Beginning Vibration Analysis with Basic Fundamentals

BY
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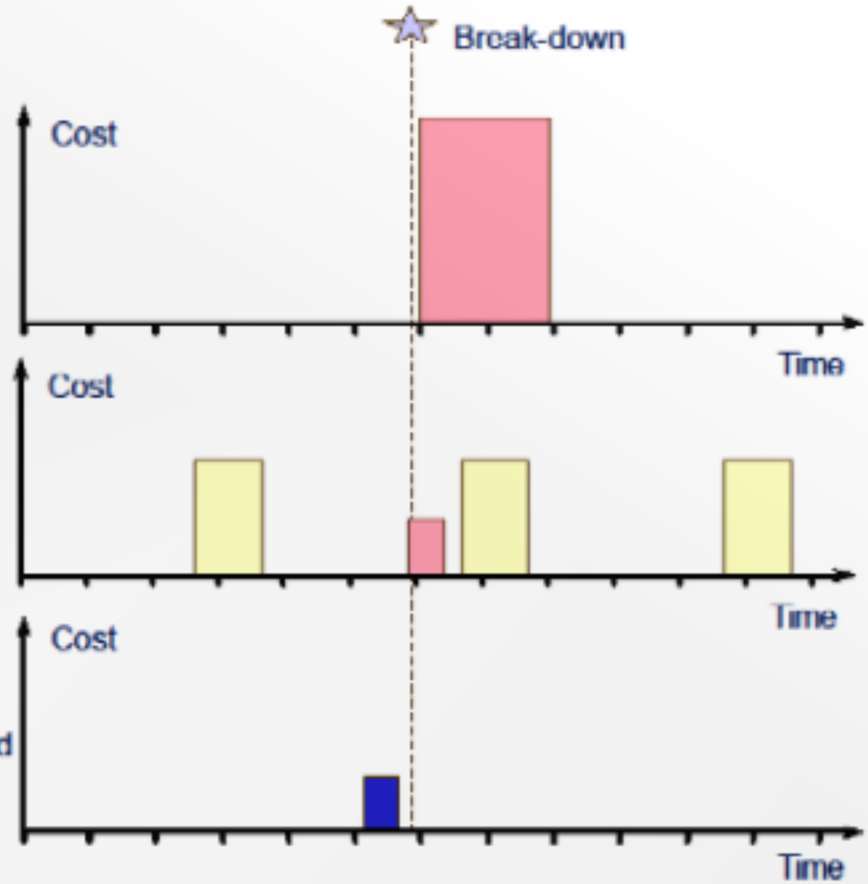
Types of Maintenance

Corrective Maintenance
(Run-to-breakdown)

Repair it when it fails

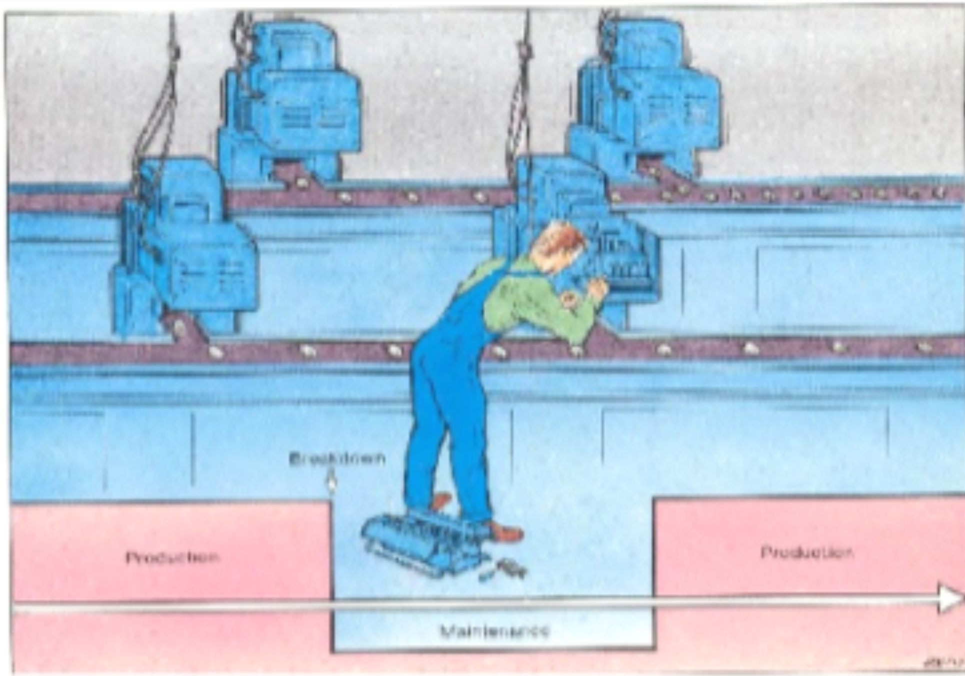
Preventive Maintenance
(Time Based Maintenance)
Maintenance at regular intervals

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



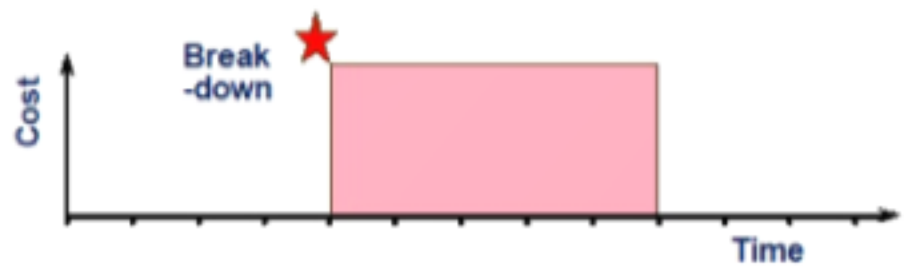
Corrective Maintenance

-Run to Breakdown-



Corrective Maintenance leads to:

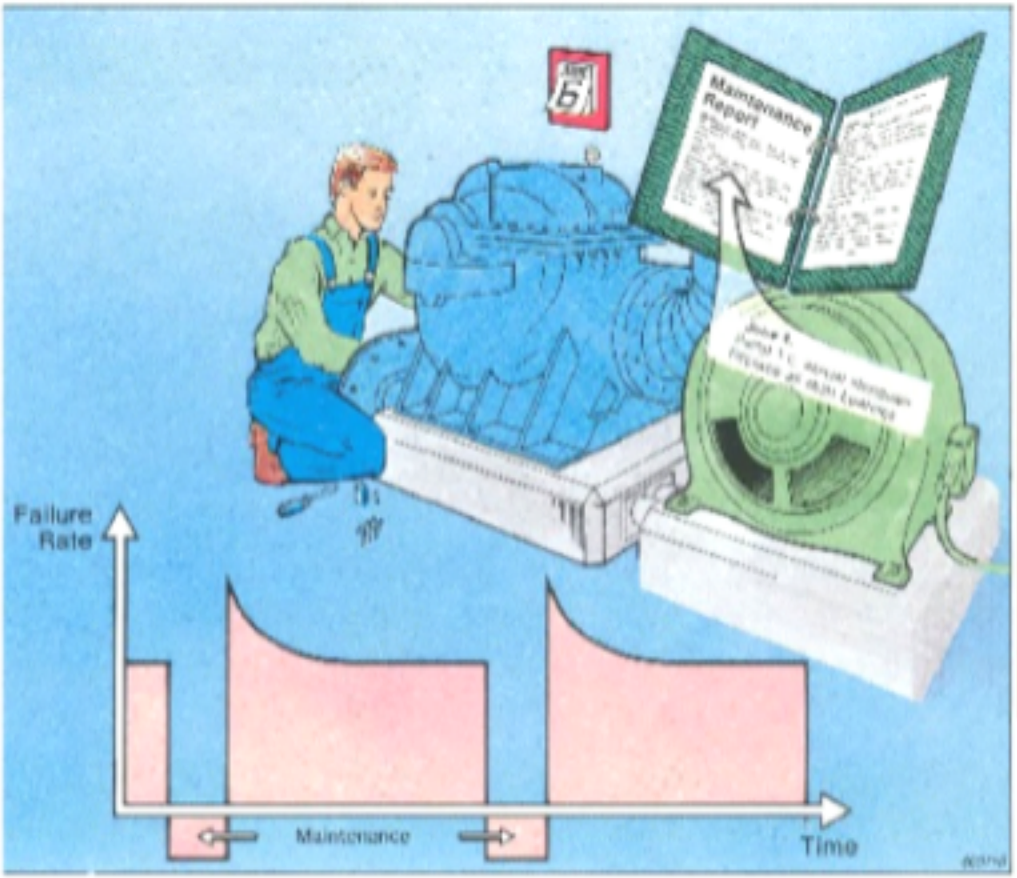
- Secondary damage
- Safety risk
- Unplanned downtime
- Unplanned maintenance
- Product waste
- Spares inventory



Preventive Maintenance

Time Based Maintenance

Not recommended for critical machines

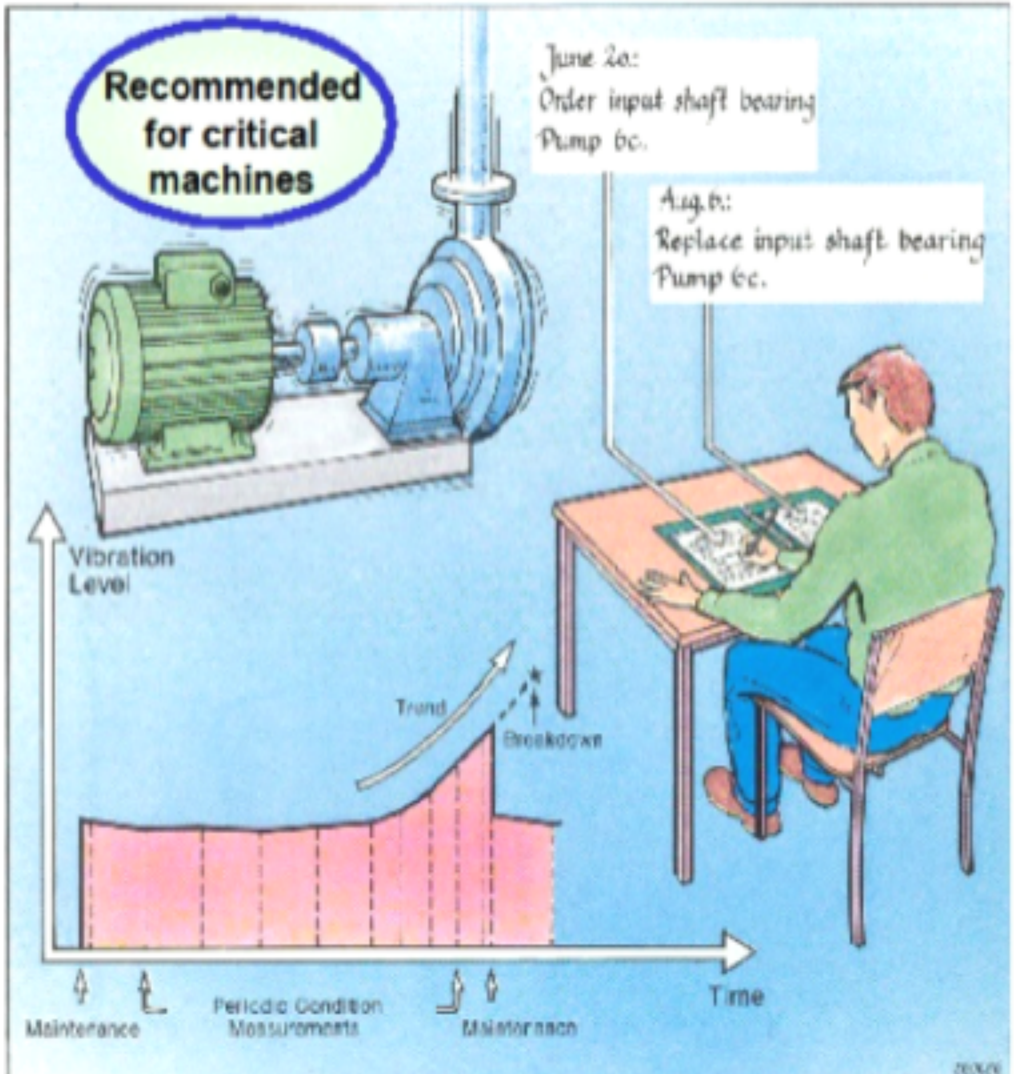


Time-based Preventive Maintenance involves:

