Machinery Fault Diagnosis

A basic guide to understanding vibration analysis for machinery diagnosis.

LUDECA

Keep it running.

Preface

This is a basic guide to understand vibration analysis for machinery diagnosis. In practice, many variables must be taken into account. PRUFTECHNIK Condition Monitoring and/or LUDECA are not responsible for any incorrect assumptions based on this information.
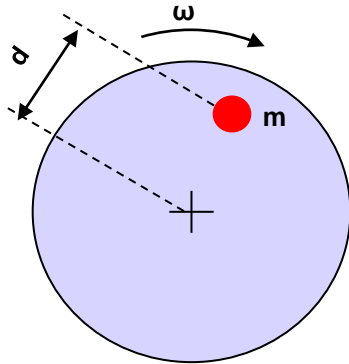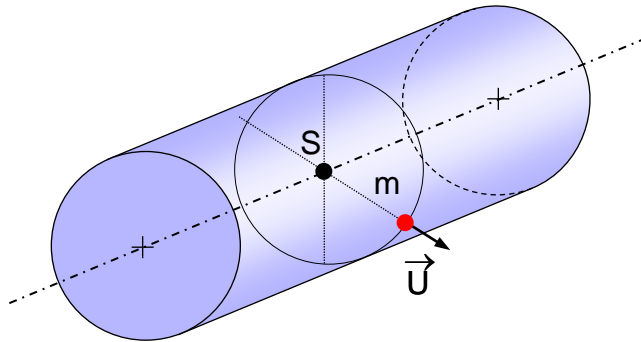
Unbalance

Unbalance is the condition when the geometric centerline of a rotation axis doesn't coincide with the mass centerline.

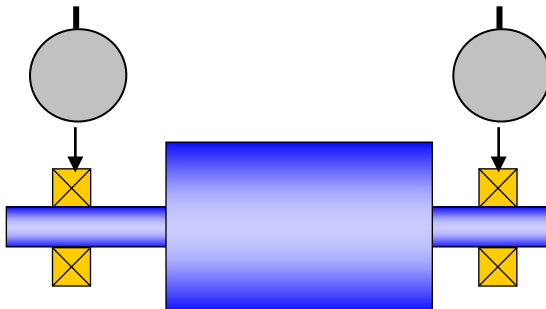$$F_{unbalance} = m \, d \, \omega^2$$

1X
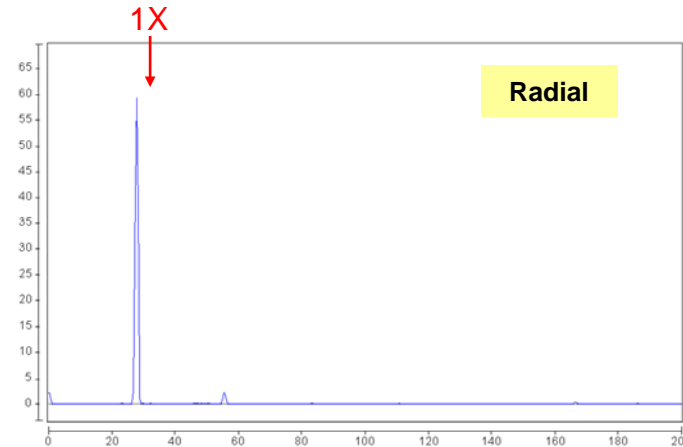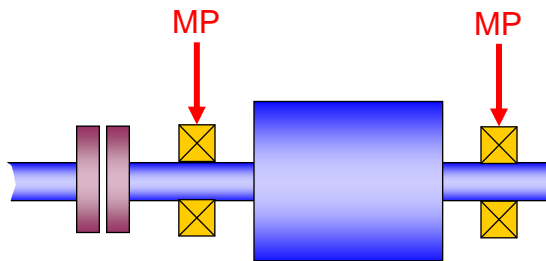
Radial

A pure unbalance will generate a signal at the rotation speed and predominantly in the radial direction.
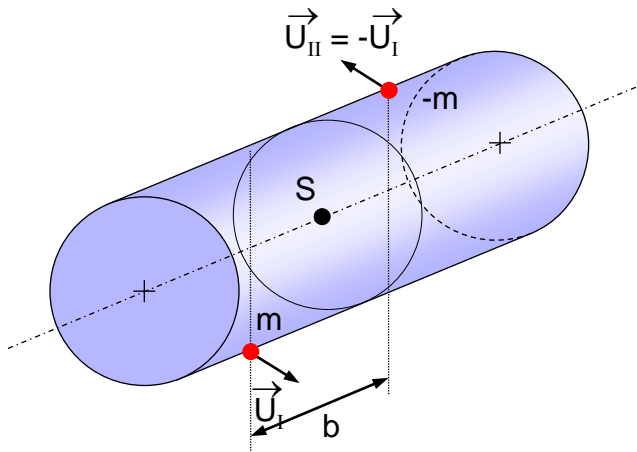
Static Unbalance

Static unbalance is caused by an unbalance mass out of the gravity centerline.

The static unbalance is seen when the machine is not in operation, the rotor will turn so the unbalance mass is at the lowest position.
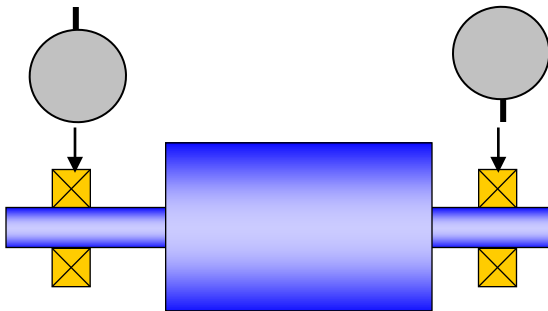
The static unbalance produces a vibration signal at 1X, radial predominant, and in phase signals at both ends of the rotor.

$$\vec{U}_{II} = -\vec{U}_{I}$$
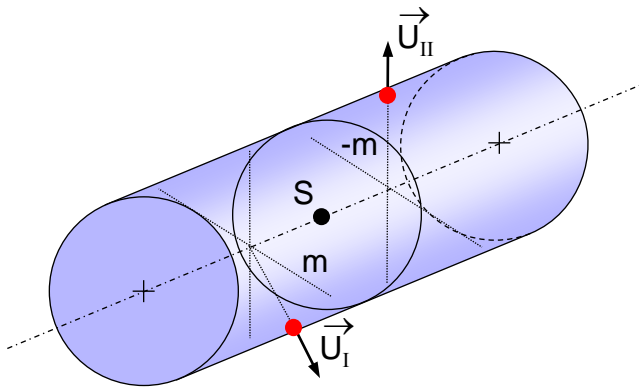
-m

S

+

m

$\vec{U}_I$

b

Pure couple unbalance is caused by two identical unbalance masses located at 180° in the transverse area of the shaft.
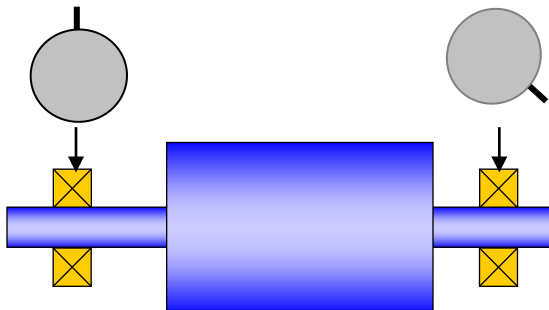
Pure couple unbalance may be statically balanced.

When rotating pure couple unbalance produces a vibration signal at 1X, radial predominant and in opposite phase signals in both ends of the shaft.

Dynamic Unbalance

**Dynamic unbalance is static and couple unbalance at the same time.**

In practice, dynamic unbalance is the most common form of unbalance found.
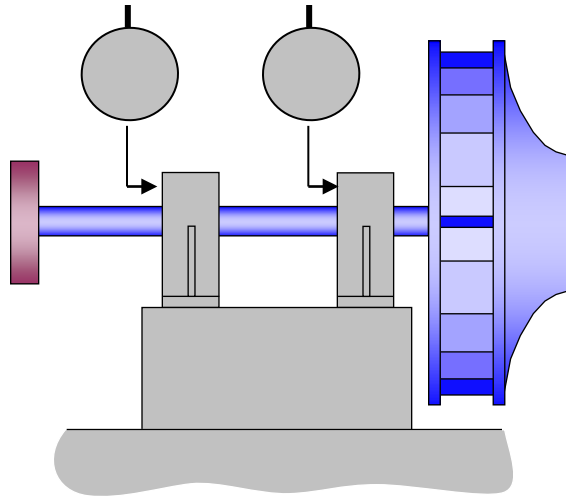
When rotating the dynamic unbalance produces a vibration signal at 1X, radial predominant and the phase will depend on the mass distribution along the axis.

