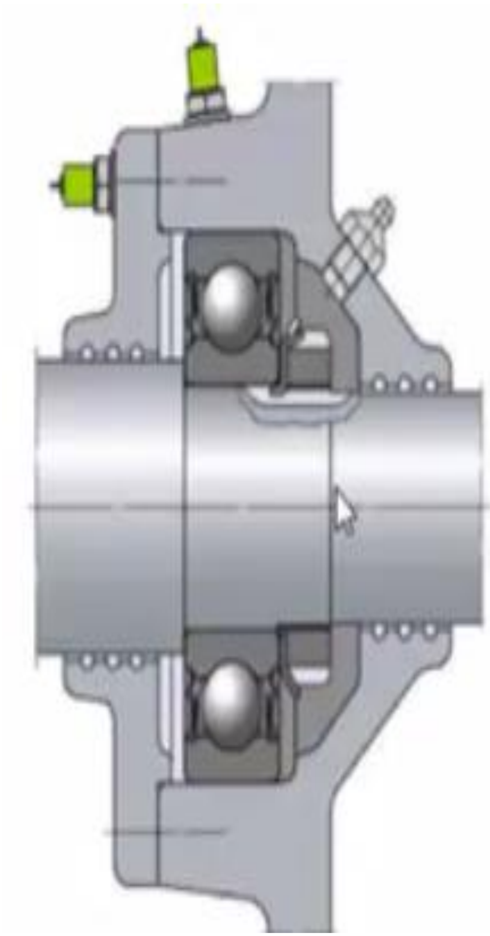
Angular contact bearings or any bearing absorbing thrust have a radial-to-axial coupling that requires an axial measurement for accurate condition monitoring.

Sensor Location

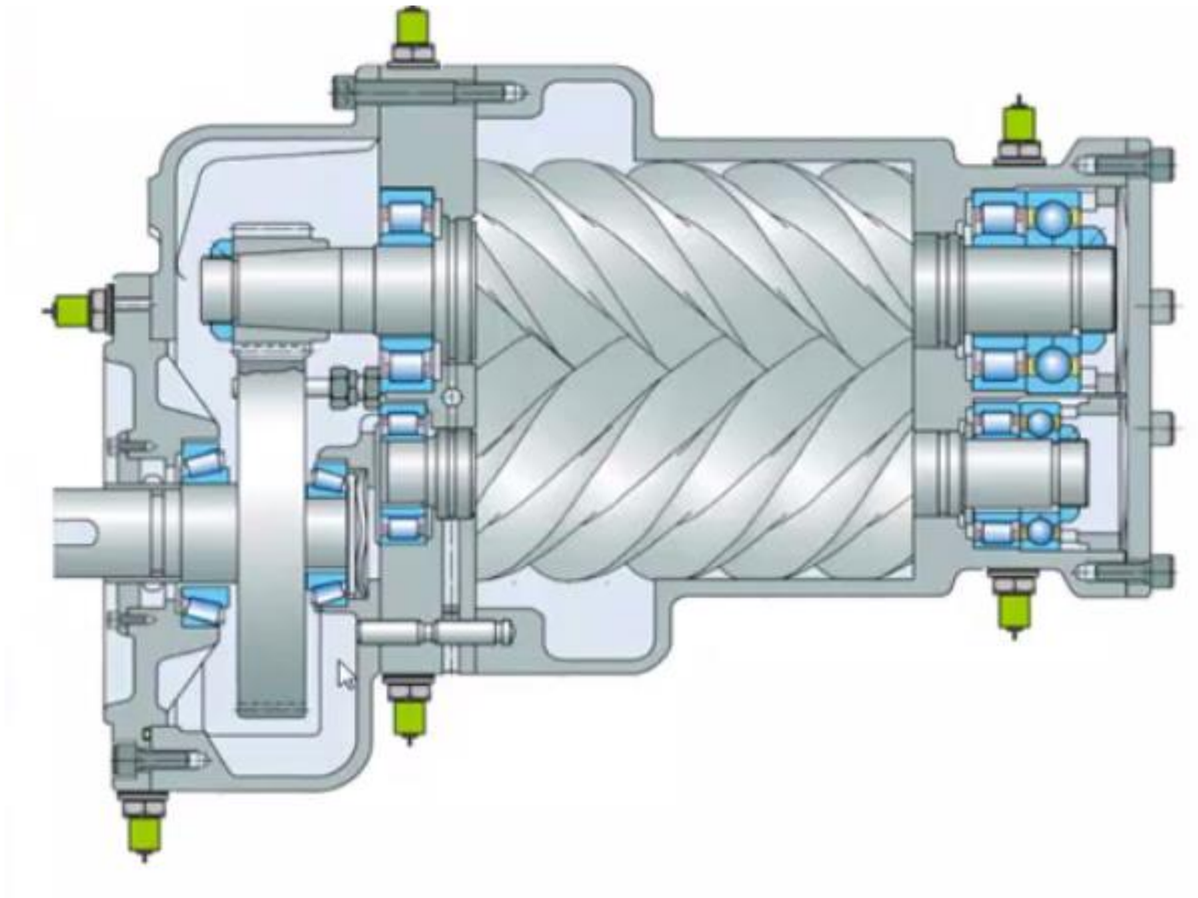


Sensor Location

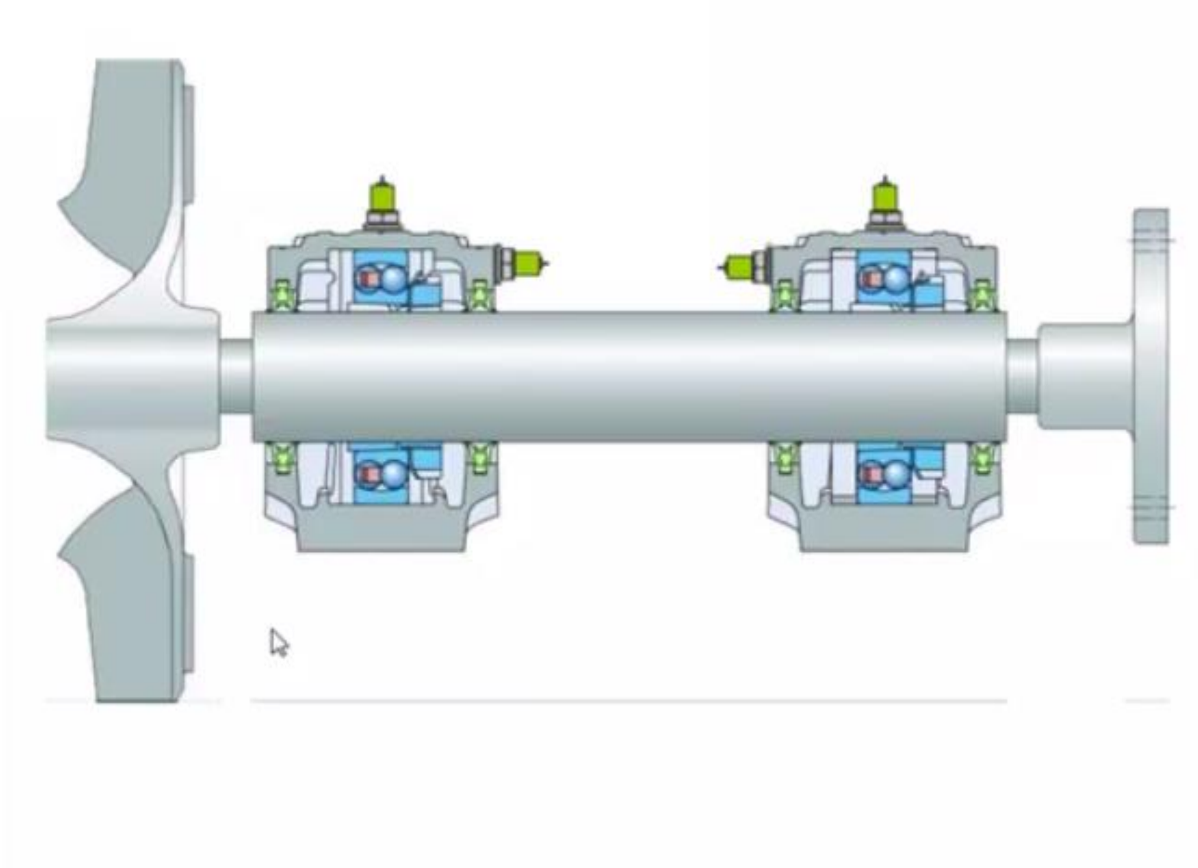
Sensor Location



Sensor Location



Sensor Location



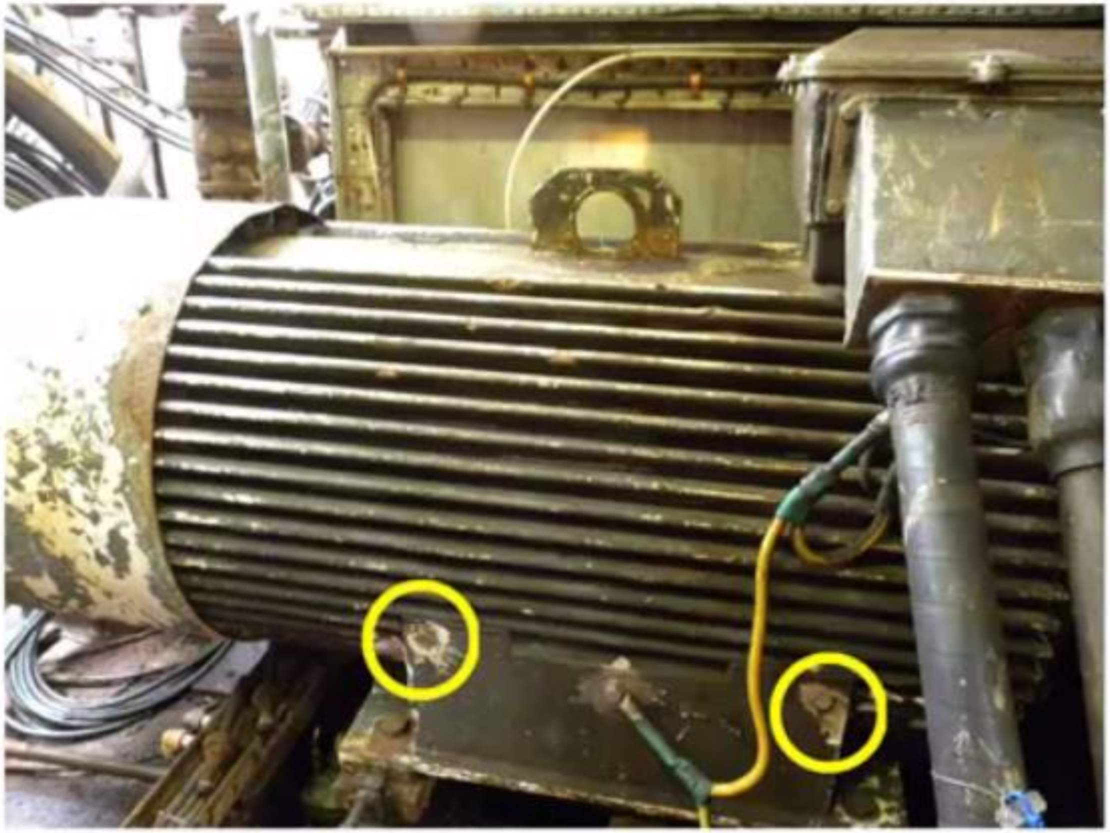
Sensor Location

Sensor Location

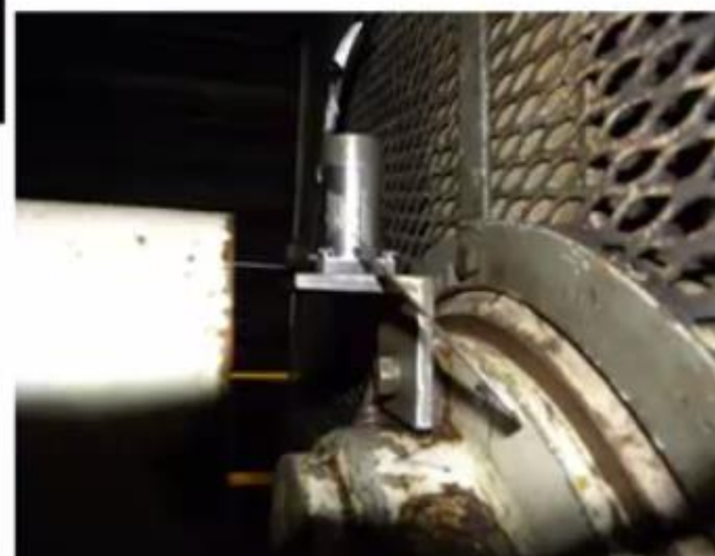
Sensor Location

Sensor Location

Sensor Location



Sensor Location



Sensor Location



Sensor Location

Sensor Location

Sensor Location

Figure 4-26 Typical junction box and labels

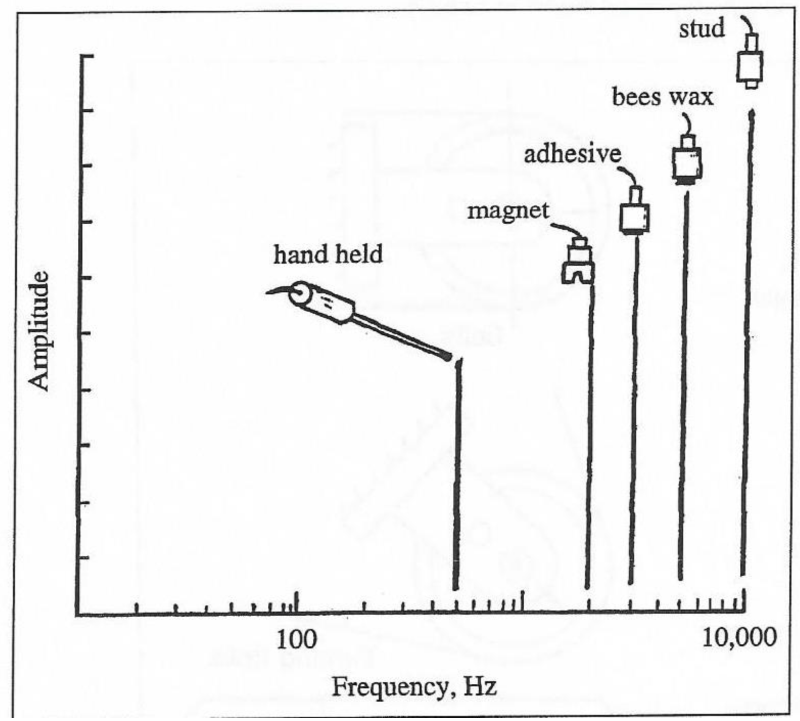
Sensor Location

Sensor Mounting

The method used to mount a vibration sensor can affect the frequency response because the natural frequency of an accelerometer can decrease, depending on the mounting method used - hand-held, magnetic, adhesive, threaded stud (Figure 3.9).

Approximate Frequency Spans for 100 mv/g Accelerometers.

Method	Frequency Limit
Hand Held	500 Hz
Magnet	2,000 Hz
Adhesive	2,500-4,000 Hz
Bees Wax	5,000 Hz
Stud	6,000-10,000 Hz



Sensor Mounting

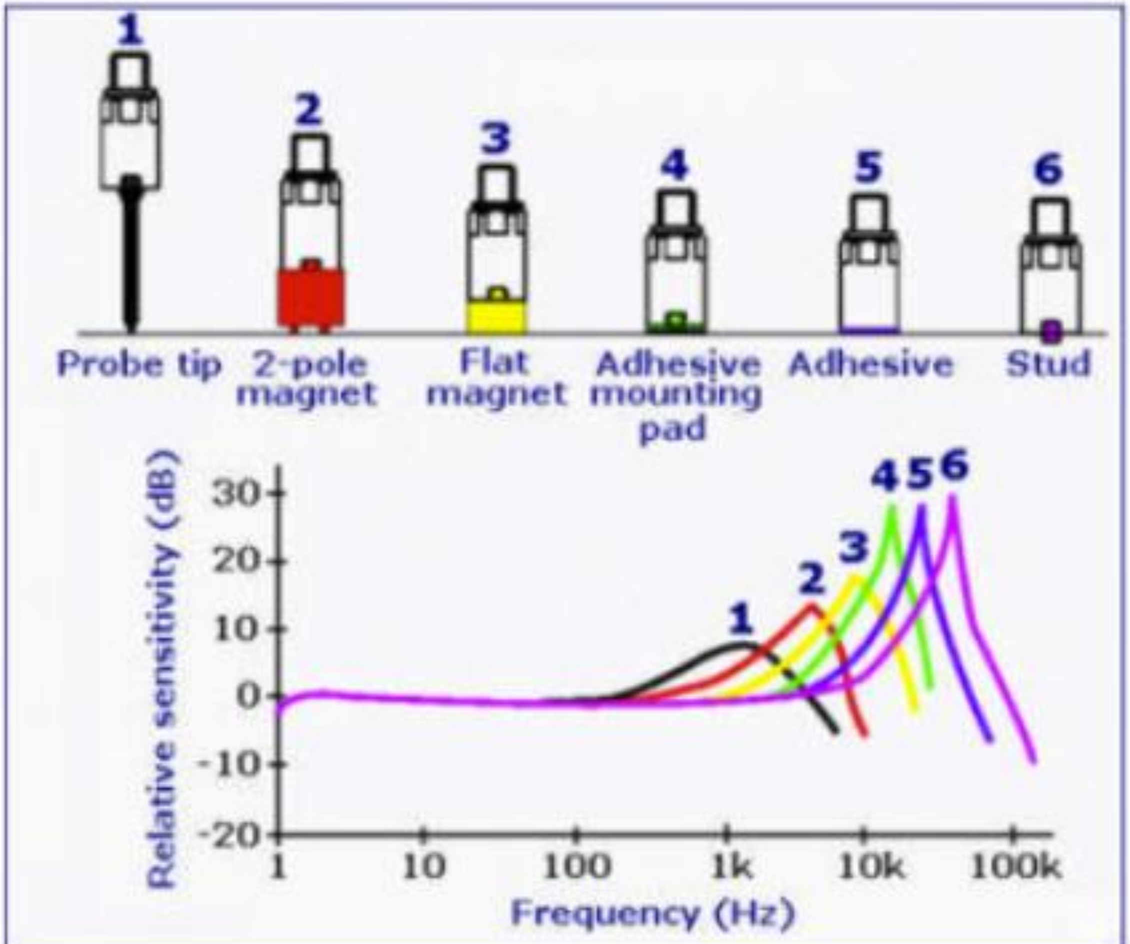
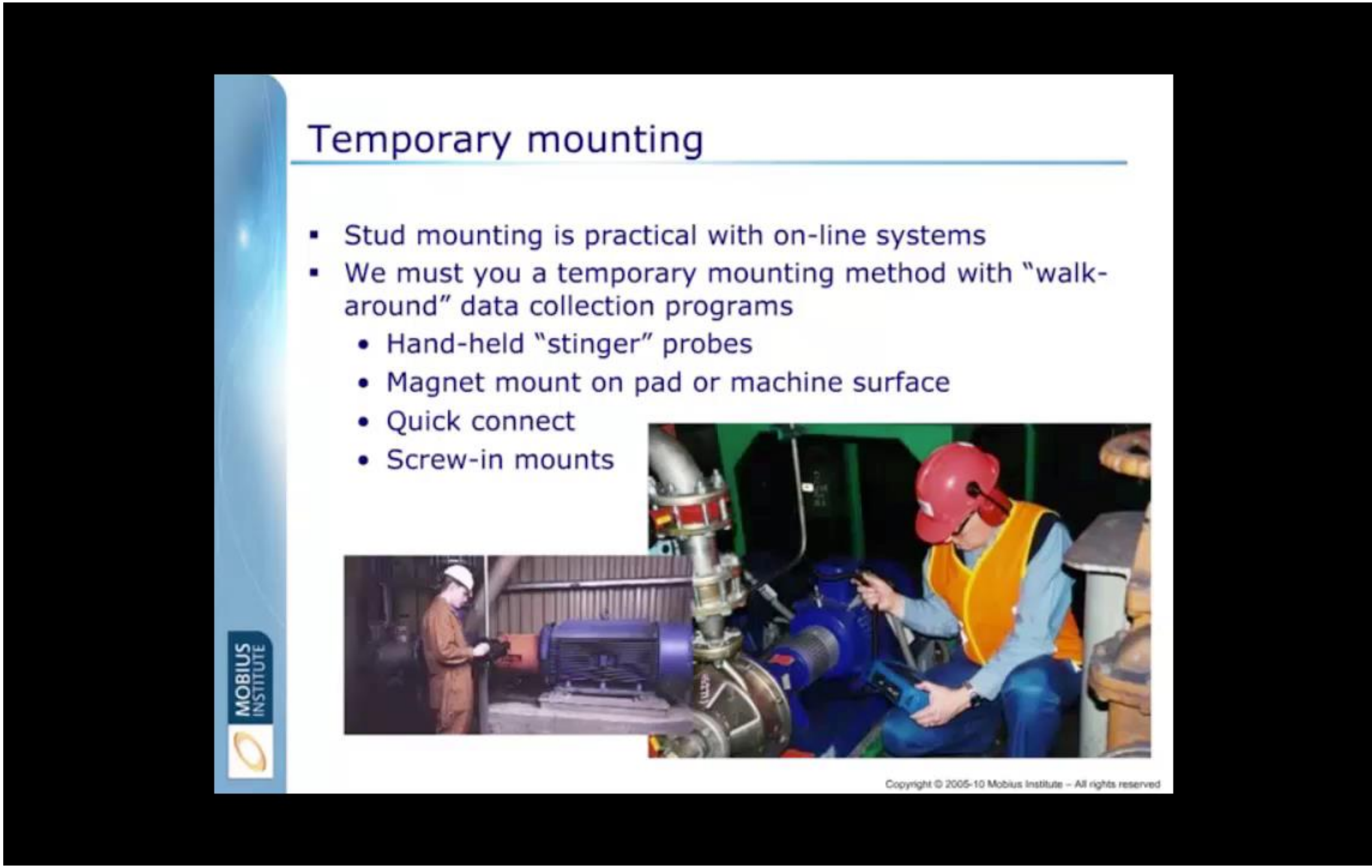


Figure 4-47 Frequency response ranges of various transducer mountings.

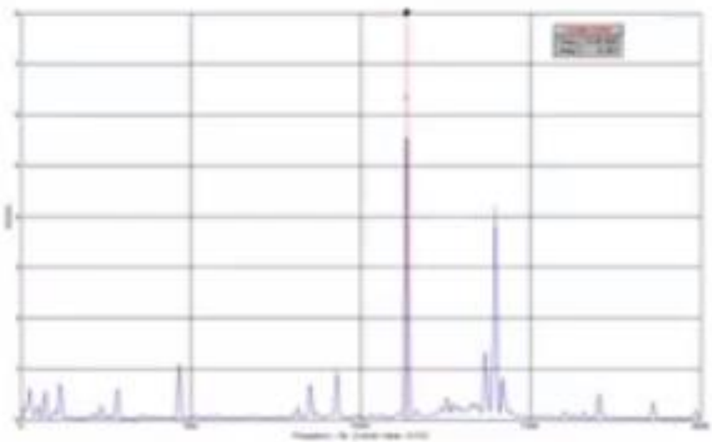
Sensor Mounting

Sensor Mounting

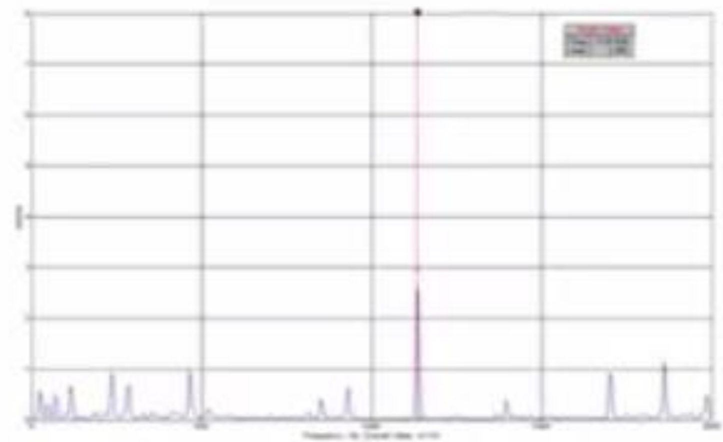
Sensor Mounting



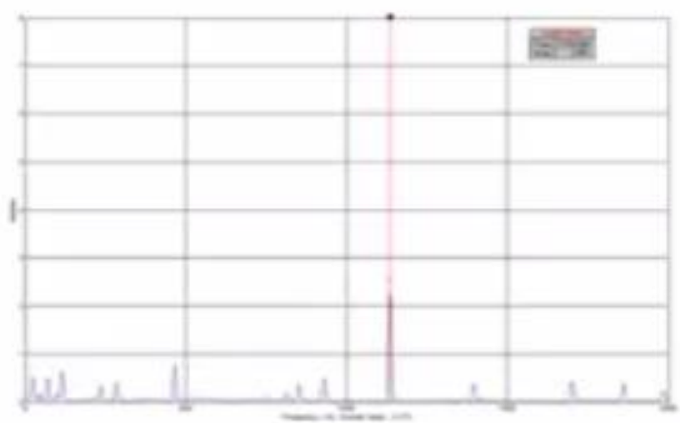
Sensor Mounting



Probe: 1135 Hz – 6.34 mm/s RMS



Magnetic base. 1135 Hz – 2.95 mm/s RMS

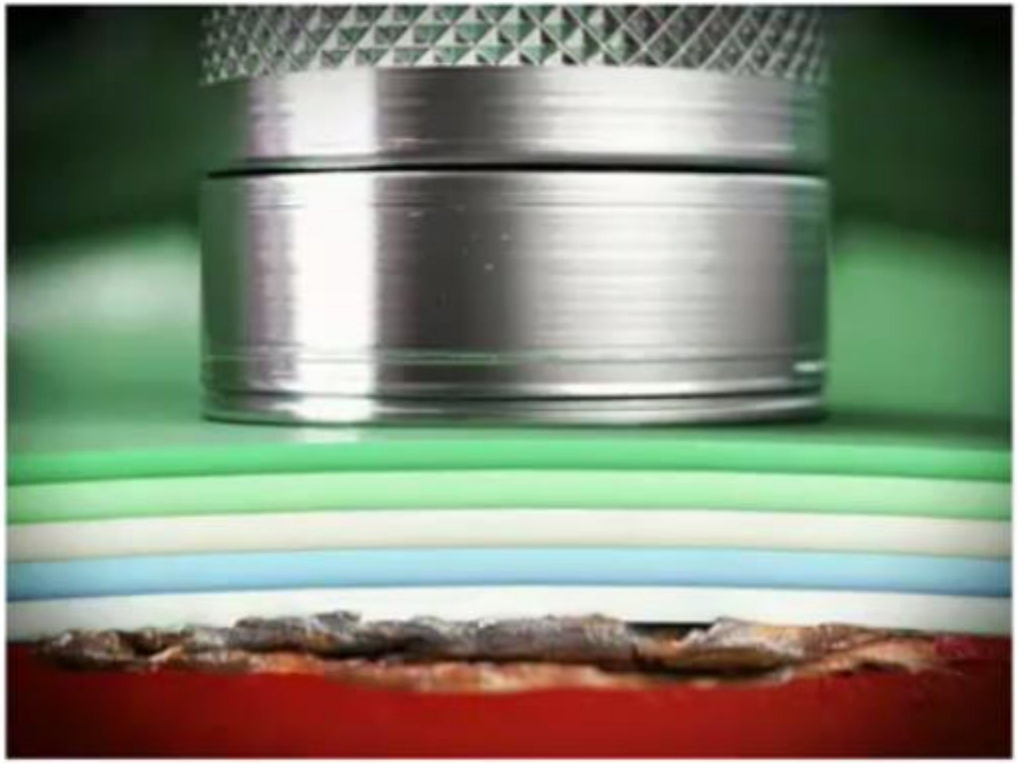


Adhesive mounting: 1135 Hz – 2.55 mm/s RMS

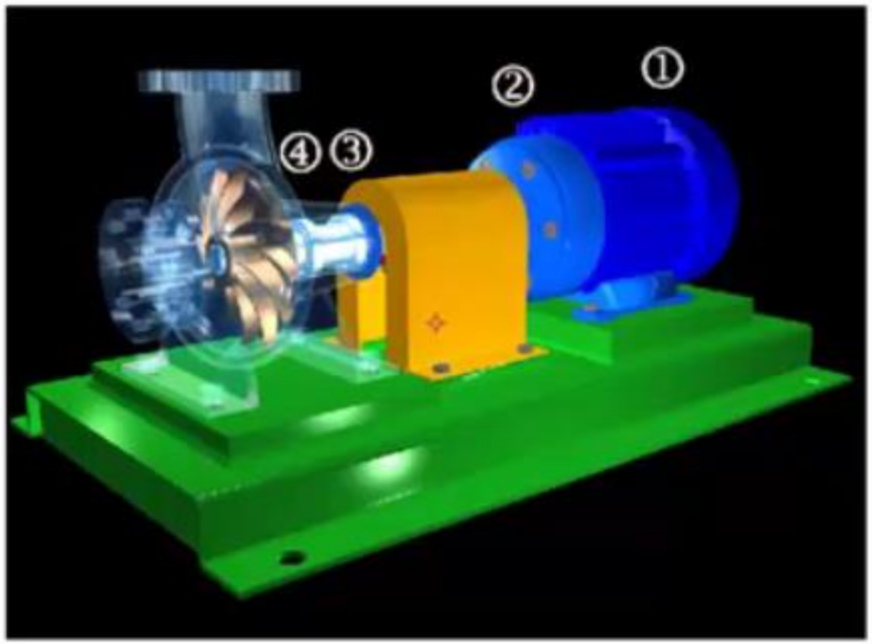
Sensor Mounting

	Model	Description
	SF5	Cementing pad, anodized aluminum, 10-32 integral stud, 0.5 inch hex
	SF8	Cementing pad, 1/4-28 integral stud, 1 inch diameter
	SF8-2	Cementing pad, 1/4-28 tapped hole, 1 inch diameter
	SF11	Magnetic mounting pad, 416 stainless steel - will accept <1 inch magnetic base
	SF20-3	Cemeting pad 3/8 - 24 integral stud
	QB-1	QuickLINK® sensor base, adapter mates to 1/4-28 sensor, for walkaround data collection. Allow quick mount of sensors in less than one turn.
	QP-1	QuickLINK® mounting pad, 1/4-28 tapped hole base, for use with QB-1
	QP-2	QuickLINK® cementing pad, flat base, for use with QB-1