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

Predictive Maintenance

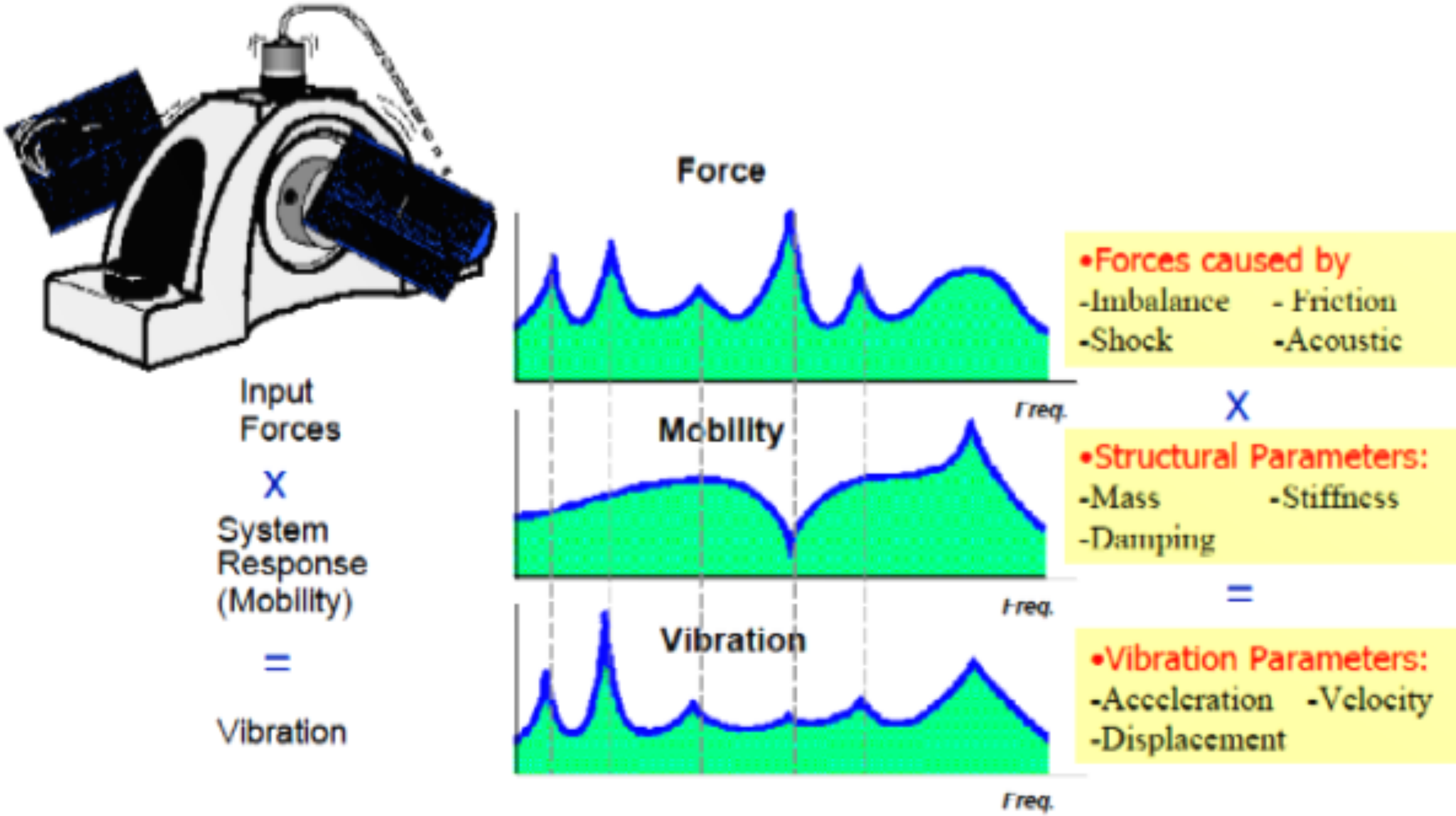
On-condition -Maintenance



- 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

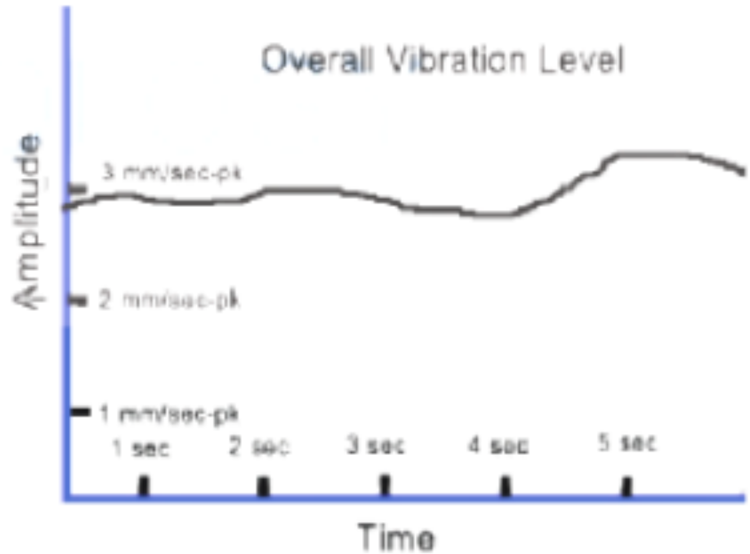
- Higher plant availability, performance and reliability
- Greater safety
- Better product quality
- Attention to environment
- Longer equipment life
- Greater cost effectiveness

What is Vibration



Overall Amplitude

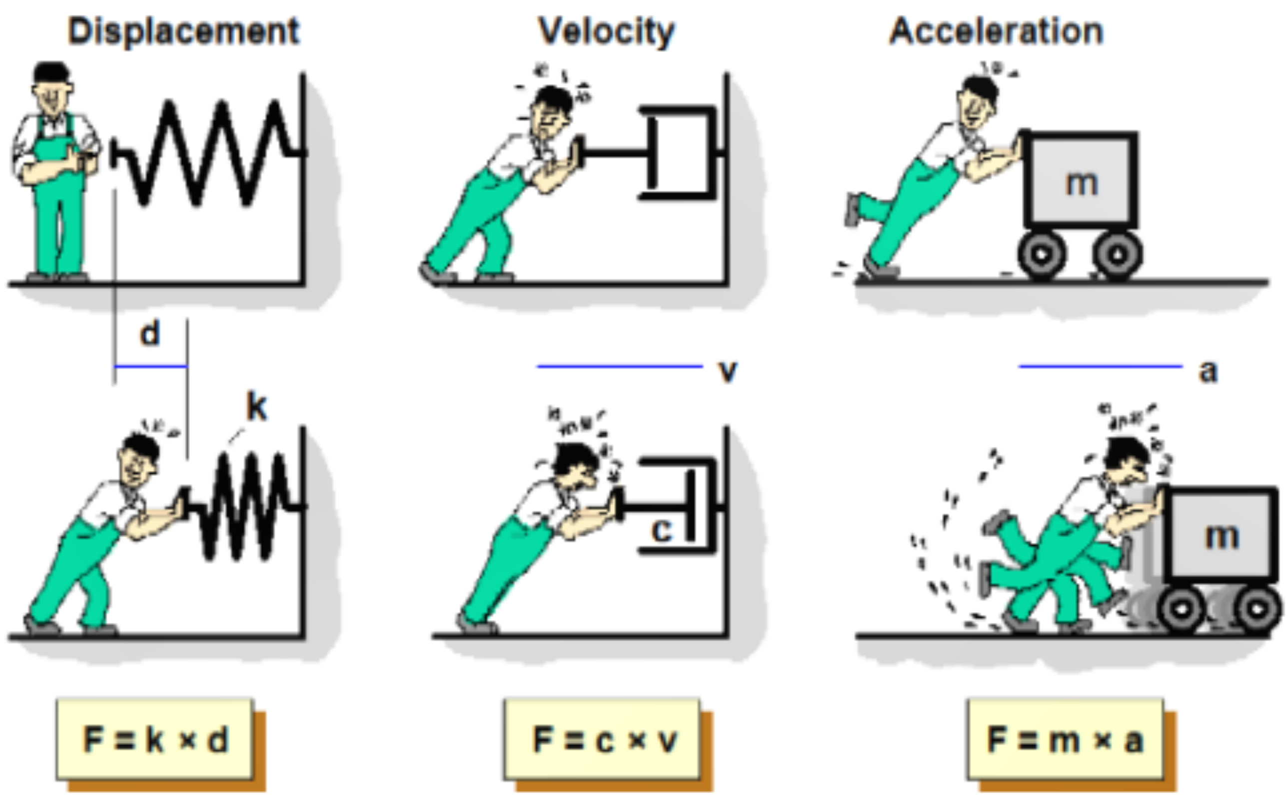
- It is the total vibration amplitude over a wide range of frequencies.
- Acceleration, Velocity, or Displacement.



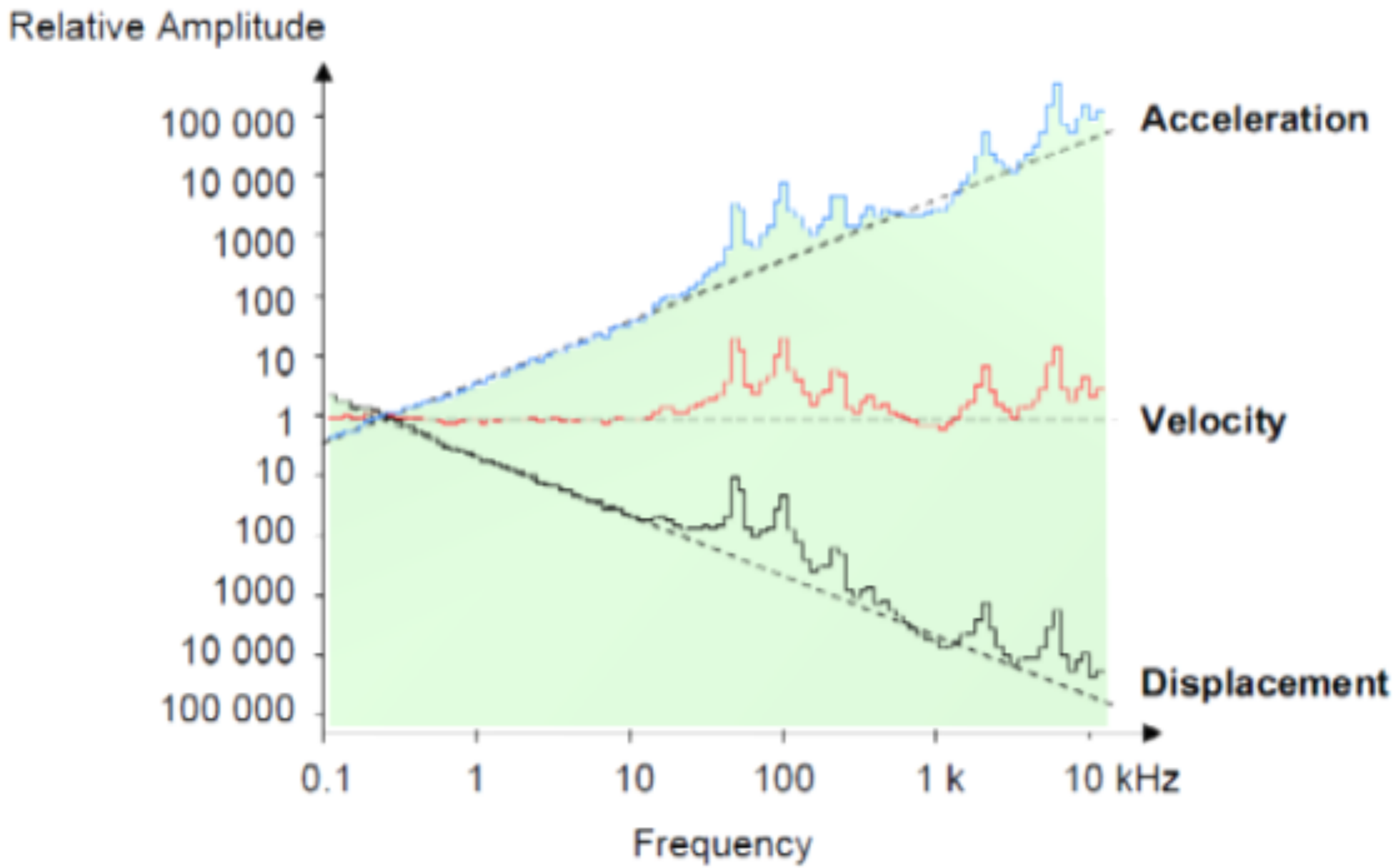
Vibration Terminology

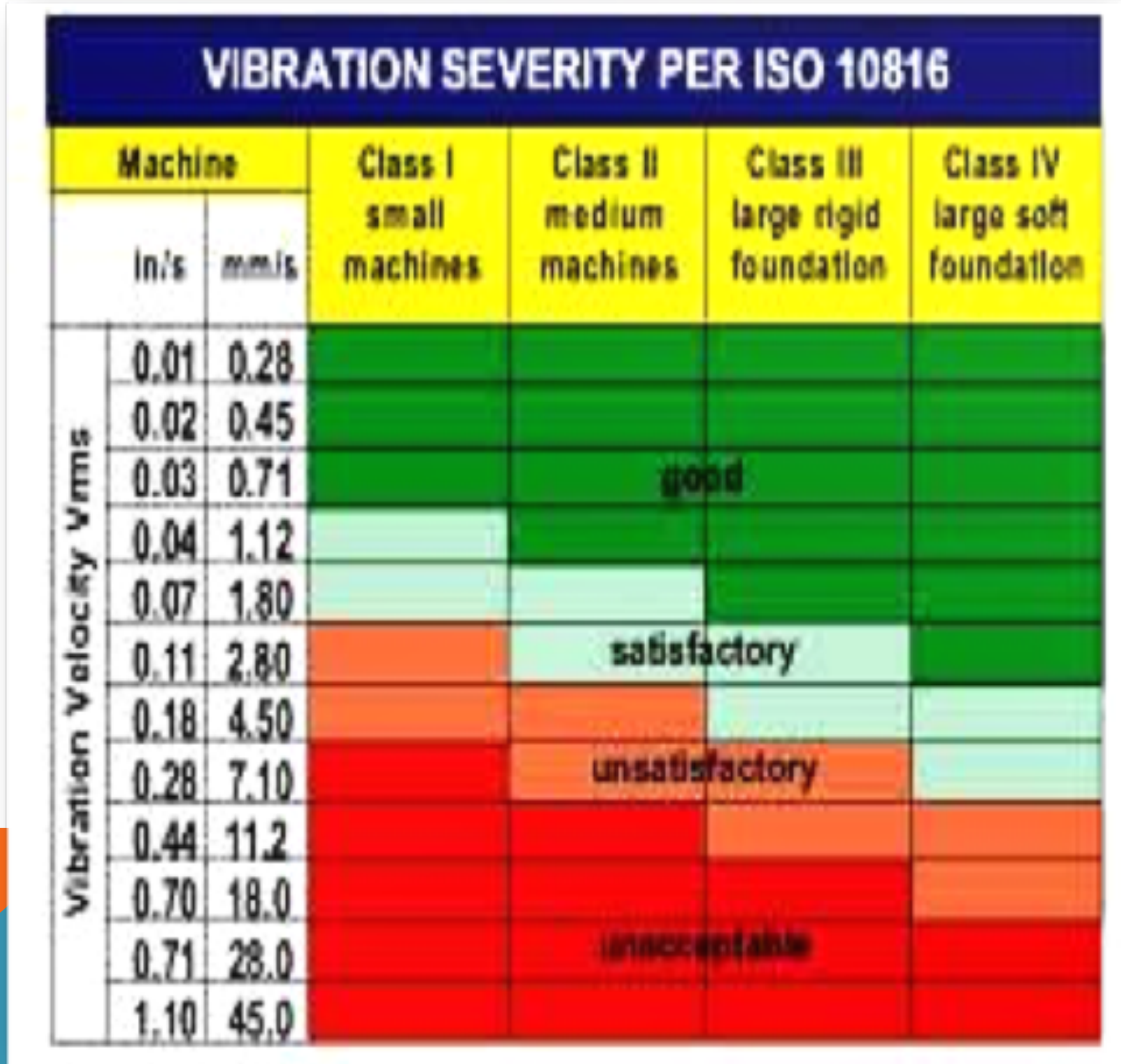
- Displacement [peak-peak]
- Velocity [peak]
- Velocity [rms]
  - Velocity rms tends to provide the energy content in the vibration, whereas the Velocity peak depicts more of the intensity of vibration.
- Acceleration - peak