Documentation of balancing

Frequency spectra before/after balancing and balancing diagram.

Balancing diagram

.. after balancing

before ..
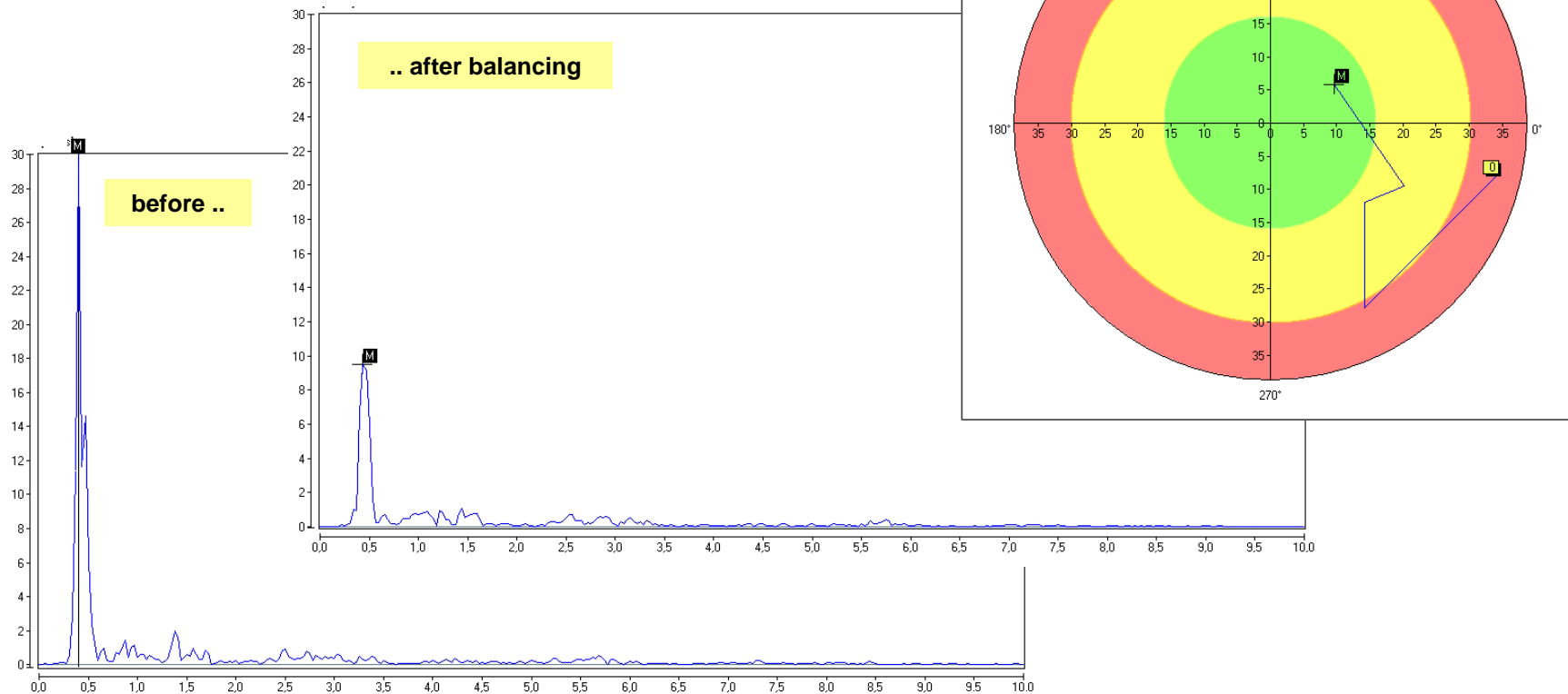
Overhung Rotors

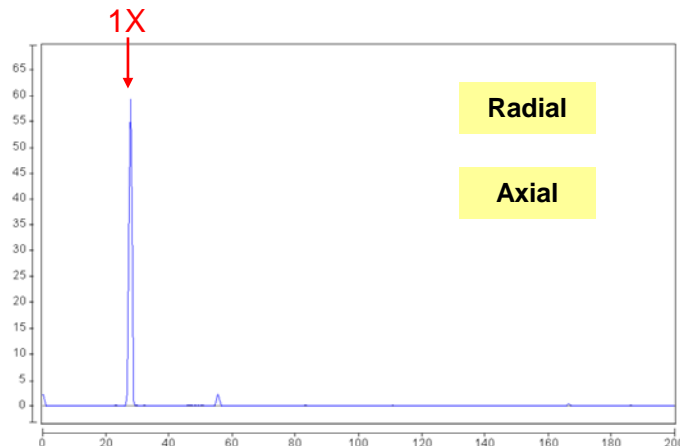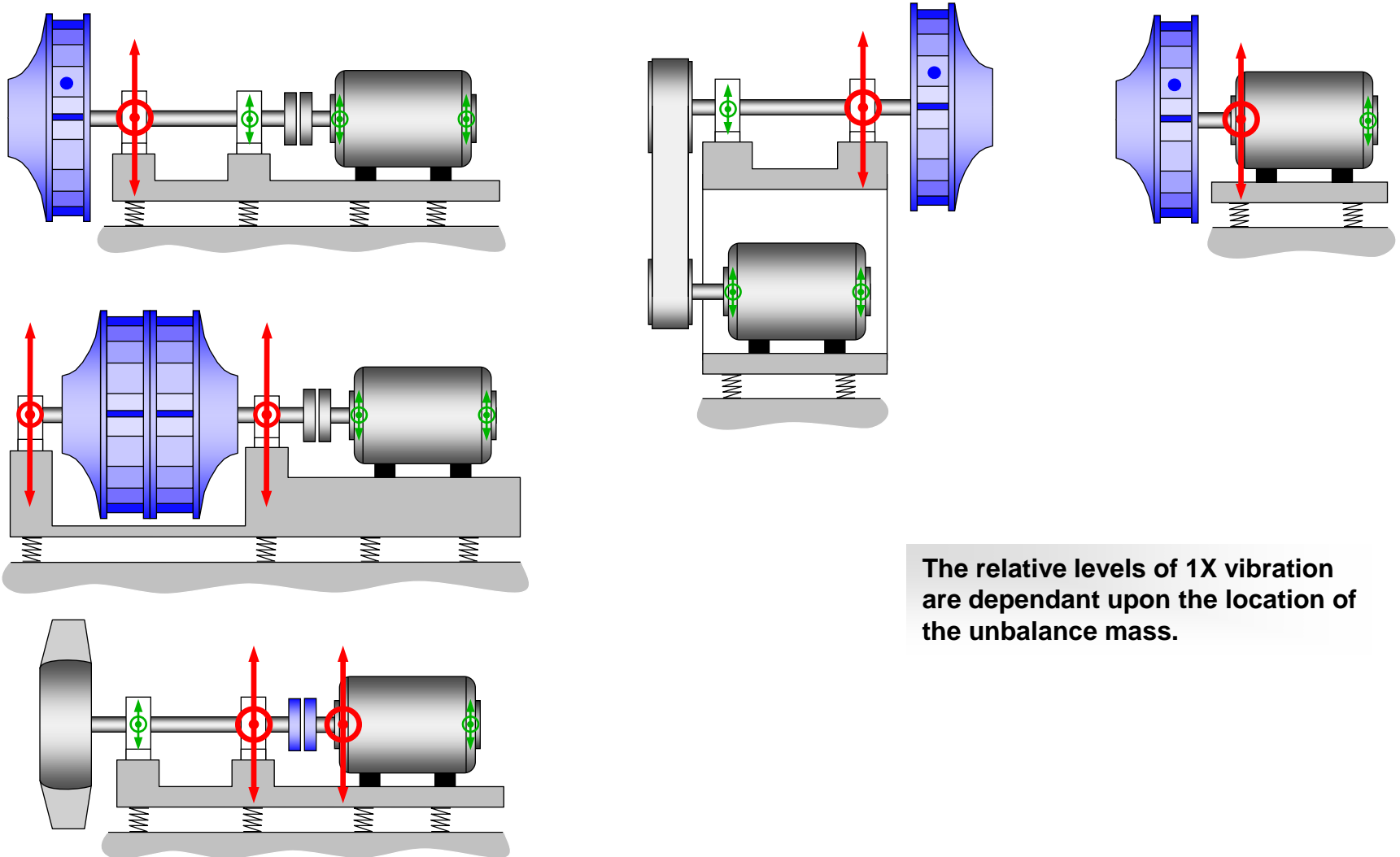**A special case of dynamic unbalance can be found in overhung rotors.**