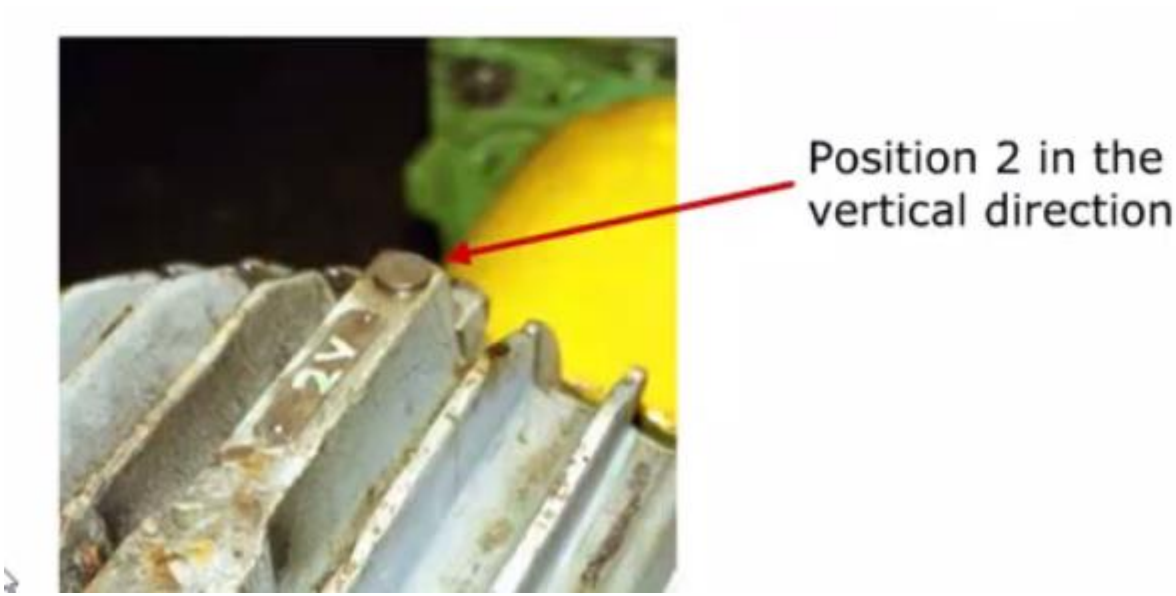
Sensor Mounting



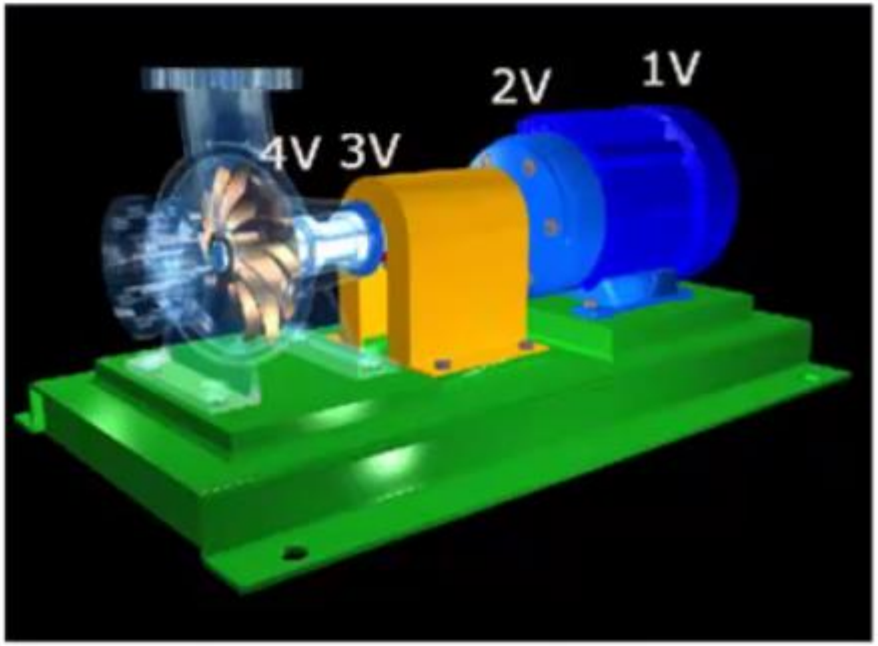
Naming Convention



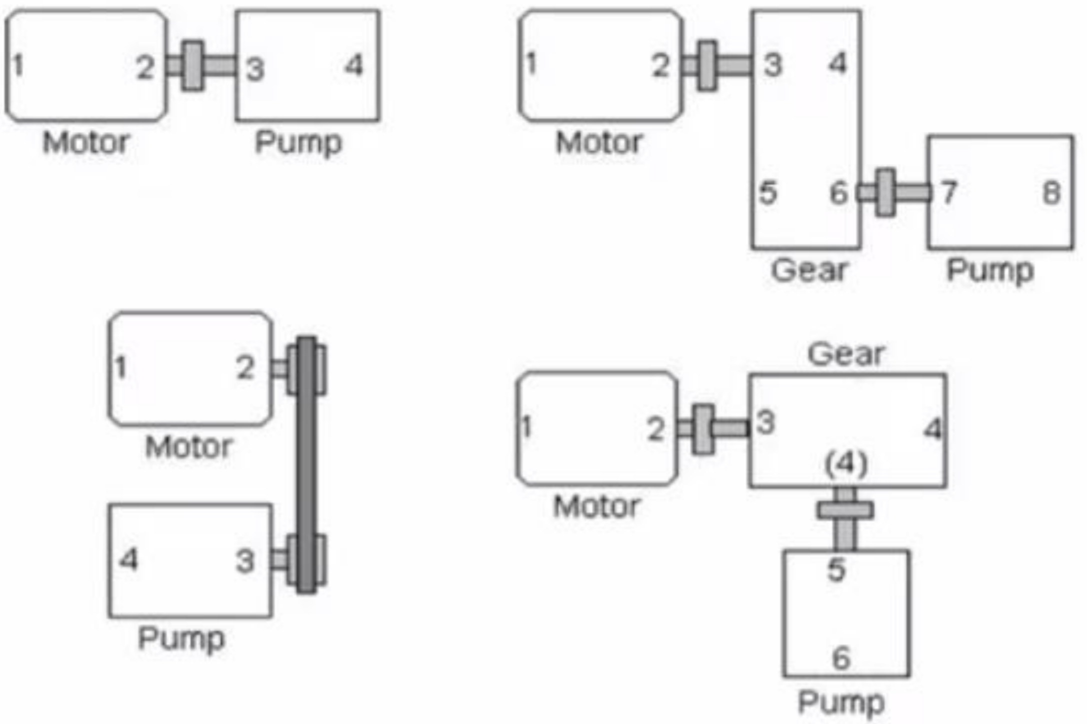
Naming Convention



Naming Convention

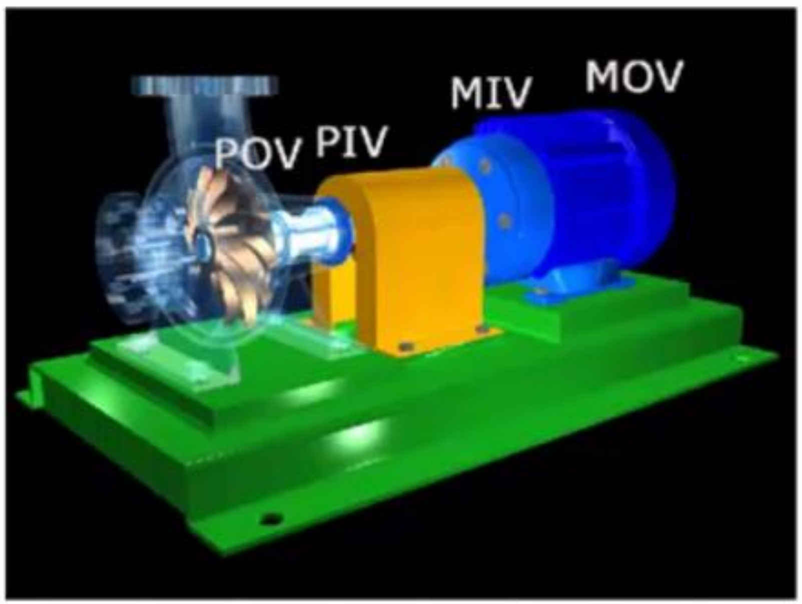


Naming Convention



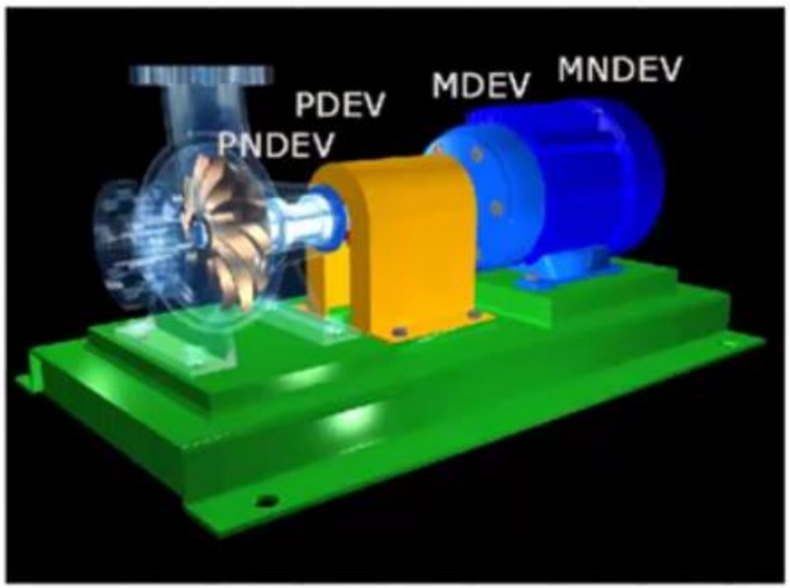
Naming Convention

- Another method uses a combination of an identifier for the component (M:Motor, P:Pump, G:Gearbox, F:Fan (or air handling unit), C:Compressor, and T:Turbine and the designator for inboard "I" and outboard "O".



Naming Convention

- Another method: identify the point on the component as either drive-end "DE" or non-drive-end "NDE" (or free end).



For example, "MDE" is the drive-end bearing on the motor.

Naming Convention

- We must identify the measurement axis: Vertical “V”, Horizontal “H” and Axial “A”.
- *Radial* and *tangential* are occasionally used: Horizontal machine:
  - Radial = Vertical
  - Tangential = Horizontal

Naming Convention

- Vertical machines are a little different:
  - Axial is axial.
  - Radial and tangential are used.
- Convention used:
  - "Vertical" is in line with discharge pipe.
  - "Horizontal" 90° from discharge pipe.

Design and Function

Component	Frequency
antifriction bearings	ball pass frequency, outer race ball pass frequency, inner race fundamental train frequency rotating unit frequency ball spin frequency
hydrodynamic journal bearings	frictional frequency, whirl frequencies
gears	rotating unit frequency gear-mesh frequencies and harmonics harmonics of gear-mesh frequencies assemblage frequencies system natural frequencies (gear-tooth defects)
Blade wheels and impellers	Rotating unit frequencies vane and blading frequencies harmonics of vane and blading frequencies

Design and Function

Component	**Frequency**
rotors	trapped fluid rotational frequency directional natural frequencies higher harmonics
couplings and universal joints	orders of rotating frequency
reciprocating mechanisms	rotating frequency and its orders
Electric motor rotors	sidebands at no poles x slips

Chapter 5 – VIBRATION INSTRUMENTS

The sensor which changes the mechanical motion of the machine to an electrical signal is connected to an instrument which provides an analytical read out and/or print out. The read out can be as simple as a single number from a meter or a waveform from an oscilloscope. More elaborate analyzers provide spectra (amplitude versus frequency) and digital time waveforms. Data collectors provide overall values, filtered values, phase readings, spectra, and time waveforms.

Chapter 5 – VIBRATION INSTRUMENTS

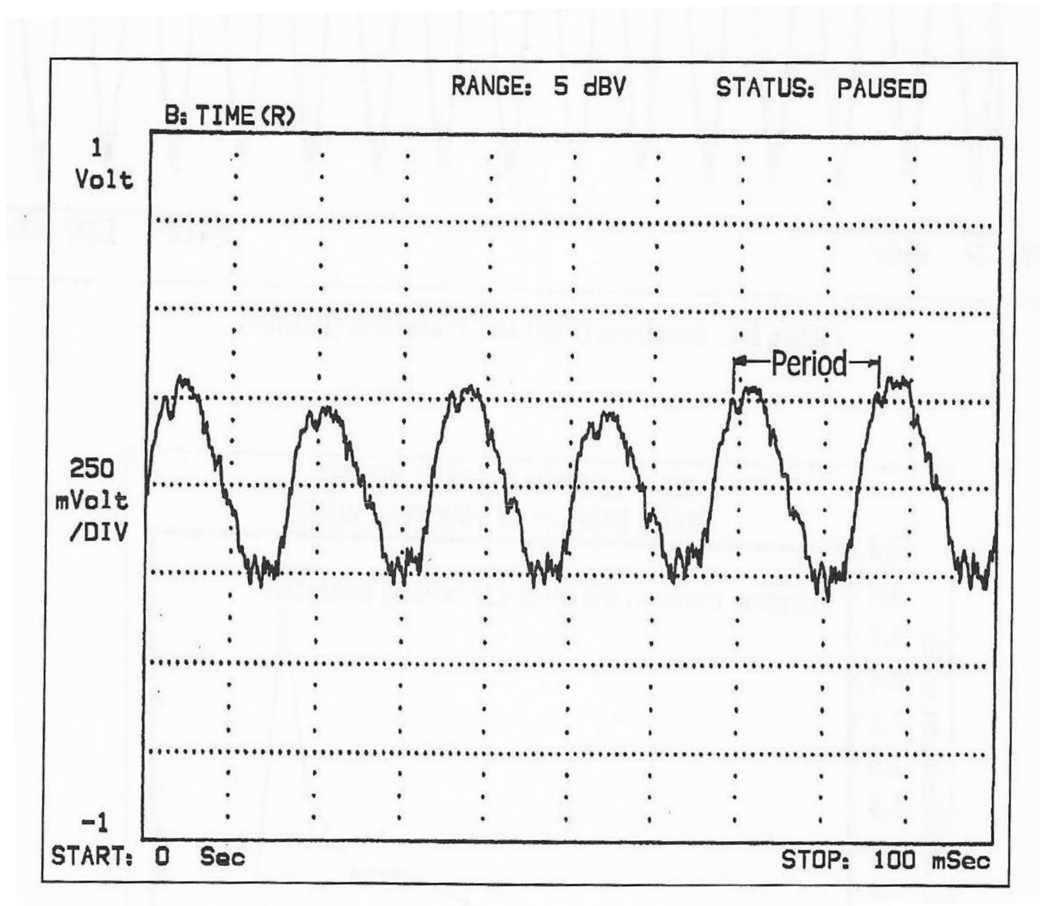


Figure 5.1. Time Waveform

Chapter 5 – VIBRATION INSTRUMENTS

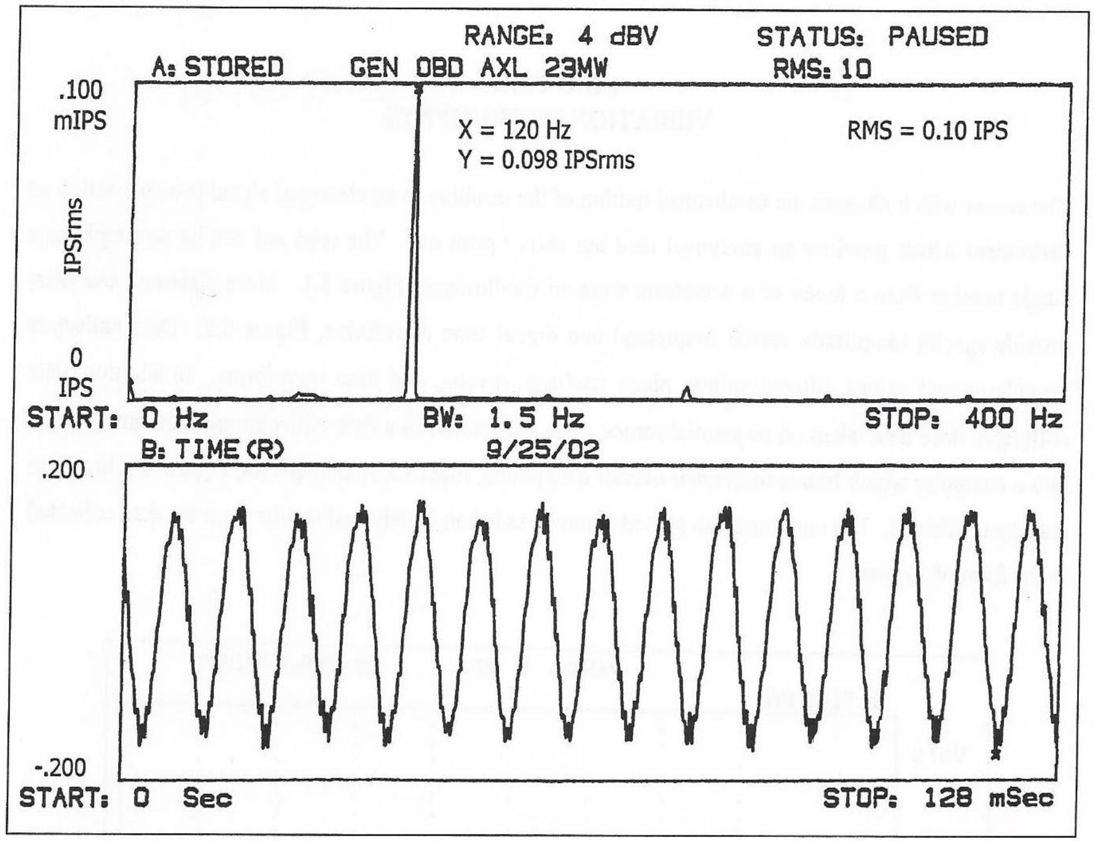


Figure 5.2. Spectrum (Top) and Waveform (Bottom).

Chapter 5 – VIBRATION INSTRUMENTS

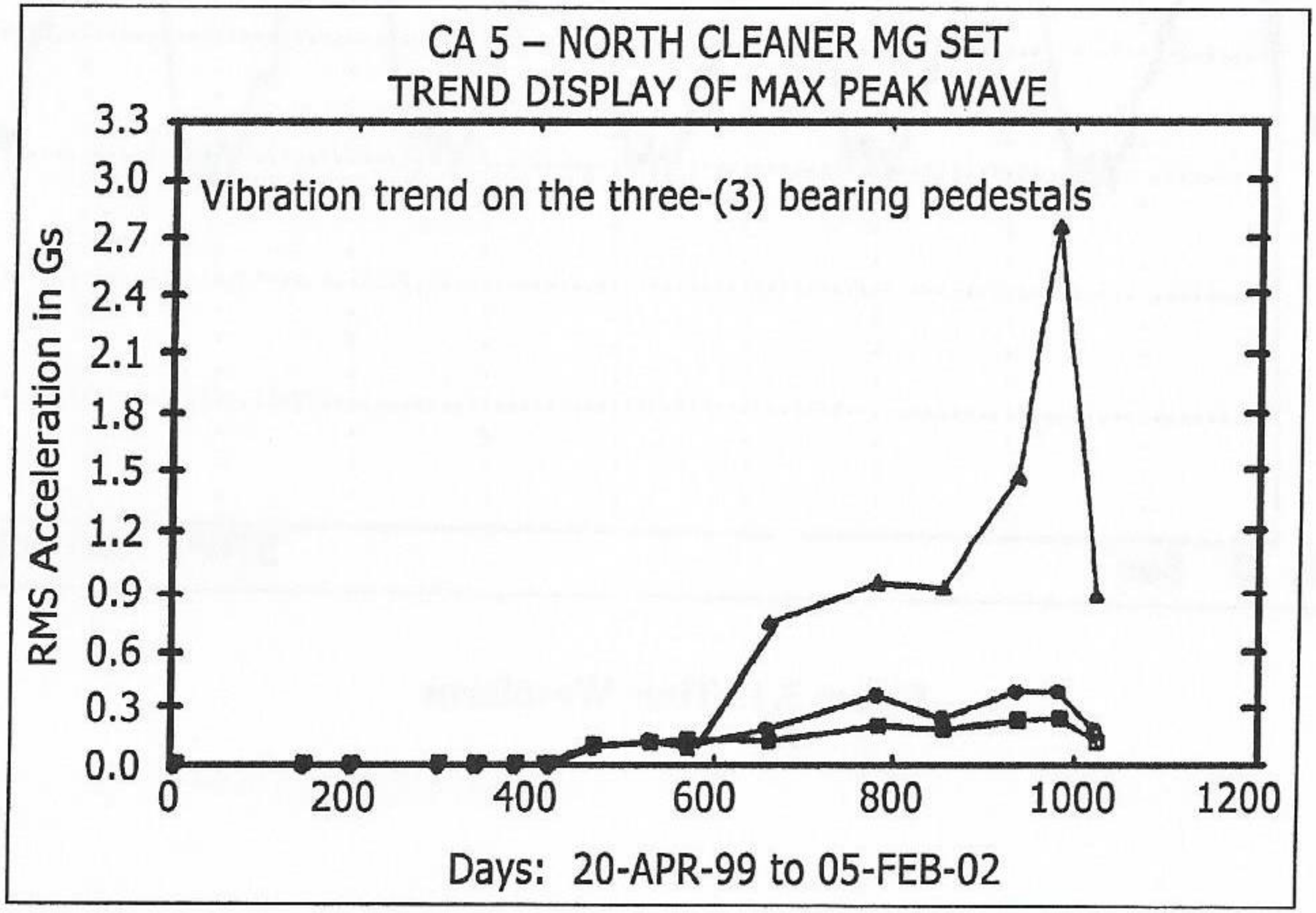
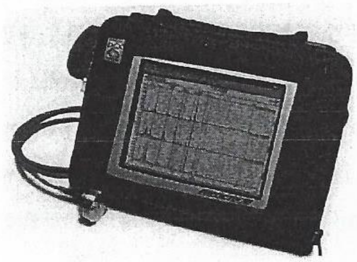


Figure. Trend on Three Bearing Pedestals.

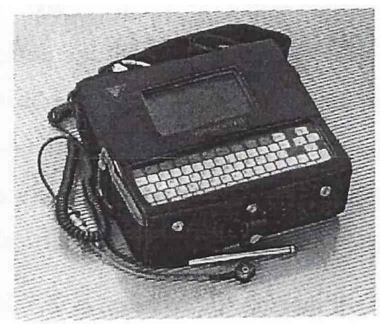
Data Collectors and Analyzers

The data collector (and analyzers are all Fast Fourier Transform (FFT) based calculated off a digitized waveform that is obtained from a sensor.



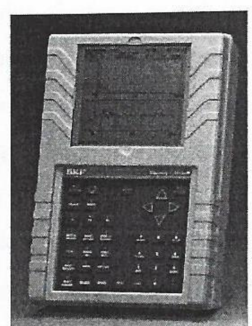
DLI Watchman® DCX™ Hammerhead Diagnostic Data Collector/Real Time Analyzer

CSI Model 2115/2120



Vibration Specialty

Bently Nevada Snapshot™



SKF Microlog

Entek

Spectrum Orders

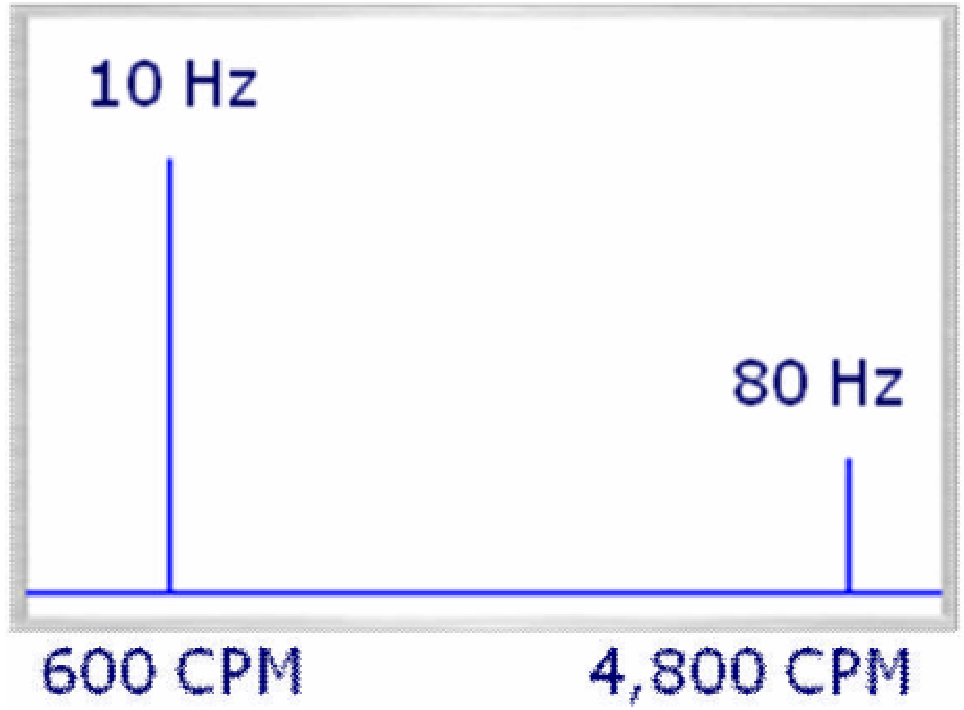


Figure 3-27 2 Frequency units are CPM and Hz.

The coin on the fan blade produced a vibration once every time the shaft turned. It didn't matter how fast it the shaft turned. The vibration occurred at the same frequency as the shaft.

Spectrum Orders

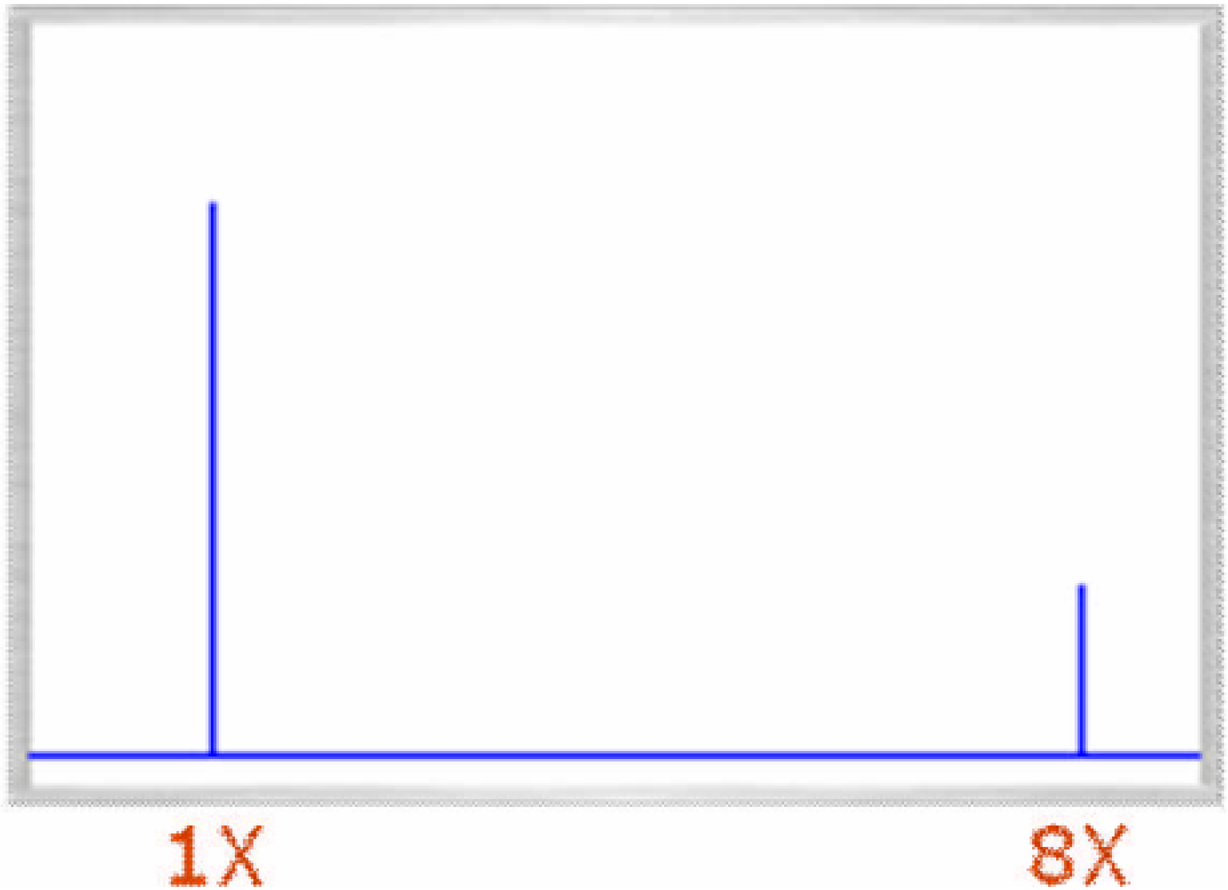


Figure 3-28 Frequency expressed in Orders.

Spectrum Order

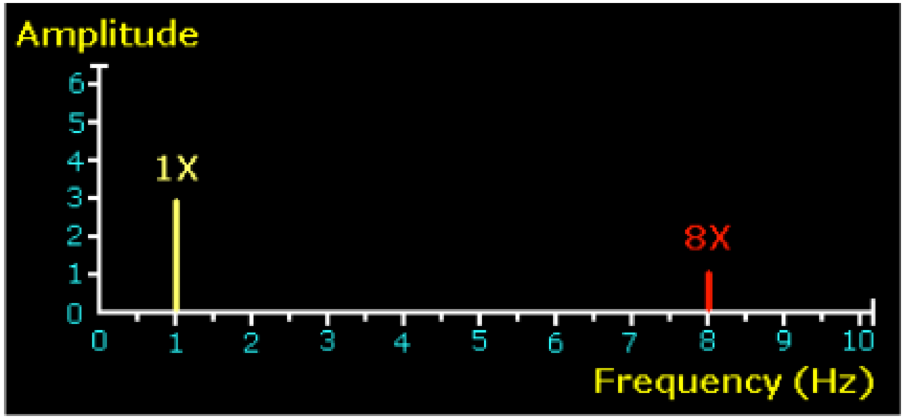


Figure 3-29 1X is at 1 Hz or 60 CPM. Blade pass of 8X is at 8 Hz or 480 CPM.

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

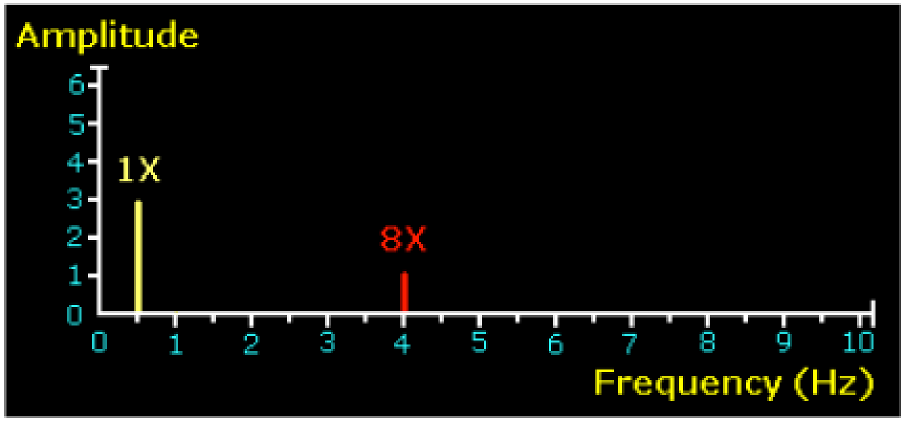


Figure 3-30 1X reduced to 0.5 Hz or 30 CPM. Blade pass of 8X is at 4 Hz or 240 CPM.

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

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

Spectrum Order

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

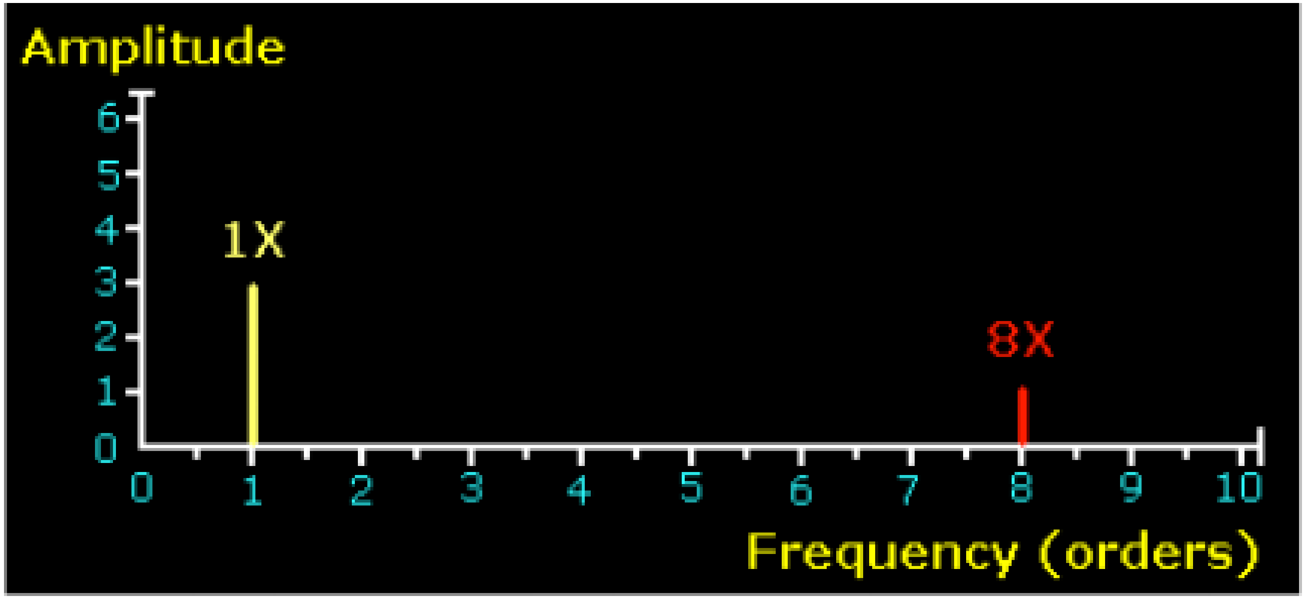


Figure 3-31 Spectrum with frequency in Orders.

Spectrum Order

Units of Orders is so useful in trying to find the source of peaks in a spectrum. When a peak is present at 5 orders it is easier to relate it to a physical occurrence such as vane pass on a pump. A peak at 38 orders could be rotor bars in a motor. A peak at 3X could relate to a 3 jaw coupling. Of course there are peaks that are not integer (whole number) multiples of turning speed.

Spectrum Order

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

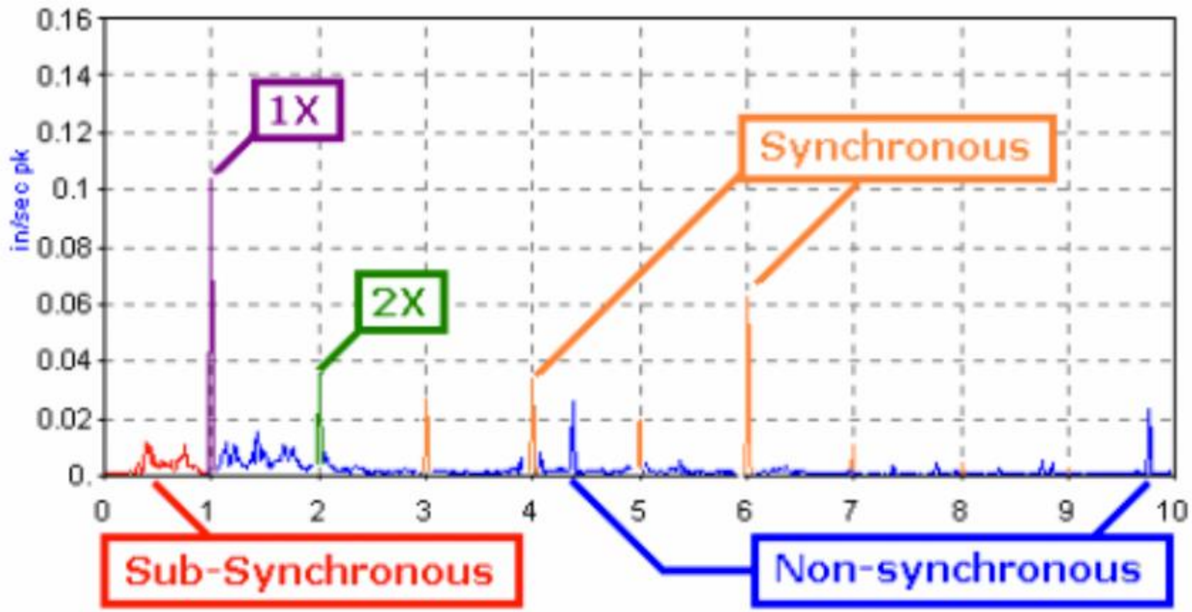


Figure 3-32 Spectral energy can be categorized into one of three groups.