Vibration Parameters



Vibration Parameters



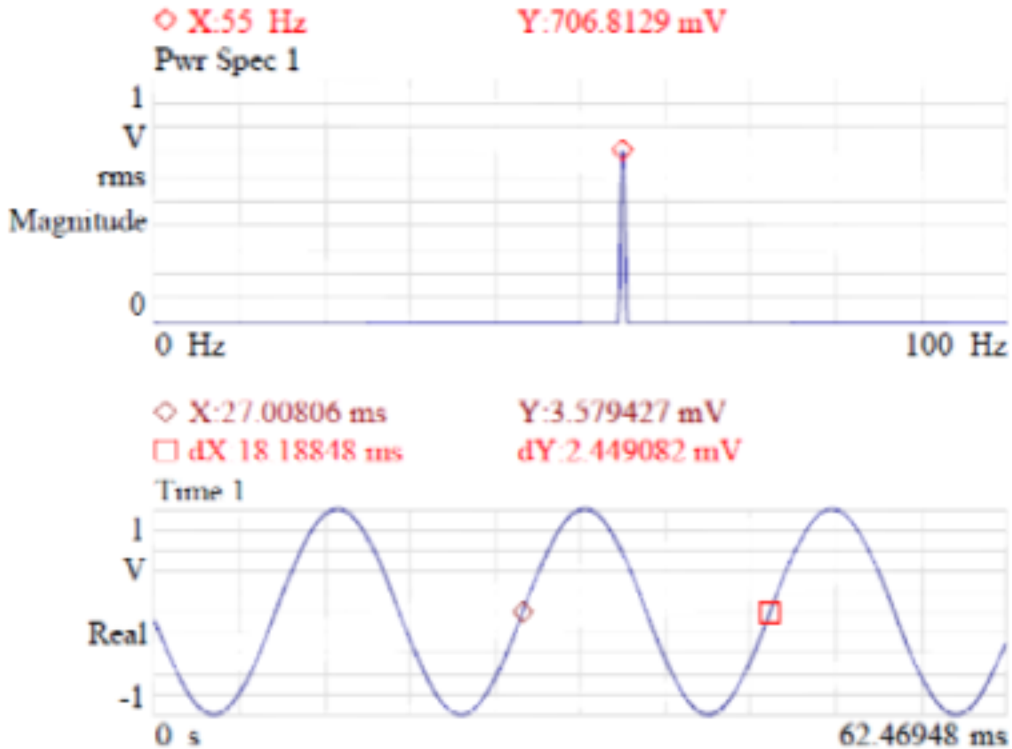


VIBRATION SEVERITY PER ISO 10816

Machine in/s	Machine mm/s	Class I small machines	Class II medium machines	Class III large rigid foundation	Class IV large soft foundation
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

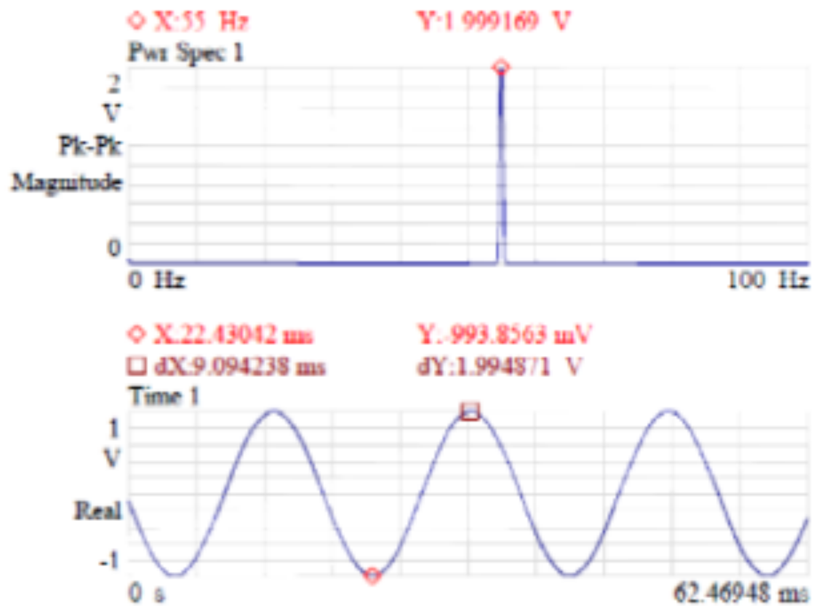
Single Frequency



Pk-Pk (Peak - Peak)

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

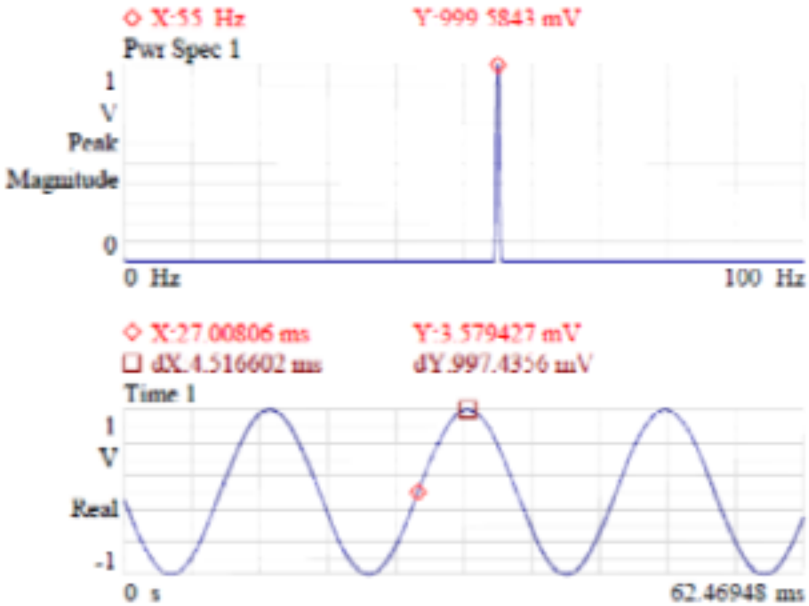
The spectrum value uses the suffix “Pk-Pk” to denote this.



Pk (Peak)

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

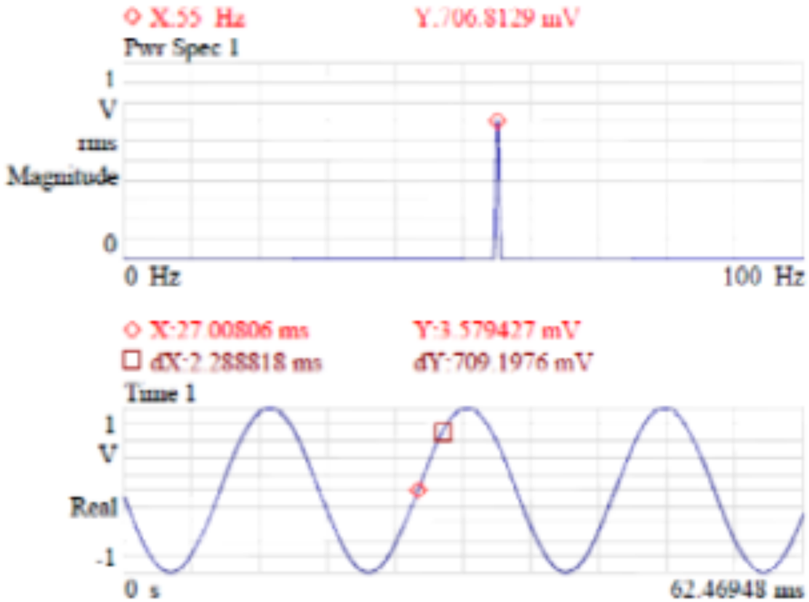
The spectrum value uses the suffix "Peak" to denote this.



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.