The unbalance creates a bending moment on the shaft.
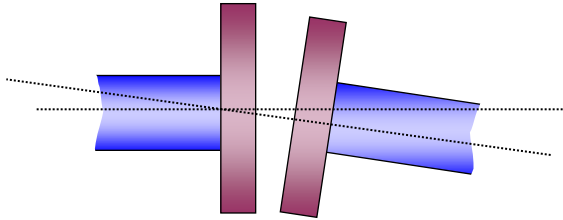

1X

Radial

Axial

Dynamic unbalance in overhung rotors causes high 1X levels in radial and axial direction due to bending of the shaft. The axial bearing signals in phase may confirm this unbalance.
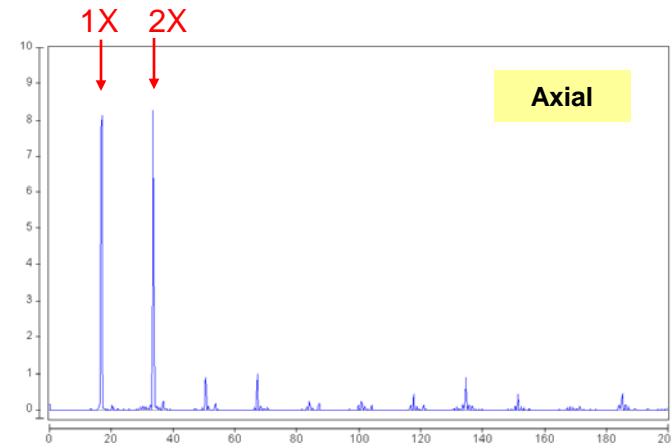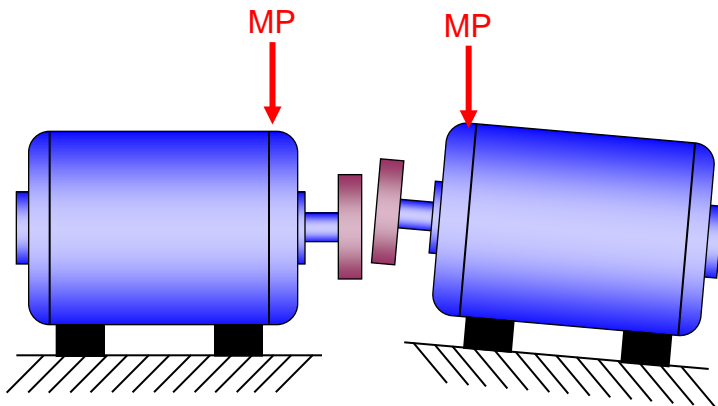
The relative levels of 1X vibration are dependant upon the location of the unbalance mass.

Misalignment

Misalignment is the condition when the geometric centerline of two coupled shafts are not co-linear along the rotation axis of both shafts at operating condition.


A 1X and 2X vibration signal predominant in the axial direction is generally the indicator of a misalignment between two coupled shafts.

Angular Misalignment
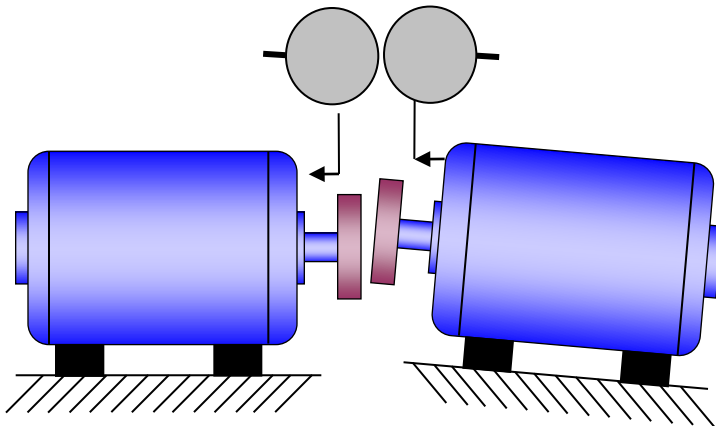
Angular misalignment is seen when the shaft centerlines coincide at one point along the projected axis of both shafts.

The spectrum shows high axial vibration at 1X plus some 2X and 3X with 180° phase difference across the coupling in the axial direction.

These signals may be also visible in the radial direction at a lower amplitude and in phase.


1X 2X 3X
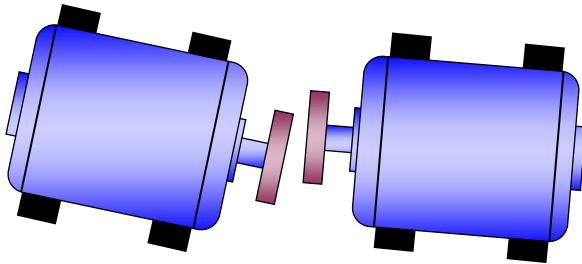
Axial
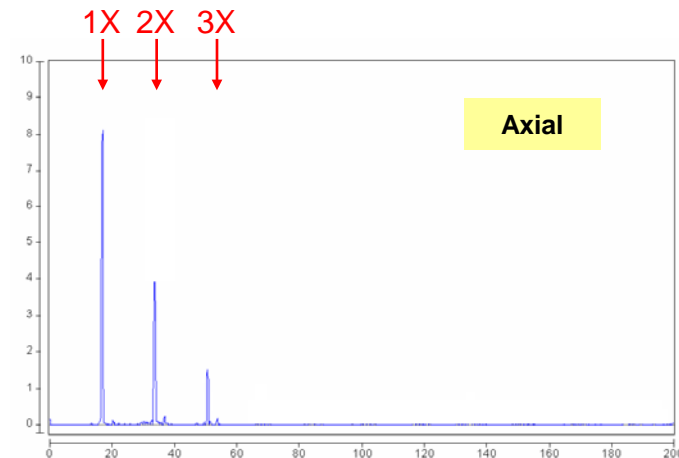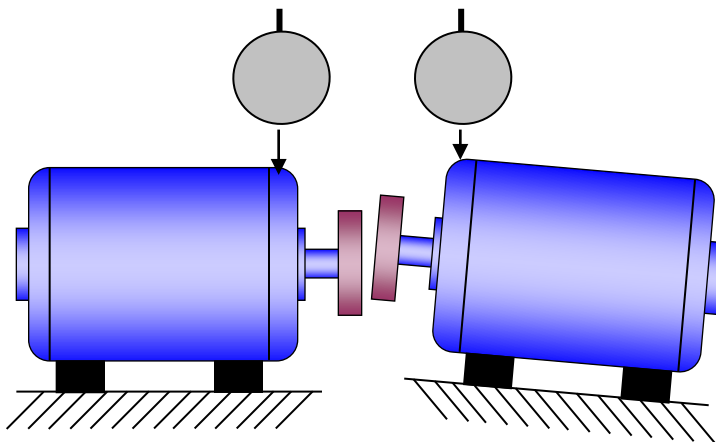
Parallel Misalignment
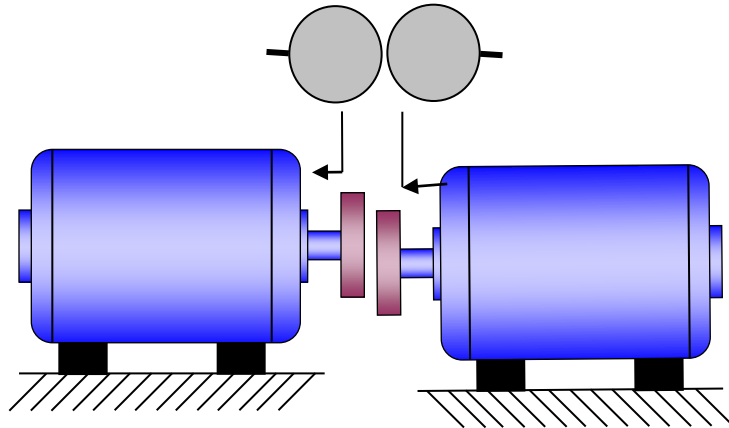
**Parallel misalignment is produced when the centerlines are parallel but offset .**

The spectrum shows high radial vibration at 2X and a lower 1X with 180° phase difference across the coupling in the radial direction.

These signals may be also visible in the axial direction in a lower amplitude and 180° phase difference across the coupling in the axial direction.