Spectrum
Forcing Frequency

Examples of Forcing Frequencies include:

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

Spectrum
Forcing Frequency

Forcing Frequency Example:

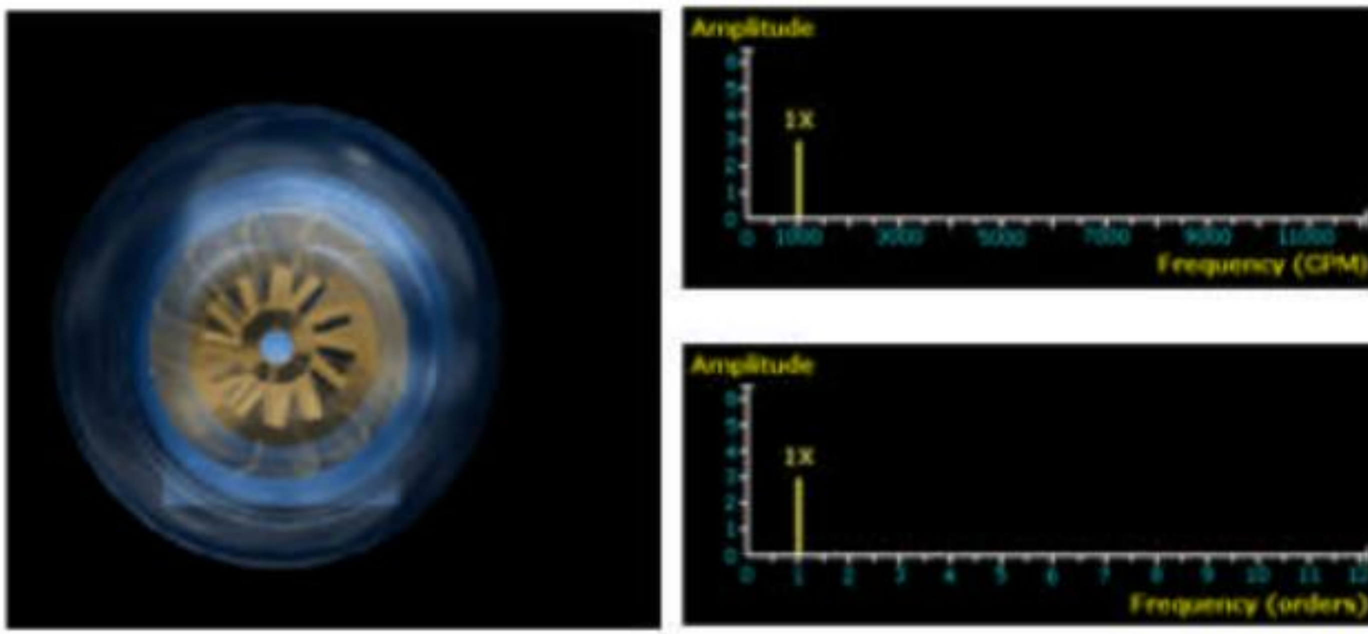


Figure 3-34 The motor running speed is 1000 CPM. A peak is expected at 1000 CPM which is also 1X.

Spectrum
Forcing Frequency

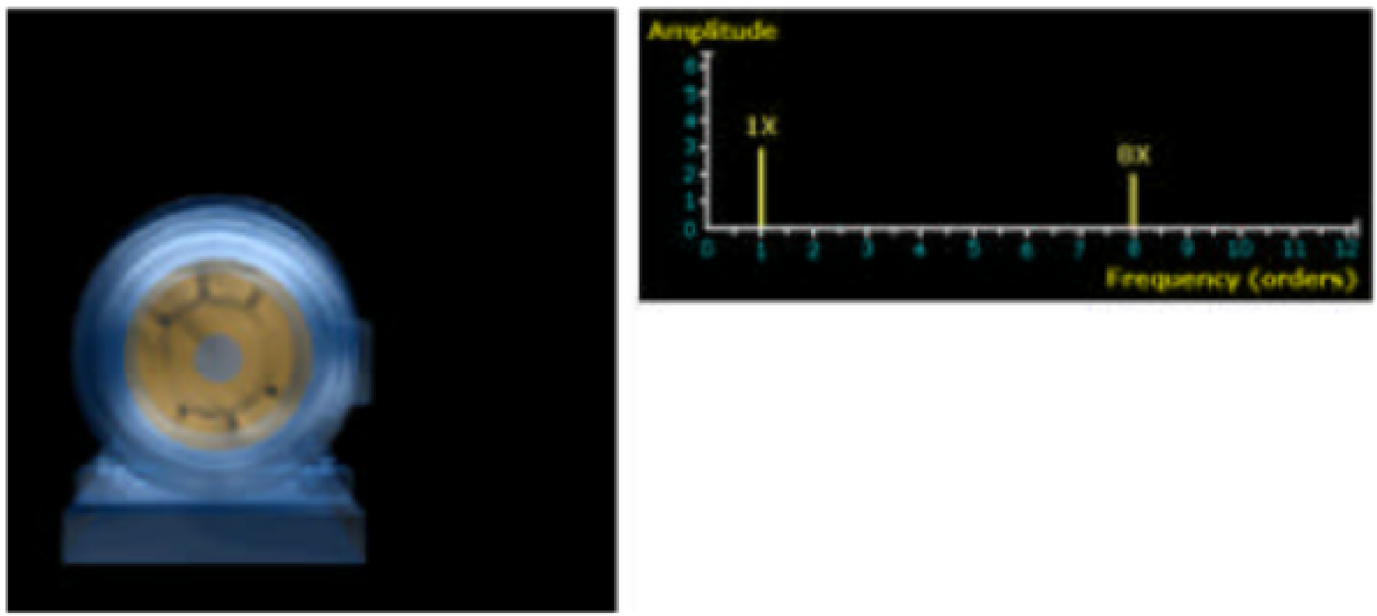


Figure 3-35 The motor cooling fan has 8 blades. Blade pass is 8X

Spectrum
Forcing Frequency

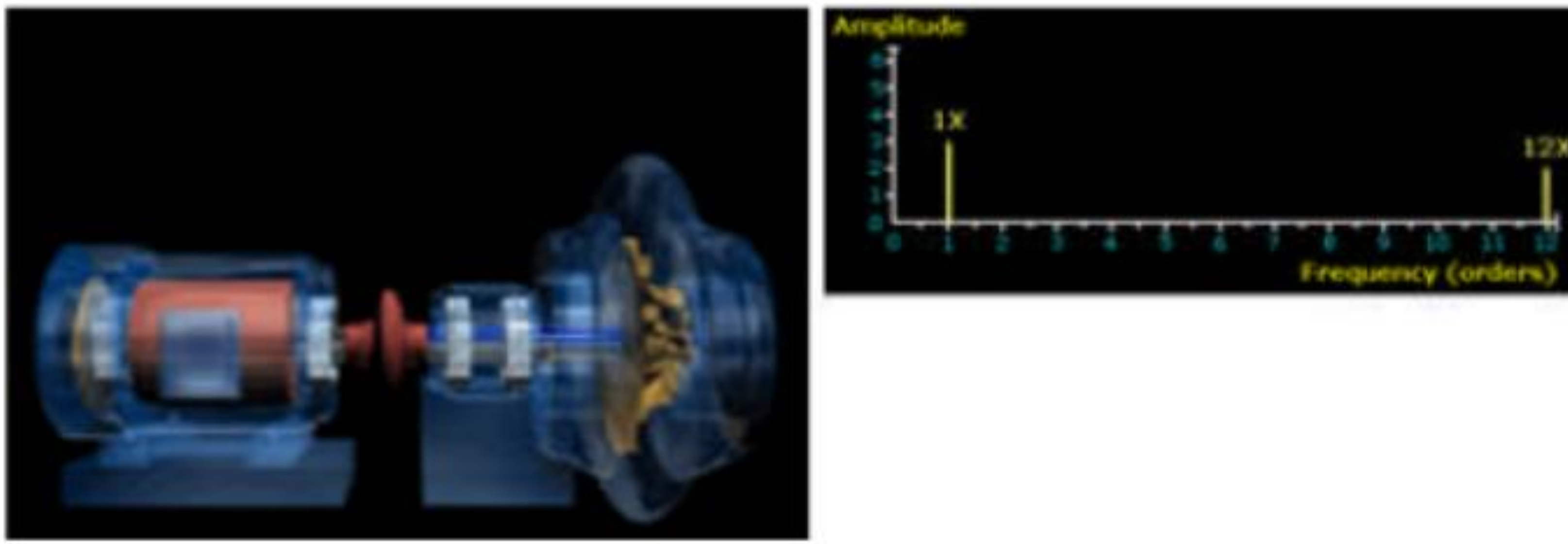


Figure 3-36 The compressor impeller has 12 vanes. A 12X peak is expected.

Spectrum
Forcing Frequency

So far, on the whole machine there are 3 frequencies that are expected; the motor and compressor shaft speed, the motor fan blade speed, and the compressor impeller vane speed.

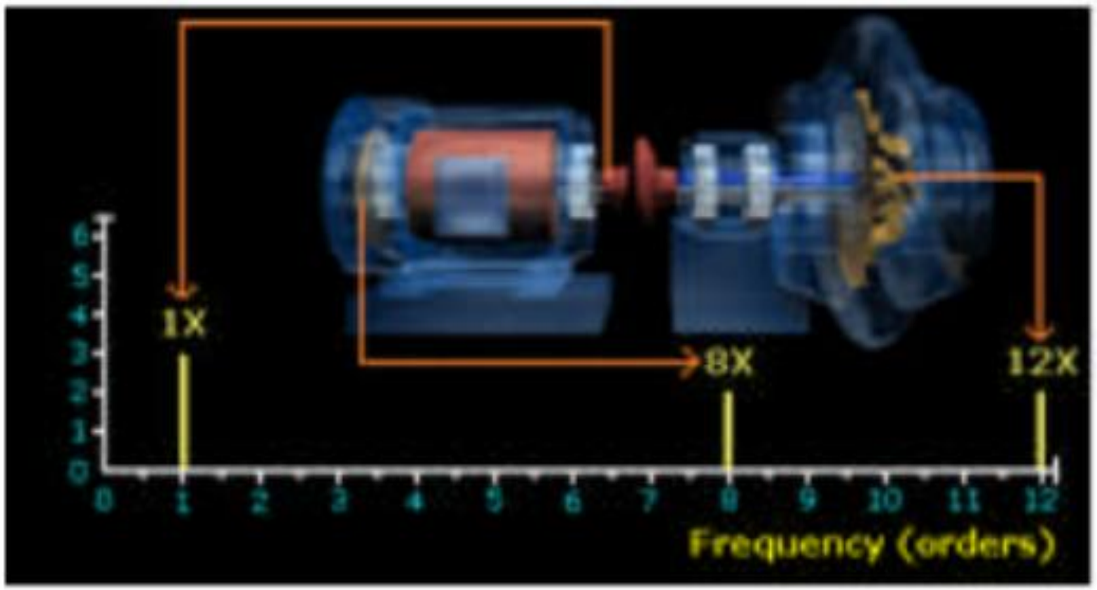
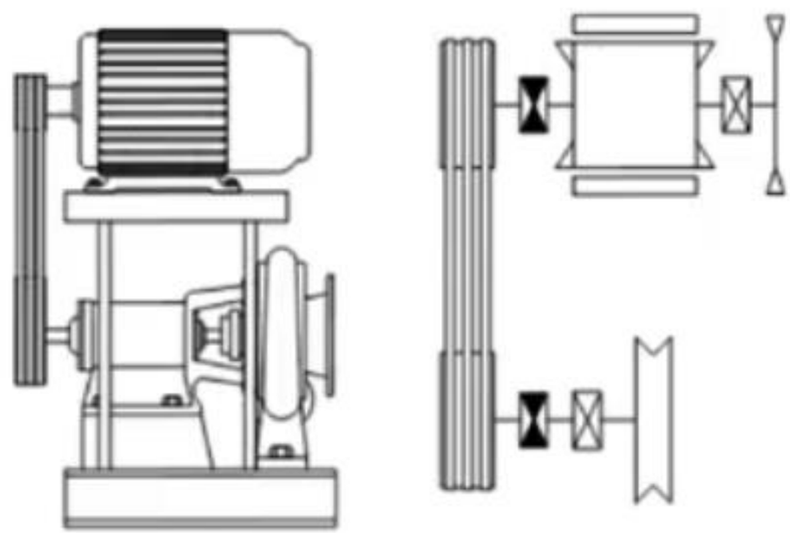


Figure 3-37 There are 3 expected frequencies for the machine.

Spectrum
Forcing Frequency

[4] If the pulley on this motor had a 70 mm diameter, and the pulley on the pump had a 350 mm diameter, and the motor speed was 2970 RPM, calculate the speed of the pump: _____CPM_____Hz______x (orders)



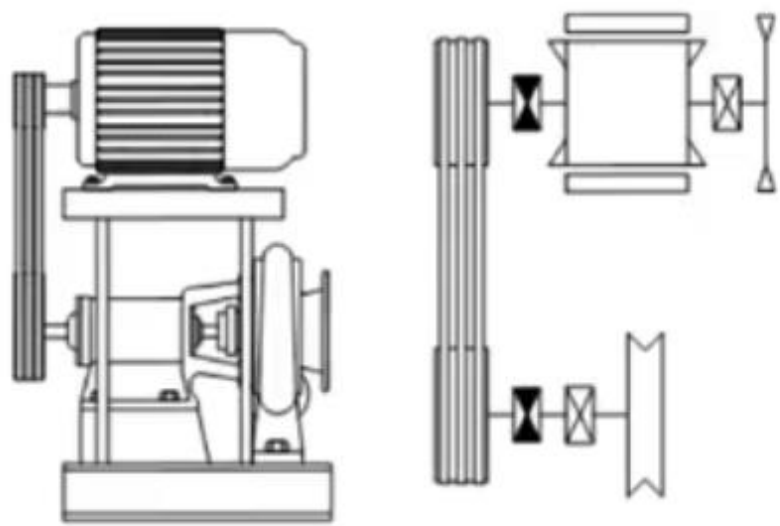
Spectrum
Forcing Frequency

[4] If the pulley on this motor had a 70 mm diameter, and the pulley on the pump had a 350 mm diameter, and the motor speed was 2970 RPM, calculate the speed of the pump:

2970 x 70 / 350 = 594 CPM

594 / 60 = 9.9 Hz

1x x (70/350) = 0.2 X

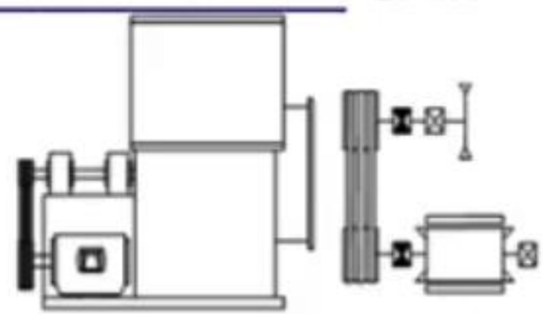


Spectrum
Forcing Frequency

[5] If the pulley on the 1760 RPM motor had a diameter of 160 mm, and the pulley on the fan had a 110 mm diameter, and the length of the belt is 1700 mm, and there were 15 blades on the fan, calculate the following frequencies:

- Fan shaft speed: ______________________ CPM
- Belt rate frequency: ___________________ CPM
- Fan blade pass rate: ___________________ CPM

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

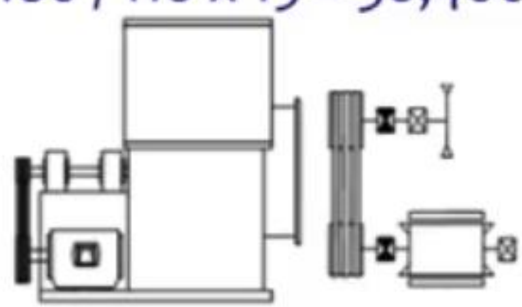


Spectrum
Forcing Frequency

[5] If the pulley on the 1760 RPM motor had a diameter of 160 mm, and the pulley on the fan had a 110 mm diameter, and the length of the belt is 1700 mm, and there were 15 blades on the fan, calculate the following frequencies:

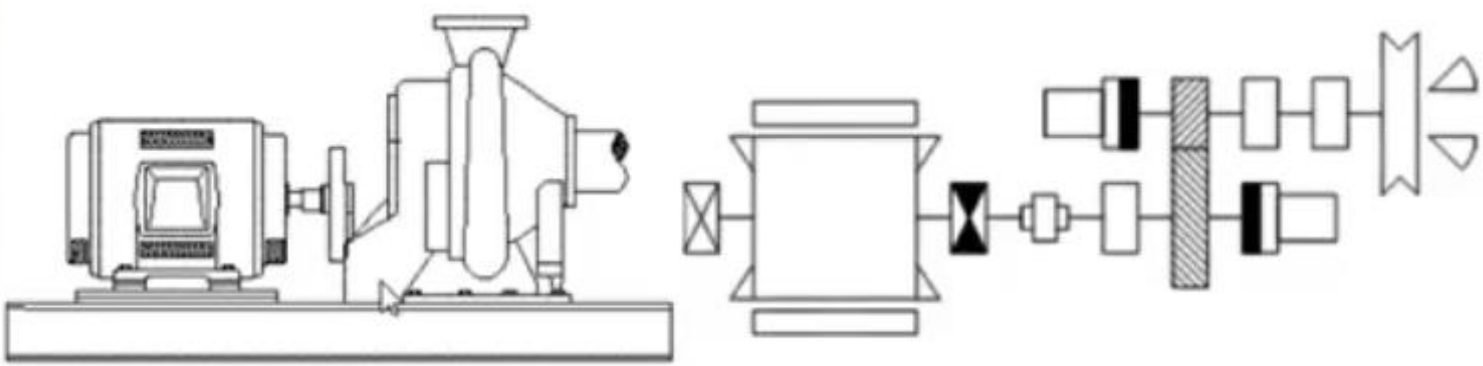
- Fan shaft speed: 1760 x 160 / 110 = 2,560 CPM
- Belt rate frequency: 3.1416 x 160 x 1760 / 1700 = 520 CPM
- Fan blade pass rate: 1760 x 160 / 110 x 15 = 38,400 CPM

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



Spectrum
Forcing Frequency

[6] If a compressor was driven by a gearbox with a 1:2 ratio, and the motor speed was 1480 CPM, calculate the compressor speed: ________ CPM____X

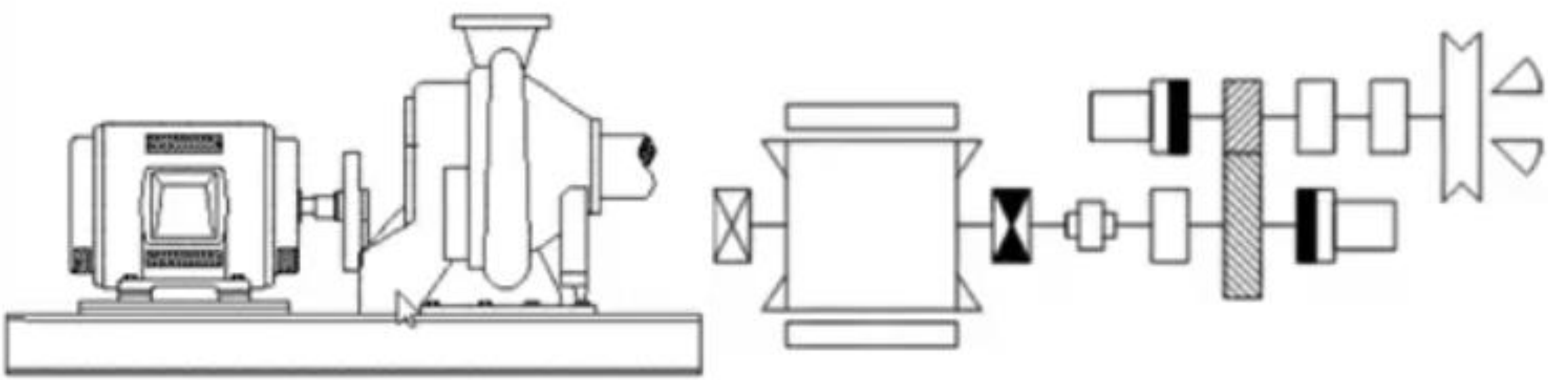


Spectrum
Forcing Frequency

[6] If a compressor was driven by a gearbox with a 1:2 ratio, and the motor speed was 1480 CPM, calculate the compressor speed:

1480 x 2 = 2,960 CPM

1X x 2 = 2 X



Identifying Running Speed

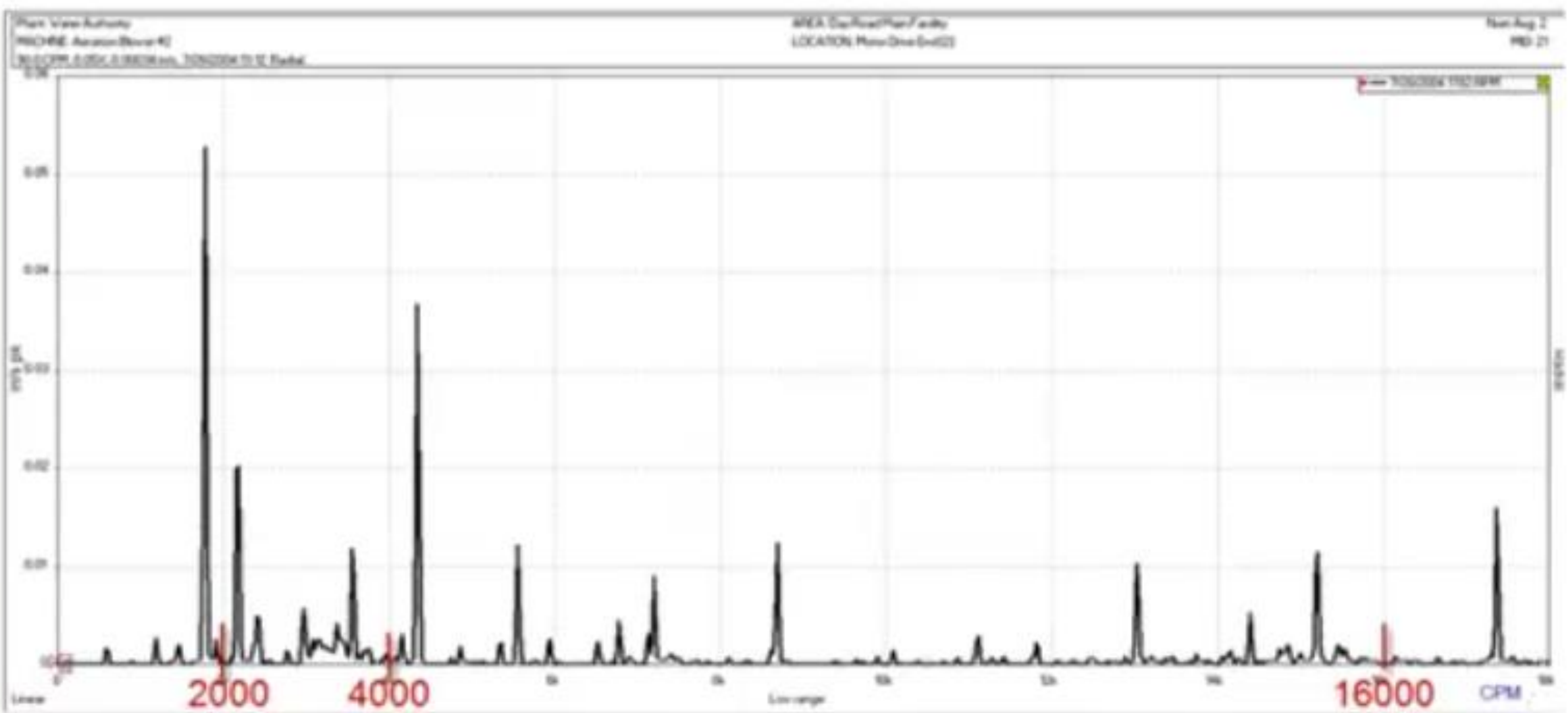
- You should know what speed it is running at if you are controlling the test conditions.

Motor: 1800 RPM

	Code	Name	On Secondary	Elements	Final Ratio
▶	BR	BELT ROTATION	No	0.335	0.335
	1XM	1 X MOTOR SHAFT	No	1	1
	1XW	1 X BLOWER SHAFT	Yes	1	1.2169
	2XM	2 X MOTOR SHAFT	No	2	2
	BL	BLOWER LOBES	Yes	2	2.4338
	3XM	3 X MOTOR SHAFT	No	3	3
	2BL	2 X BLOWER LOBES	Yes	4	4.8676
	3BL	3 X BLOWER LOBES	Yes	6	7.3014
	4BL	4 X BLOWER LOBES	Yes	8	9.7352
	MB	MOTOR ROTOR BARS	No	64	64

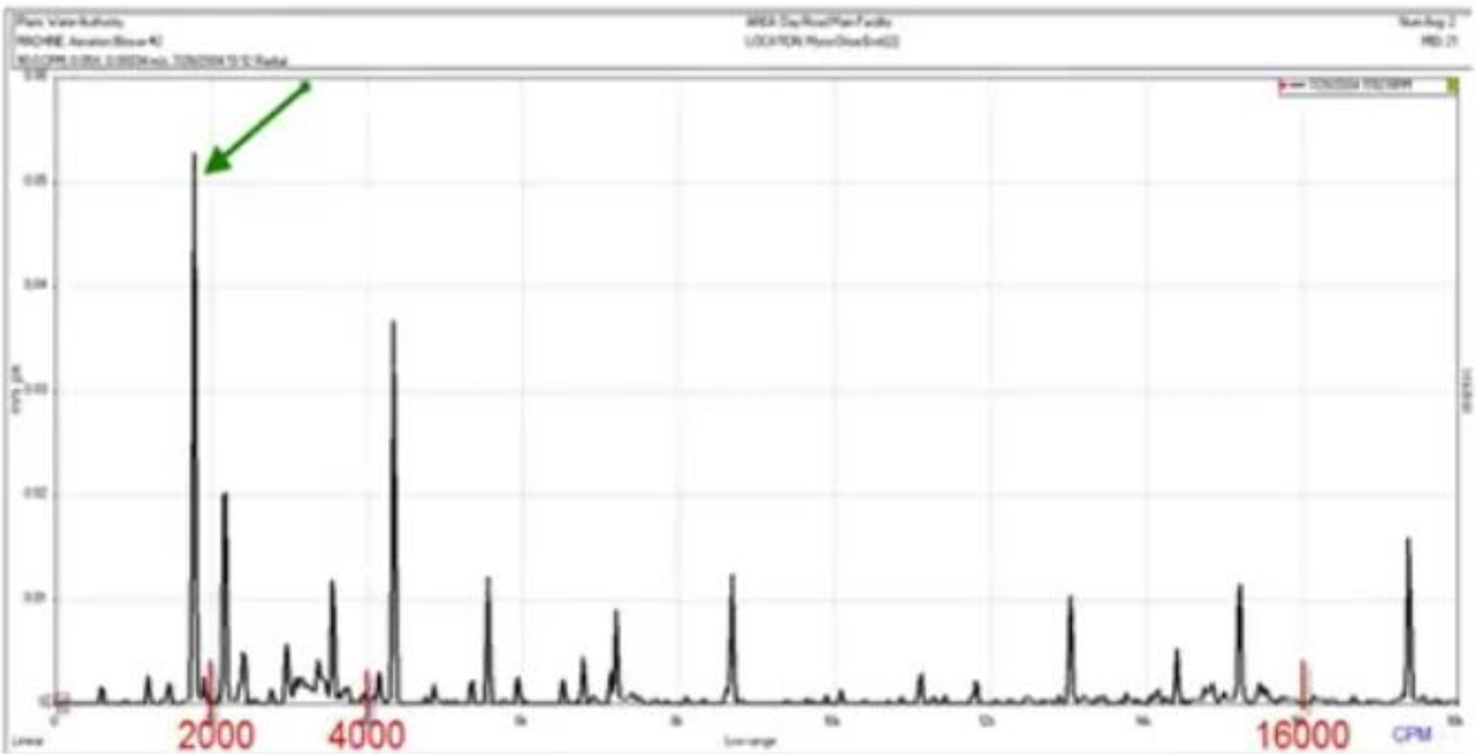
Identifying Running Speed

- You must determine the speed of the machine
- Find the 1X peak (1800 RPM)
- X-axis is in CPM



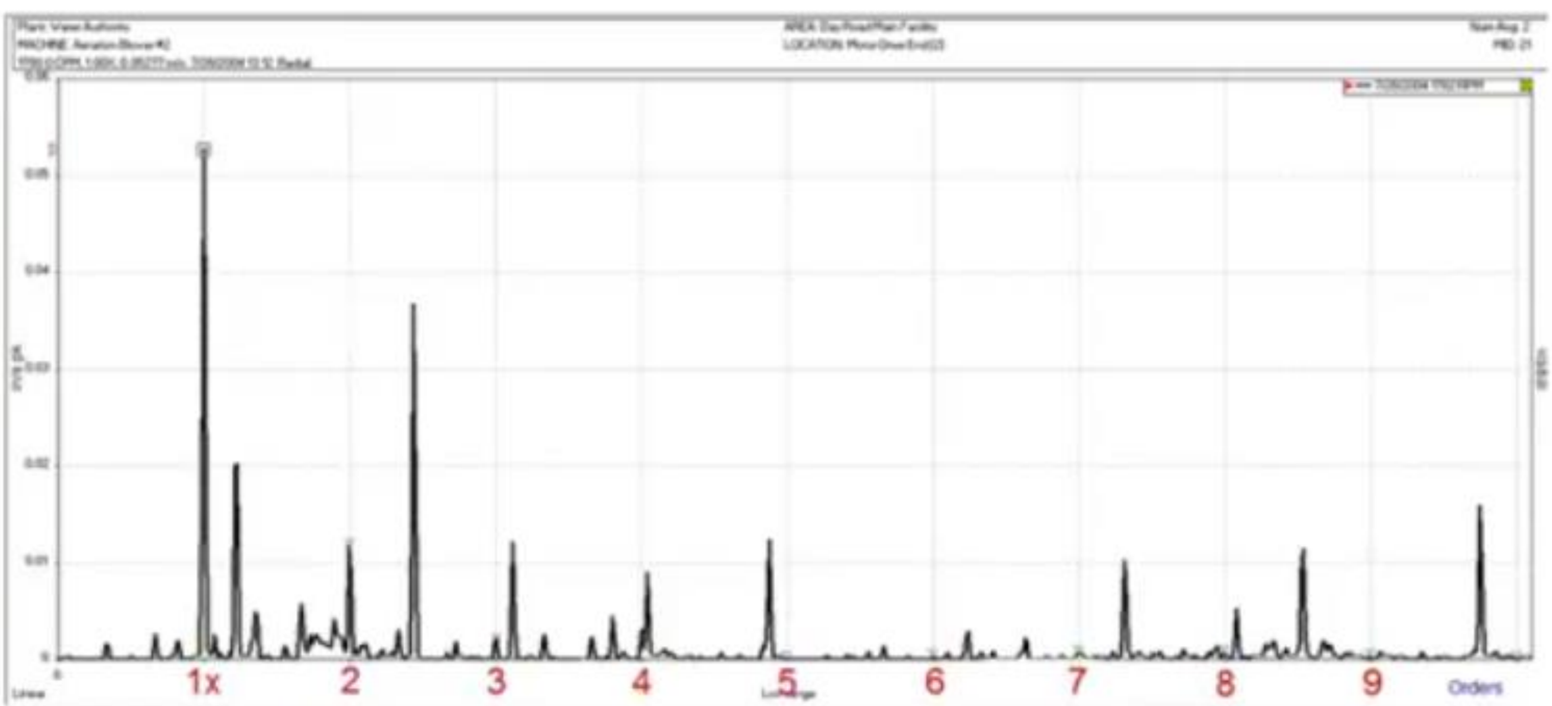
Identifying Running Speed

- The large peak is just to the left of 2000 CPM
  - Put a cursor on in and see it is a bit less than 1800 RPM



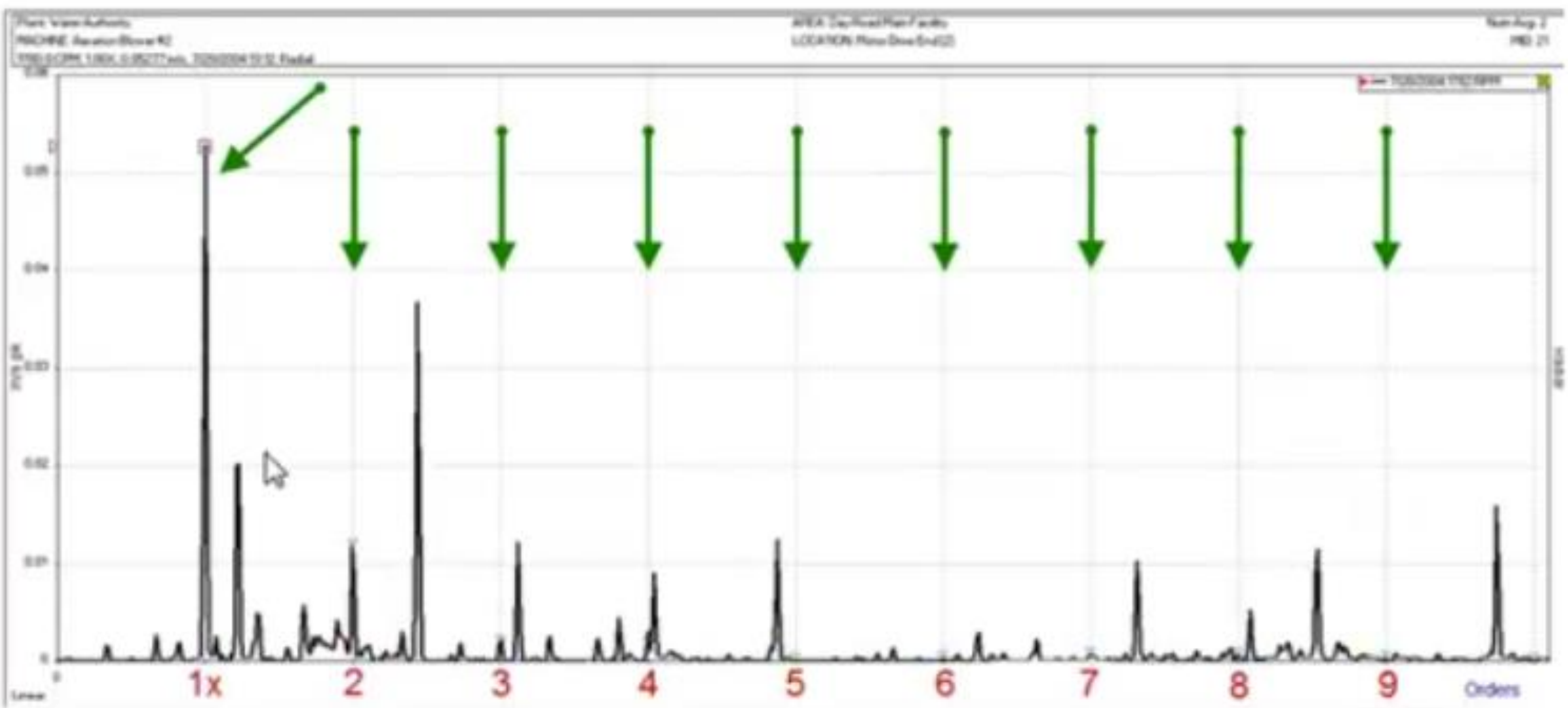
Identifying Running Speed

- Identify 1x peak
- Change X-axis to "Orders"