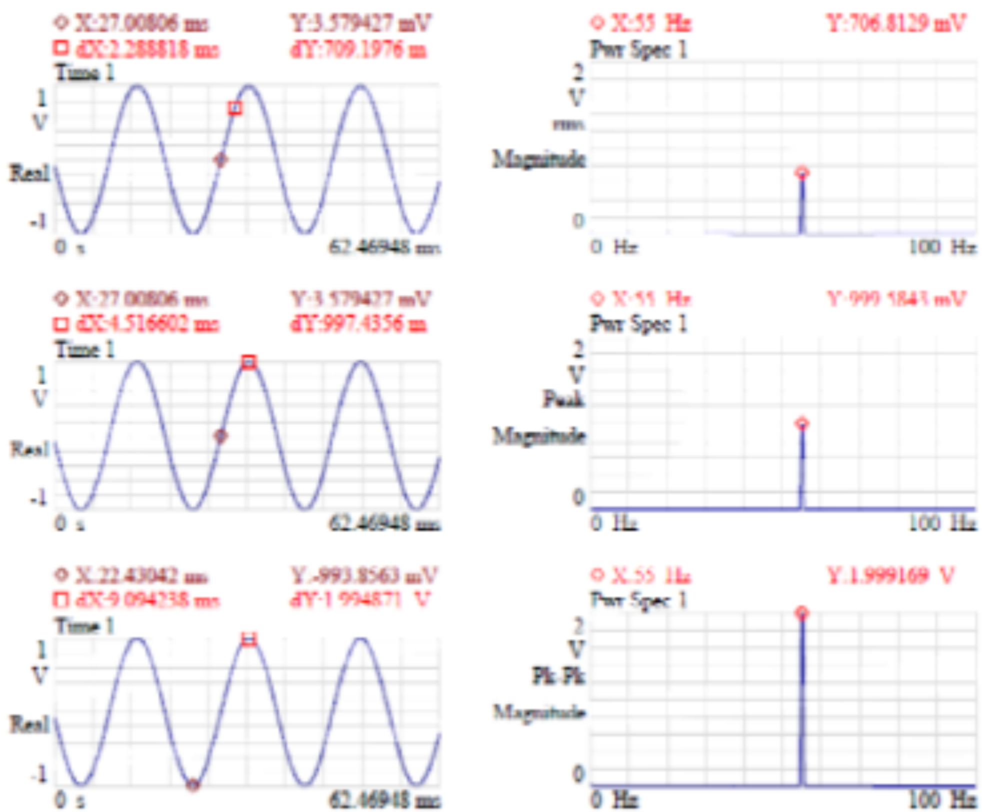


Suffix Comparison

RMS

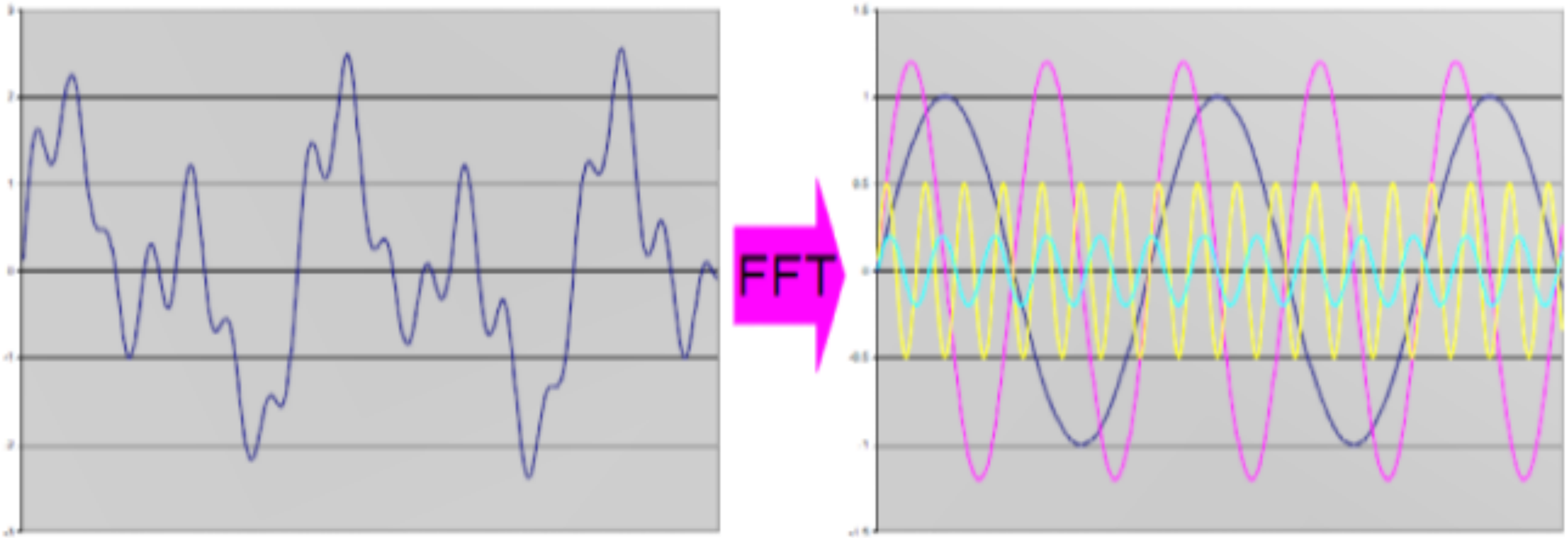
Peak

Peak - Peak



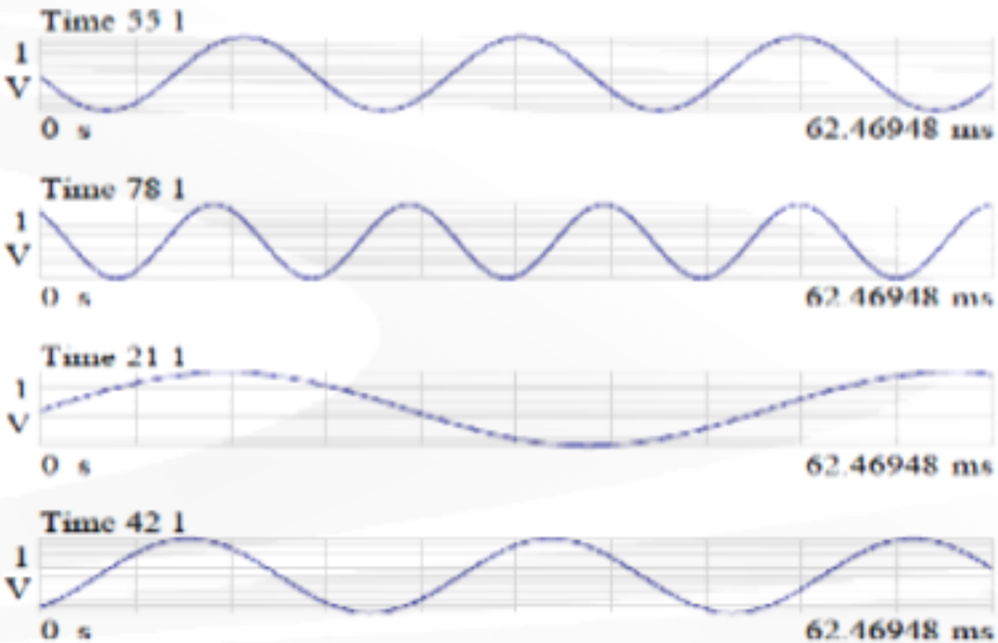
Signal Processing – break down complex waveform in to waveform components

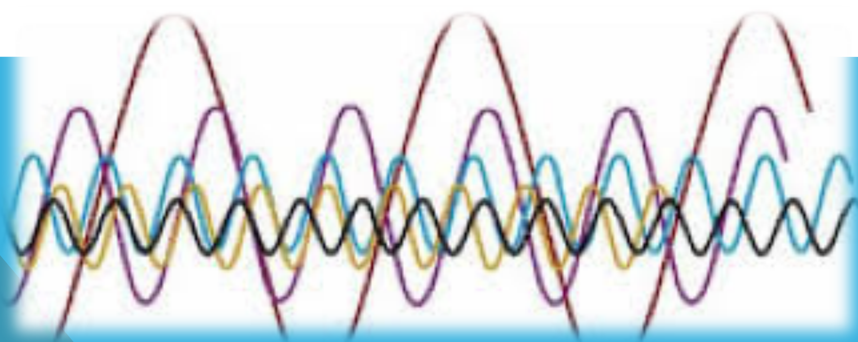
The Fast Fourier Transform (FFT) takes the complex waveform and breaks it down into the component sine waves



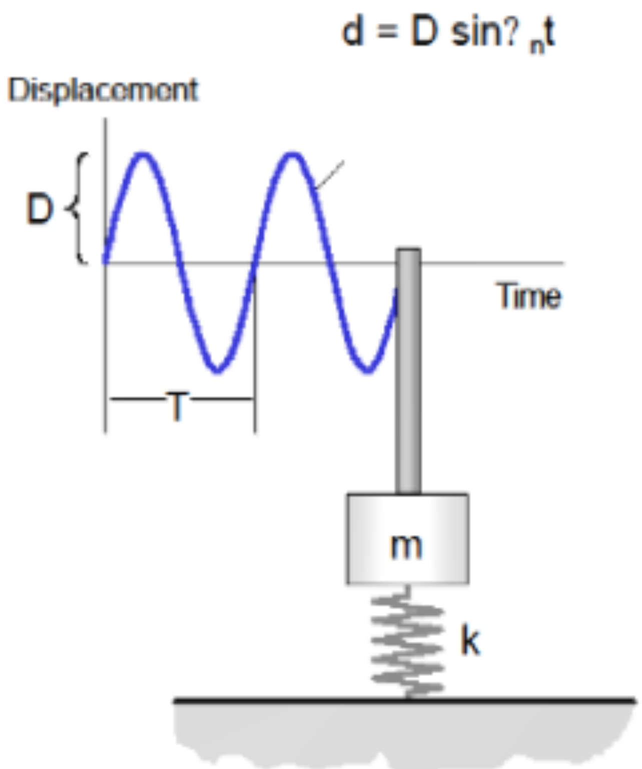
The amplitudes for each sine wave is then plotted at the frequency of the sine wave, creating the Spectrum

Multiple Time





FFT transformation

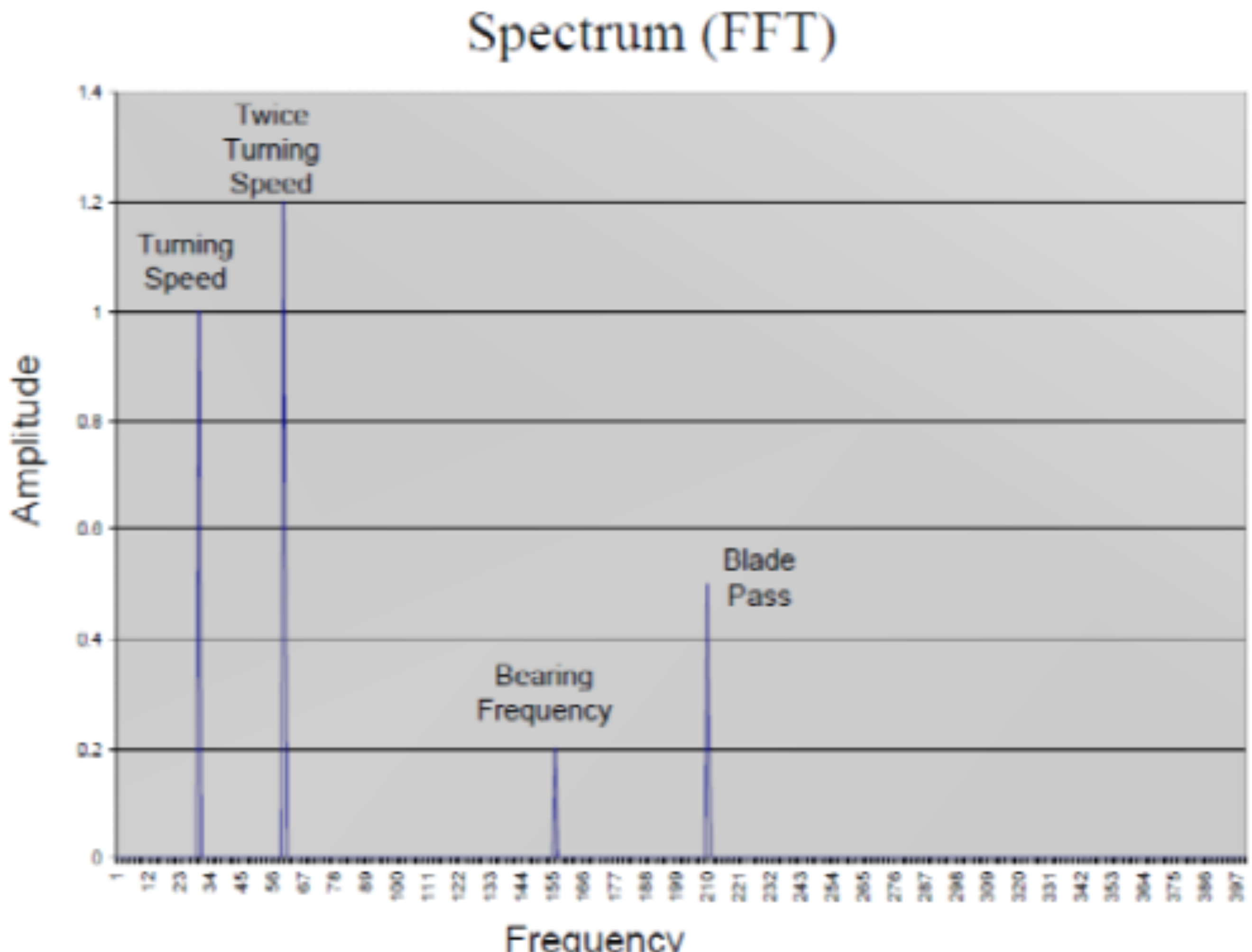


Period, T_n in [sec]

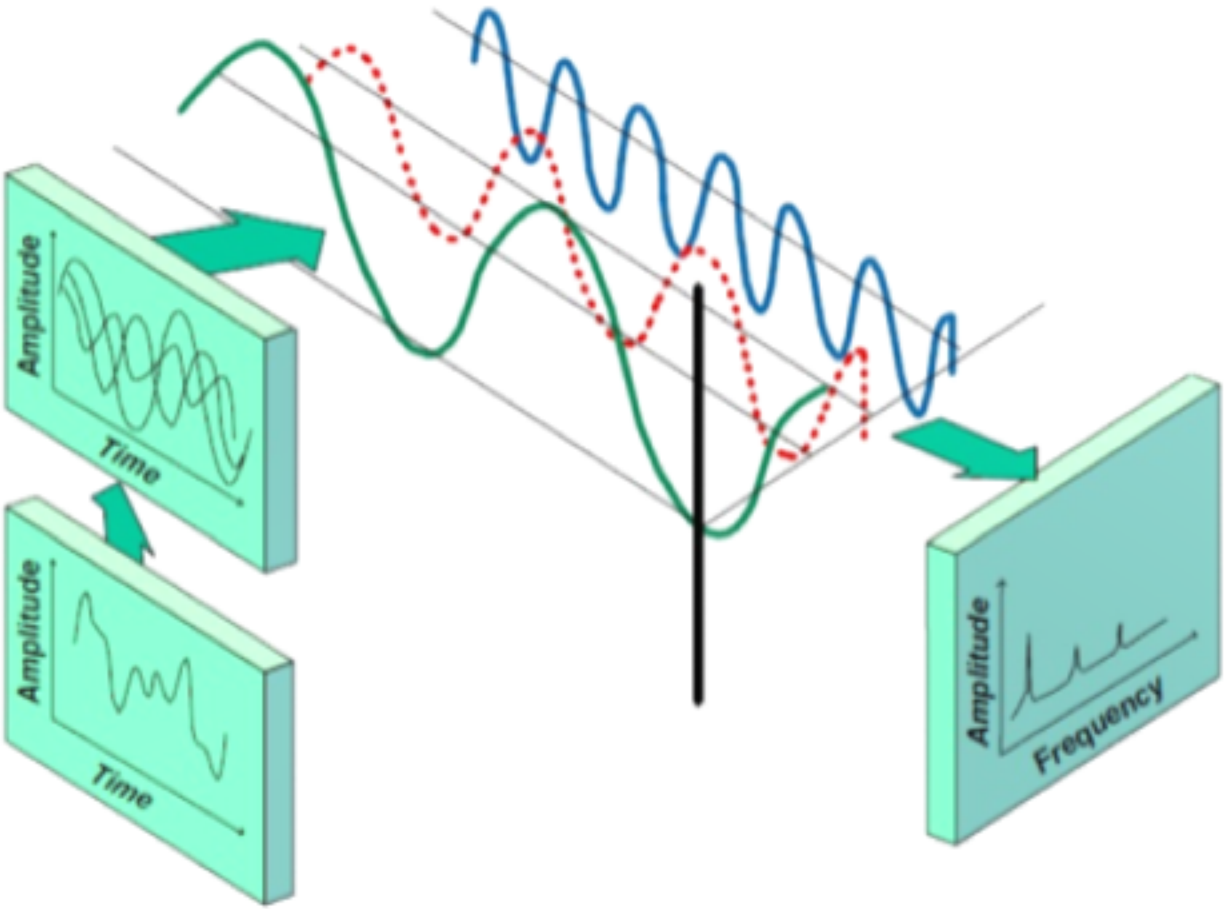
Frequency, $f_n = \frac{1}{T_n}$ in [Hz = 1/sec]