1X 2X 3X
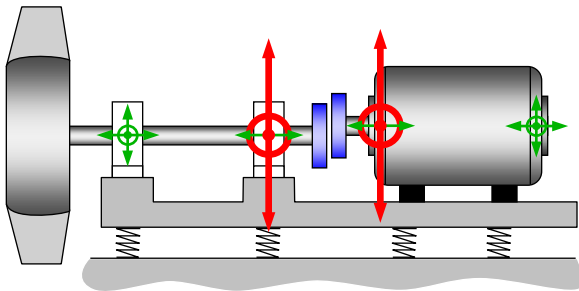
Radial

Misalignment Diagnosis Tips
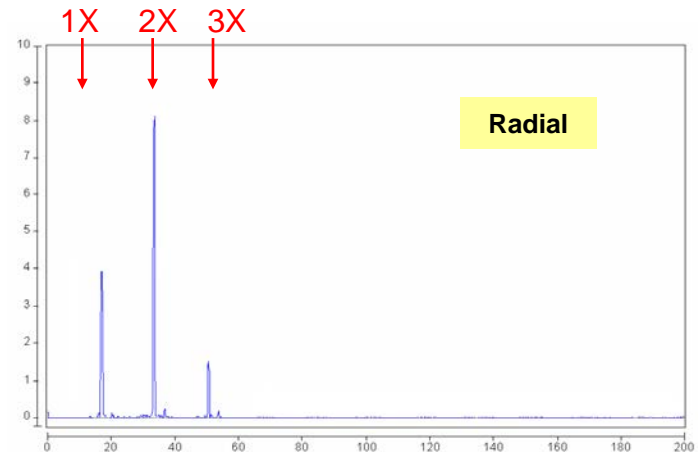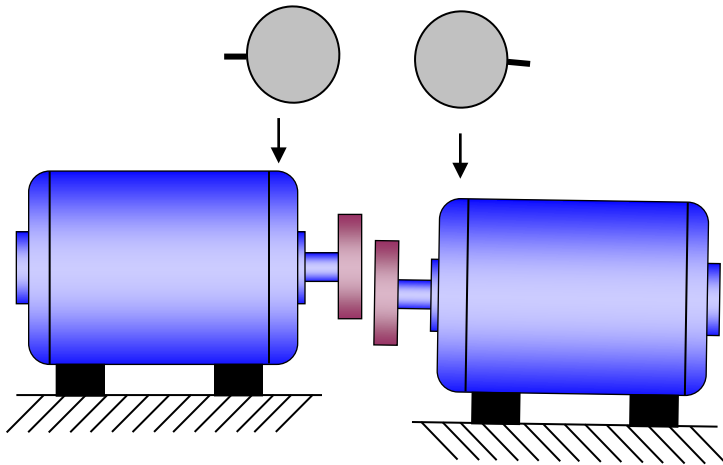
In practice, alignment measurements will show a combination of parallel and angular misalignment.
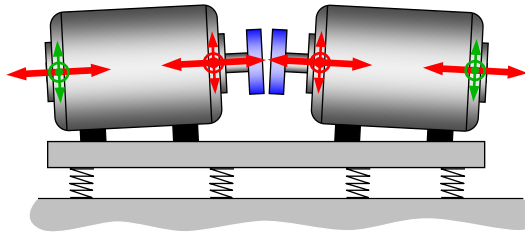
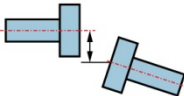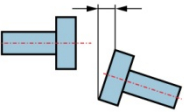Diagnosis may show both a 2X and an increased 1X signal in the axial and radial readings.
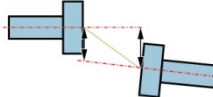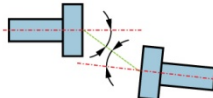
The misalignment symptoms vary depending on the machine and the misalignment conditions.

The misalignment assumptions can be often distinguished from unbalance by:

• Different speeds testing

• Uncoupled motor testing

Temperature effects caused by thermal growth should also be taken into account when assuming misalignment is the cause of increased vibration.

Alignment Tolerance Table

Short "flexible" couplings		Alignment Tolerance inch [mils]	
	RPM	acceptable	excellent
Offset	900	6.0	3.0
	1200	4.0	2.5
	1800	3.0	2.0
	3600	1.5	1.0
Angularity (gap difference at coupling edge per 10" diameter)	900	10.0	7.0
	1200	8.0	5.0
	1800	5.0	3.0
	3600	3.0	2.0
Spacer shafts and membrane (disk) couplings **Offset** (per inch spacer length)	900	2.0	1.2
	1200	1.5	0.9
	1800	1.0	0.6
	3600	0.5	0.3
Angularity [mrad]	900	2.0 [mrad]	1.2
	1200	1.5	0.9
	1800	1.0	0.6
	3600	0.5	0.3
Soft foot	any	2.0 mils	

The suggested alignment tolerances shown above are general values based upon experience and should not be exceeded.
They are to be used only if existing in-house standards or the manufacturer of the machine or coupling prescribe no other values.
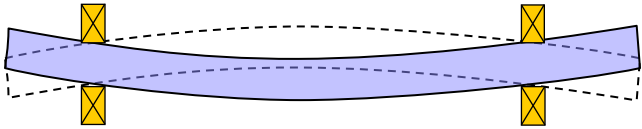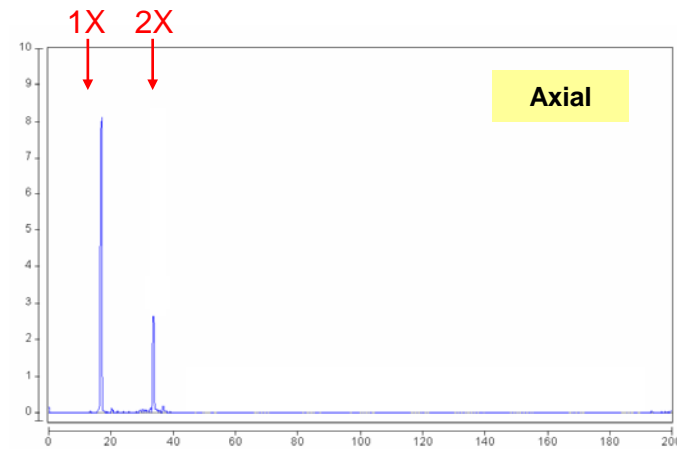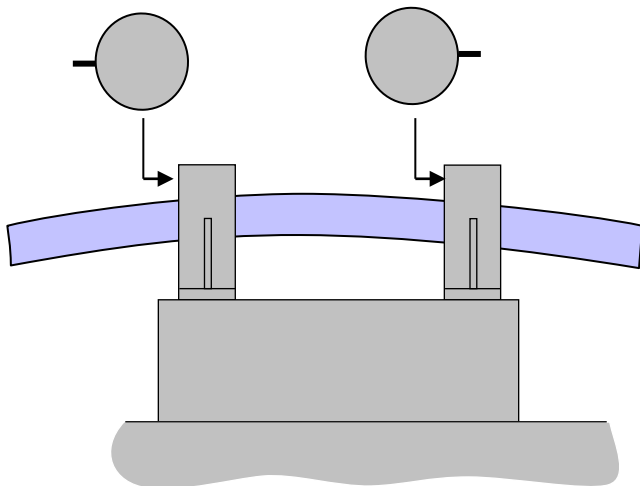
Shaft Bending

A shaft bending is produced either by an axial asymmetry of the shaft or by external forces on the shaft producing the deformation.
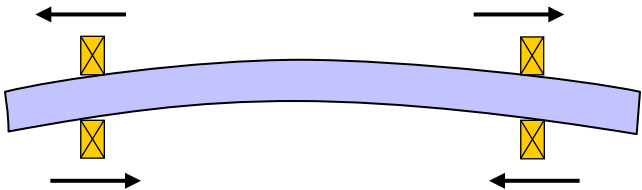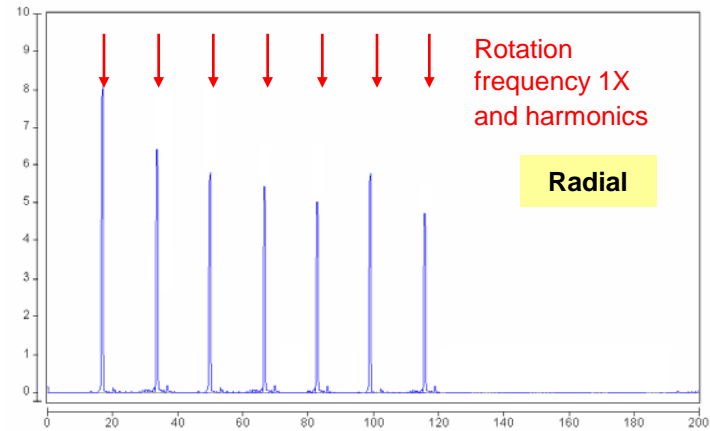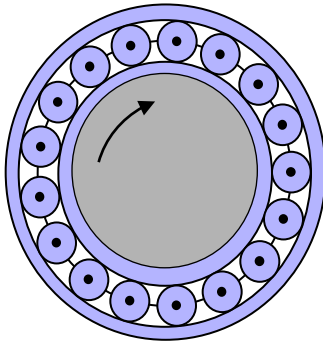

A bent shaft causes axial opposed forces on the bearings identified in the vibration spectrum as 1X in the axial vibration.