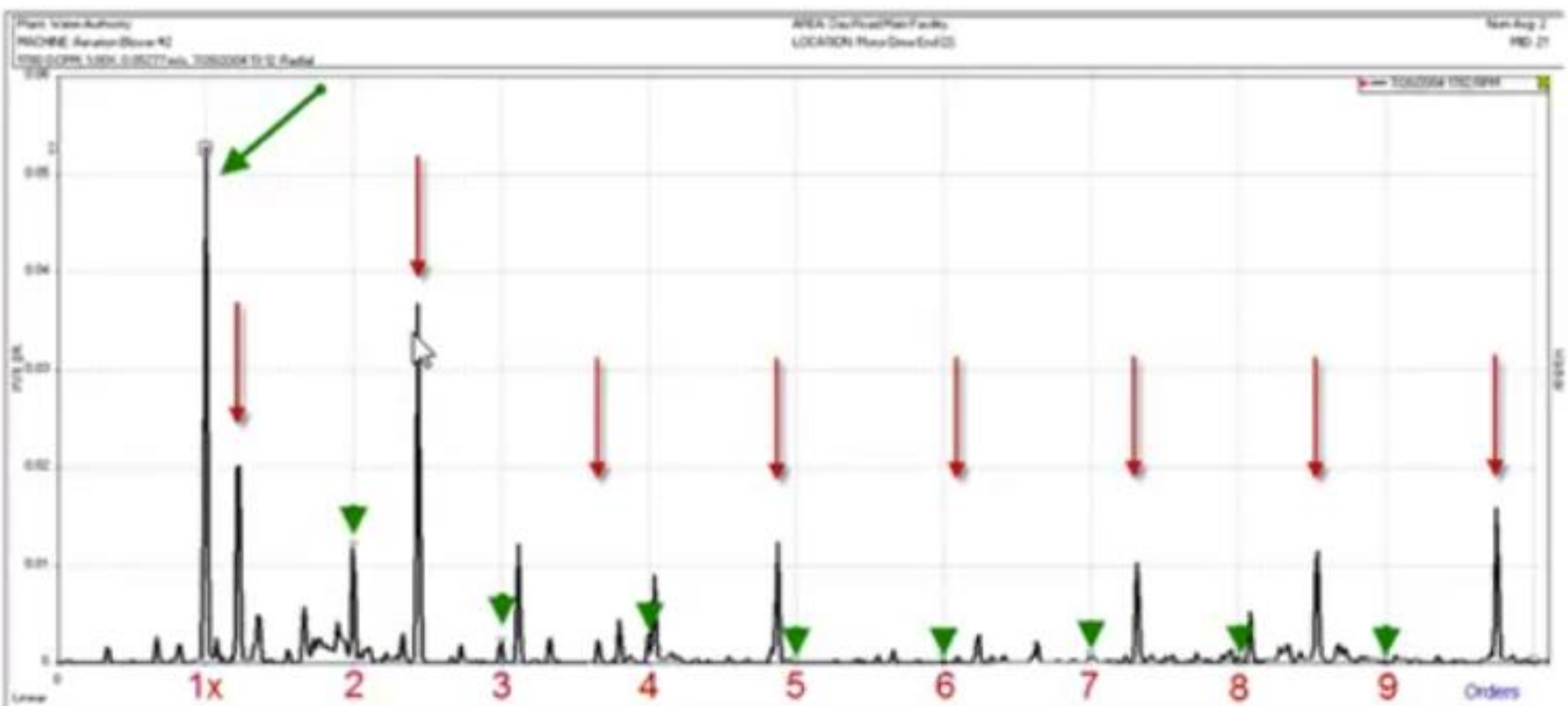
Identifying Running Speed

- All of these marked peaks are related to the motor shaft
  - Next we will look for the blower shaft at 1.22x



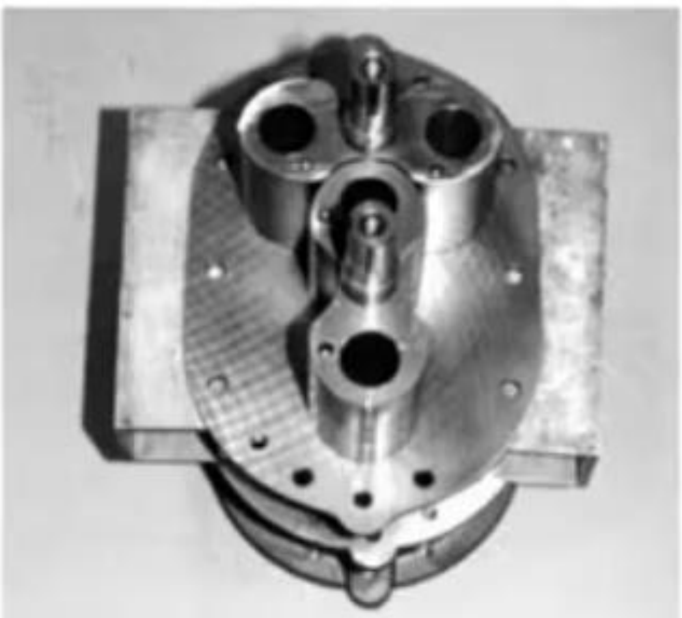
Identifying Running Speed

- The peaks marked by red arrows are harmonics of the blower shaft



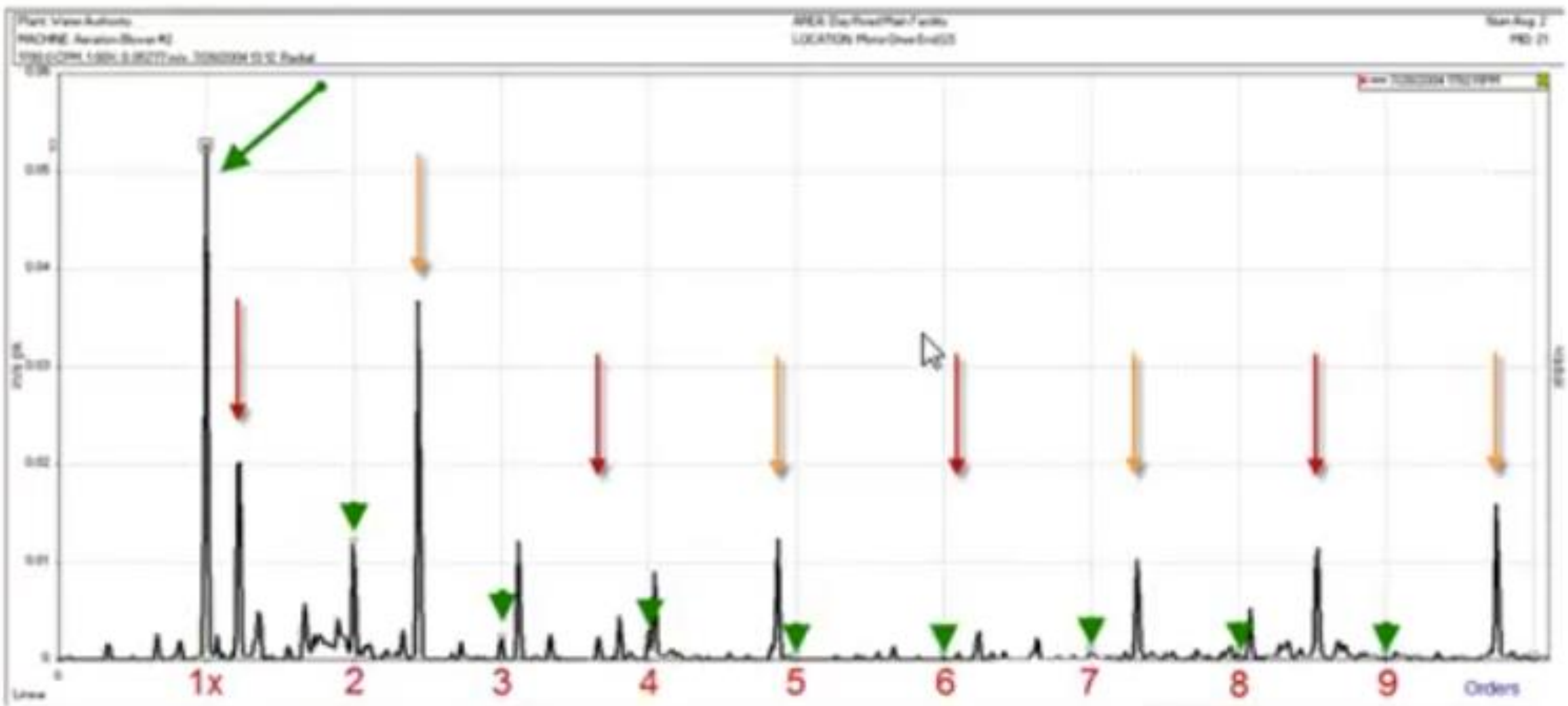
Identifying Running Speed

- The blower has 2 lobes
- What is the "lobe pass" frequency?



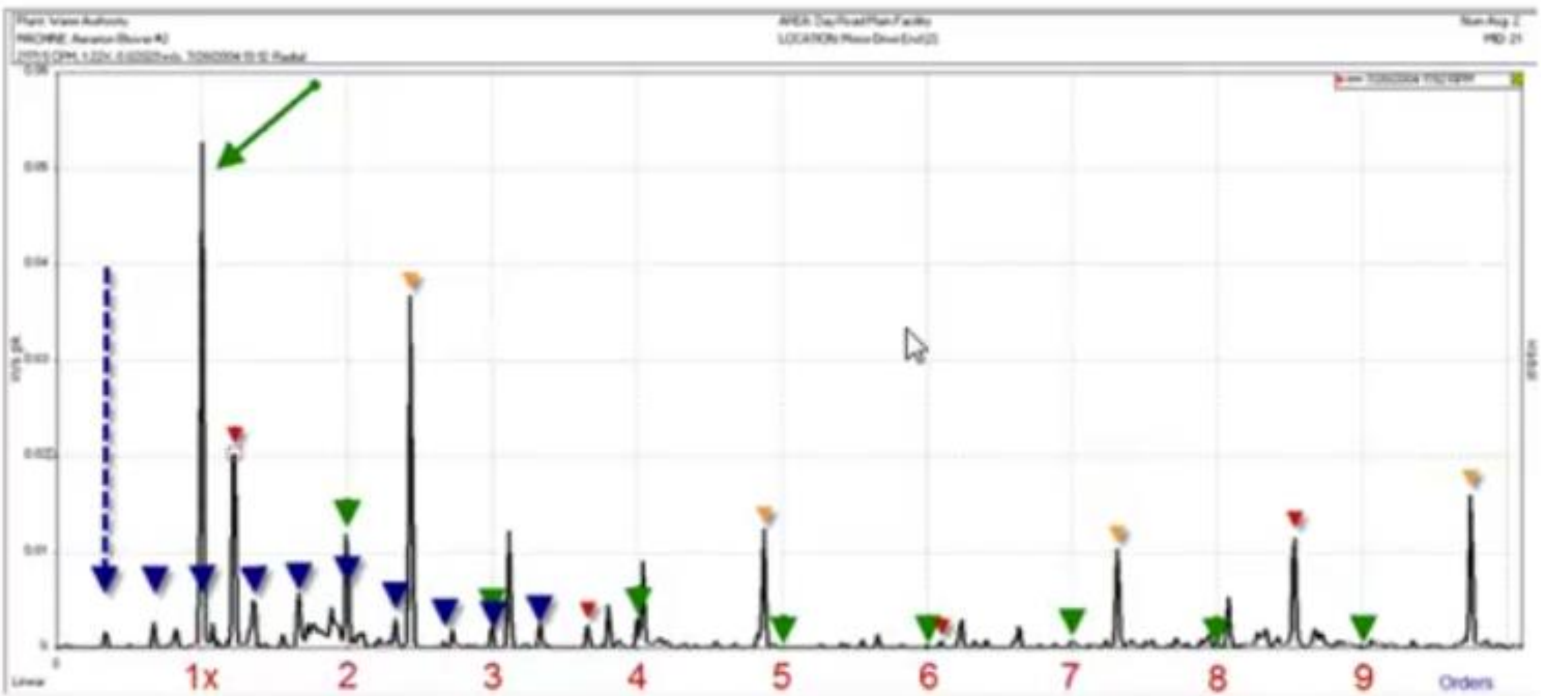
Identifying Running Speed

- The blower has 2 lobes – 2x the blower shaft rate
  - The orange peaks are blower lobe harmonics
- Next we will look for the belt at 0.36x



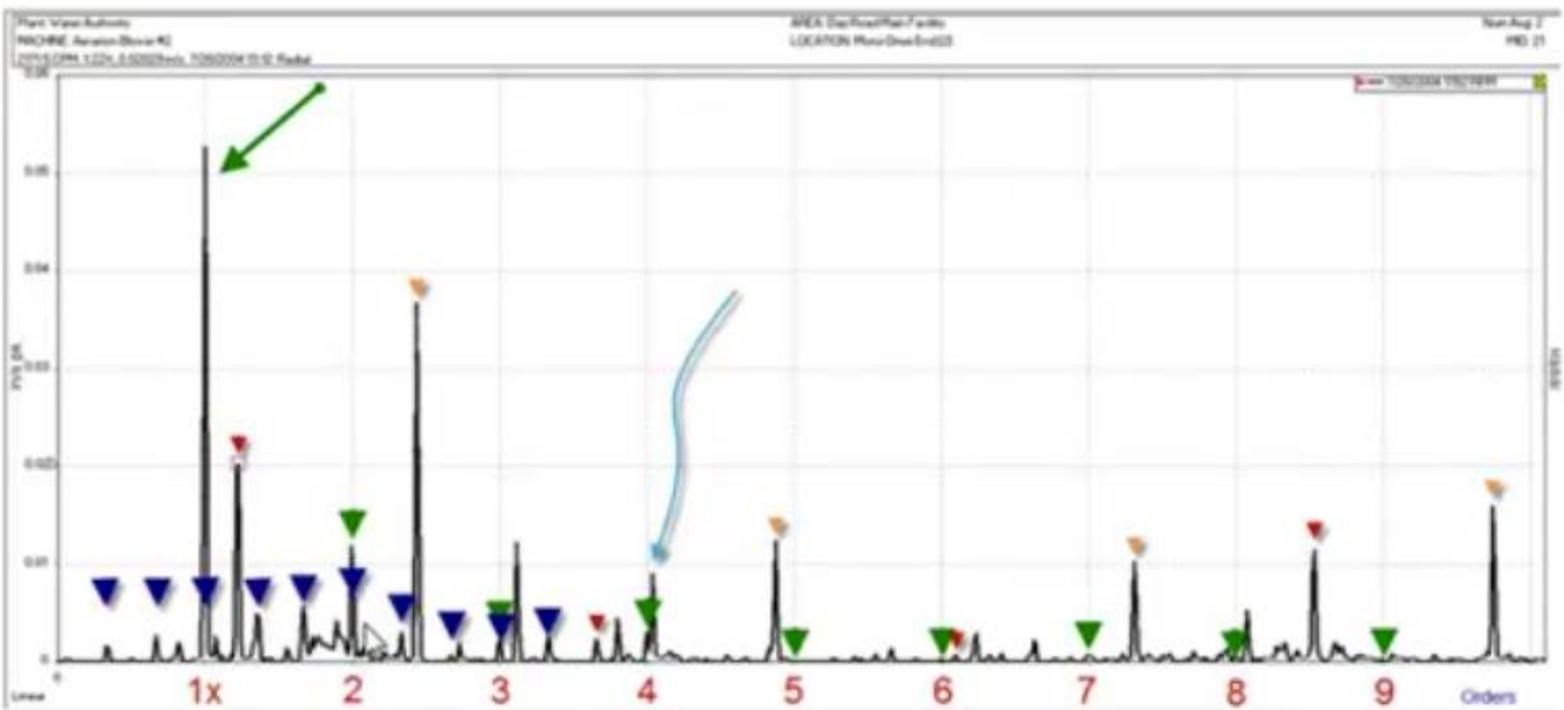
Identifying Running Speed

- The belt is at 0.36x
  - The belt rate will also have harmonics



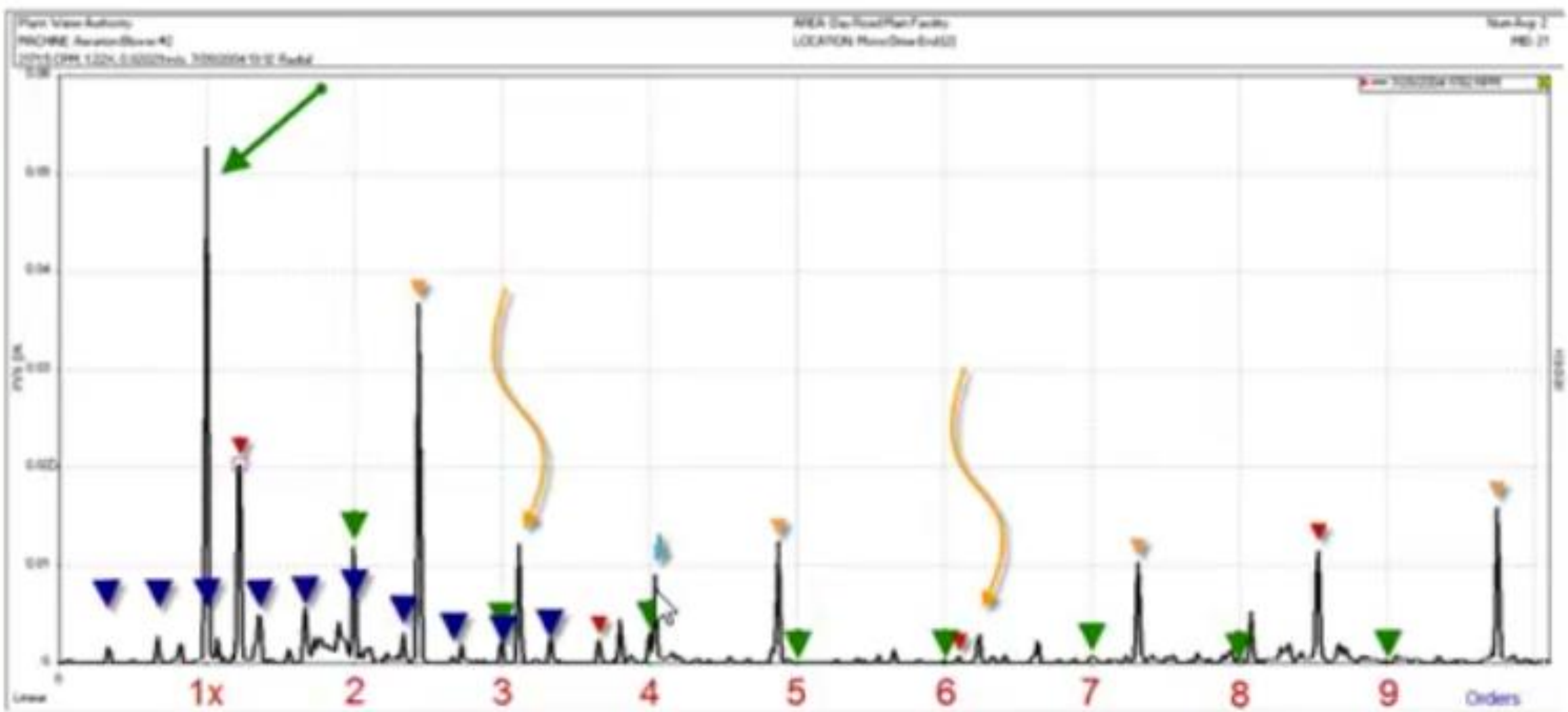
Identifying Running Speed

- There is a peak just over 4X
  - What frequency is this in Hz or CPM?
- Are there any other unidentified peaks?



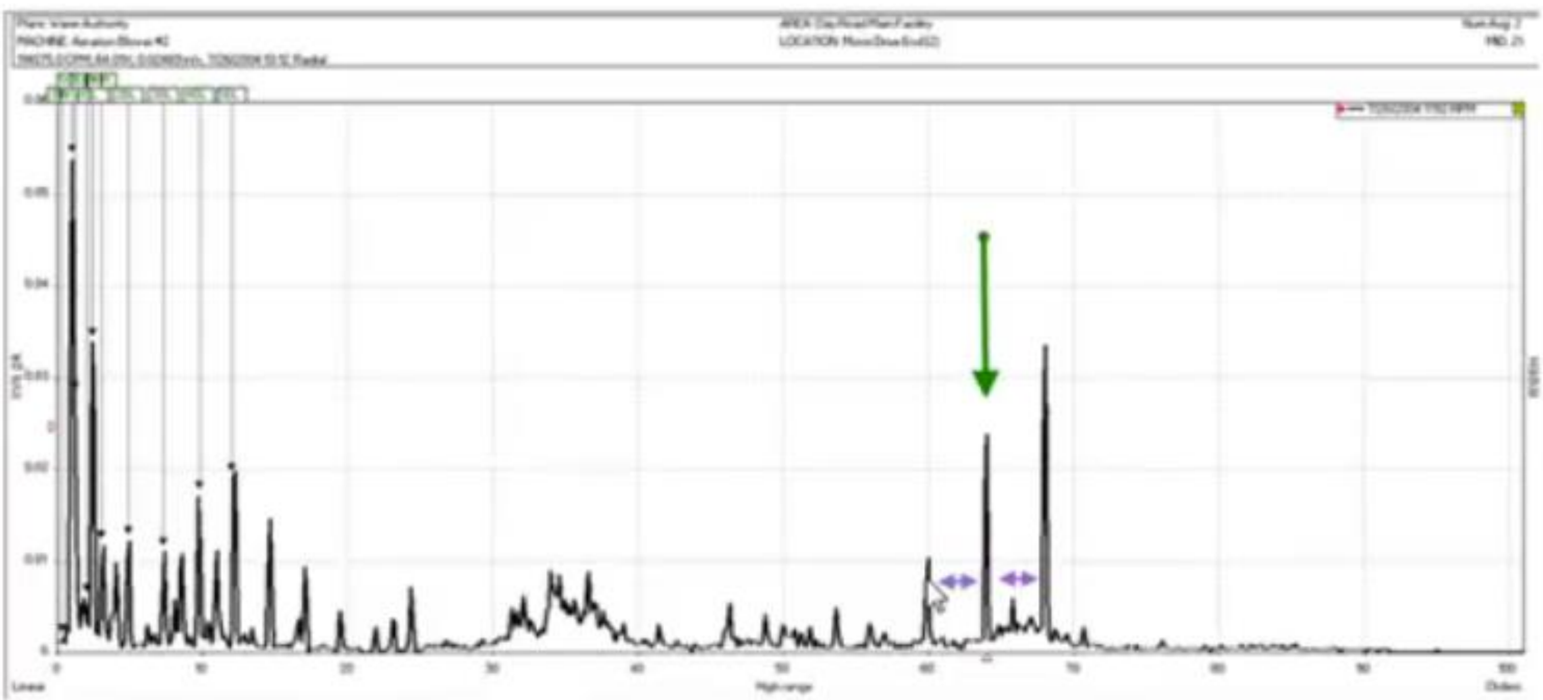
Identifying Running Speed

- There is a peak at 3.1x with a harmonic at 6.2x
  - Ideas???
  - We'll talk more about it later…



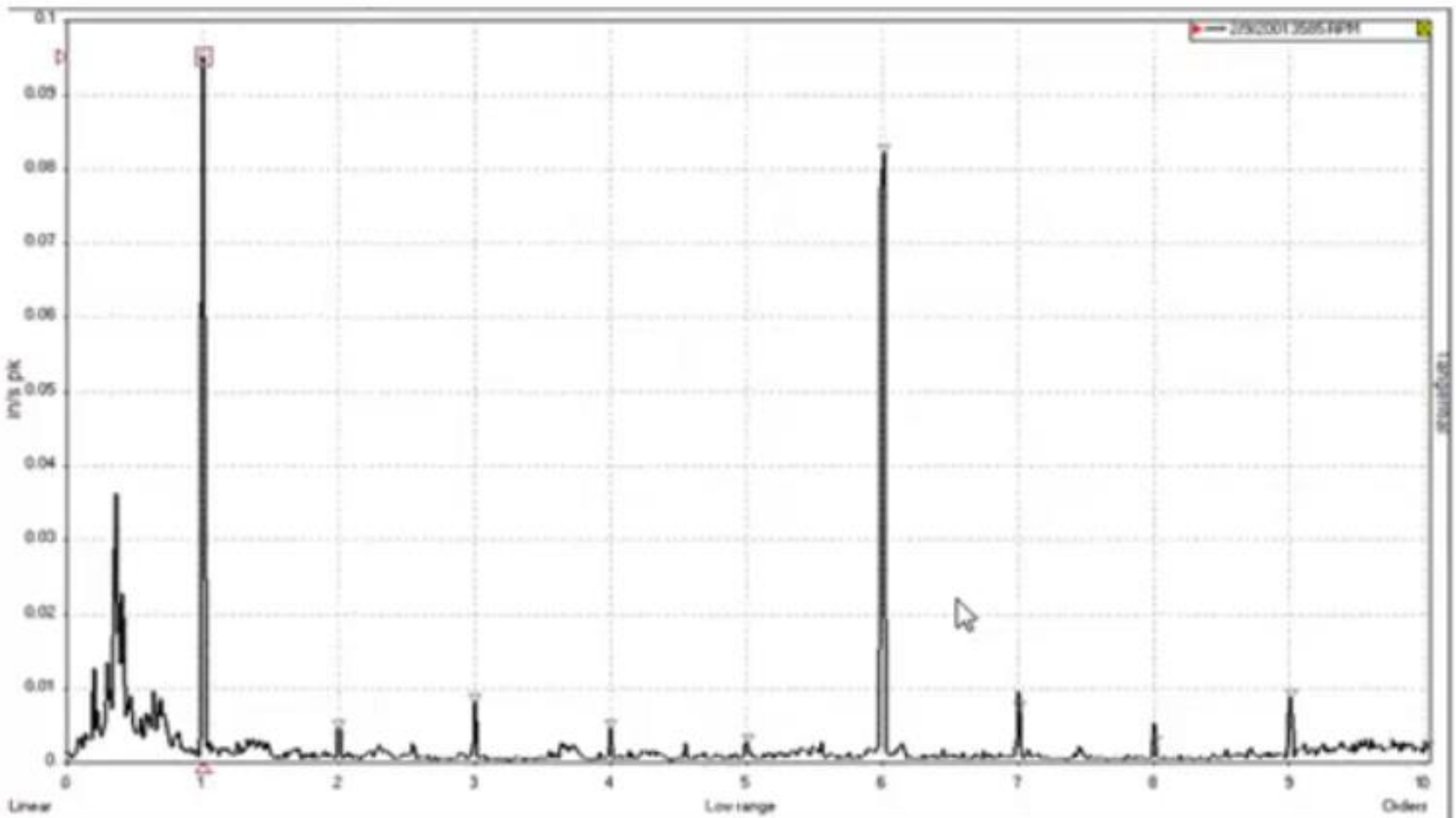
Identifying Running Speed

- These equally spaced peaks are separated by 120 Hz
  - Now we have identified the motor rotor bars



Identifying Running Speed

- This is a centrifugal pump. 1X is marked. How many pump vanes does the pump have?



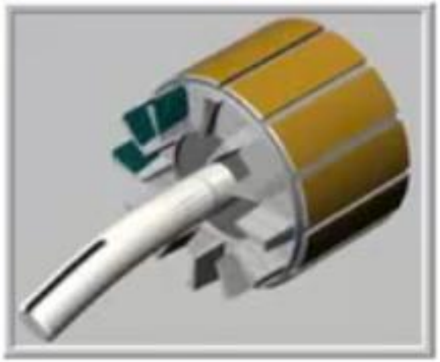
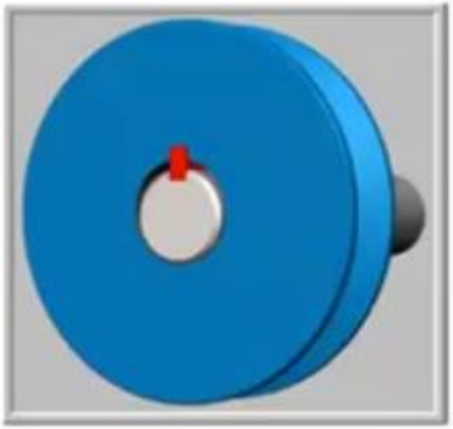
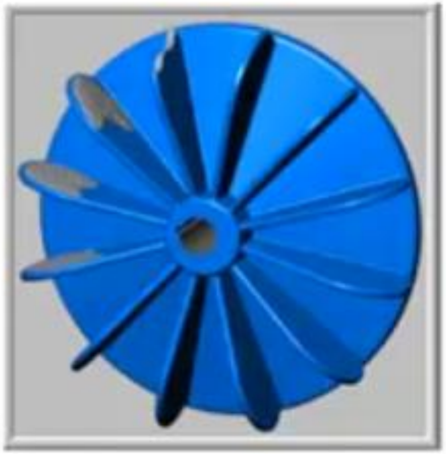
Gearbox failures

- There are a number of reasons why a gearbox may fail:
  - Tooth wear
  - Tooth load
  - Gear eccentricity
  - Backlash
  - Gear misalignment
  - Broken or cracked teeth
  - And others…

Vibration Analysis
Unbalance

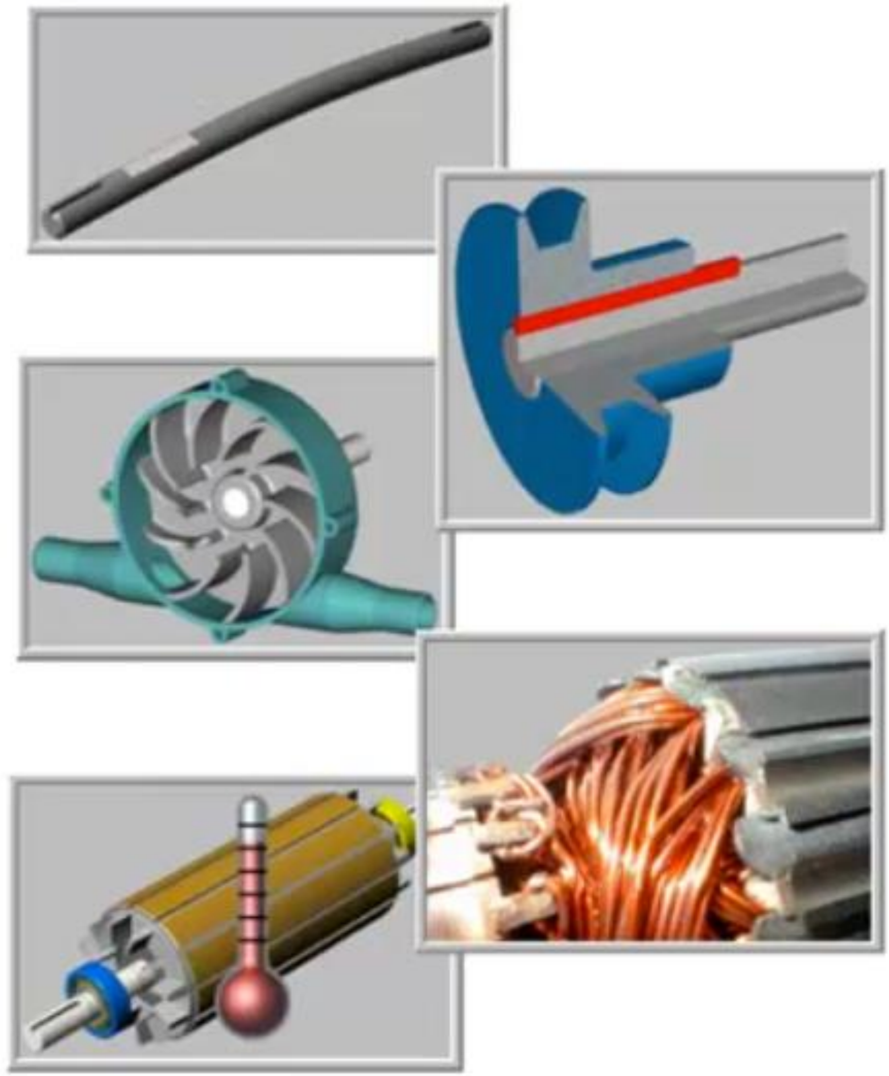
Causes of Unbalance

- Uneven dirt accumulation on fan rotors
- Lack of homogeneity in materials, especially in castings (e.g. bubbles, porous sections, blow-holes)
- Difference in dimension of mating parts (e.g. shaft, bore…)
- Eccentric rotor (which will be discussed shortly)
- Cracked rotor

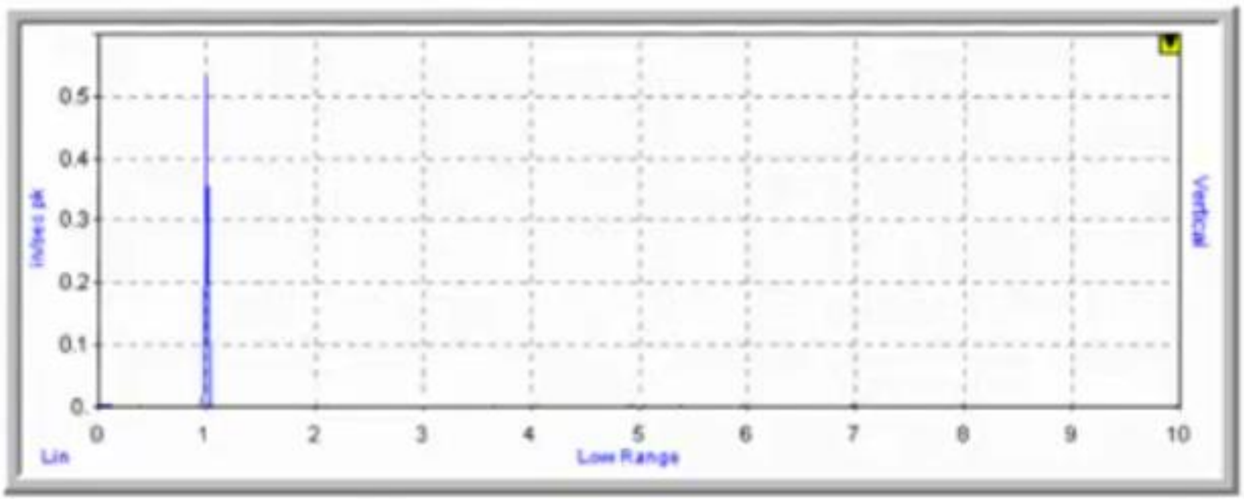
Causes of Unbalance

- Roller deflection (e.g. paper mill rolls)
- Machining errors
- Uneven mass distribution in electrical windings
- Uneven corrosion or erosion of rotors
- Missing balance weights
- Incorrect key
- Uneven or excessive heating