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

Signal Processing – The FFT or Spectrum



FFT Signal Processing



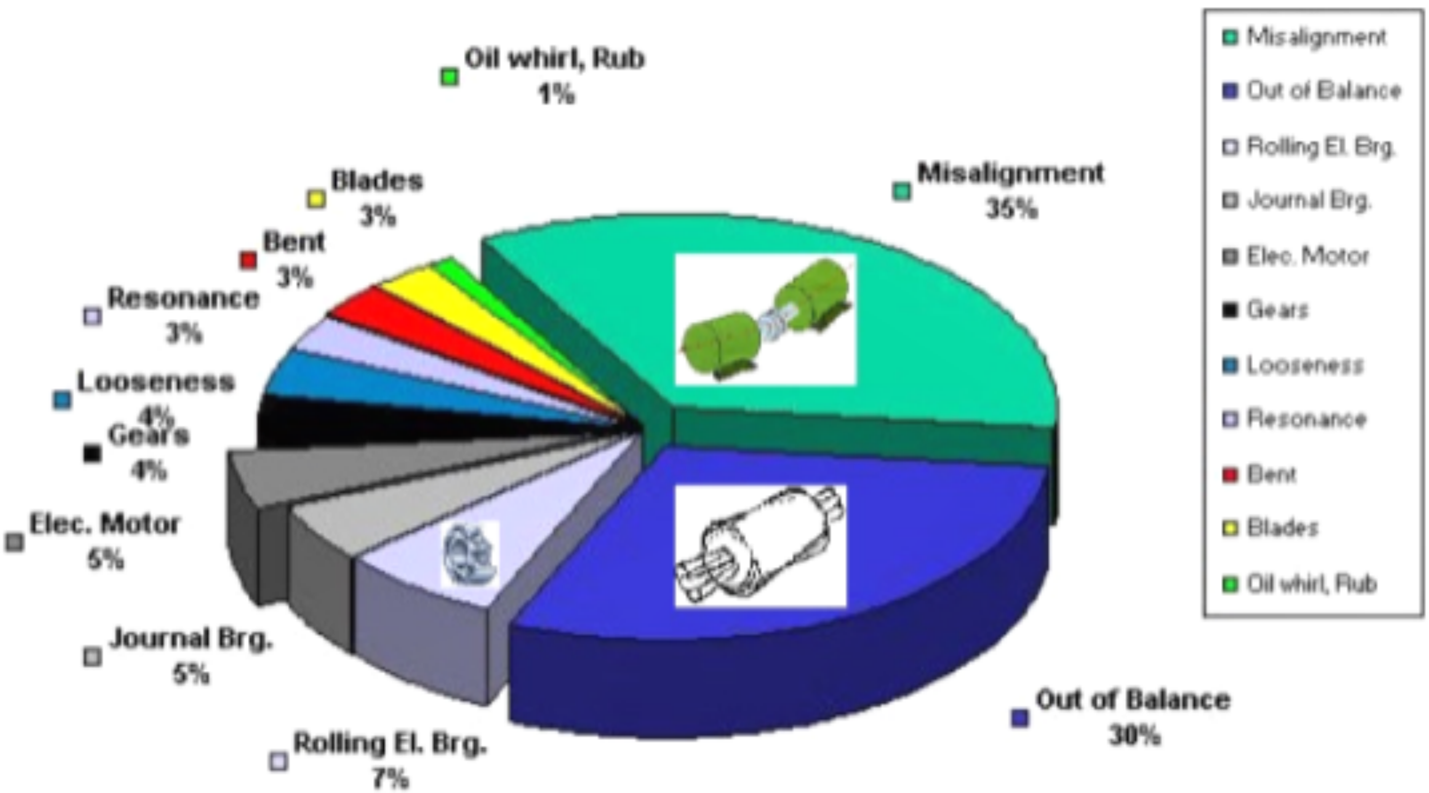
Common Machinery Faults

- 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

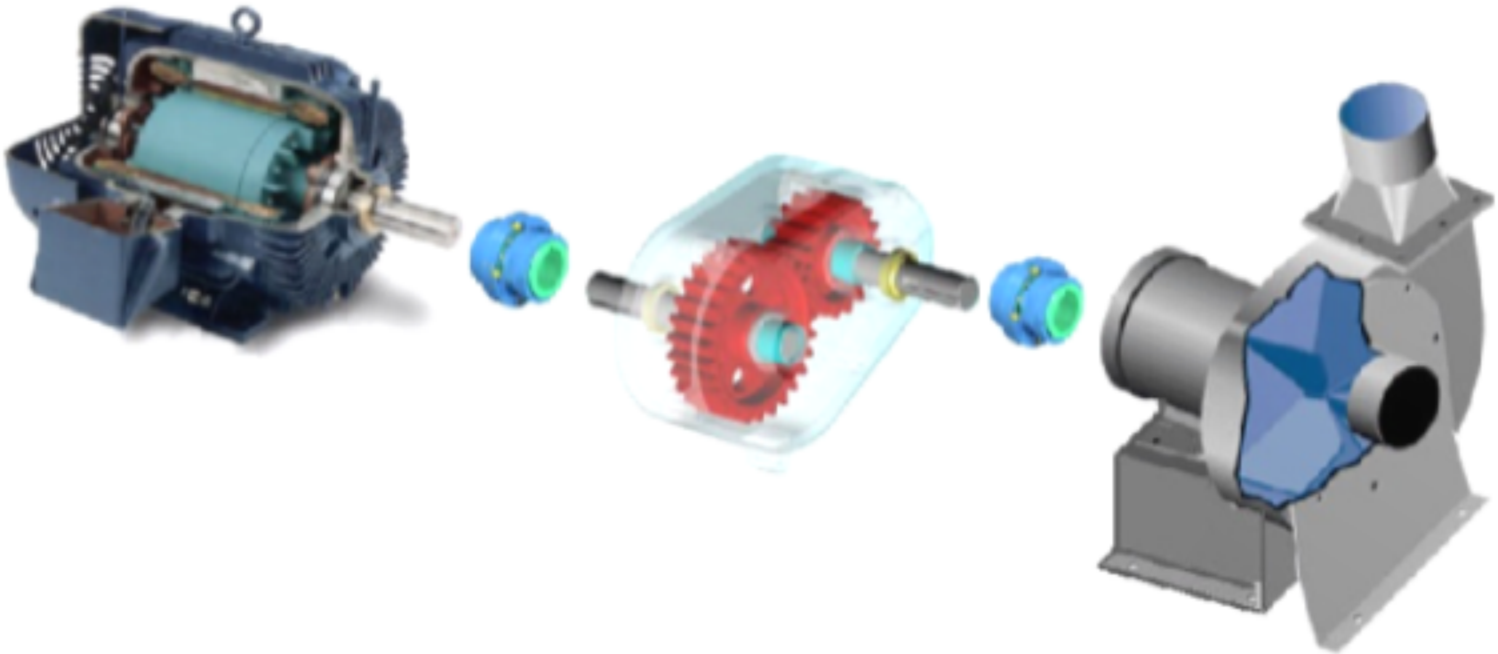
Vibration Sources Identification Guide

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.