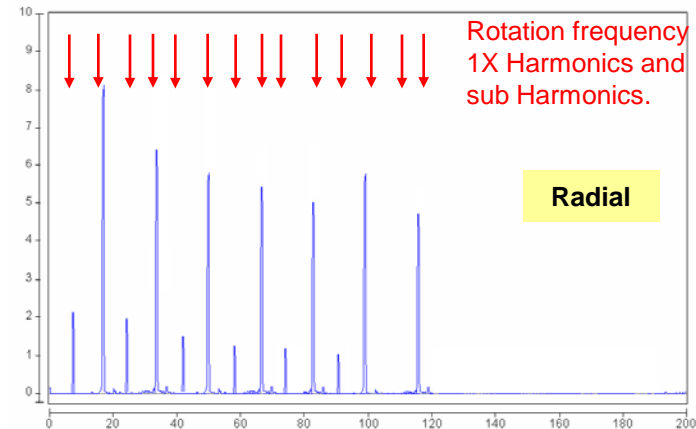**2X and radial readings can also be visible.**
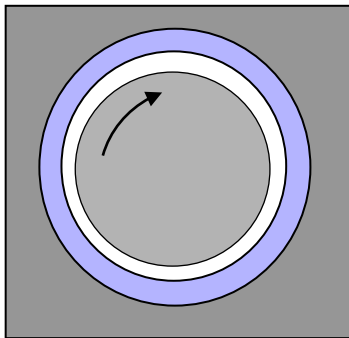
Rotating Looseness

Rotating looseness is caused by an excessive clearance between the rotor and the bearing

Rolling element bearing:

Rotation frequency 1X and harmonics

Radial

Journal bearing:
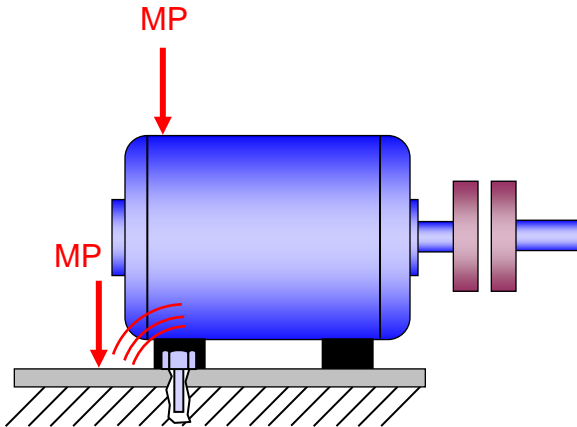

Rotation frequency 1X Harmonics and sub Harmonics.

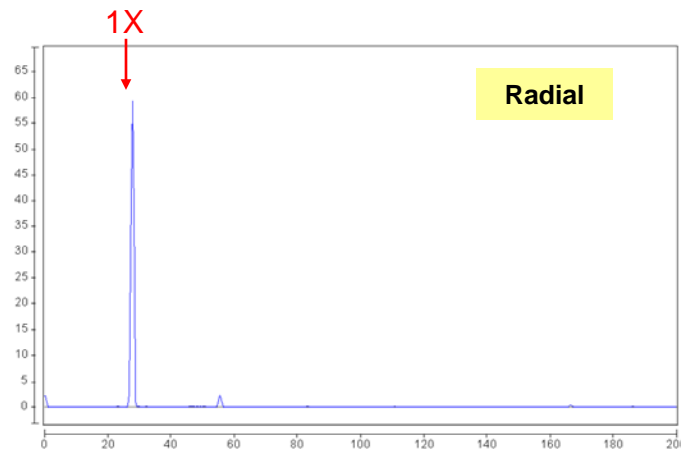**Radial**

Structural Looseness

Structural looseness occurs when the machine is not correctly supported by, or well fastened to its base.

MP

MP

MP MP
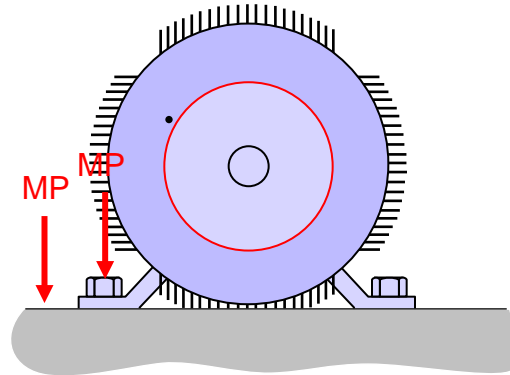
- **Poor mounting**
- **Poor or cracked base**
- **Poor base support**
- **Warped base**

1X

Radial

Structural looseness may increase vibration amplitudes in any measurement direction. Increases in any vibration amplitudes may indicate structural looseness.

Measurements should be made on the bolts, feet and bases in order to see a change in the amplitude and phase. A change in amplitude and 180° phase difference will confirm this problem.

Resonance

Resonance is a condition caused when a forcing frequency coincides (or is close) to the natural frequency of the machine's structure. The result will be a high vibration.

1st form of natural flexure

2nd form of natural flexure

3rd form of natural flexure
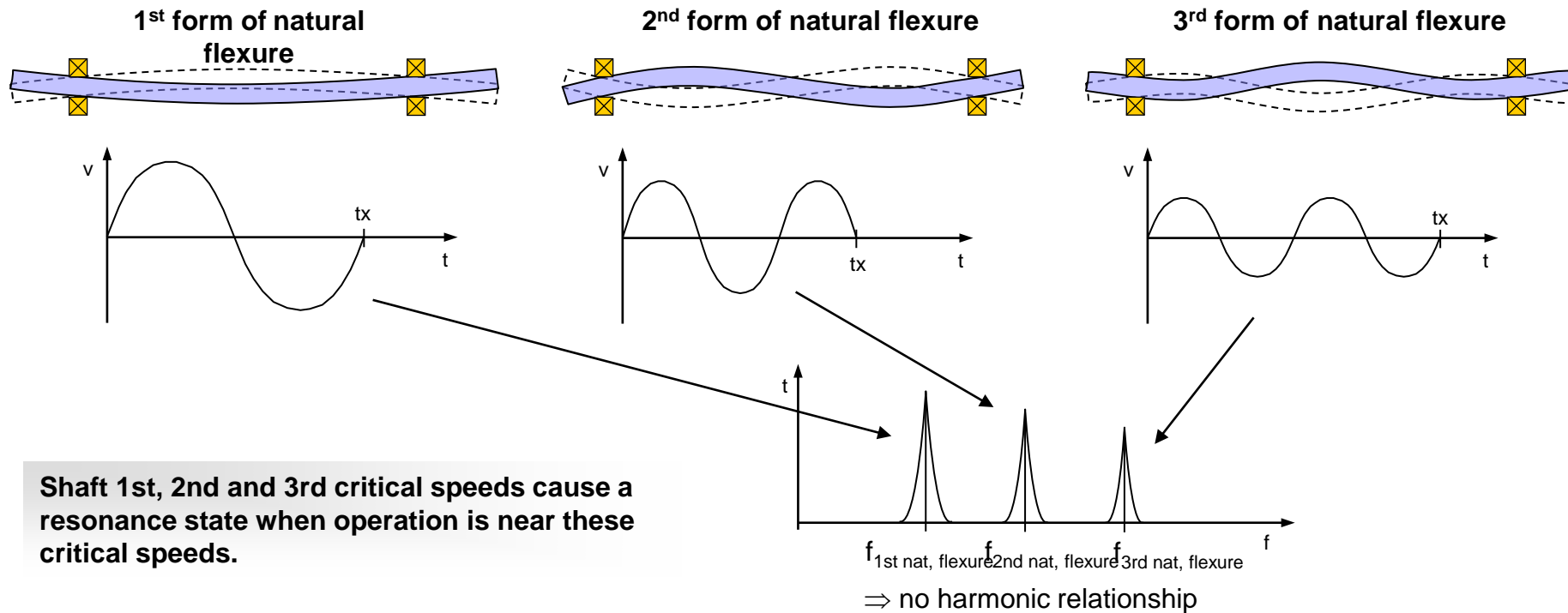
Shaft 1st, 2nd and 3rd critical speeds cause a resonance state when operation is near these critical speeds.