Mass Unbalance

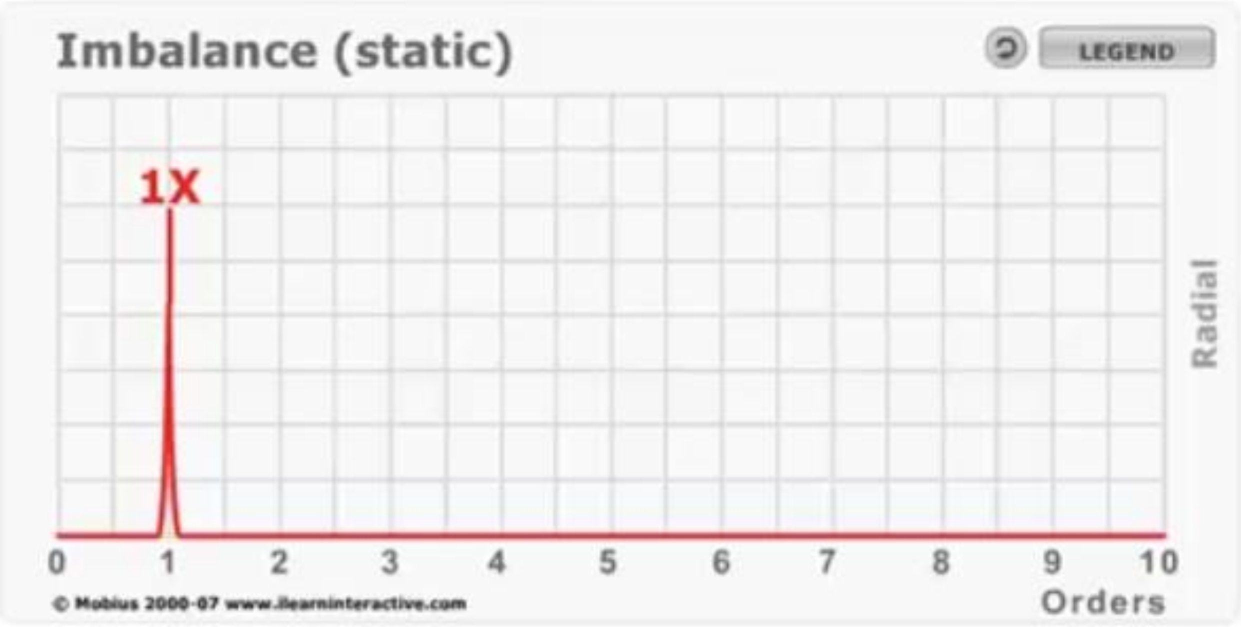
We expect to see a high 1X peak in the spectrum in the radial direction



Mass Unbalance

- There is always some "residual" imbalance
  - There is almost always a 1X peak

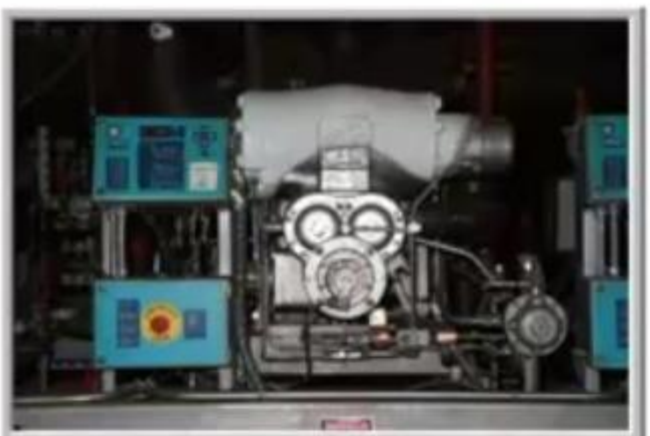
- If the spectrum is dominated by 1X, and the amplitude is high – suspect mass unbalance



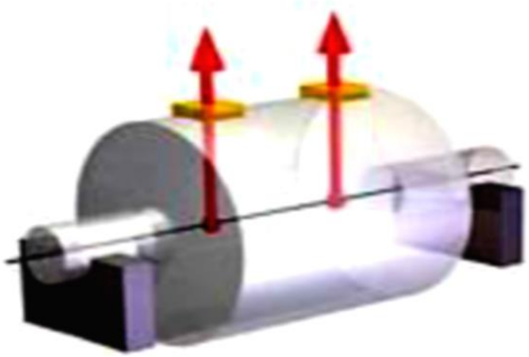
The importance of Mass Unbalance

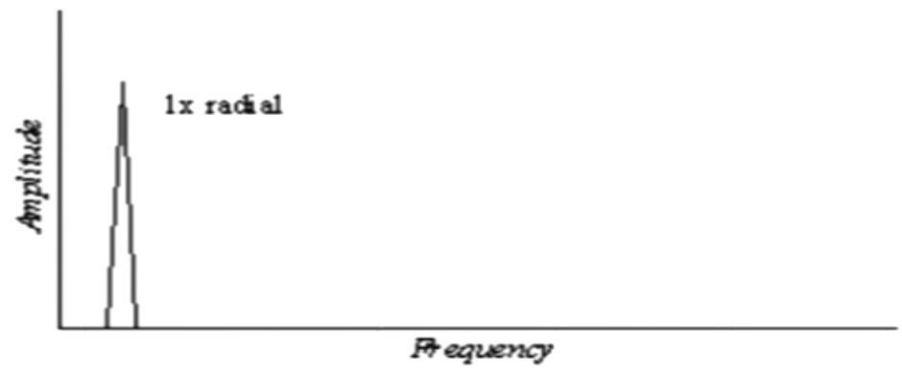
- Imbalance forces:
  - Put stress on bearings and seals
  - Excite resonances
  - Exacerbate looseness problems

- Critically important for high speed machines



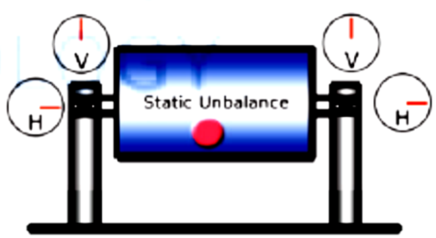
Static Unbalances



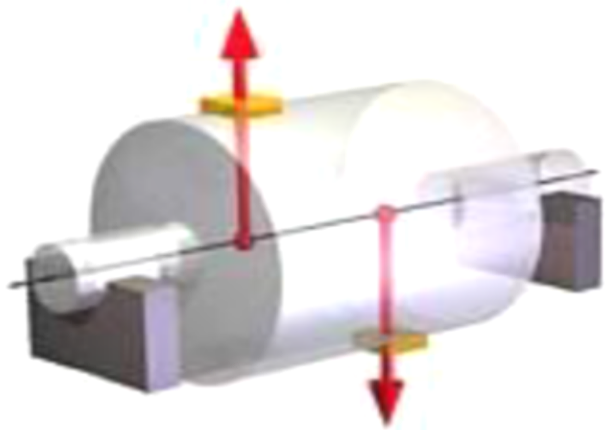


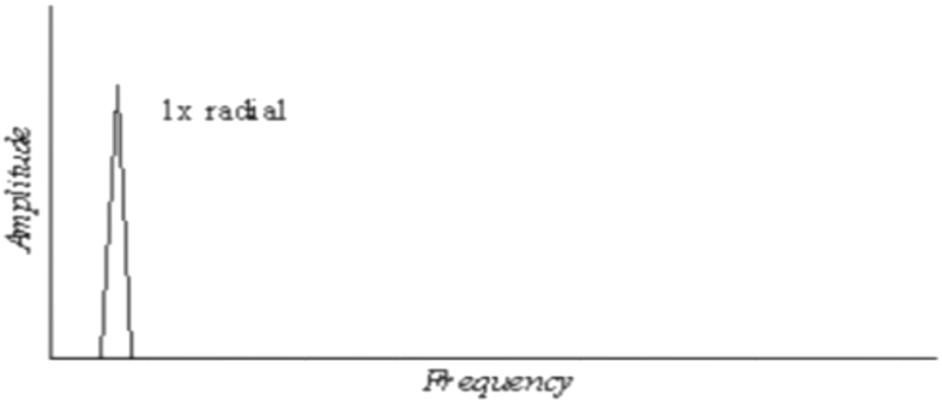
Unbalance - Static

- Amplitude due to unbalance will vary with the square of speed.
- The FFT will show 1 × rpm frequency of vibration.
- It will be predominant.
- Phase difference is as shown



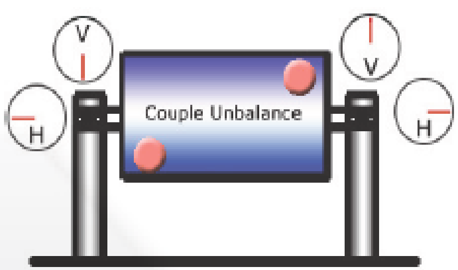
Couple Unbalances



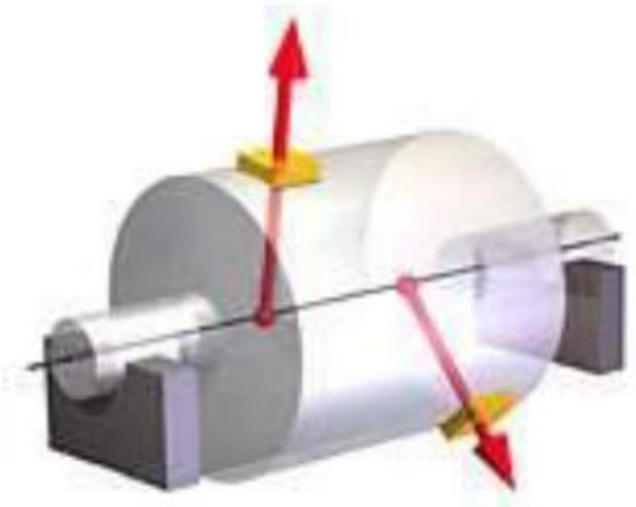


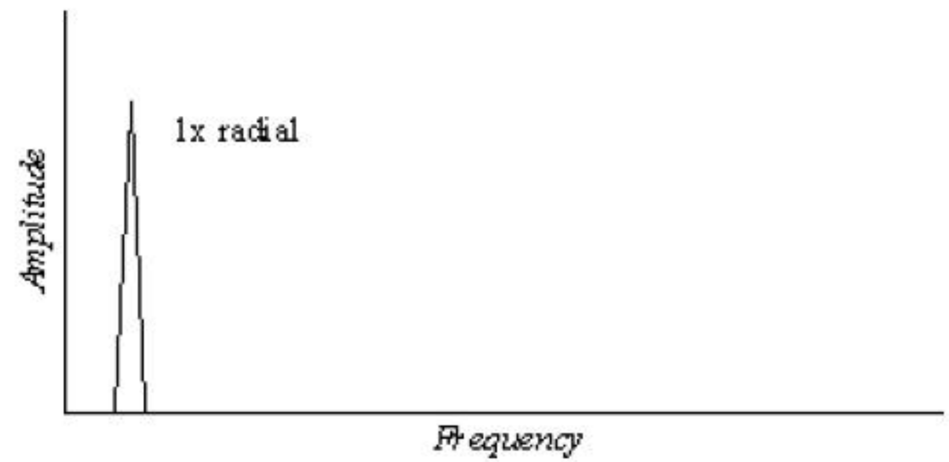
Unbalance - Couple

- Amplitude varies with square of speed.
- Predominant 1 × peak.
- May cause high axial along with radial vibrations.
- Phase difference is 180º on shaft ends in both planes.

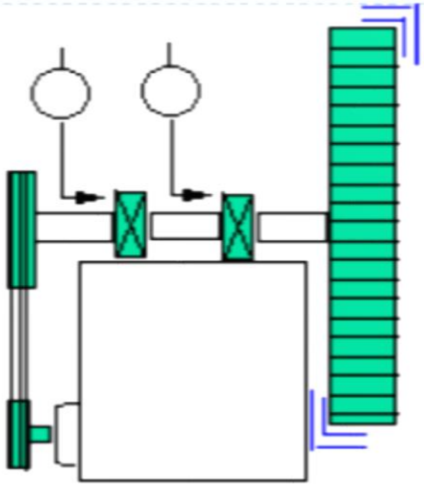


Dynamics Unbalances





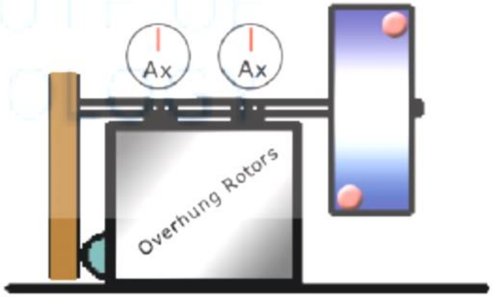
Overhung Rotor Unbalance



Special case of Dynamics Unba

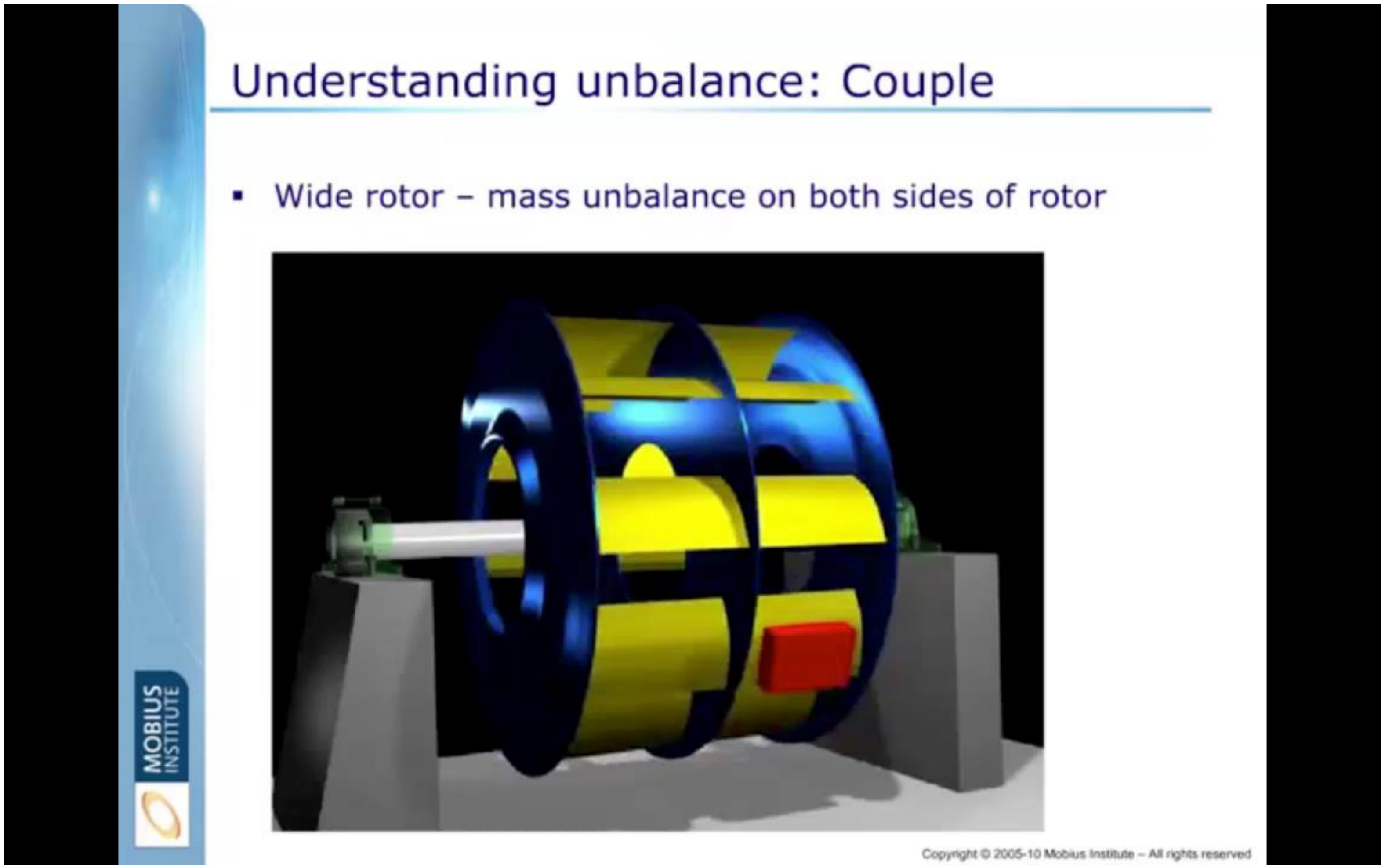
Unbalance - Overhung Rotors

- Amplitude varies with square of speed.
- Predominant 1× peak.
- May cause high axial along with high radial vibrations.
- Axial plane phase difference is 0º. Radial direction phase is unsteady.



Static Unbalance

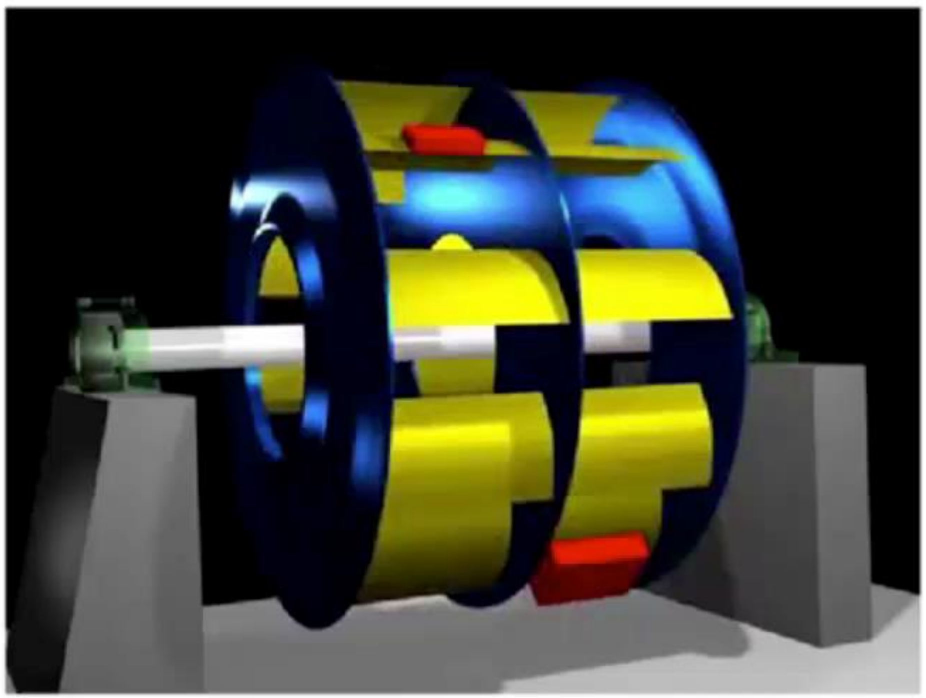
Couple Unbalance



Static and Couple Unbalance

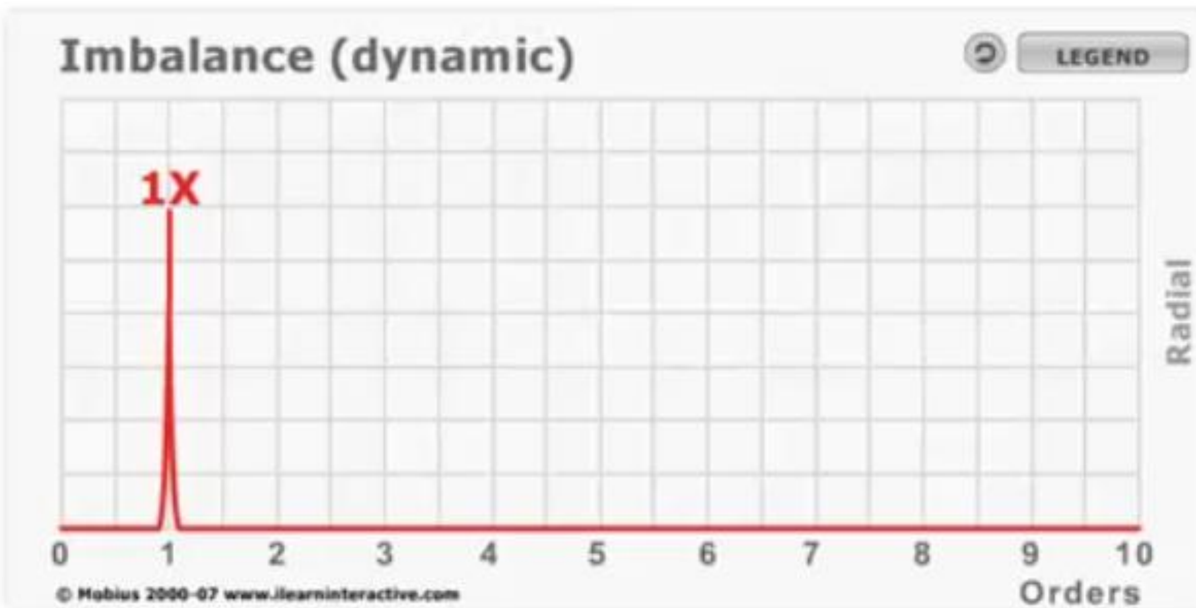
Understanding unbalance: Dynamic

- Static and couple unbalance



Static and Couple Unbalance

- "Dynamic" unbalance: most common
  - High 1X peak in spectrum
  - Often higher in horizontal due to increased flexibility
    - Some axial vibration at 1X

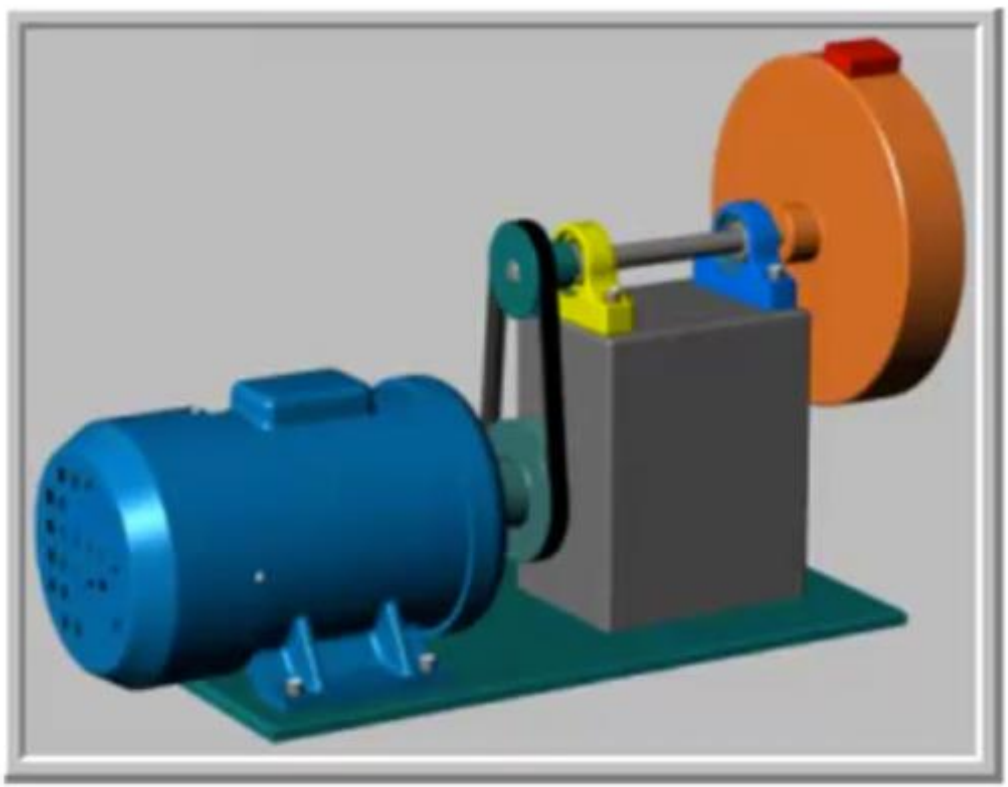


Vibration Analysis
Unbalance

- Look for a high 1X peak.
- Often higher in the horizontal axis, depending on foundations.
- Sinusoidal time waveform.
- Phase analysis is very useful.

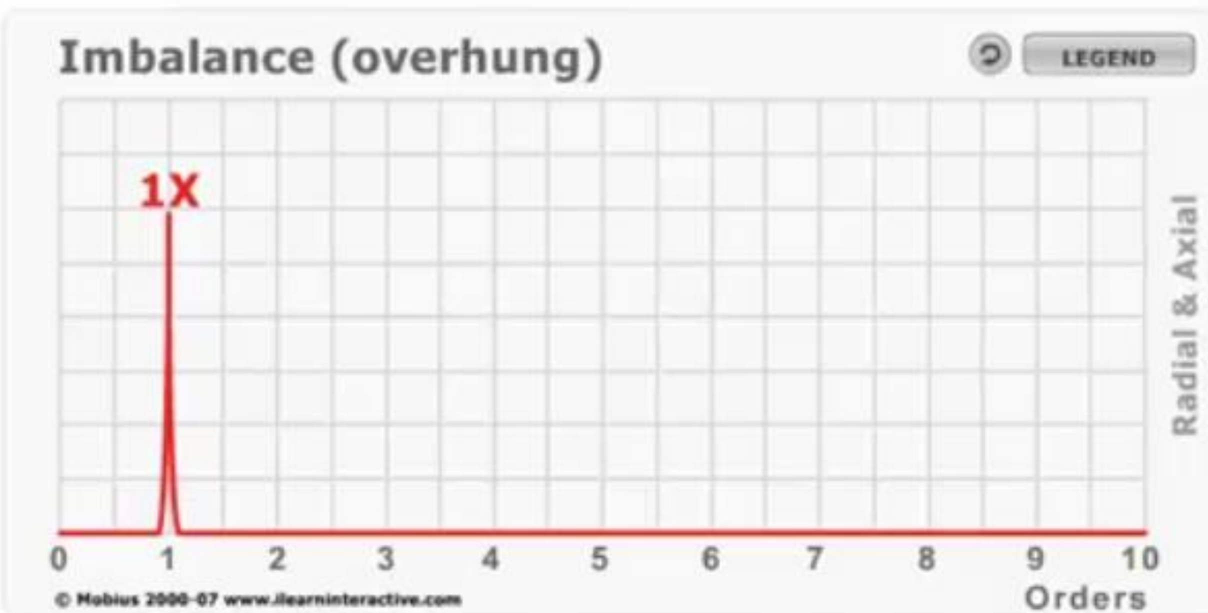
Vibration Analysis
Unbalance in overhung machines

- Forces are different in an overhung machine.
- Axial as well as radial forces are observed.



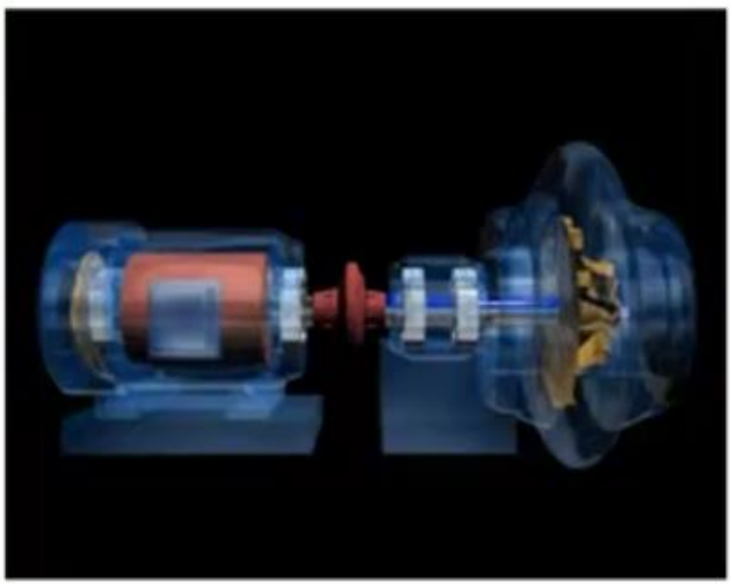
Vibration Analysis
Unbalance in overhung machines

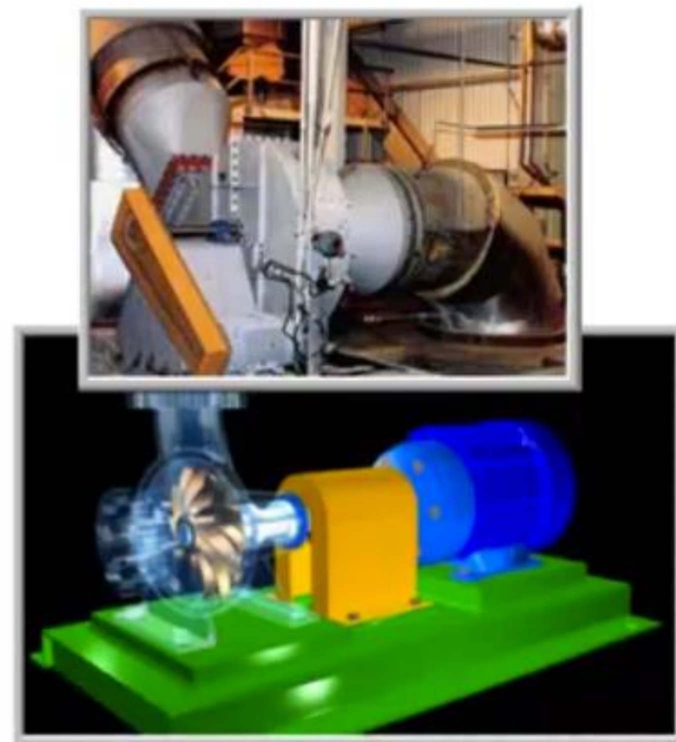
- High 1X vibration
  - Predominantly in the axial direction
  - Vertical and horizontal
- Collect data close to impeller vanes/fan blades



Examples of overhung machines

Examples of overhung rotors are close-coupled pumps, axial flow fans, and small turbines.



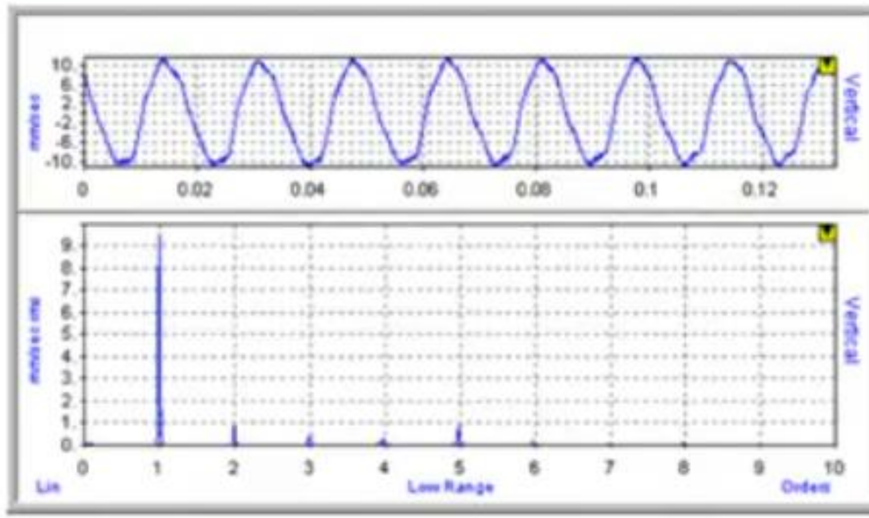


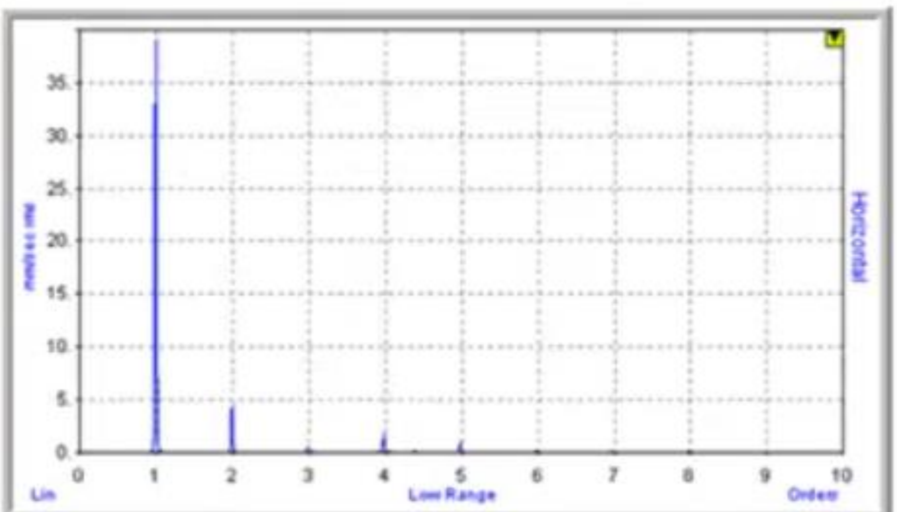
Case Study: Pump

Speed = 3600 CPM
150 HP motor,
Centrifugal-supported
245 GPM @ 385' head
Paraflex coupling

Case Study: Pump

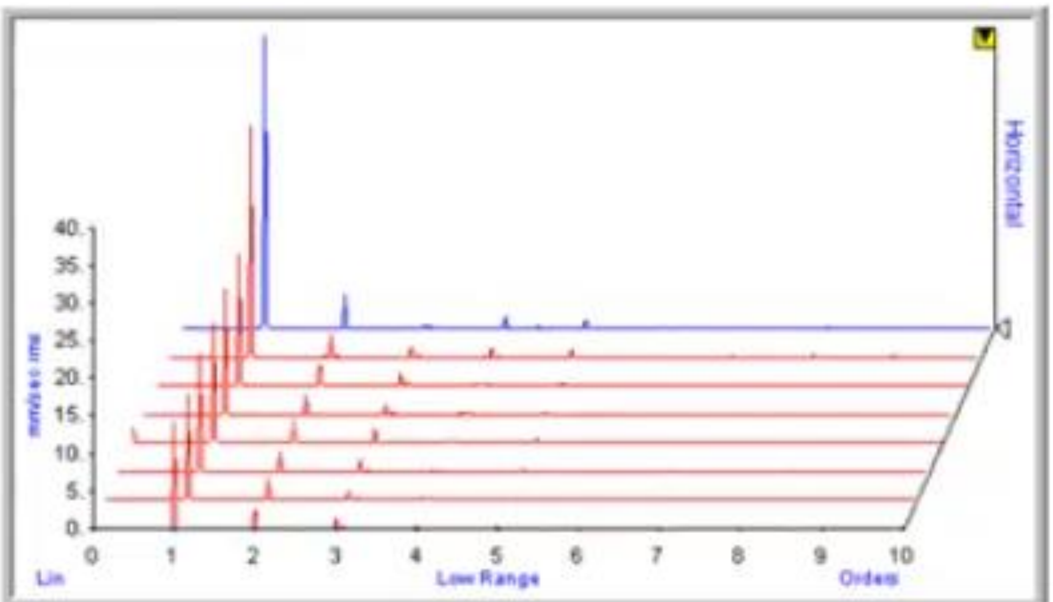
- High 1X
- Highest in horizontal

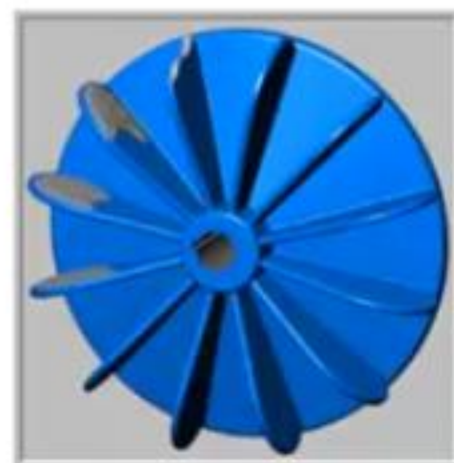




Case Study: Pump

- The vibration level at 1X has increased over time.
- That may indicate that there has been a build-up of ash on the impeller, or there has been uneven corrosion.