Machine Potential Failures Analysis

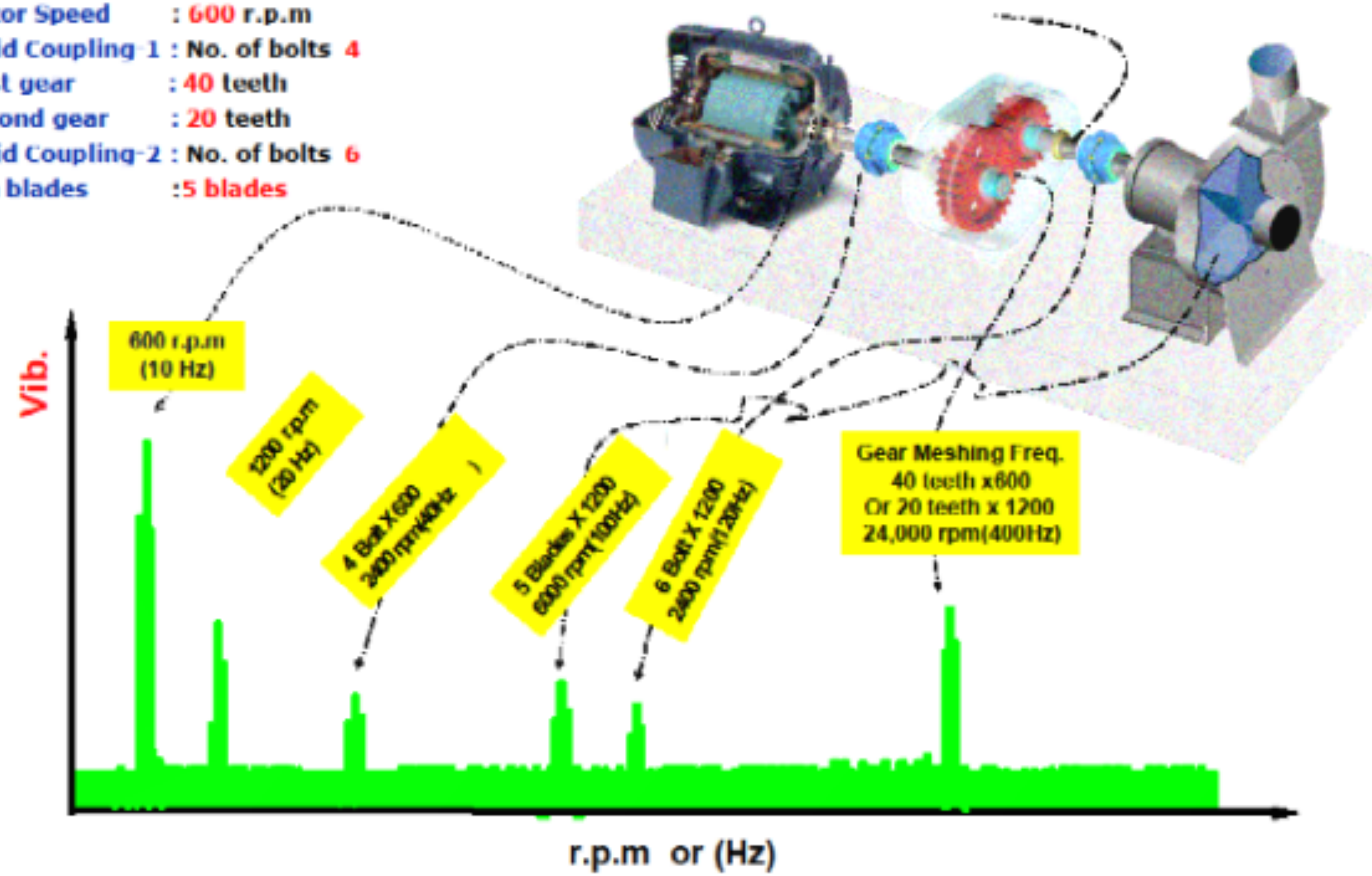


Example of Machine Component

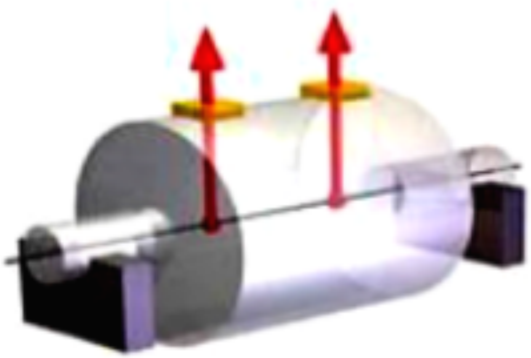


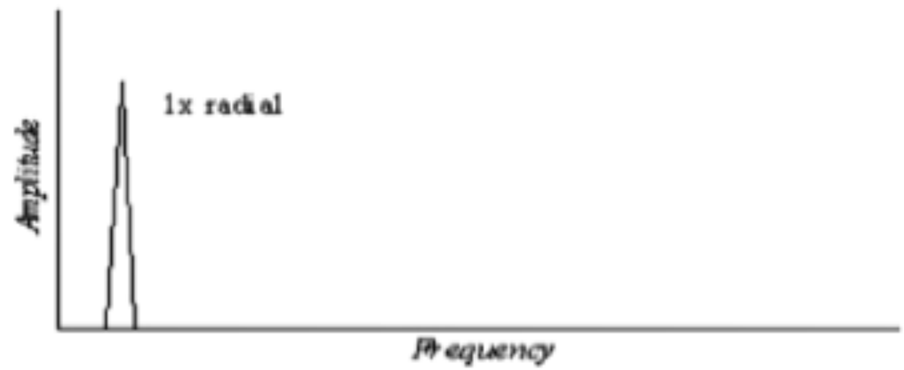
Frequency Spectrum interpretation

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



Static Unbalances

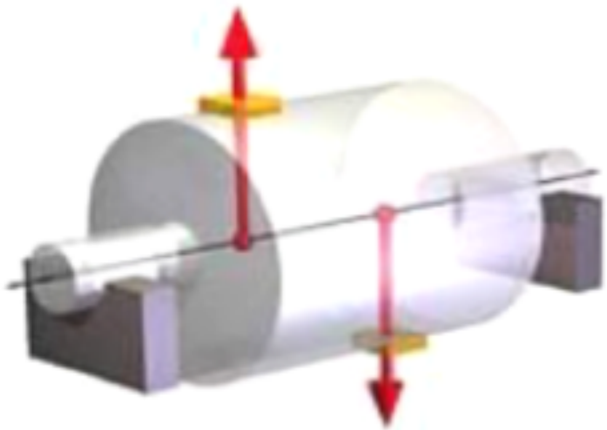


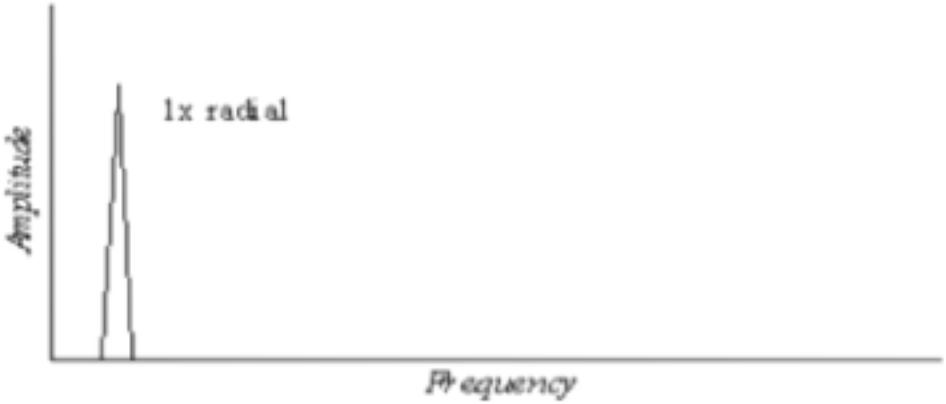


Unbalance - Static

- 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

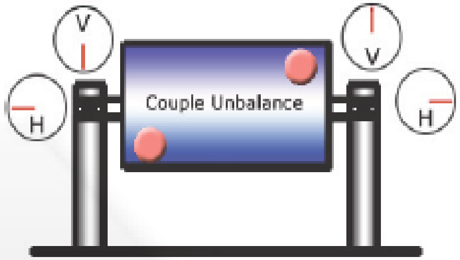
Couple Unbalances



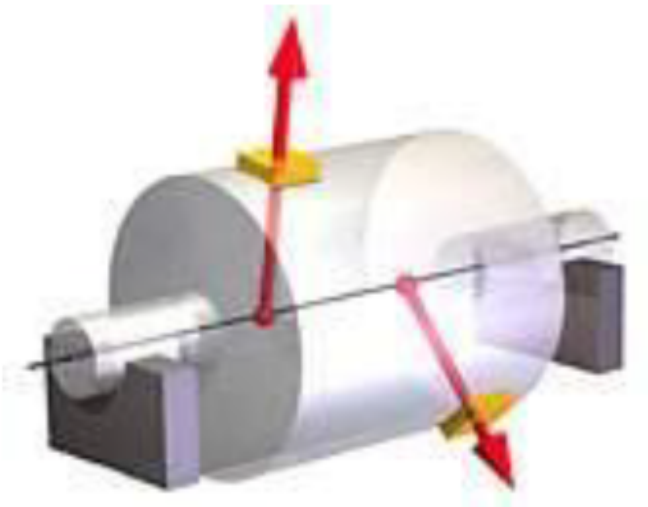


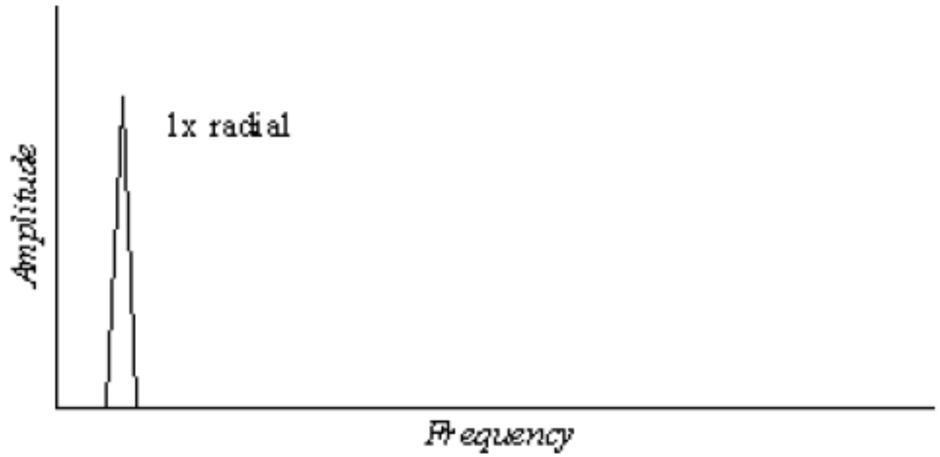
Unbalance - Couple

- 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.

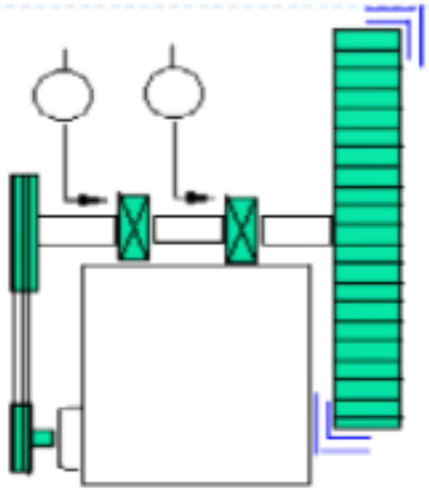


Dynamics Unbalances





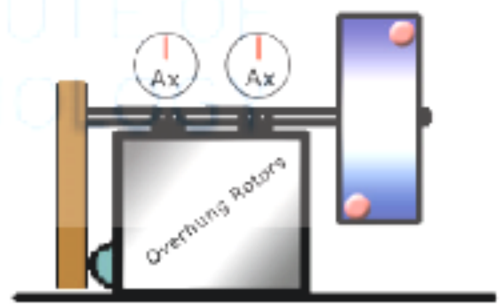
Overhung Rotor Unbalance



Special case of Dynamics Unba

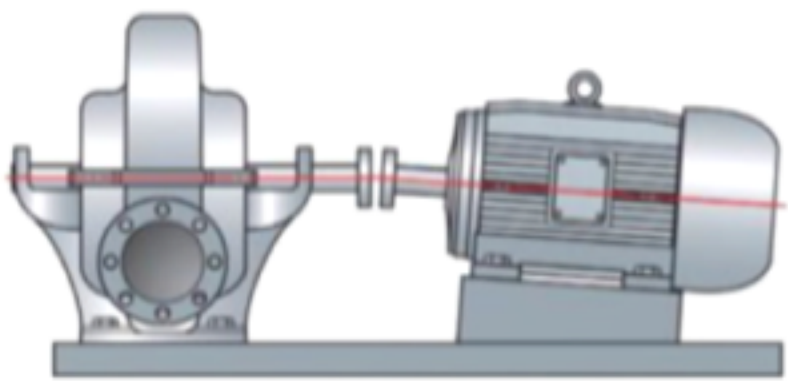
Unbalance - Overhung Rotors

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



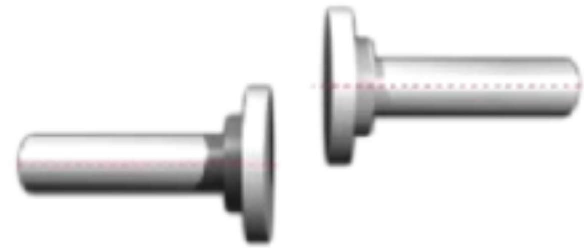
Misalignment

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



Type of Misalignment

Parallel Offset



Parallel Offset Misalignment

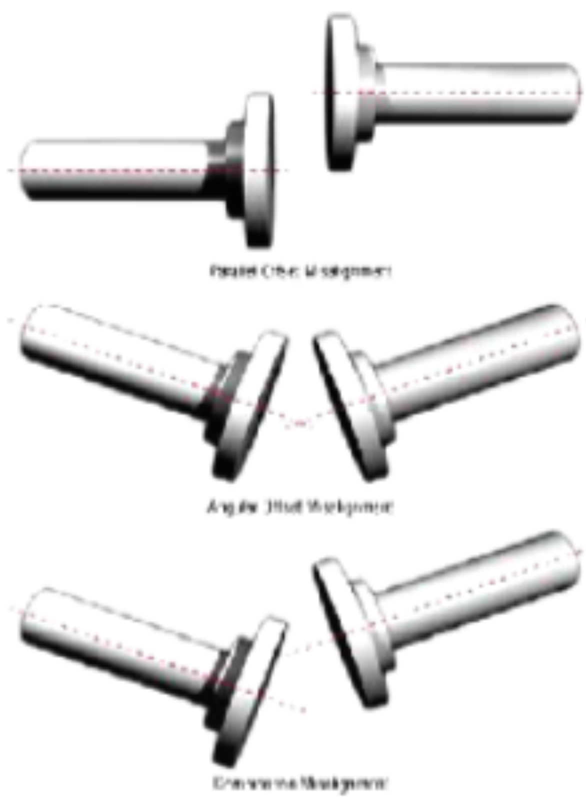
Angular Offset

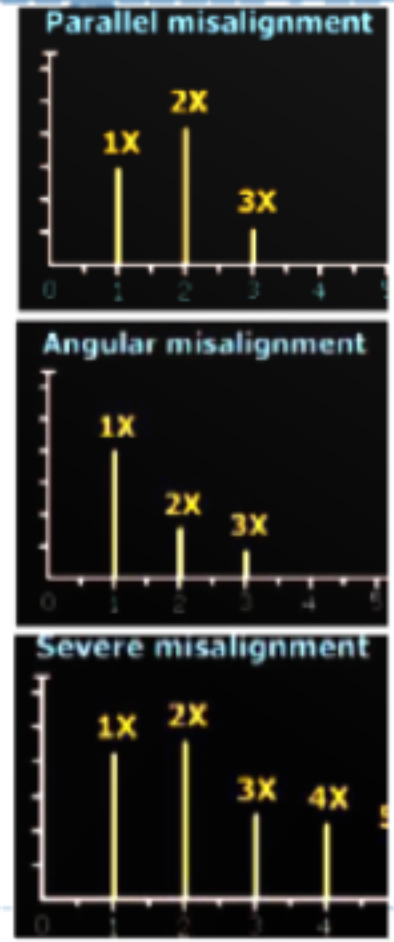
Angular Offset Misalignment

Combination Offset

Combination Misalignment

Type of Misalignment

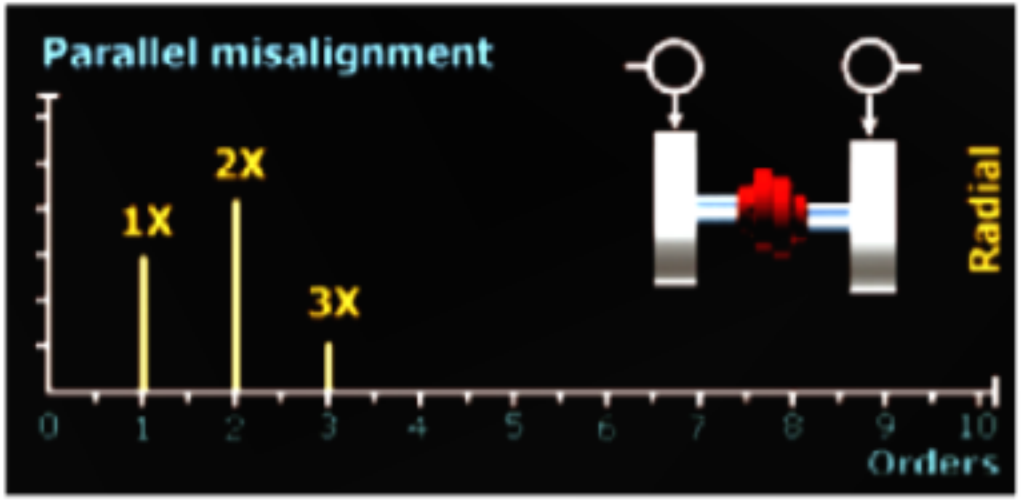




Type of

Misalignment

Symptoms: 2X radial, smaller 1X *radial*

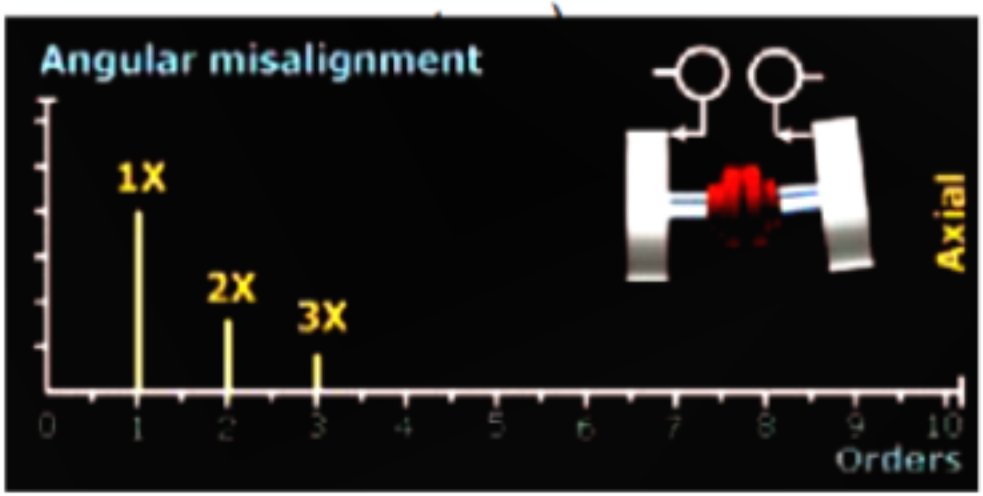


- Shear force
- Bending moment
- 2X component will be higher than 1X.
- Depend on the coupling type
- Axial 1X and 2X levels will be low