$f_{\text{1st nat, flexure}}$ $f_{\text{2nd nat, flexure}}$ $f_{\text{3rd nat, flexure}}$

$\Rightarrow$ no harmonic relationship

Resonance

- **Resonance can be confused with other common problems in machinery.**

- **Resonance requires some additional testing to be diagnosed.**

MP   MP

Resonance
Step-up

rev/min

Grad

Phase jump
by 180°

rev/min
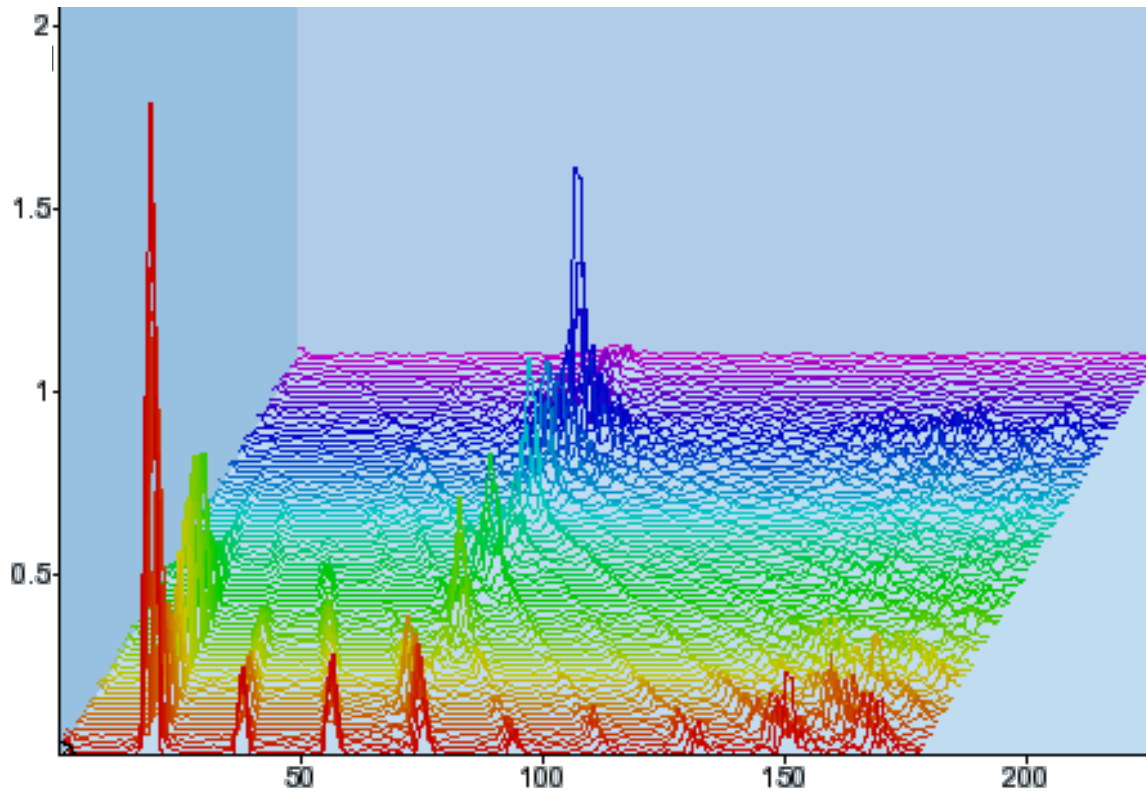
$\varphi_1 = 240°$

$\varphi_2 = 60...80°$
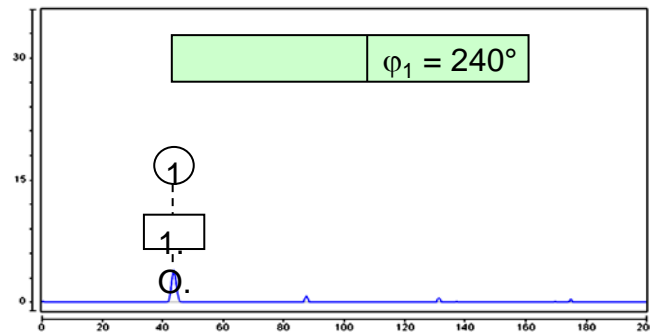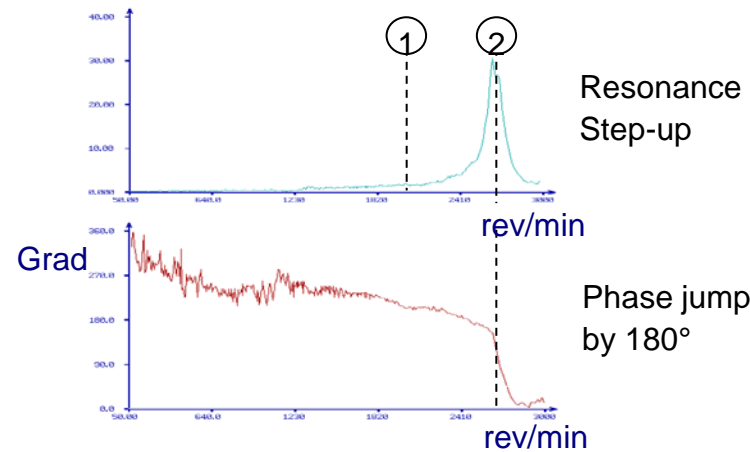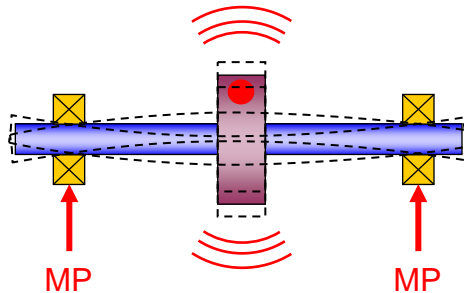
Amplitude at rotation frequency fn by residual unbalance on rigid rotor.
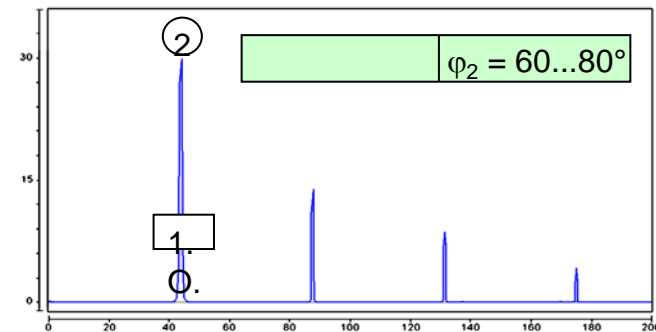
Strong increase in amplitude of the rotation frequency fn at the point of resonance, step-up dependent on the excitation (unbalanced condition) and damping.
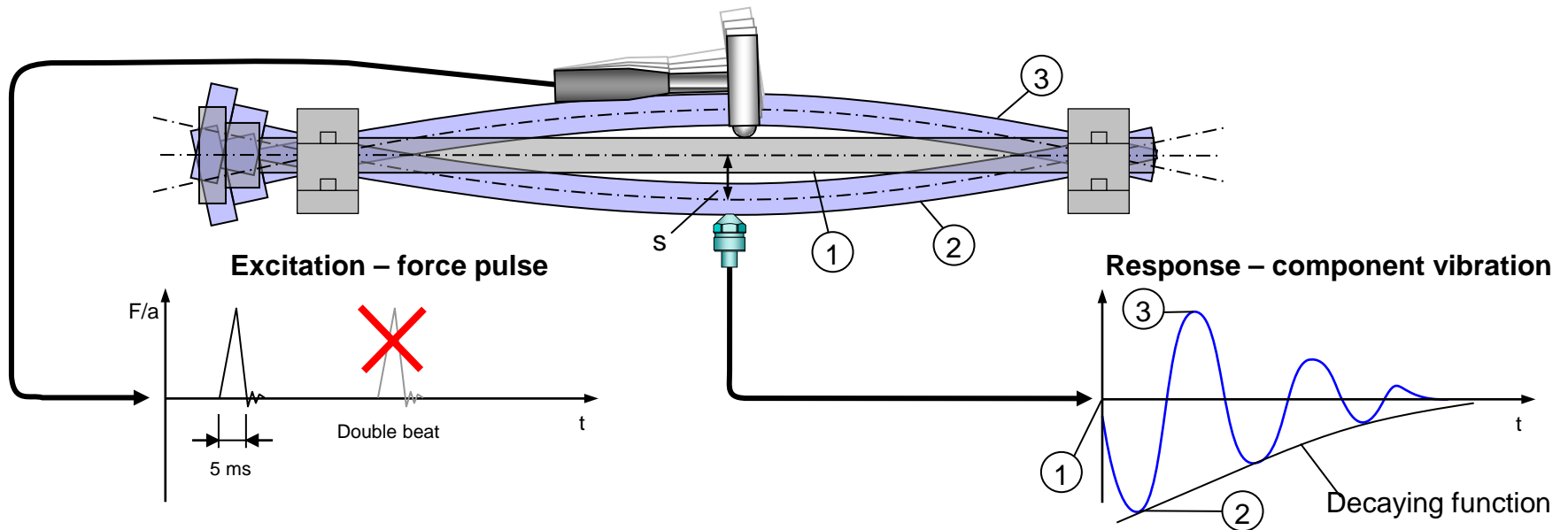
Resonance Diagnosing Tests
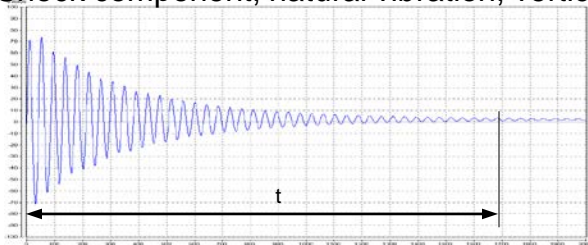
Run Up or Coast Down Test:

- **Performed when the machine is turned on or turned off.**

- **Series of spectra at different RPM.**

- **Vibration signals tracking may reveal a resonance.**

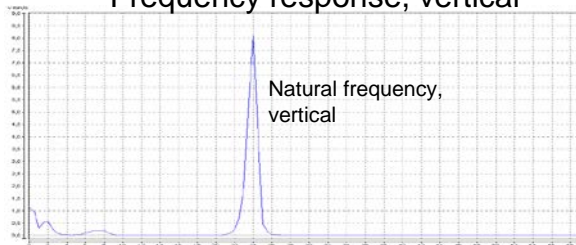**The use of tachometer is optional and the data collector must support this kind of tests.**

Bump Test:

Excitation – force pulse

F/a

Double beat

5 ms

t

Response – component vibration

Decaying function

t

Shock component, natural vibration, vertical

t

Frequency response, vertical

Natural frequency, vertical

Frequency response, horizontal

2nd mod.

1st mod.

Natural frequency, horizontal
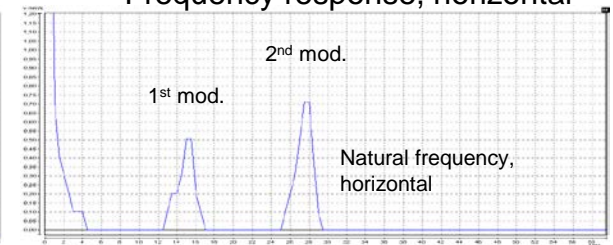
Journal Bearings

Journal bearings provides a very low friction surface to support and guide a rotor through a cylinder that surrounds the shaft and is filled with a lubricant preventing metal to metal contact.
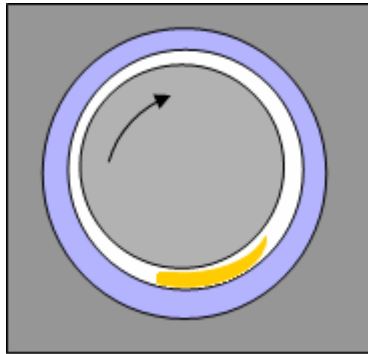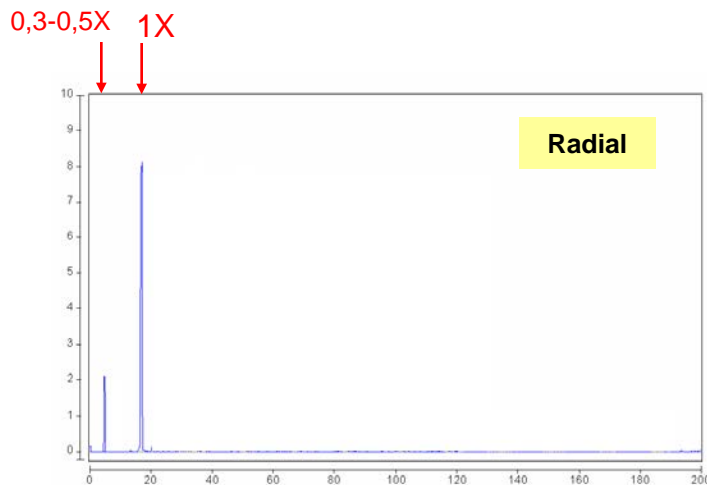

High vibration damping due to the oil film:

• **High frequencies signals may not be transmitted.**

• **Displacement sensor and continuous monitoring recommended**
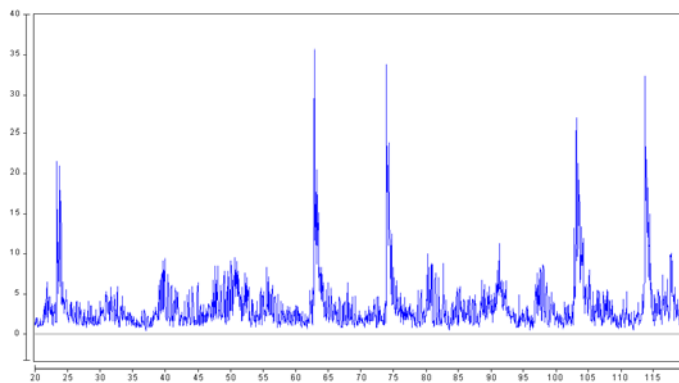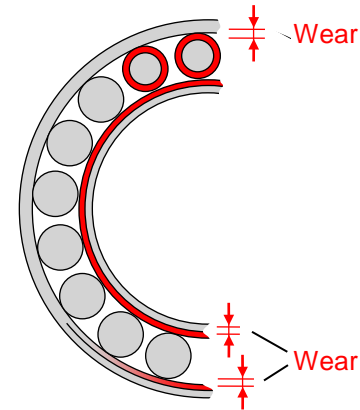
Clearance problems (rotating mechanical looseness).

0,3-0,5X   1X

Radial

Oil whirl

• **Oil-film stability problems.**

• **May cause 0.3-0.5X component in the spectrum.**

Rolling Element Bearings

1. Wear:

- **Lifetime exceeded**
- **Bearing overload**
- **Incorrect assembly**
- **Manufacturing error**
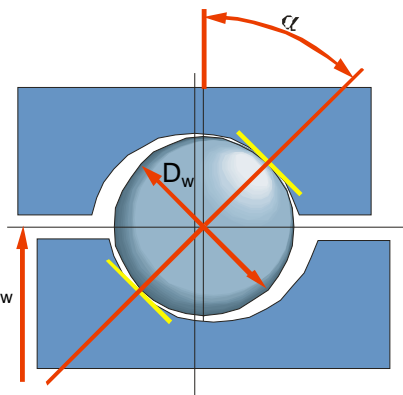- **Insufficient lubrication**

Wear

Wear

The vibration spectrum has a higher noise level and bearing characteristic frequencies can be identified.

Increased level of shock pulses.

2. Race Damage:
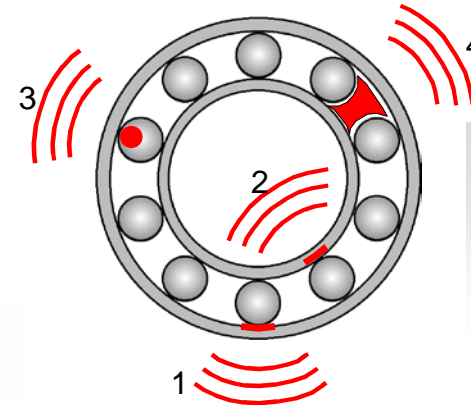
Roller bearing geometry and damage frequencies:

1 - Outer race damage

2 - Inner race damage

3 - Rolling element damage

4 - Cage damage

α **Angle of contact**

D Arc diameter

d Rolling element diameter

Z Number of rolling elements

n Shaft RPM

Example of rollover frequencies:

Ball bearing SKF 6211

RPM, n = 2998 rev/min

Ball pass frequency, outer race:

$$\text{BPFO} = \frac{Z \cdot n}{2 \cdot 60} \left(1 - \frac{d}{D} \cos \alpha \right)$$

Ball pass frequency, inner race:

$$\text{BPFI} = \frac{Z \cdot n}{2 \cdot 60} \left(1 + \frac{d}{D} \cos \alpha \right)$$

Ball spin frequency:

$$\text{BSF} = \frac{D \cdot n}{d \cdot 60} \left(1 - \left[\frac{d}{D} \cos \alpha \right]^2 \right)$$