Before Balancing

Before attempting to balance:

Remember there are multiple reasons for a high 1X amplitude component.

Perform a complete analysis prior to balancing to ensure that other mechanical faults are not the cause of the 1X response.

Potential 1X causes

- Misalignment
- Thermal Effects
- Product buildup on rotor
- Erosion or corrosion of rotor
- Bowed, bent, or eccentric shaft
- Bearing or seal wear
- Roller Deflection – Paper Machines
- Machining errors/incorrect assembly
- Not properly balanced in shop
- Looseness in built-up rotor components

BALANCING PITFALLS

- Not mass unbalance
- Loose supports
- Frame misalignment
- Stiffness asymmetry
- Inaccurate data
- Thermal sensitivity
- Resonance
- Unbalance distribution

Always Remember

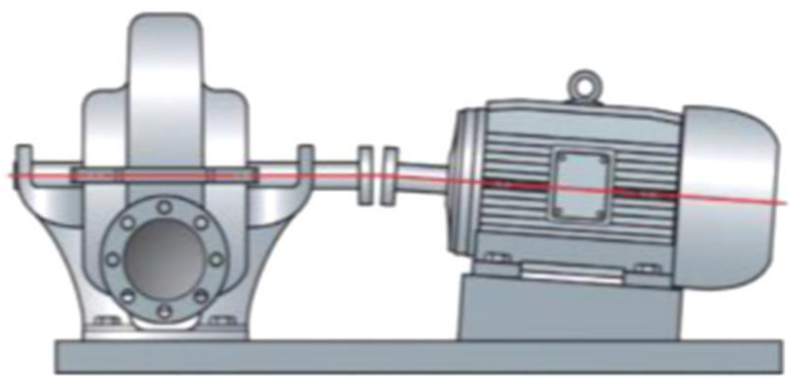
1. The rotating component did not go out of balance by itself.

2. Remain skeptical during the balancing process and use the balancing procedure as a diagnostic tool.

3. If balance attempt is not working as anticipated – STOP and THINK about what is going on. It may not be unbalance.

Misalignment

Misalignment is a condition where the centerlines of coupled shafts do not coincide.

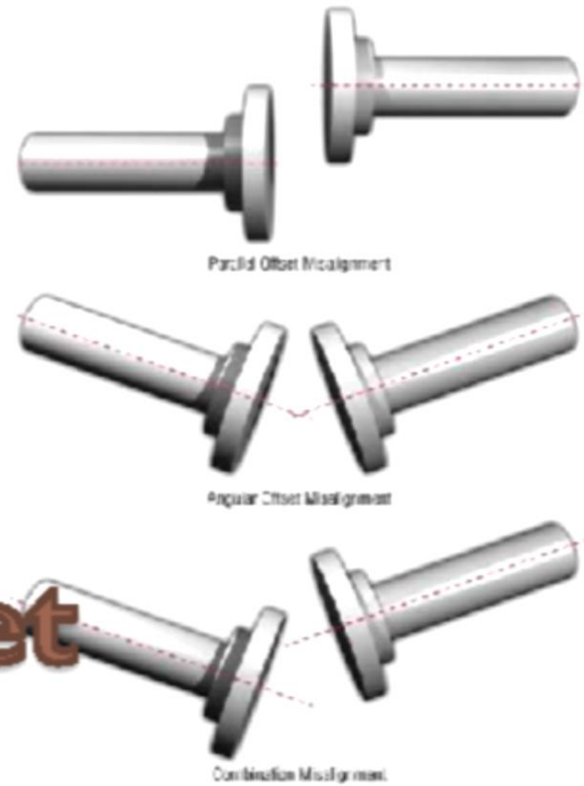


Type of Misalignment

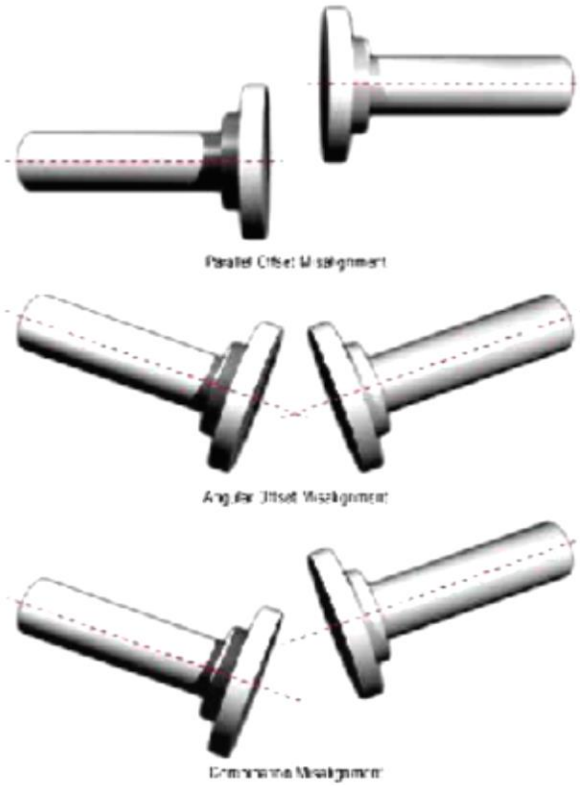
Parallel Offset

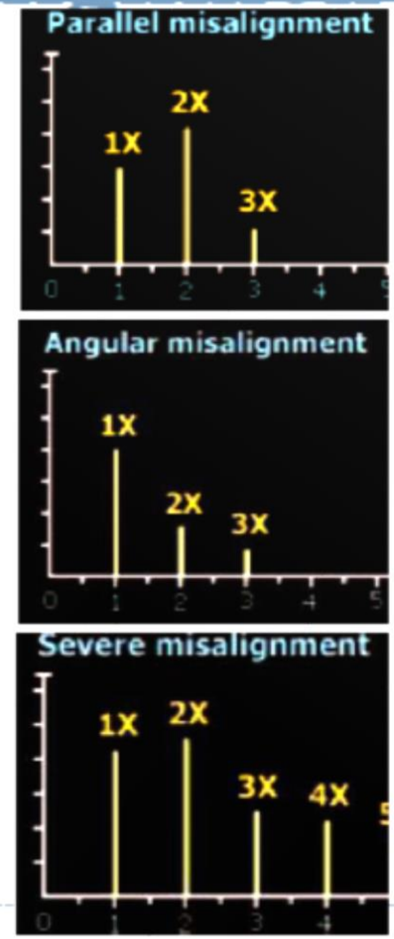
Angular Offset

Combination Offset



Type of Misalignment

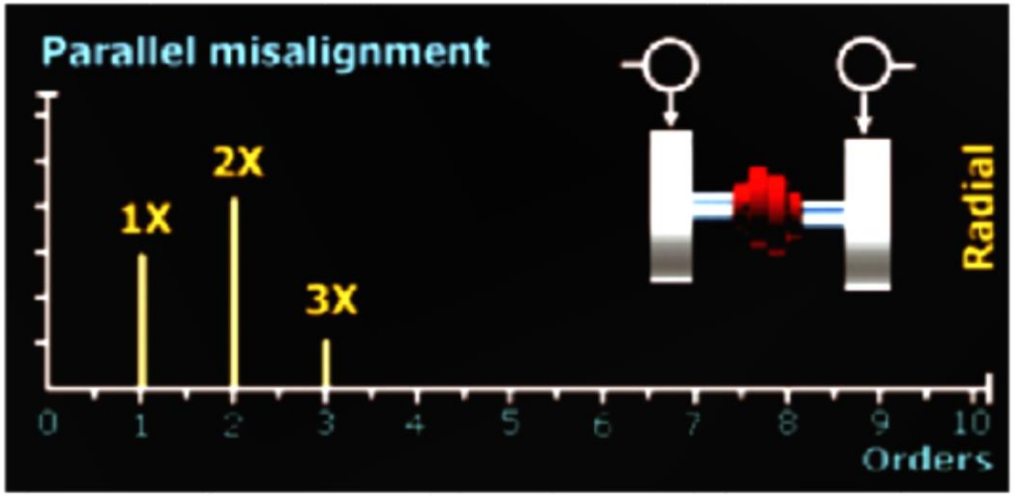




Type of Misalignment

Symptoms: 2X radial, smaller 1X

radial

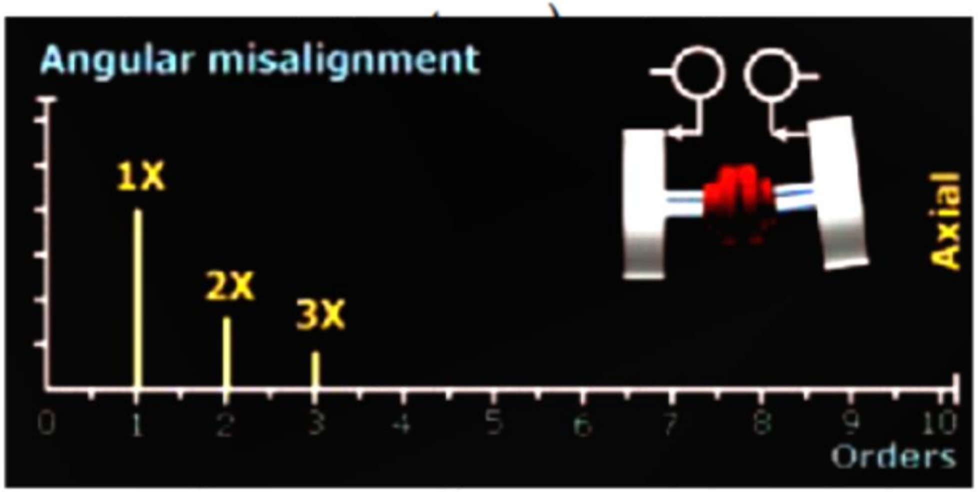


- Shear force
- Bending moment
- 2X component will be higher than 1X.
- Depend on the coupling type
- Axial 1X and 2X levels will be low

Type of

Misalignment

Symptoms: High *axial* vibration: 1X strong (but 2X and 3X can also be strong)

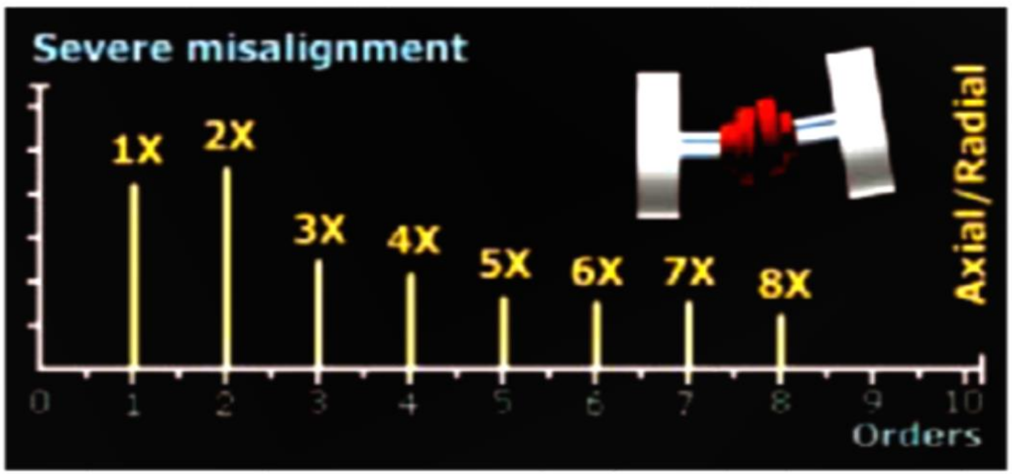


- Bending moment
- Also be fairly strong ***radial*** 1X and 2X levels

Type of

Misalignment

Frequency: IX and 2X (and 3X and 4X…) *axial and radial*



- Can be 3X, 4X all the way up to 8X peaks
- The noise floor is not raised(unlike rotating looseness)

Misalignment

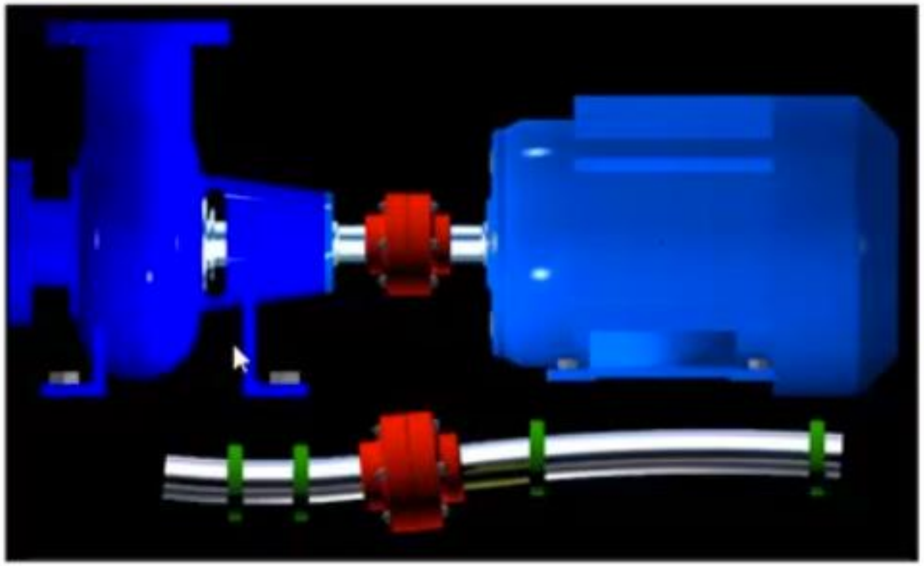
- Misalignment is responsible for a large proportion of machine failure – up to 50%.
- Machines should be precision aligned.
- Vibration analysis can be used to detect misalignment.

Misalignment

- If you increase the load on a bearing by 20% the life is halved! If you double the load, you reduce the life to one-seventh of its design life.
- Seals will be damaged by misalignment.
- Shafts and couplings can break.

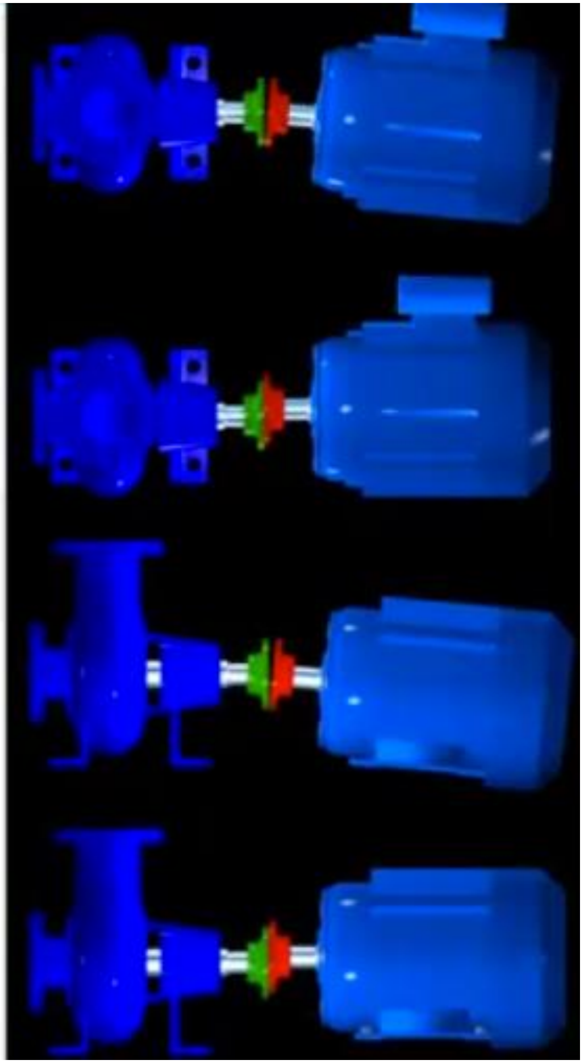
Misalignment

- Precision alignment = less stress on seals, bearings, shafts and couplings
- *"Shafts are misaligned when their rotational centerlines are not collinear when the machines are operating under normal conditions."*

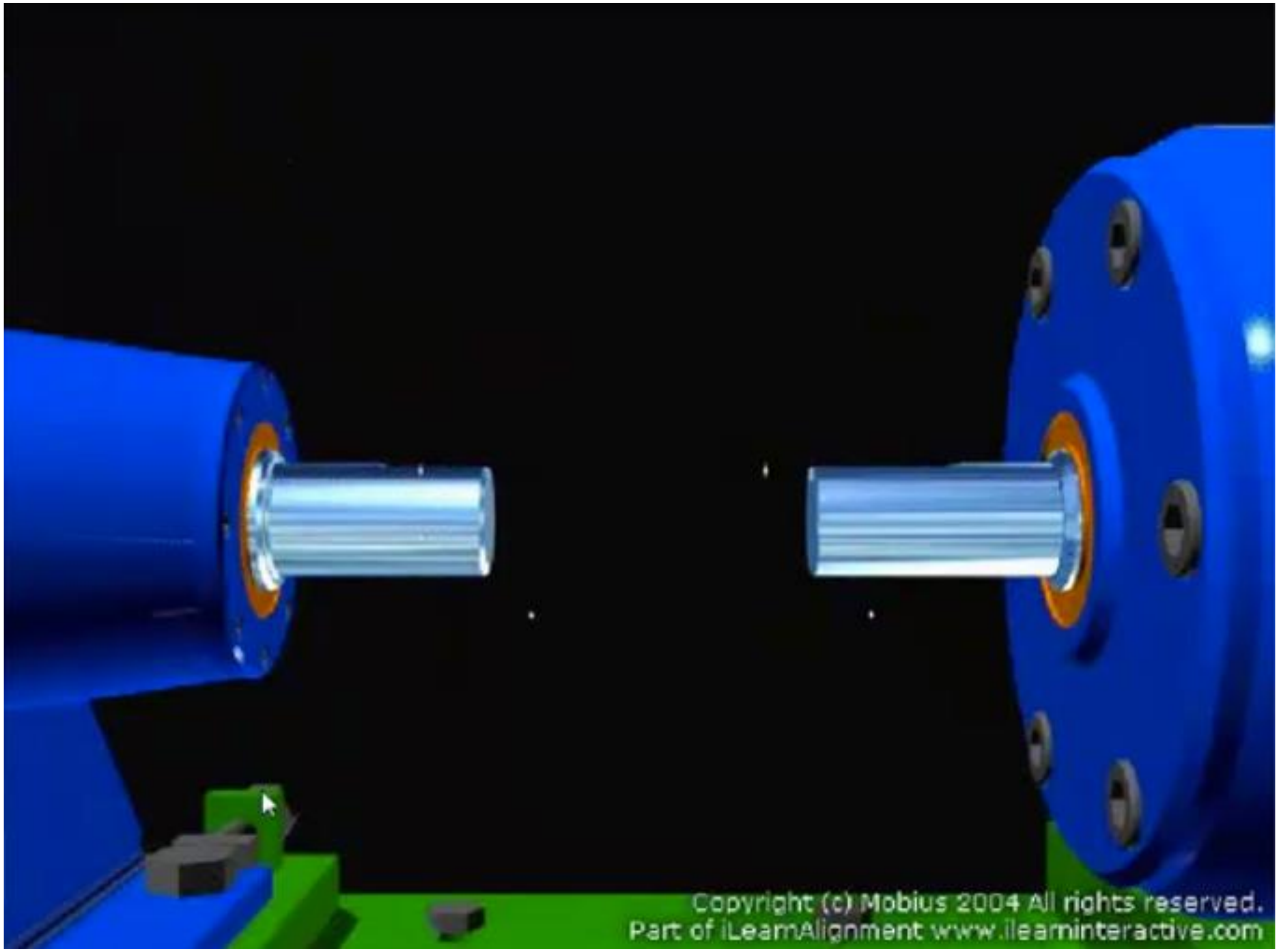


Misalignment

- Unless we take special precautions the shafts will not be collinear.
- There will be angular and offset misalignment.
- The misalignment will exist in the vertical and horizontal direction.

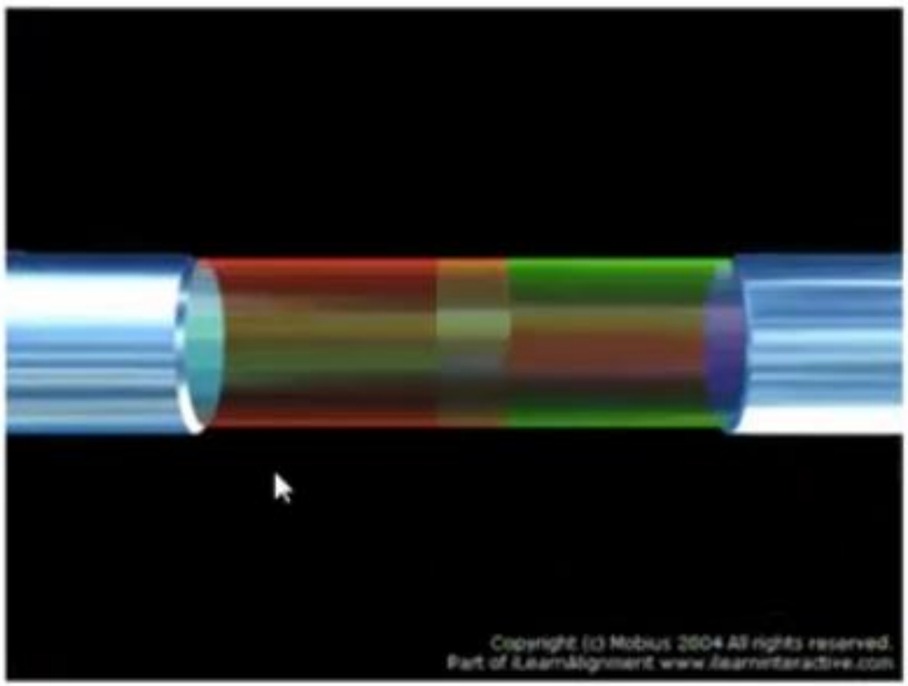


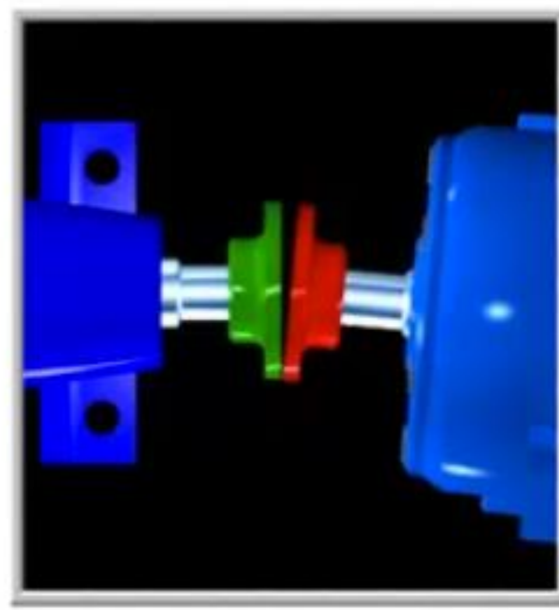
Misalignment



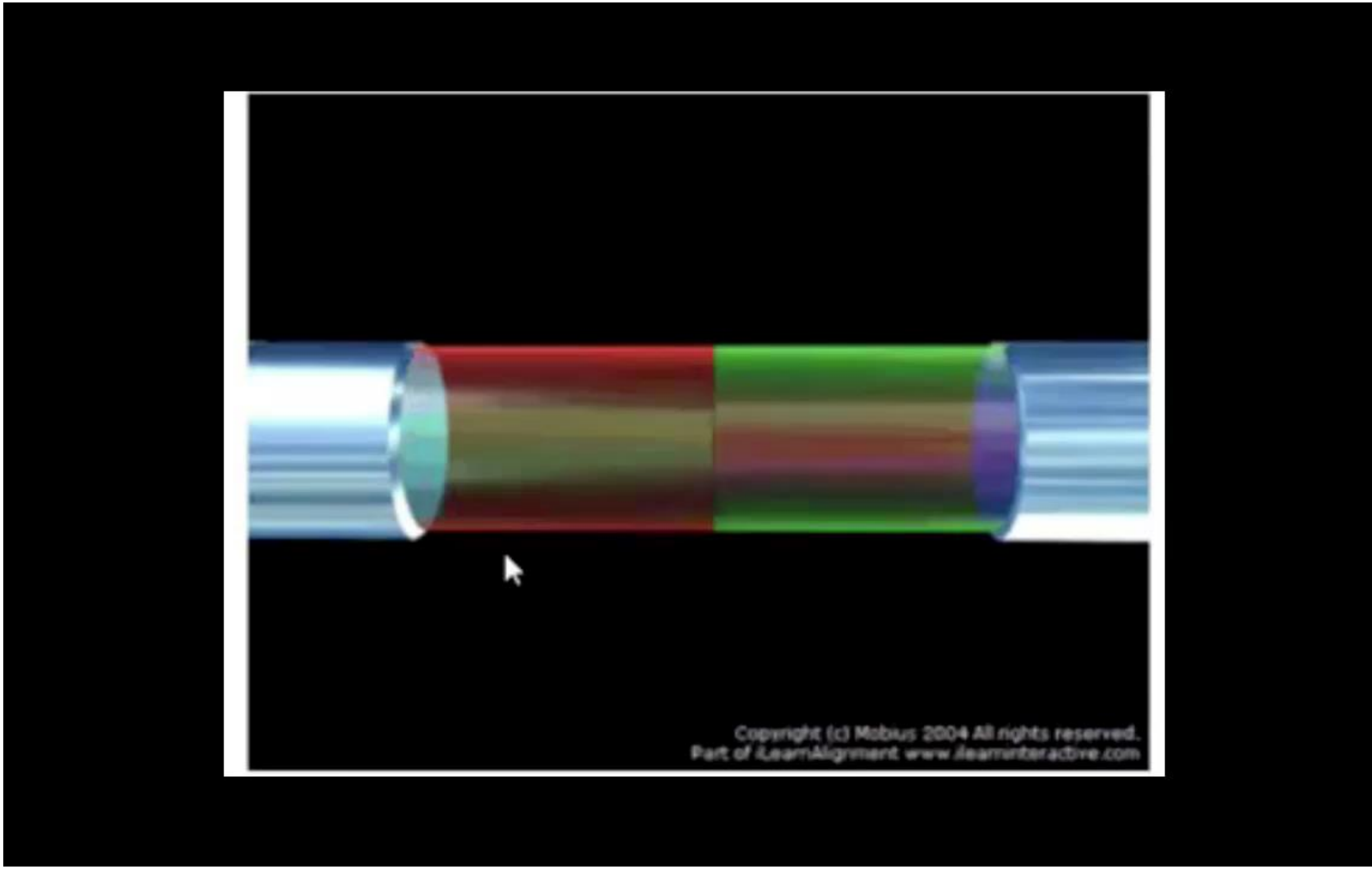
Misalignment

- If the misaligned shafts meet at a point but are not parallel, then the misalignment is called angular (or gap) misalignment.





Misalignment



Diagnosing Angular Misalignment

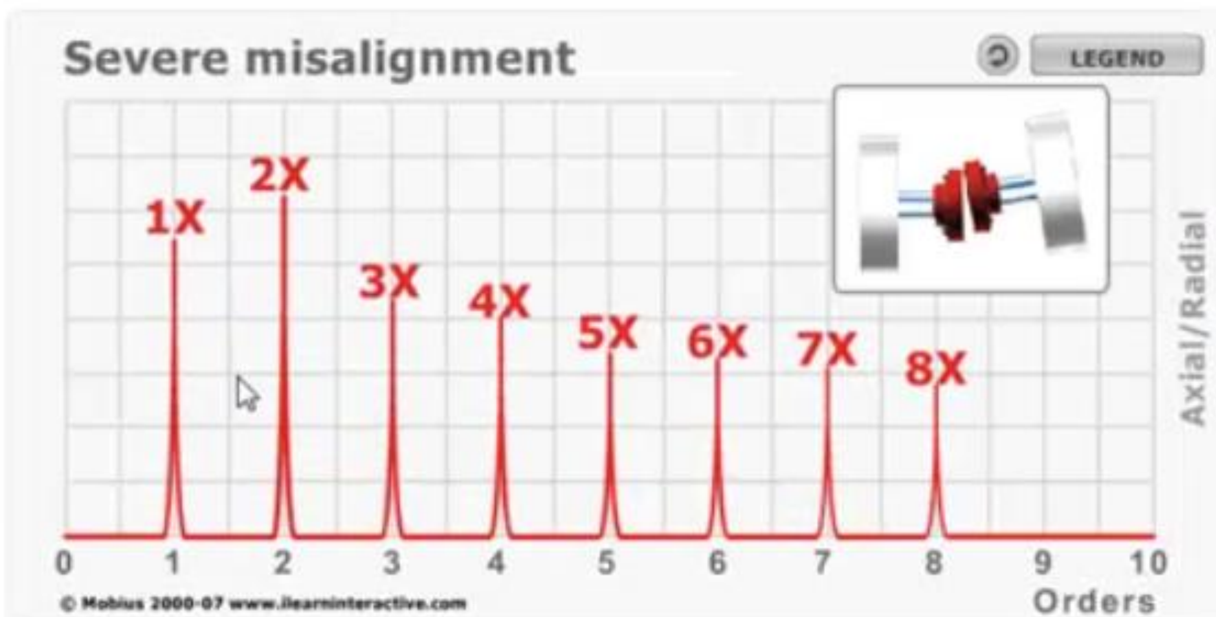
- High 1X axial (and some 2X in axial)
- 1X and 2X can be high in radial directions.
- 3X and 4X peaks are not unusual.

Diagnosing Offset Misalignment

- High 2X radial (and some 1X in radial)
- Axial 1X and 2X levels will be low for pure offset misalignment.