Type of

Misalignment

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

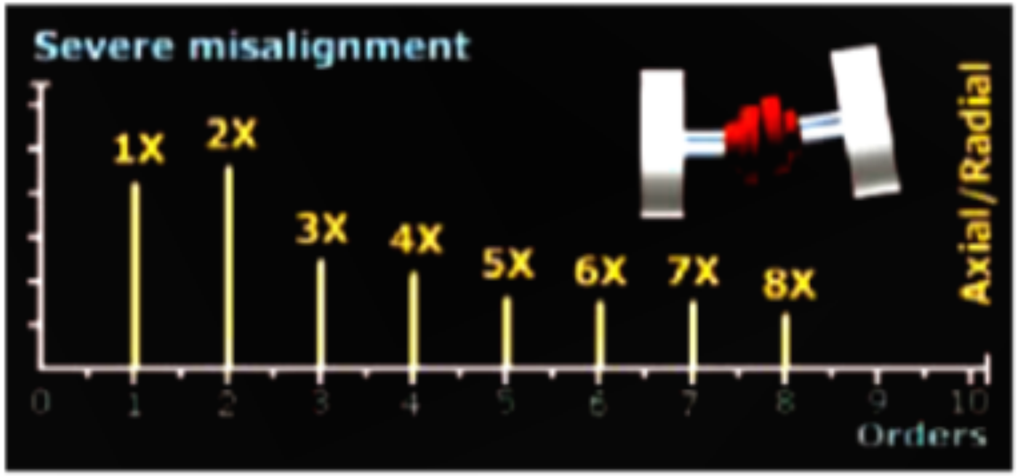


- Bending moment
- Also be fairly strong ***radial*** 1X and 2X levels

Type of

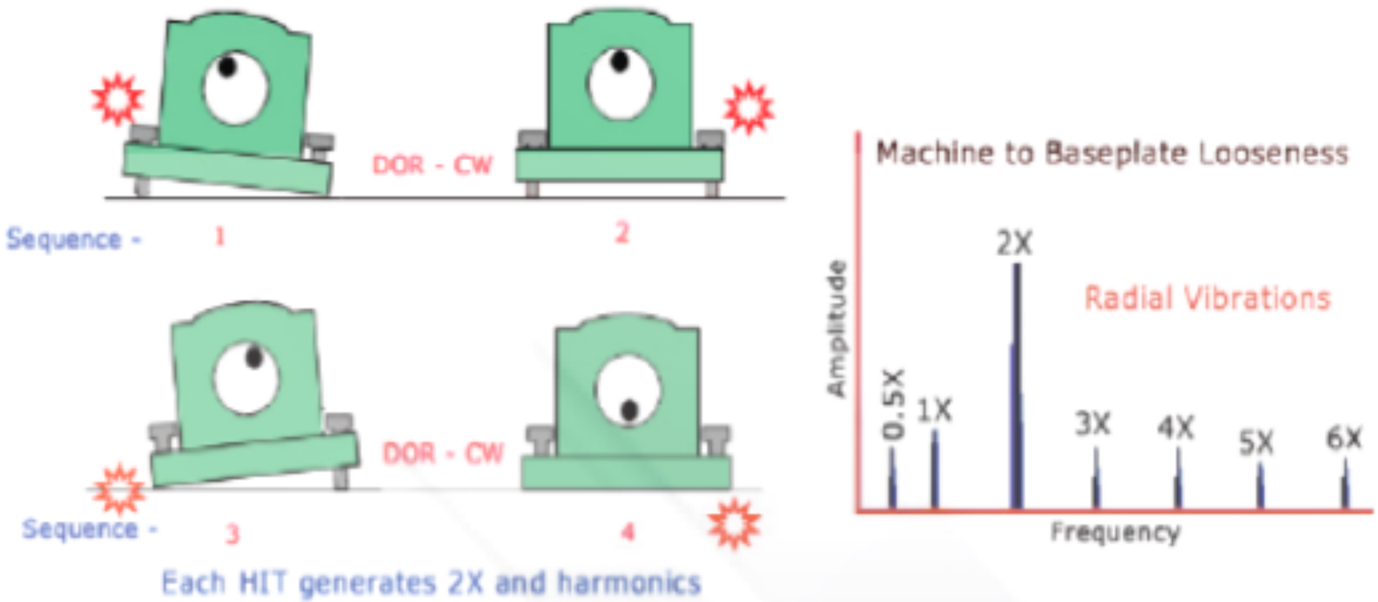
Misalignment

Frequency: 1X 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)

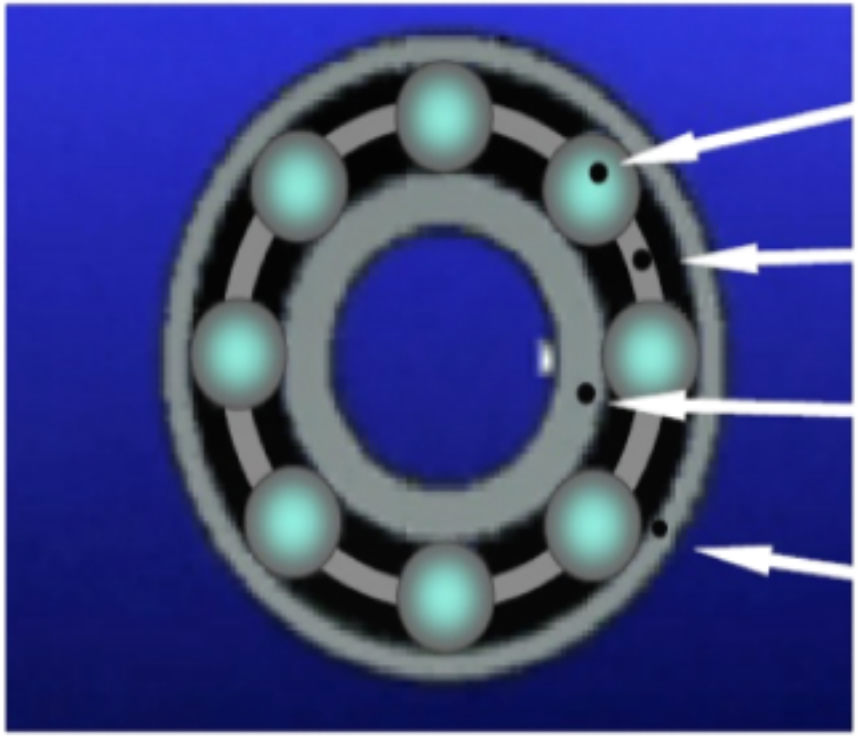
Looseness at Machine to Base Plate interface



Mechanical Looseness

Roller Bearing Faults

Four different bearing frequencies



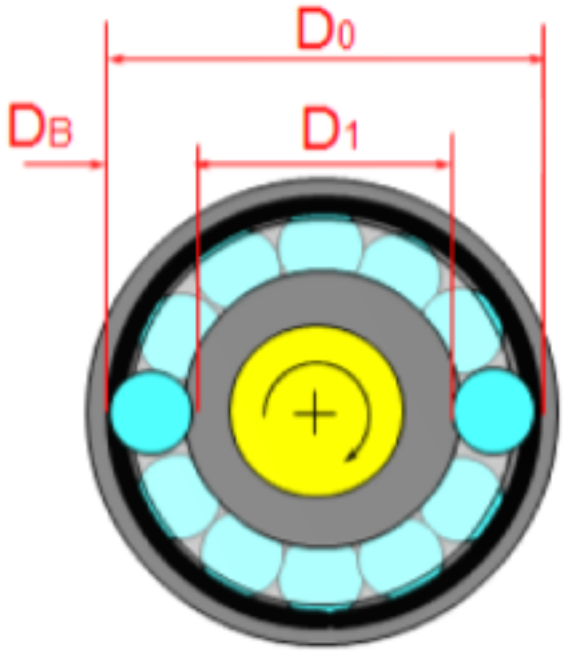
Ball Spin Frequency (BSF)

Fundamental Train Frequency (FTF)

Ball Pass Frequency Inner Race (BPFI)

Ball Pass Frequency Outer Race (BPFO)

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

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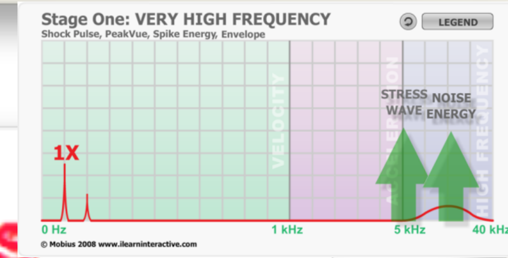
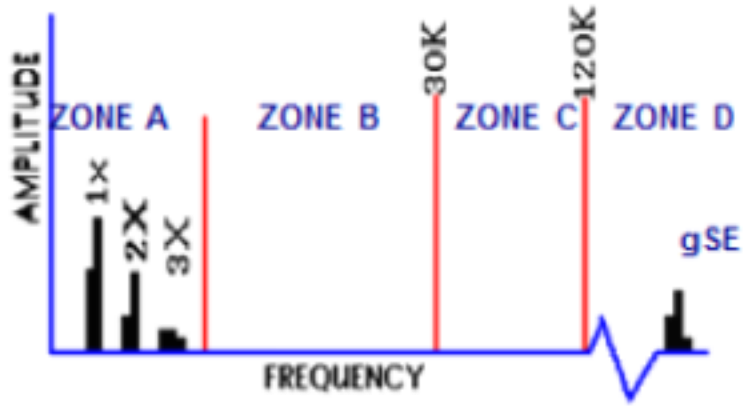
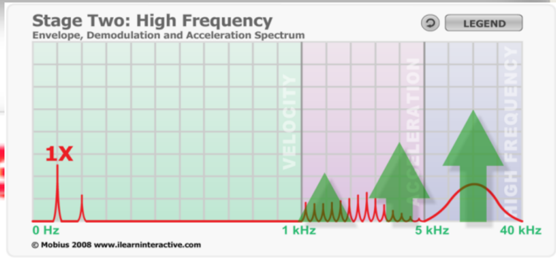


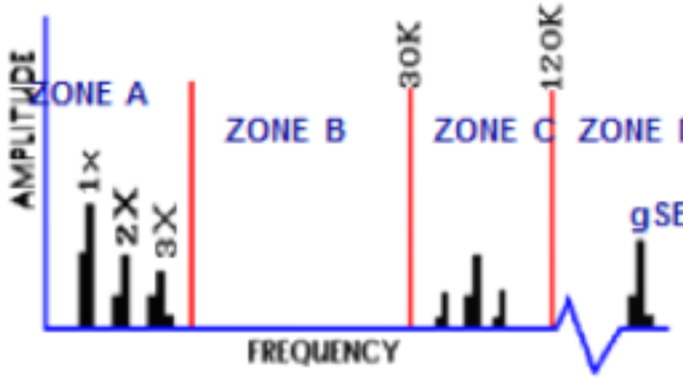
ROLLING ELEMENT BEARINGS STAGE 1 FAILURE MODE



- Earliest indications in the ultrasonic range
- These frequencies evaluated by Spike Energy™ gSE, HFD(g) and Shock Pulse
- Spike Energy may first appear at about 0.25 gSE for this first stage



ROLLING ELEMENT BEARING STAGE 2 FAILURE MODE



- Slight defects begin to ring bearing component natural frequencies
- These frequencies occur in the range of 30k-120k CPM
- At the end of Stage 2, sideband frequencies appear above and below natural frequency
- Spike Energy grows e.g. 0.25-0.50gSE

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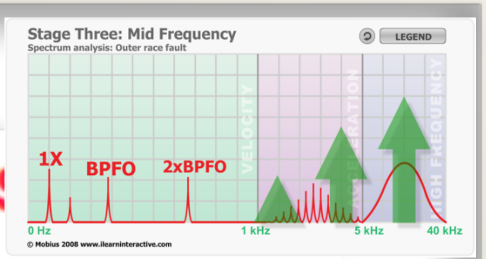
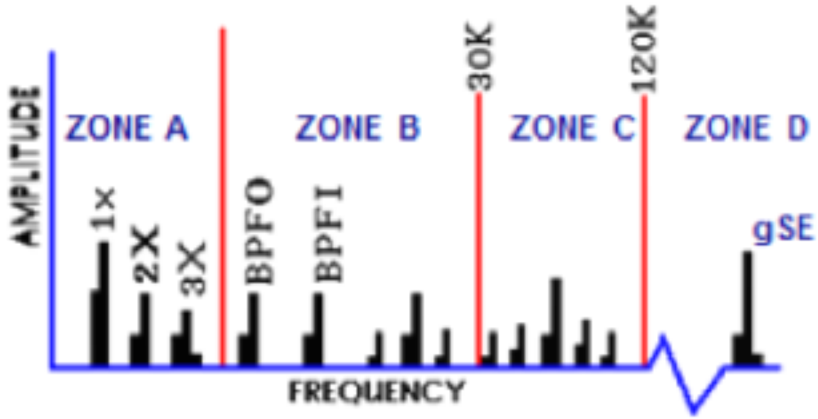


ROLLING ELEMENT BEARINGS
STAGE 3 FAILURE MODE



- Bearing defect frequencies and harmonics appear
- Many defect frequency harmonics appear with wear the number of sidebands grow
- Wear is now visible and may extend around the periphery of the bearing
- Spike Energy increases to between 0.5 -1.0 gSE