Severe Misalignment

- It is quite common to see high 3X and 4X peaks, and additional harmonics.
- It can be confused with looseness, however:
    - the harmonics will not be as strong, and
    - the noise floor will not be raised

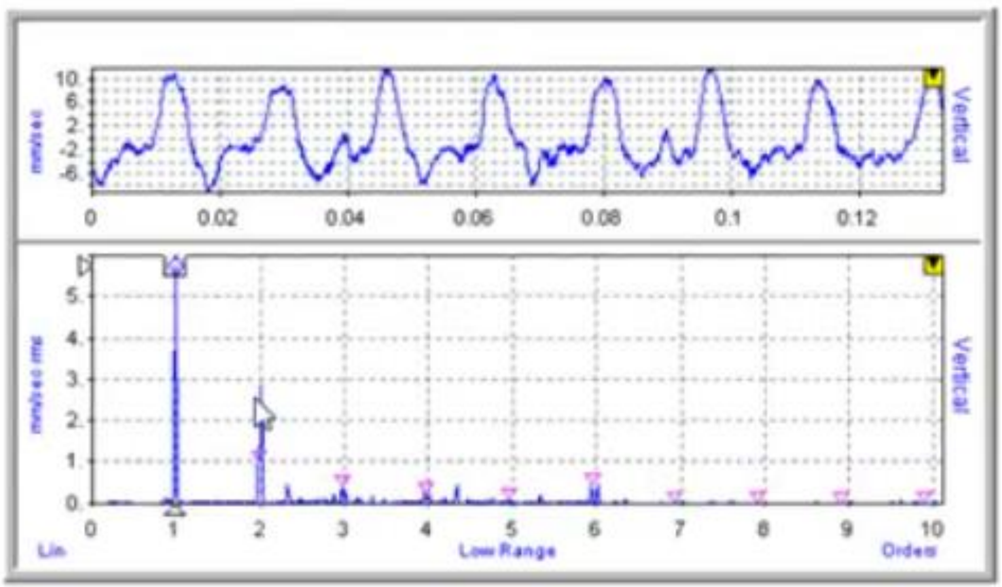


Confusing Misalignment with Unbalance

- Misalignment can be confused with imbalance.
- You can verify the diagnoses by:
  - Changing speed (if imbalance, amplitude will change)
  - Check phase readings carefully
  - Run the motor uncoupled
  - Perform a quick alignment measurement

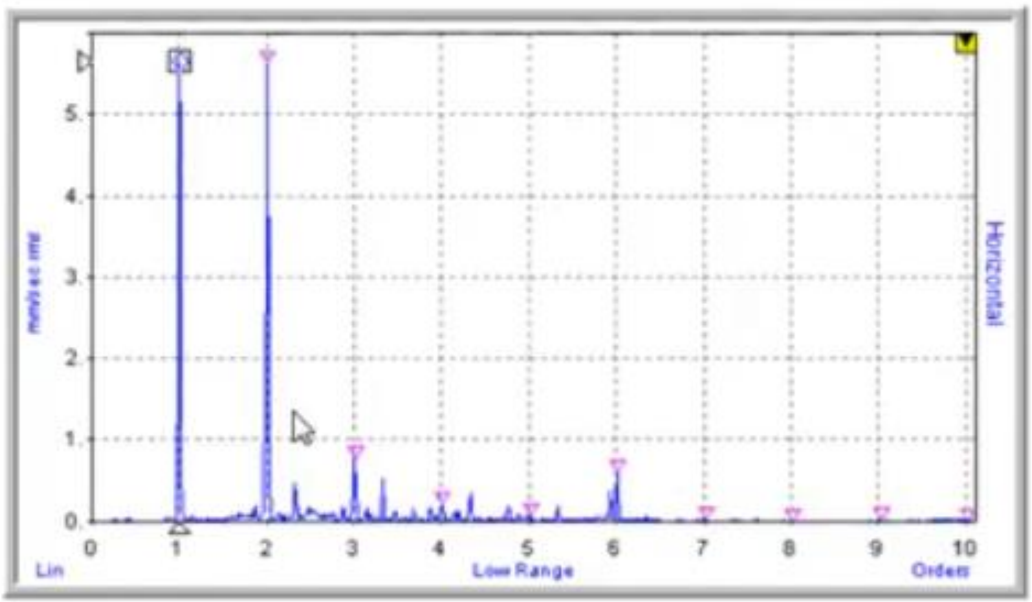
Misalignment

- High 1X and 2X
  - Data from pump bearing (2X is not 2xLF)



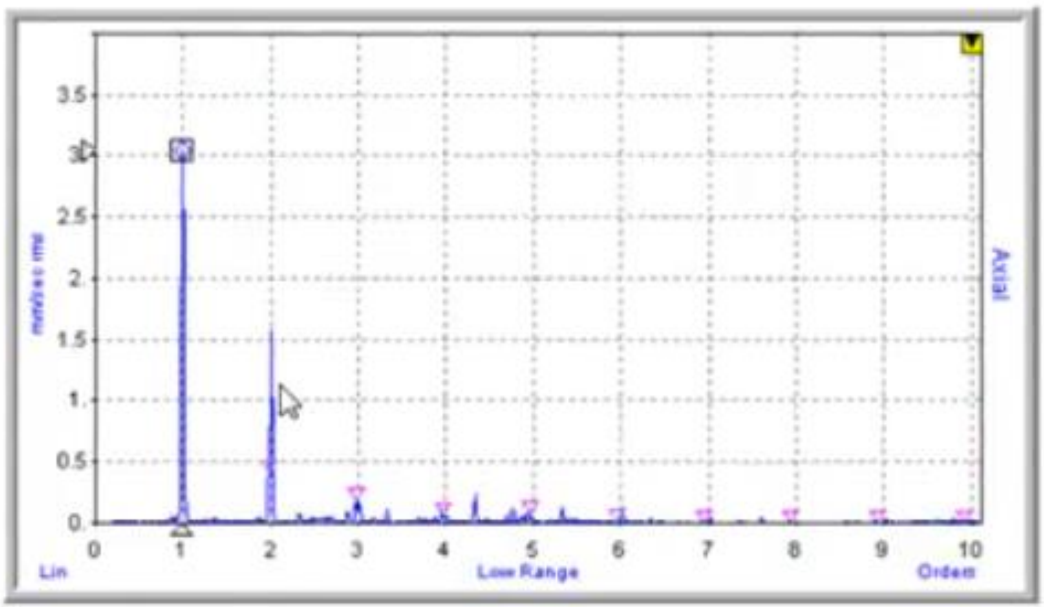
Misalignment

- Very high 2X in horizontal axis



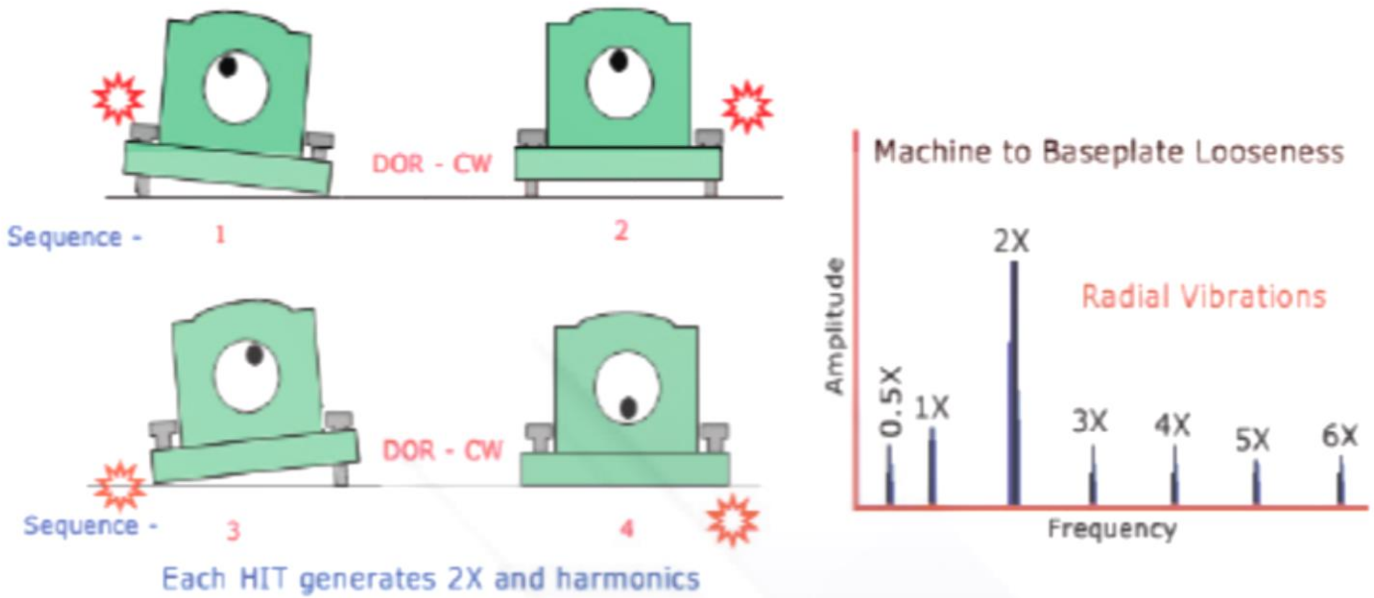
Misalignment

- 1X is high in the axial direction



Note that the graph scale is different - the amplitude of the 1X peak is actually lower than the levels observed in the vertical and horizontal axes.

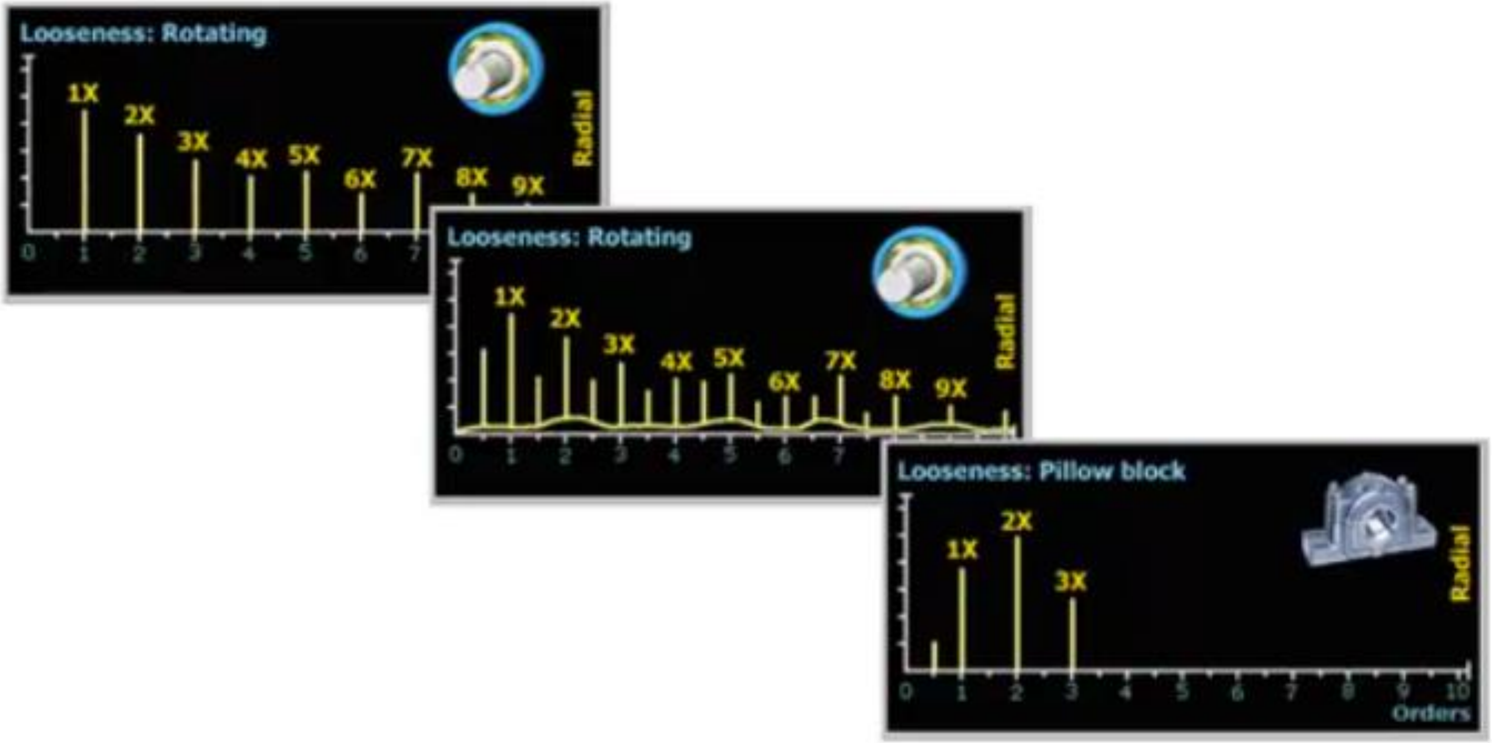
Looseness at Machine to Base Plate interface



Mechanical Looseness

Looseness

- We will consider rotating looseness, non-rotating looseness and foundation flexibility.



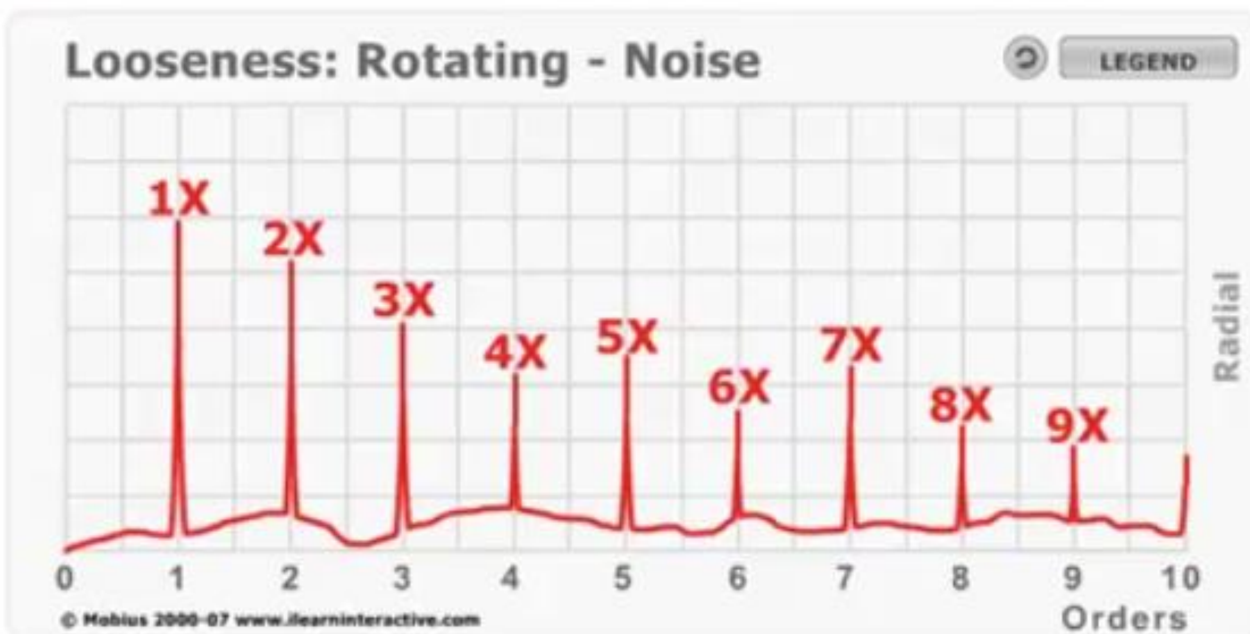
Rotating Looseness

- Rotating looseness can occur due to :
    - Improper fit, bearing loose on shaft, and excessive clearance.
    - Can also occur due to significant bearing wear.
        - Other symptoms of bearing fault will precede this state.

Rotating Looseness

Rotating Looseness

- Harmonics can extend beyond 10X.
- The noise floor can be raised.



Rolling Element Bearing

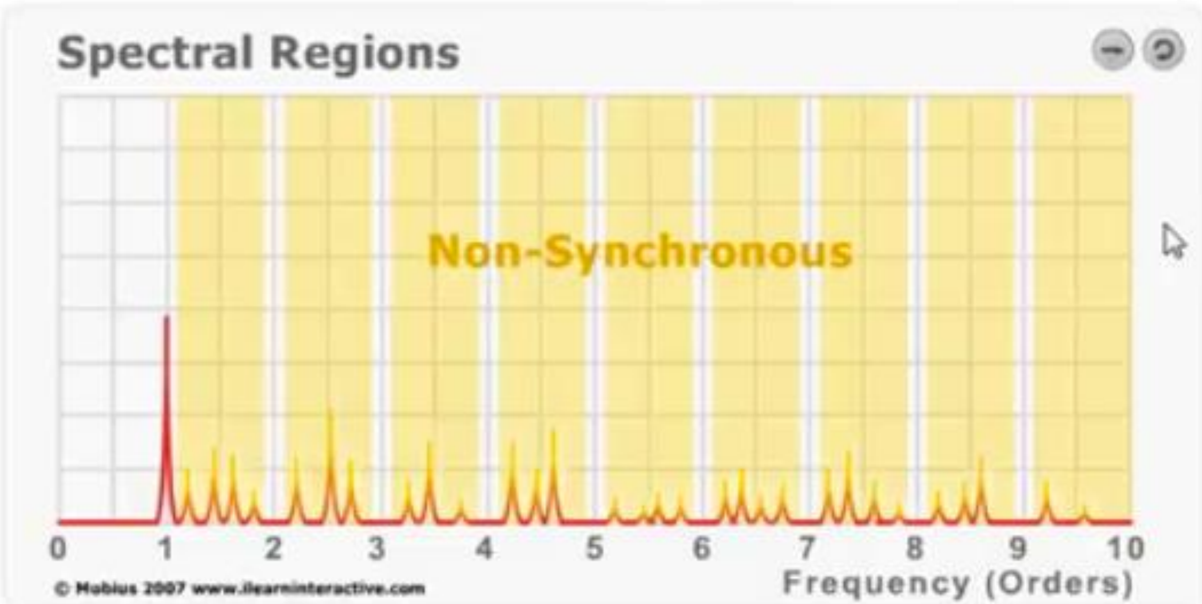
- Rolling element bearings are essential in industry.
- ***Less than 10%*** reach their design life time.





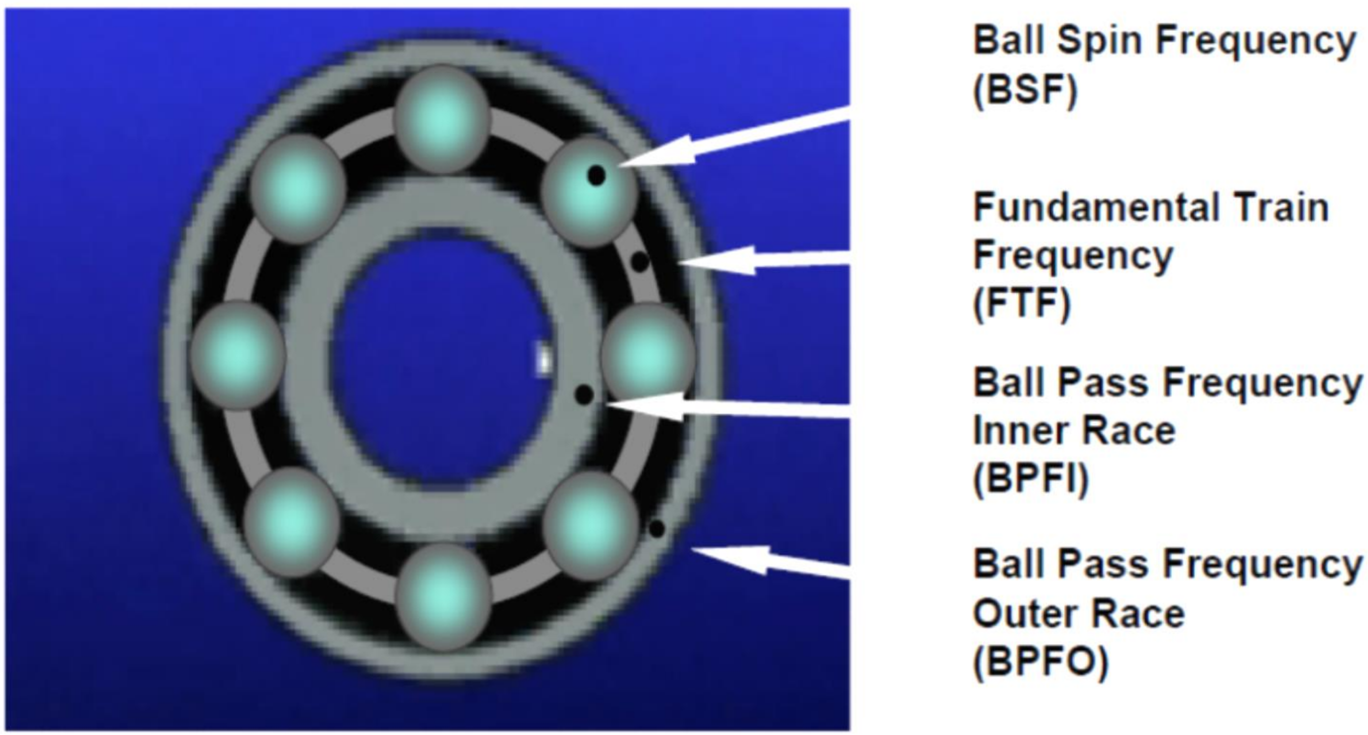
Rolling Element Bearing

- Soon we will learn that all of the frequencies generated by rolling element bearings are "non-synchronous".
- If you see a non-synchronous peak, and there are harmonics (and there are 1X or sub-synchronous sidebands) there is a good chance there is a bearing fault.

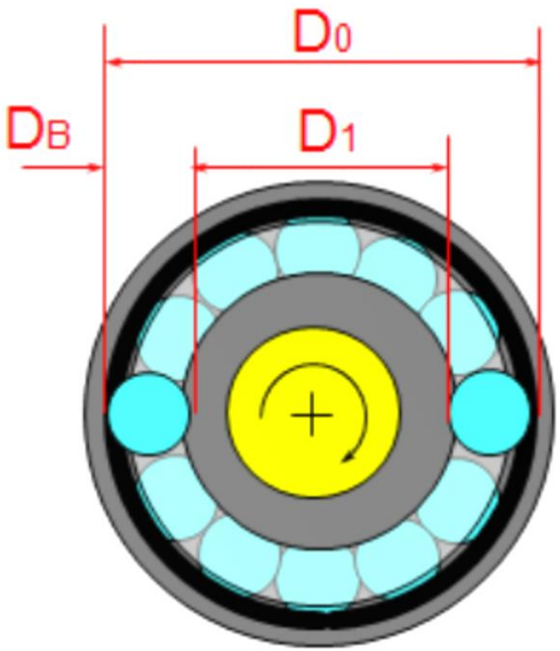


Roller Bearing Faults

Four different bearing frequencies



Bearing defect frequencies



$$BPFI = \frac{N_b}{2}\left(1 + \frac{B_d}{P_d}\cos\theta\right) X RPM$$

$$BPFO = \frac{N_b}{2}\left(1 - \frac{B_d}{P_d}\cos\theta\right) X RPM$$

$$BSF = \frac{P_d}{2B_d}\left(1 - \left(\frac{B_d}{P_d}\cos\theta\right)^2\right) X RPM$$

$$FTF = \frac{1}{2}\left(1 - \frac{B_d}{P_d}\cos\theta\right) X RPM$$

Note : shaft turning
outer race fixed

F = frequency in cpm
N = number of balls

Rolling Element Bearing

- Thanks to the geometry of the bearing, we can calculate the 'defect frequencies' - where we expect to see increased vibration levels.
- We can detect these frequencies in the time waveform and spectrum (velocity, acceleration and envelope).
- We can calculate the frequencies or find them in a database.
- Or we can *estimate* them.
  - Bearings generate 'non-synchronous' frequencies, harmonics and sidebands.

Defect Frequencies

- Four frequencies of interest:
  - Ball pass inner race (BPFI),
  - Ball pass outer race (BPFO),
  - Fundamental train (also called the cage rate) (FTF),
  - Ball spin (BSF).

Fundamental train (cage frequency)

Inner race rotating:

$$FTF = \frac{1}{2}[1-(B_D/P_D)Cos(\phi)]$$

$$FTF \approx \frac{1}{2}-(1.2/N_B)$$ Typically 0.33-0.48X

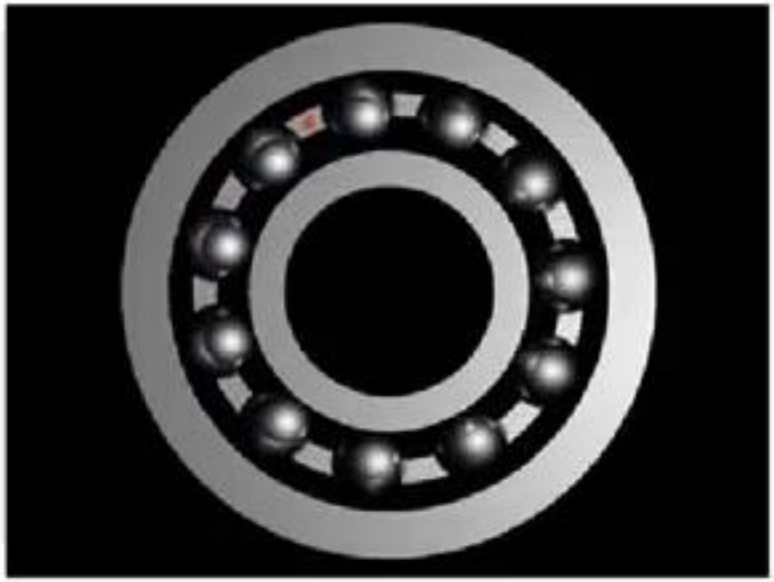
Outer race rotating:

$$FTF = \frac{1}{2}[1+(B_D/P_D)Cos(\phi)]$$

$$FTF \approx \frac{1}{2}+(1.2/N_B)$$ Typically 0.52-0.67X

Cage Frequency (FTF)

- For rolling element bearings:
  - Calculate the frequency of revolution of the cage
  - Fundamental Train Frequency: FTF
  - Also known as the Cage Rate



Cage Frequency

Ball Spin [BSF]

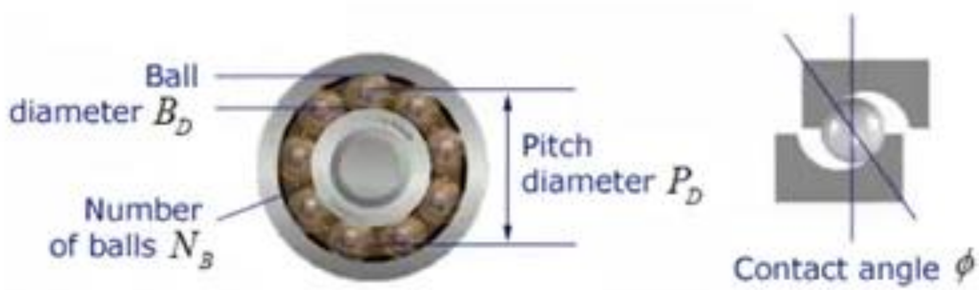
- For rolling element bearings:
  - Calculate the frequency of revolution of each ball/roller
  - Ball Spin Frequency: BSF

Ball Spin [BSF]

Ball/Roller spin frequency:

$$BSF = \frac{P_D}{2B_D}[1-(B_D / P_D)^2 Cos^2(\phi)]$$

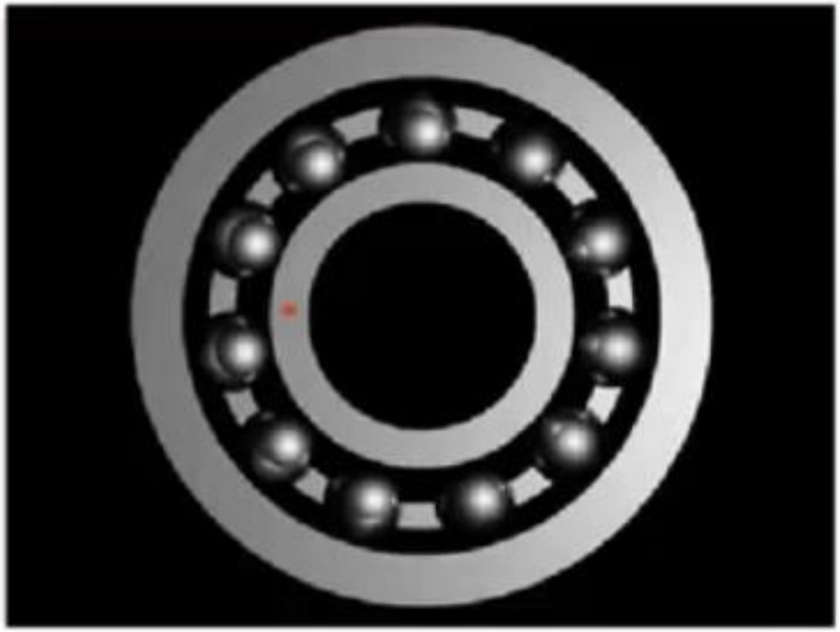
$$BSF \approx \frac{1}{2}[\frac{N_B}{2} - \frac{1.2}{N_B}]$$



Ball Spin [BSF]

Ball Pass Inner Race (BPFI)

- For rolling element bearings:
  - Calculate the rate at which the balls/rollers strike a point on the inner race
  - Ball Pass Frequency Inner: BPFI

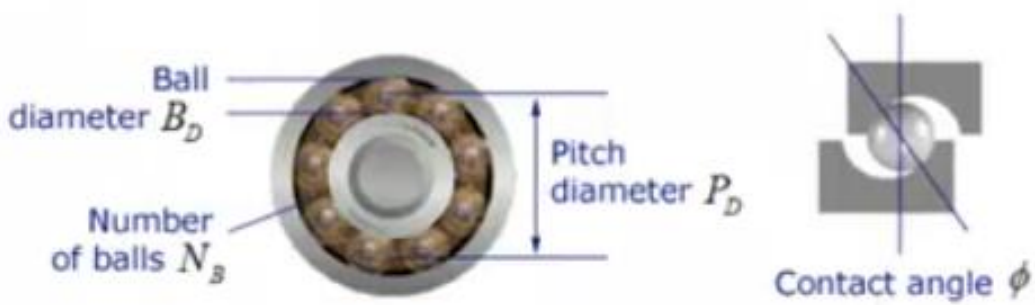


Ball Pass Inner Race (BPFI)

Inner race frequency:

$$BPFI = \frac{N_B}{2}[1 + (B_D / P_D)Cos(\phi)]$$

$$BPFI \approx \frac{N_B}{2} + 1.2$$



Ball Pass Inner Race (BPFI)

Ball Pass Outer Race (BPFO)

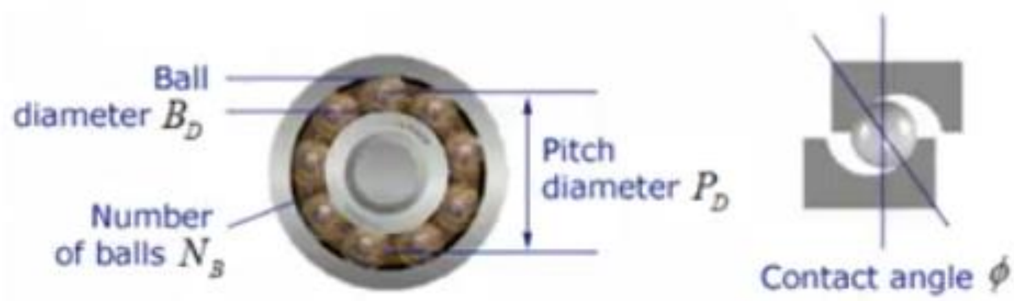
- For rolling element bearings:
  - Calculate the rate at which the balls/rollers strike a point on the outer race
  - Ball Pass Frequency Outer race: BPFO

Ball Pass Outer Race (BPFO)

Outer race frequency:

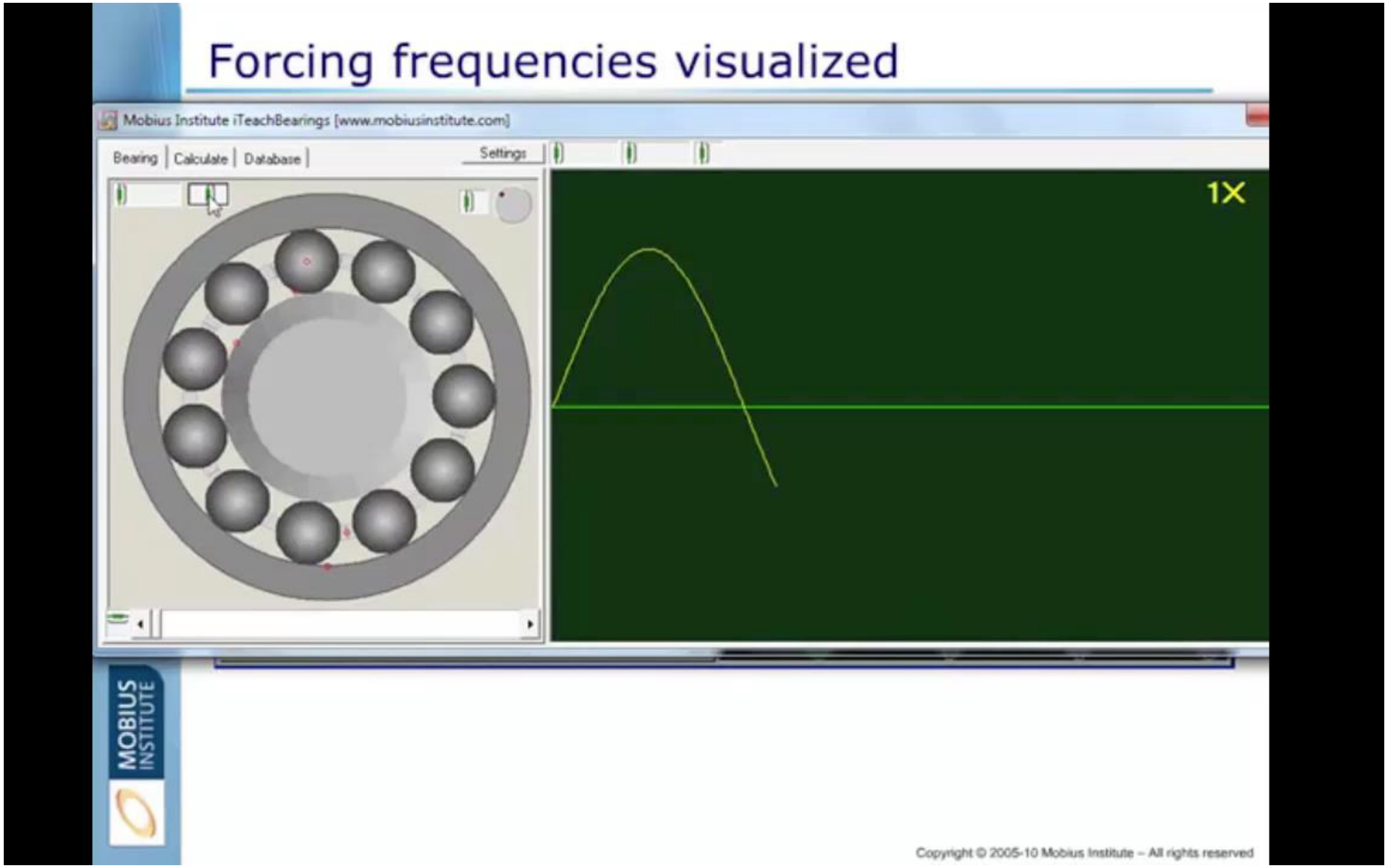
$$BPFO = \frac{N_B}{2}[1-(B_D/P_D)Cos(\phi)]$$

$$BPFO \approx \frac{N_B}{2} - 1.2$$

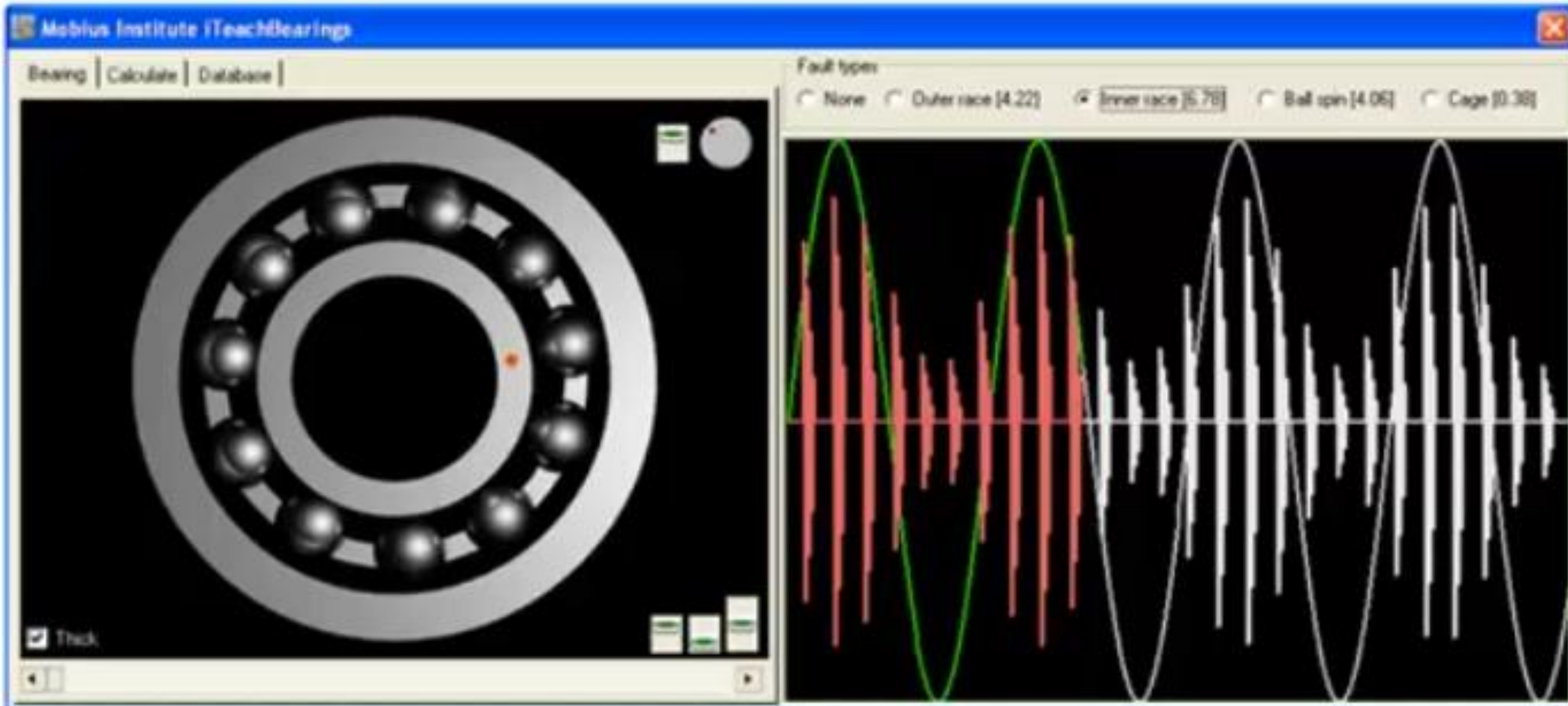


Ball Pass Outer Race (BPFO)

Forcing frequencies visualized



Forcing frequencies visualized



Vibration analysis of bearing

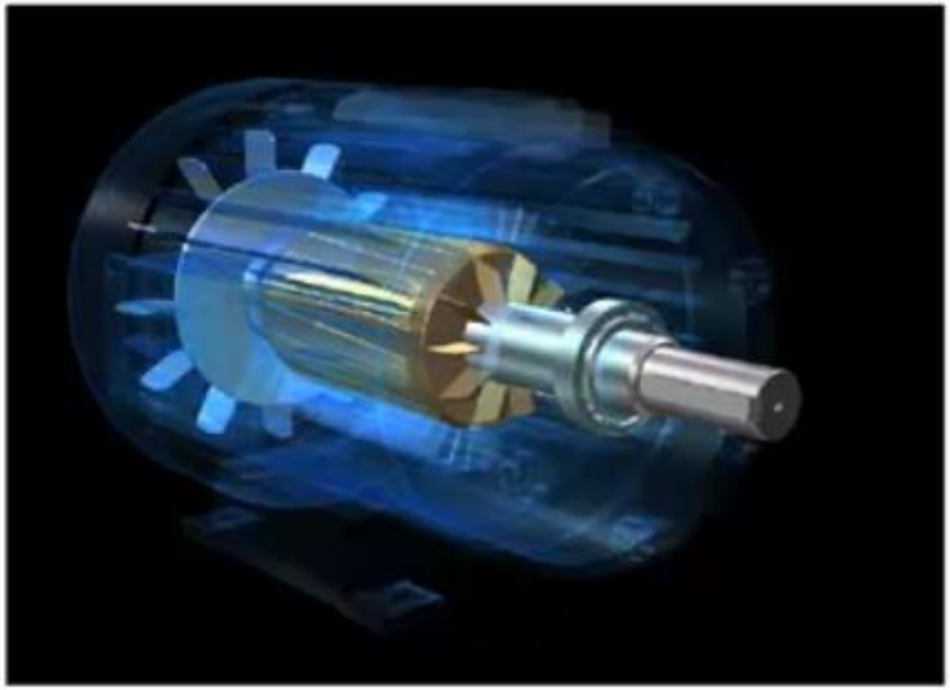
- We can use a variety of tools to detect rolling element bearings faults.
- You have to decide how early you wish to detect the fault, and what you will do when you detect the fault.
- In Category I we are not able to explain all of the detection methods, but it is important to be aware of them.
- We will summarize how the bearing fails, and explain how the vibration changes as the condition degrades.

Induction Motors

- Electric motors are the workhorse of industry
  - Faults can develop due to misalignment, imbalance, resonance, and foundation problems.
  - Motors with rolling element bearings therefore suffer from bearing failures.
  - Unique to electric motors there are a range of electrical and mechanical faults.

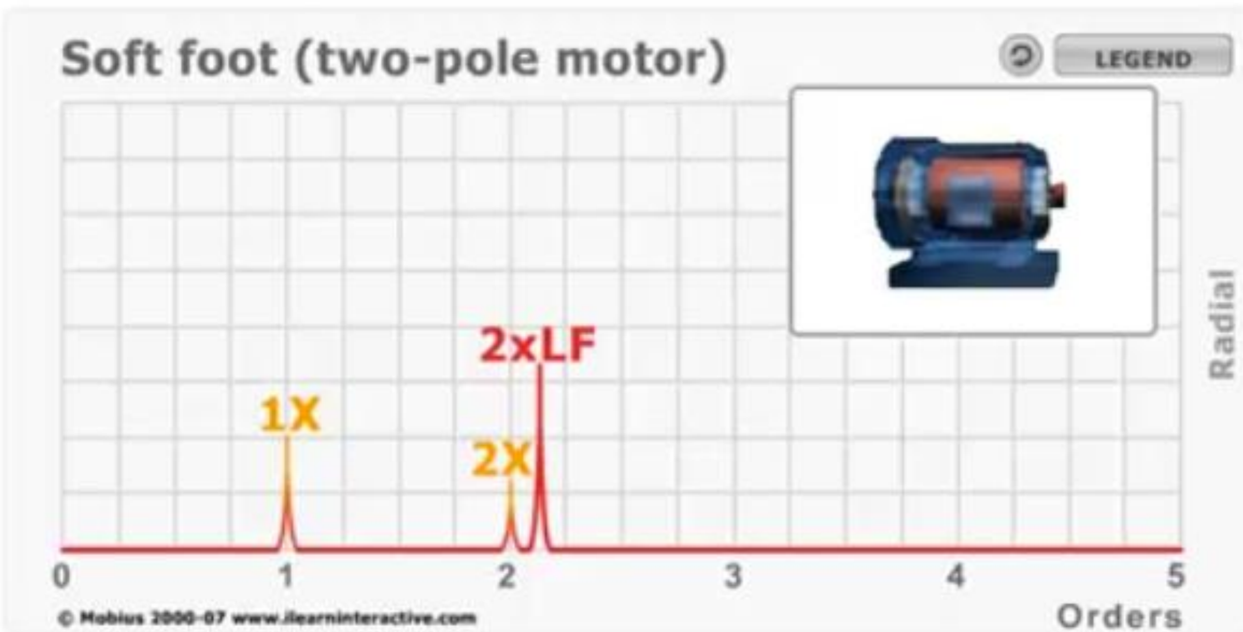
Vibration can detect a wide range of faults

- Stator eccentricity
  - Soft foot
- Rotor off-center
- Broken rotor bars
- Shorted end rings
- Bent shaft or bowed rotor

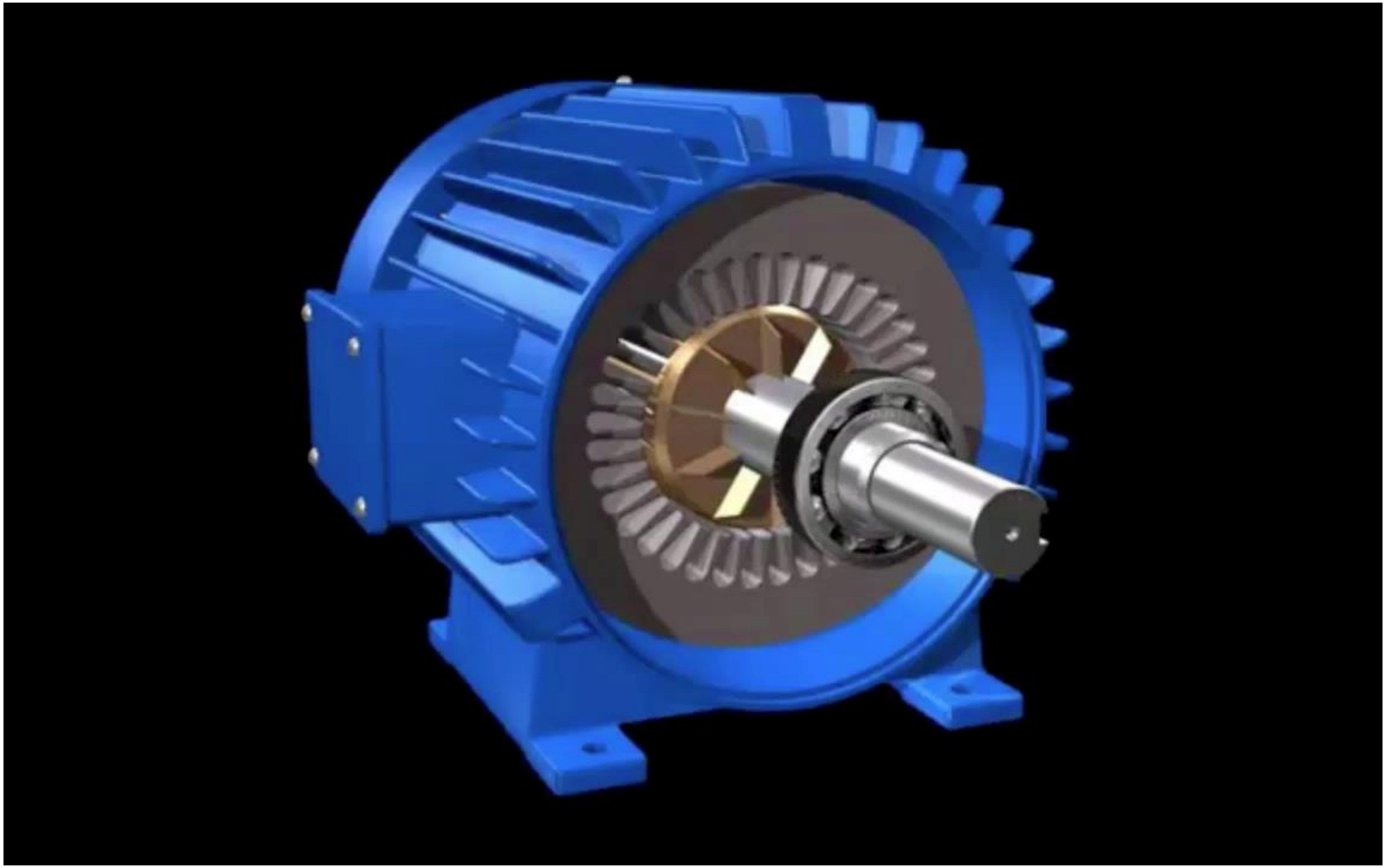


Eccentric stator: soft foot

- One of the most common faults. The stator is eccentric because it is distorted by soft foot forces.
- Peak at twice line frequency (100 or 120 Hz) will increase in amplitude.

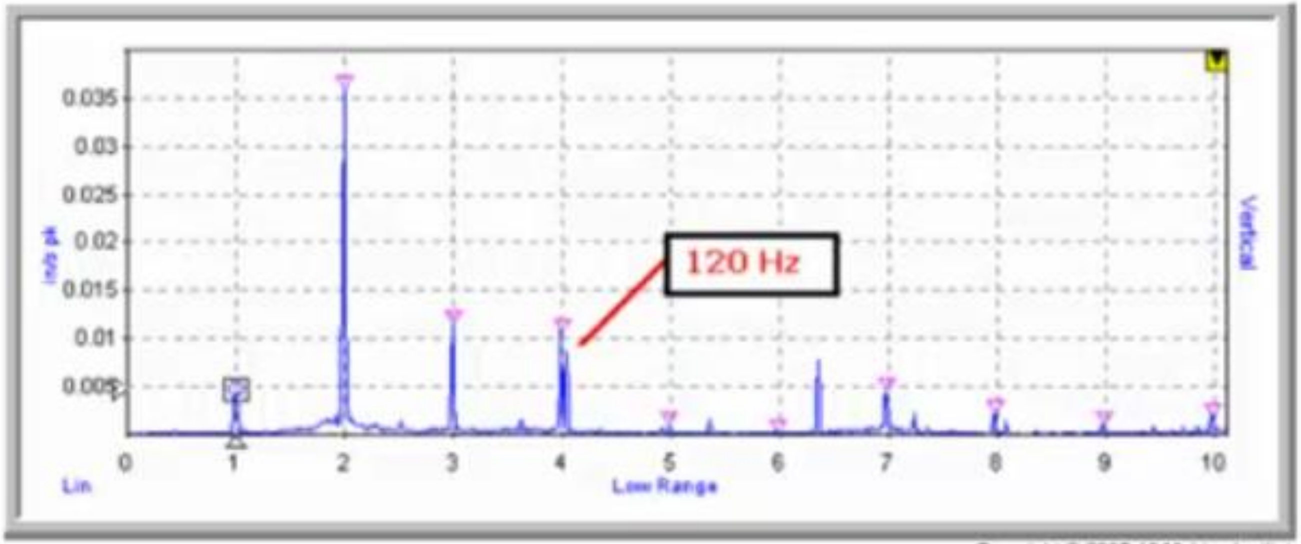


Eccentric stator: soft foot



Sources of vibration in induction motors

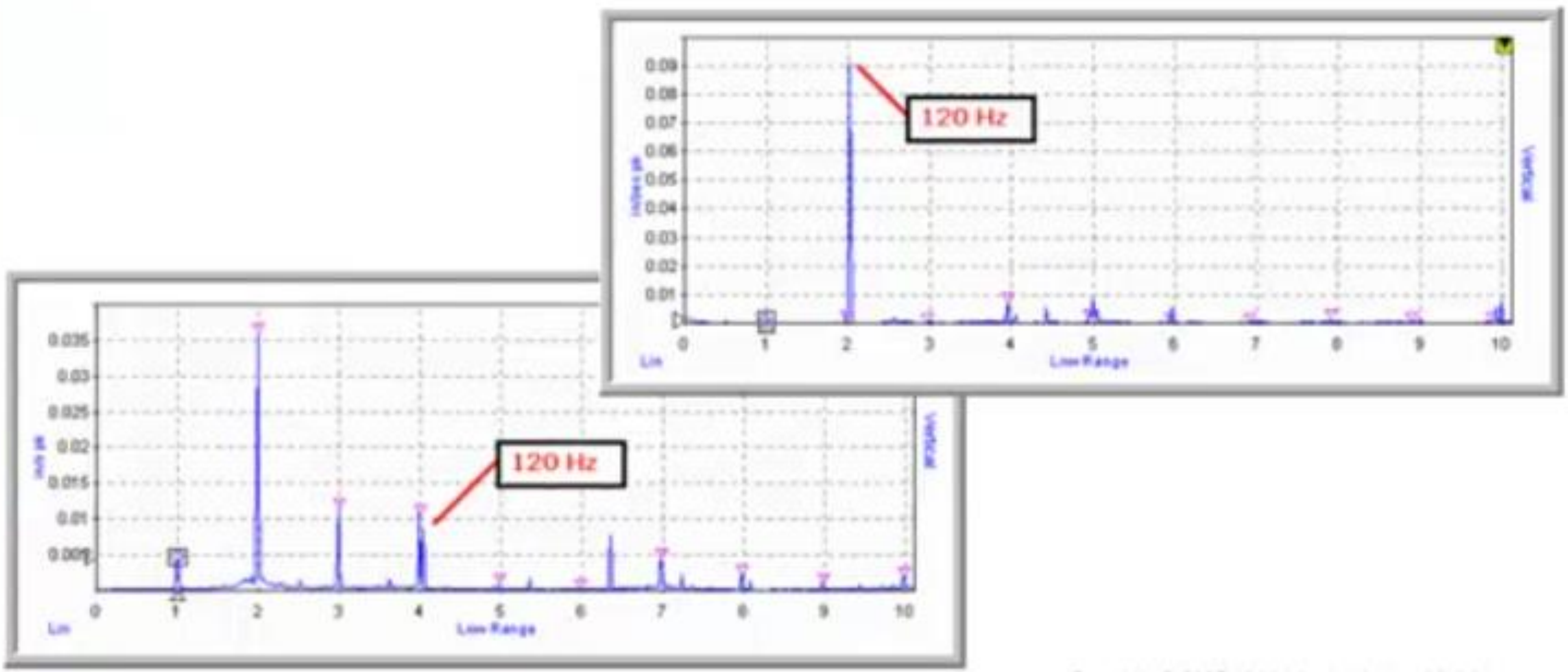
- A peak at twice line frequency is common
  - 120 Hz or 100 Hz
  - 7200 CPM or 6000 CPM
  - In electric motors, there is a magnetic attraction between the stator and rotor that varies at 100 or 120 Hz. This also causes the stator to vibrate at 100 or 120 Hz.



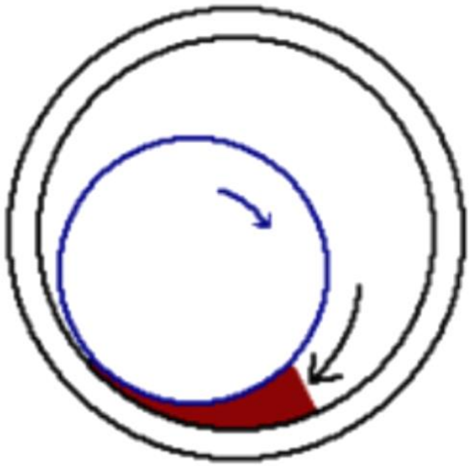
Twice line frequency

- Twice line frequency (120 Hz or 100 Hz) will be close to 2X or 4X
  - You need sufficient resolution

- Tip: cut power – 2xLF will disappear

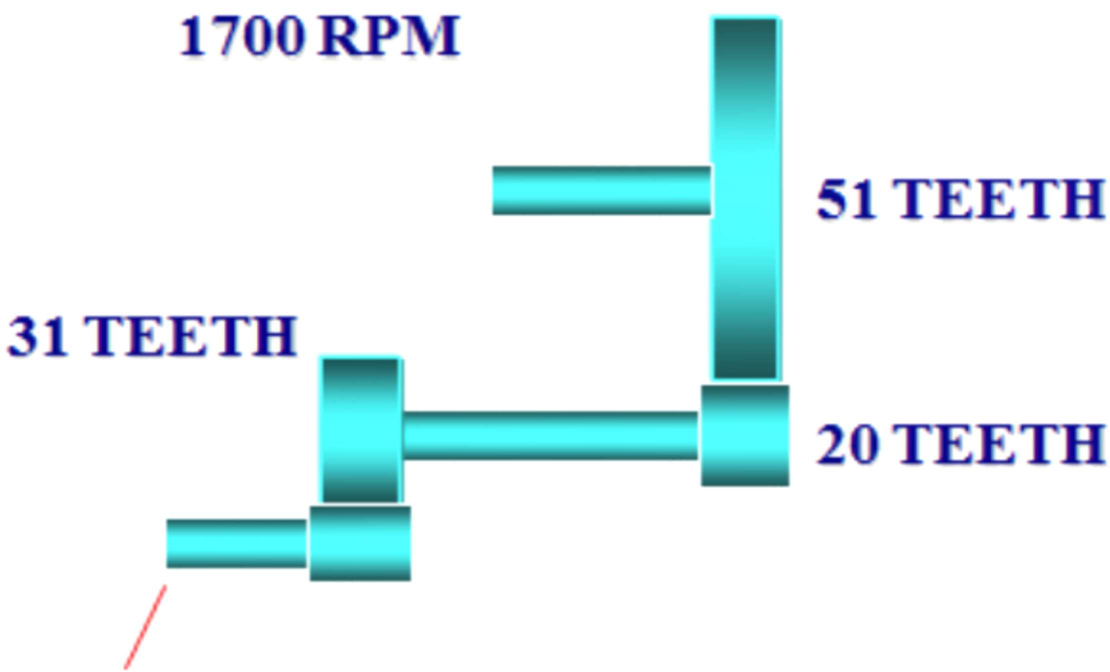


OIL WHIRL INSTABILITY



- Usually occurs at 42 - 48 % of running speed
- Vibration amplitudes are sometimes severe
- Whirl is inherently unstable, since it increases centrifugal forces therefore increasing whirl forces

CALCULATION OF GEAR MESH FREQUENCIES



8959 RPM -- HOW MANY TEETH ON THIS GEAR?

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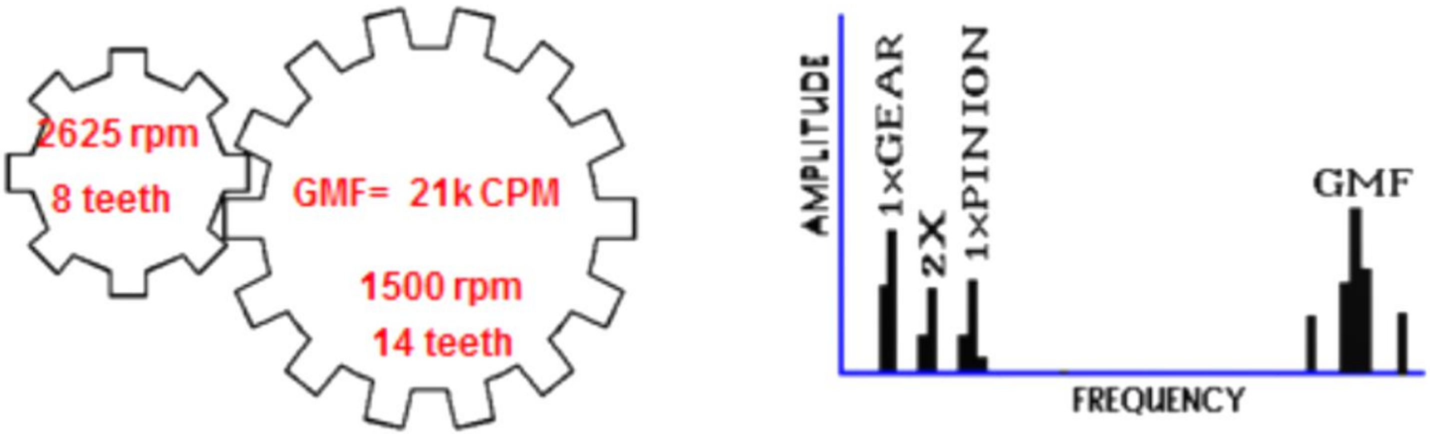
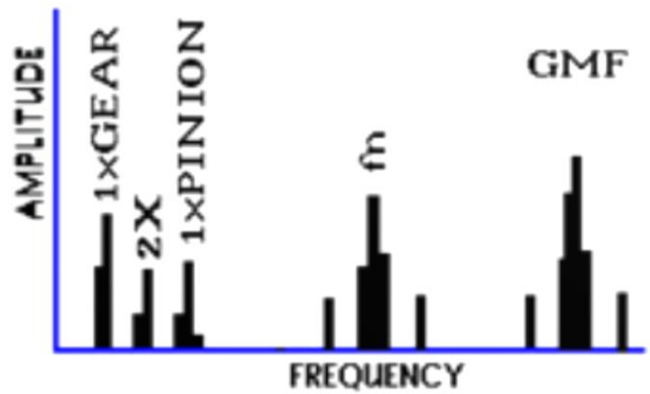
GEARS
NORMAL SPECTRUM



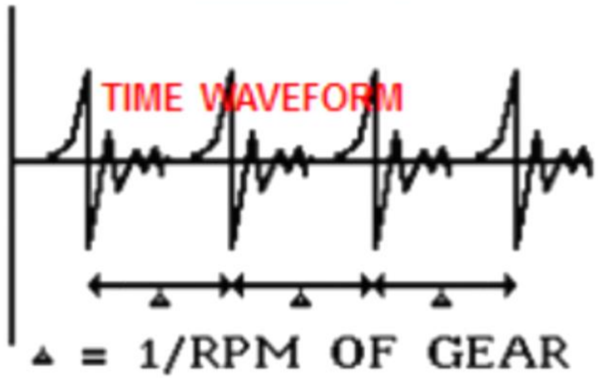
- Normal spectrum shows 1X and 2X and gear mesh frequency GMF
- GMF commonly will have sidebands of running speed
- All peaks are of low amplitude and no natural frequencies are present

GEARS
GEAR ECCENTRICITY AND BACKLASH



- Fairly high amplitude sidebands around GMF suggest eccentricity, backlash or non parallel shafts
- The problem gear will modulate the sidebands
- Incorrect backlash normally excites gear natural frequency

GEARS
CRACKED / BROKEN TOOTH



- A cracked or broken tooth will generate a high amplitude at 1X RPM of the gear
- It will excite the gear natural frequency which will be sidebanded by the running speed fundamental
- Best detected using the time waveform
- Time interval between impacts will be the reciprocal of the 1X RPM

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FREQUENCIES PRODUCED BY ELECTRICAL MOTORS.

- Electrical line frequency.(FL) = 50Hz = 3000 cpm.
  60HZ = 3600 cpm
- No of poles. (P)
- Rotor Bar Pass Frequency (Fb) = No of rotor bars x Rotor rpm.
- Synchronous speed (N_s) = $\frac{2xFL}{P}$
- Slip frequency (F_S)= Synchronous speed - Rotor rpm.
- Pole pass frequency (F_P)= Slip Frequency x No of Poles.

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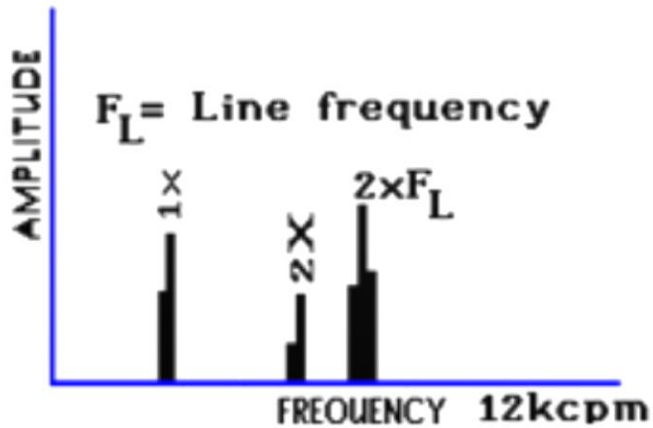
ELECTRICAL PROBLEMS

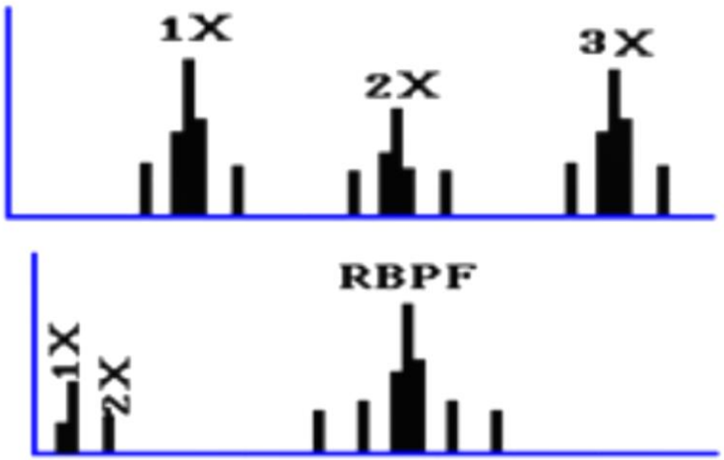
STATOR ECCENTRICITY
SHORTED LAMINATIONS
AND LOOSE IRON

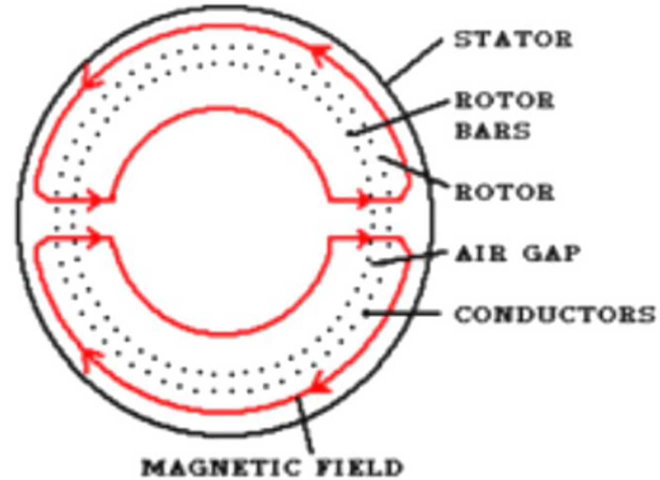


- Stator problems generate high amplitudes at $2F_L$ (2X line frequency)
- Stator eccentricity produces uneven stationary air gap, vibration is very directional
- Soft foot can produce an eccentric stator

ELECTRICAL PROBLEMS

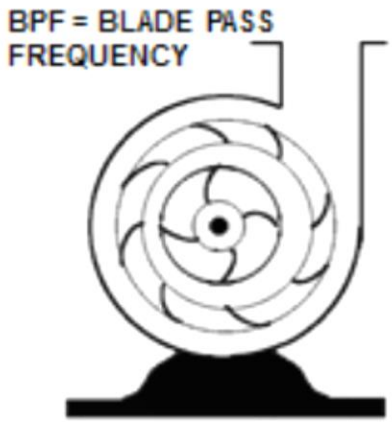
ROTOR PROBLEMS

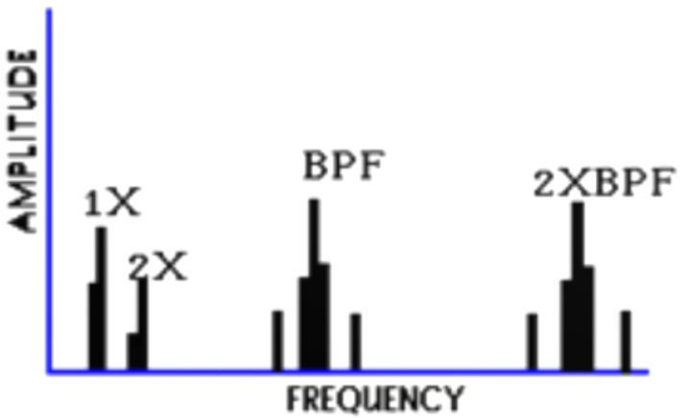




- 1X, 2X, 3X, RPM with pole pass frequency sidebands indicates rotor bar problems.
- 2X line frequency sidebands on rotor bar pass frequency (RBPF) indicates loose rotor bars.
- Often high levels at 2X & 3X rotor bar pass frequency and only low level at 1X rotor bar pass frequency.

HYDRAULIC AND AERODYNAMIC FORCES





- If gap between vanes and casing is not equal, Blade Pass Frequency may have high amplitude
- High BPF may be present if impeller wear ring seizes on shaft
- Eccentric rotor can cause amplitude at BPF to be excessive

Common Machinery Faults

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

Vibration Sources Identification Guide

CAUSE	FREQUENCY	AMPLITUDE	PHASE	COMMENTS
Unbalance	1 x RPM	Highest in Radial Direction-Proportional to Unbalance	Single Mark (Steady)	A common cause of vibration.
Defective Anti-Friction Bearings	Very High-Often From 10 to 100 x RPM	Use Velocity	Unstable	Velocity readings are highest at defective bearing. As failure approaches, the amplitude of the velocity signal will increase and its frequency will decrease. Cage frequency is approximately 0.6 x RPM x number elements.
Misalignment of Coupling or Bearing	1, 2 or 3 x RPM	High Axial Axial 50% or more of Radial	Often 2, Sometimes 1 or 3	Use phase analysis to determine relative movement of machine or bearings. Use a dial indicator if possible. Often diagnosed as a bent shaft. Can be caused by misalignment of V belts.
Sleeve Bearing	1 x RPM	Not Large Use Displacement Mode Up to 6000 CPM	Single Reference Mark	May appear to be unbalanced. Shaft and bearing amplitude should be taken. If shaft vibration is larger than the bearing, vibration amplitude indicates clearance.
Bent Shaft	1 or 2 x RPM	High Axial	1 or 2	Similar to misalignment. Use phase analysis.
Defective Gears	High No. Gear Teeth x RPM	Radial	Unsteady	Use velocity measurement. Often affected by misalignment. Generally accompanied by side band frequency. Pitting, scuffing and fractures are often caused by torsional vibrations. Frequency sometimes as high as 1 million CPM or more.
Mechanical Looseness	2 x RPM Sometimes 1 x RPM	Proportional to Looseness	1 or 2	Check movement of mounting bolts in relation to the machine base. Difference between base and machine indicates amount of looseness.
Defective Drive Belts	1 or 2 x Belt Speed	Erratic	Use Strobe to Freeze Belt in OSC Mode	Calculate the belt RPM using: $\text{Belt RPM} = \frac{\text{Pulley Diameter} \times 3.141}{\text{Belt Length}} \times \text{Pulley RPM}$ Look for cracks, hard spots, soft spots or lumps. Loose belt. Changes with belt tension.
Electrical	1 or 2 x Line Frequency (3600 or 7200 CPM for 60Hz Power) May appear at 1 x RPM	Usually Low	1 or 2 Marks Sometimes Slipping	Looks like mechanical unbalance until power is removed. Then drops dramatically.
Oil Whip	45 - 55% RPM	Radial Unsteady	Unstable	Caused by excessive clearance in sleeve bearings or by underloaded bearings. Will change with viscosity of oil (temperature).
Hydraulic-Aerodynamic	No. Blades or Vanes x RPM	Erratic	Unsteady	May excite resonance problems.
Beat Frequency	Near 1 x RPM	Variable at Beat Rate	Rotates at Beat Frequency	Caused by two machines, mounted on same base, running at close to same RPM.
Resonance	Specific Criticals	High	Single Reference Mark	Phase will shift 180° going through resonance (90° at resonance). Amplitude will peak at resonance. Resonance in frame can be removed by changing rotor operating speed or by changing the stiffness of the structure.