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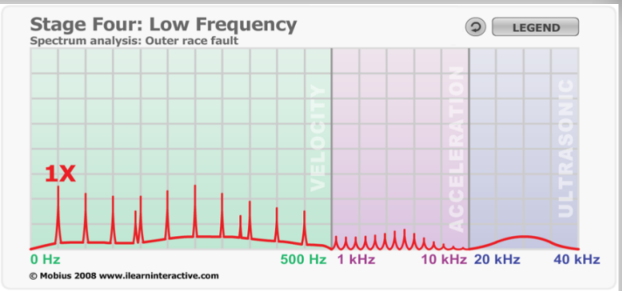
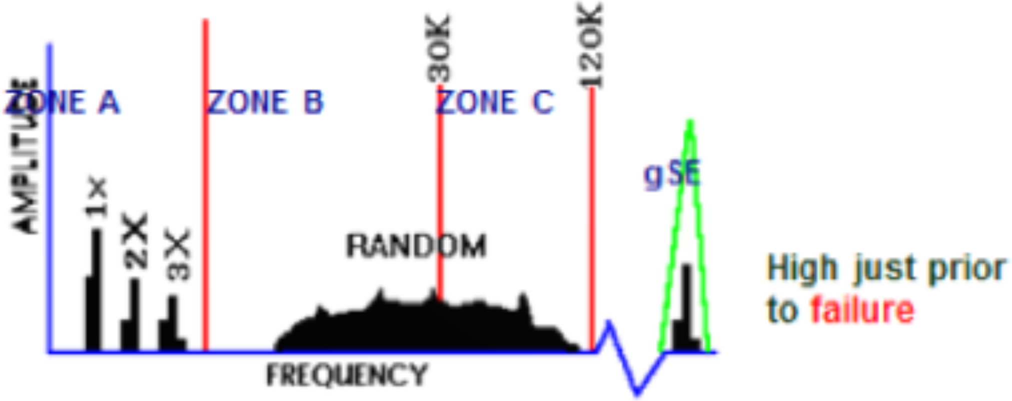


ROLLING ELEMENT BEARING STAGE 4 FAILURE MODE



- Discreet bearing defect frequencies disappear and are replaced by random broad band vibration in the form of a noise floor
- Towards the end, even the amplitude at 1 X RPM is effected
- High frequency noise floor amplitudes and Spike Energy may in fact decrease
- Just prior to failure gSE may rise to high levels

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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

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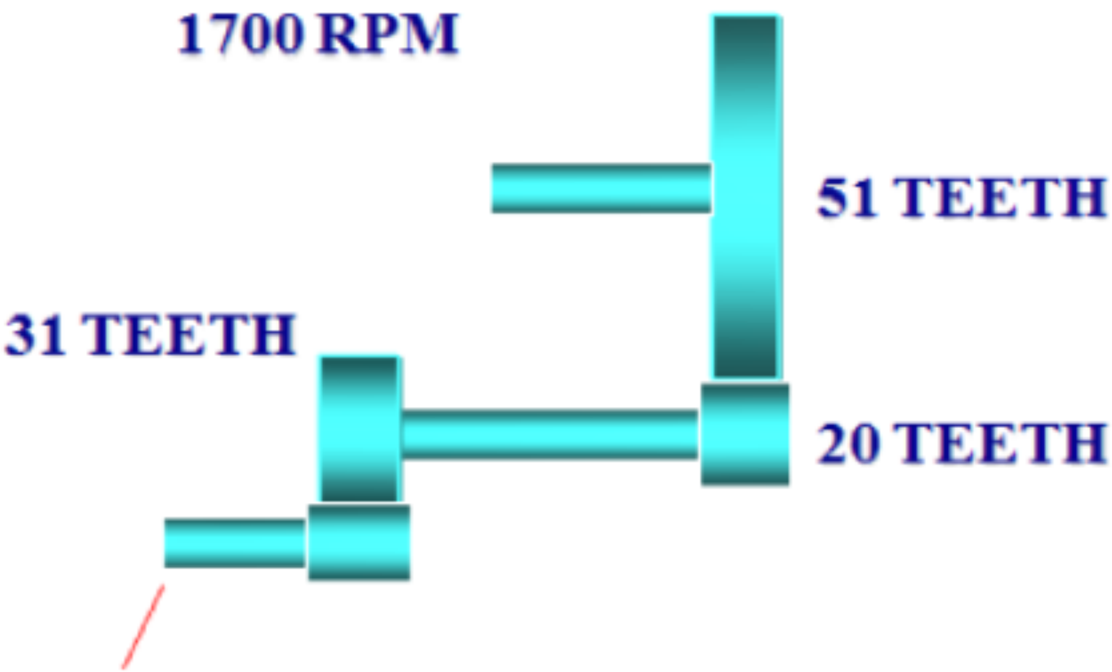
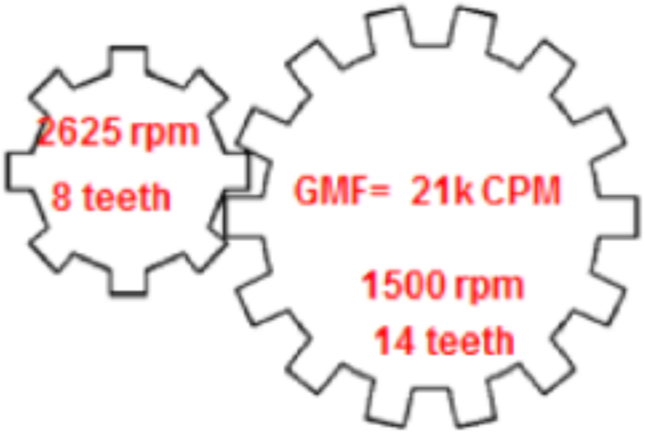
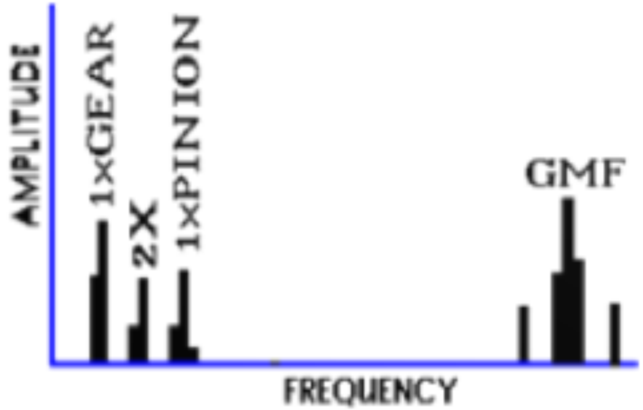
CALCULATION OF GEAR MESH FREQUENCIES



8959 RPM -- HOW MANY TEETH ON THIS GEAR?

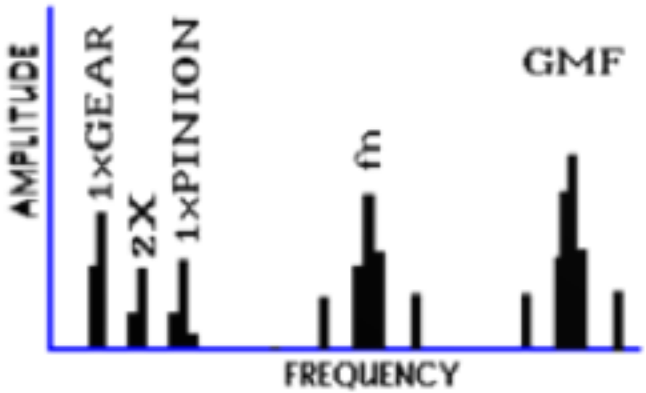
GEARS
NORMAL SPECTRUM





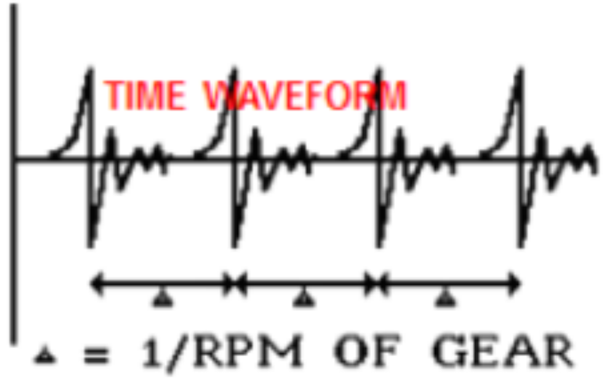
- Normal spectrum shows 1X and 2X and gear mesh frequency GMF
- GMF commonly will have sidebands of running speed
- All peaks are of low amplitude and no natural frequencies are present

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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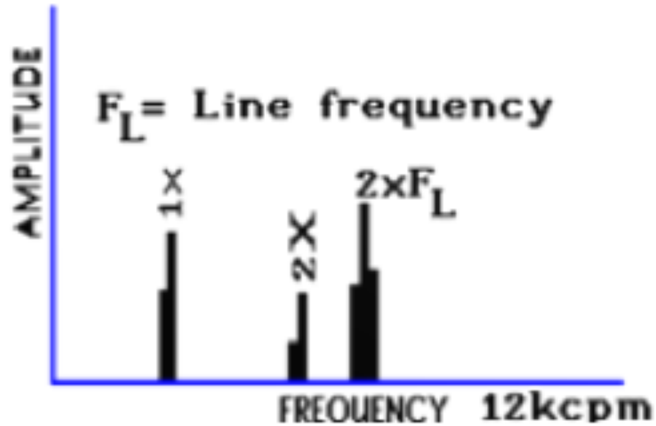
FREQUENCIES PRODUCED BY ELECTRICAL MOTORS.

- 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 (Ns) = $\frac{2xFL}{P}$
- Slip frequency (F_S)= Synchronous speed - Rotor rpm.
- Pole pass frequency (F_P)= Slip Frequency x No of Poles.

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

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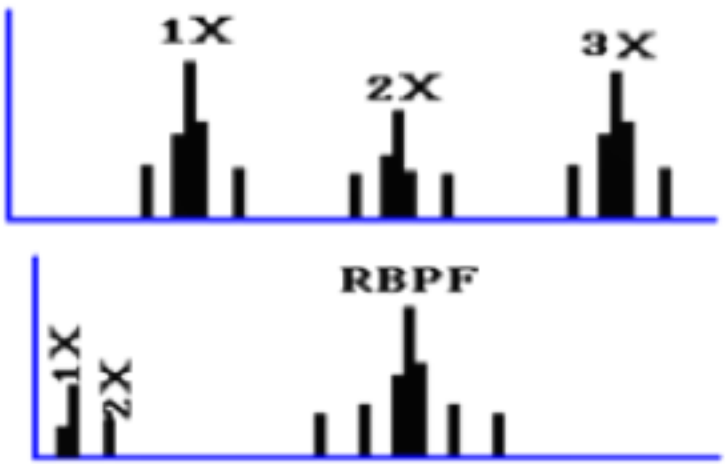
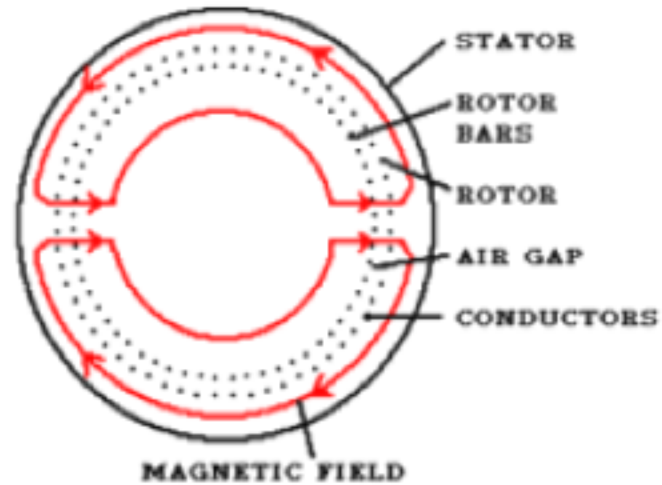
ELECTRICAL PROBLEMS

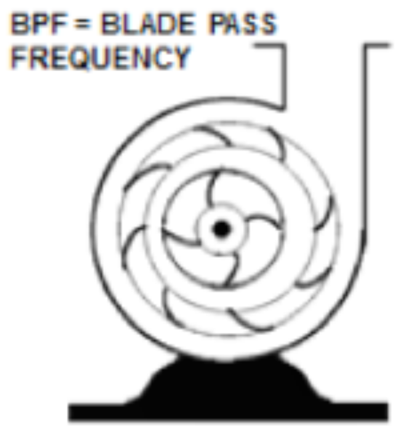
ROTOR PROBLEMS

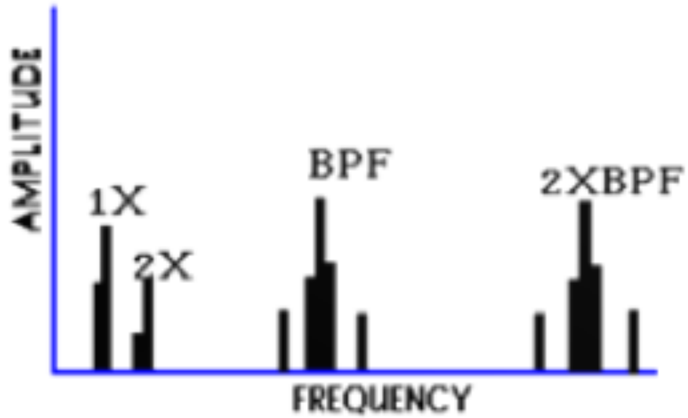




- 1X, 2X, 3X, RPM with pole pass frequency sidebands indicates rotor bar problems.
- 2X line frequency sidebands on rotor bar pass frequency (RBPF) indicates loose rotor bars.
- Often high levels at 2X & 3X rotor bar pass frequency and only low level at 1X rotor bar pass frequency.

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

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The end

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