Fundamental train frequency:

$$\text{TFT} = \frac{n}{2 \cdot 60} \left(1 - \frac{d}{D} \cos \alpha \right)$$

Dimensions	Rollover frequencies
d = 77.50 mm	BPFO = n / 60 · 4.0781 = 203.77 Hz
D = 14.29 mm	BPFI = n / 60 · 5.9220 = 295.90 Hz
Z = 10	2·f_w = n / 60 · 5.2390 = 261,77 Hz
α = 0	f_K = n / 60 · 0.4079 = 20.38 Hz

Rolling Element Bearings

Outer race damage:
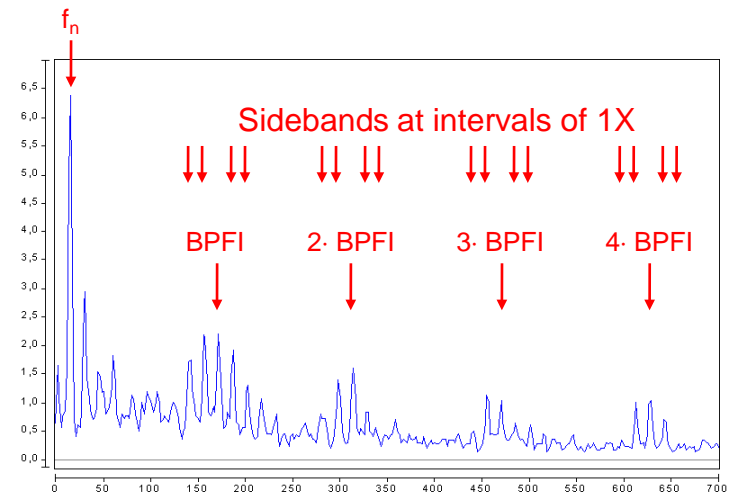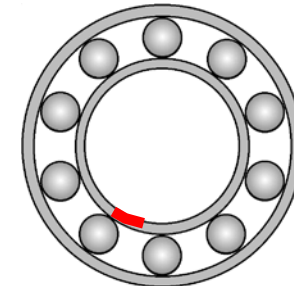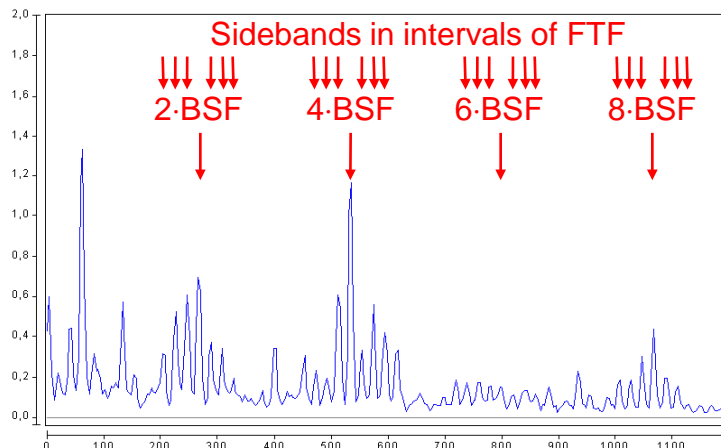(Ball passing frequency, outer range BPFO)

Inner race damage:
(Ball passing frequency, inner range BPFI)
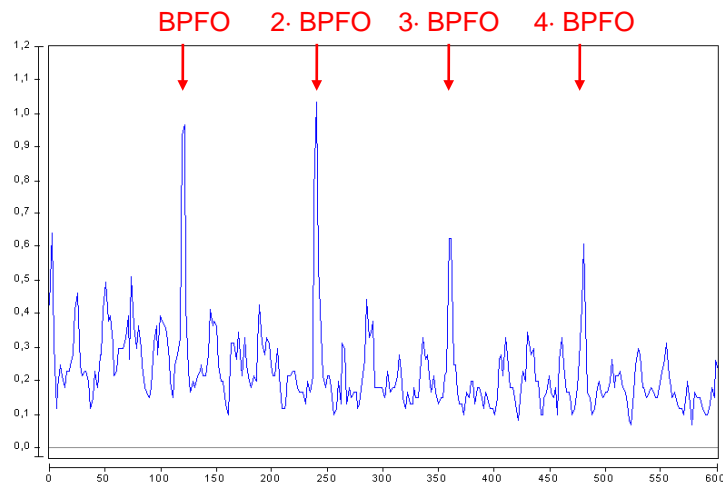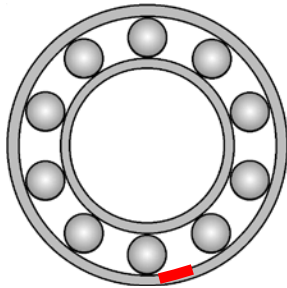
Outer race damage frequency BPFO as well as harmonics clearly visible

**Inner race damage frequency BPFI as well as numerous sidebands at intervals of 1X.**

Rolling Element Bearings

Rolling element damage:
(Ball spin frequency BSF)

Cage damage:
(Fundamental train frequency FTF)

Sidebands in intervals of FTF

$2 \cdot BSF$ $4 \cdot BSF$ $6 \cdot BSF$ $8 \cdot BSF$

FTF and 2nd, 3rd, 4th harmonics
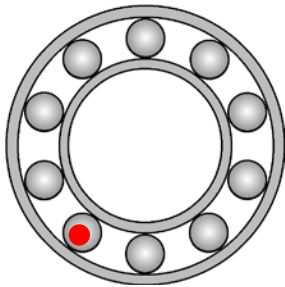
Rolling elements rollover frequency BSF with harmonics as well as sidebands in intervals of FTF
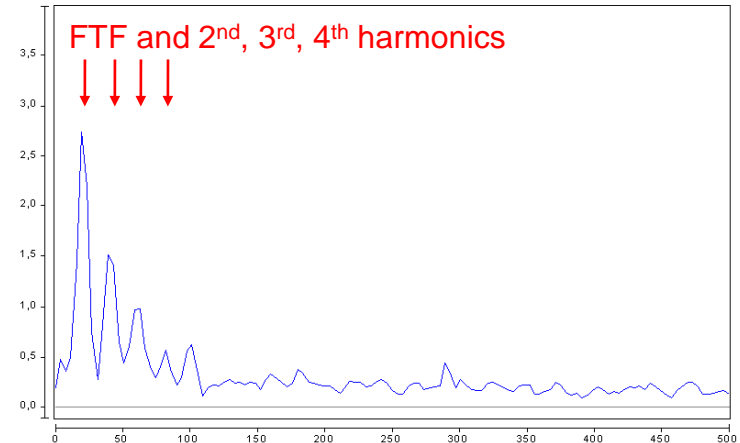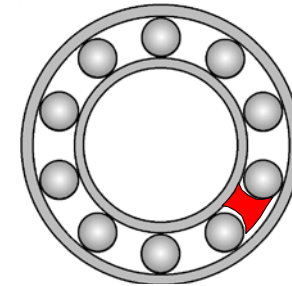
Cage rotation frequency FTF and harmonics visible
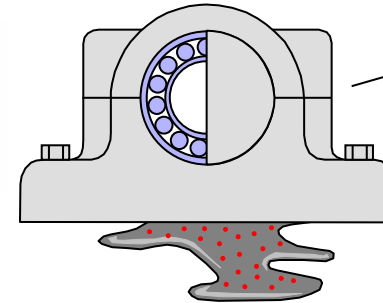
Rolling Element Bearings

Lubrication Problems:

Lubricant contamination

- **Race damage**
- **Defective sealing**
- **Contaminated lubricant used**

Major fluctuation in level of shock pulses and damage frequencies

Insufficient lubrication

- **Insufficient lubricant**
- **Underrating**

→ Subsequent small temperature increase
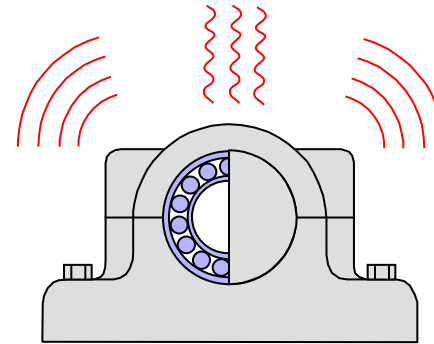
Over-greasing

- **Maintenance error**
- **Defective grease regulator**
- **Grease nipple blocked**

Large temperature increase after lubrication

# Rolling Element Bearings
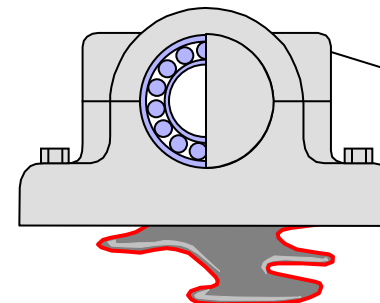
Incorrect mounting.

Bearing rings out of round, deformed.

- **Incorrect installation**
- **Wrong bearing storage**
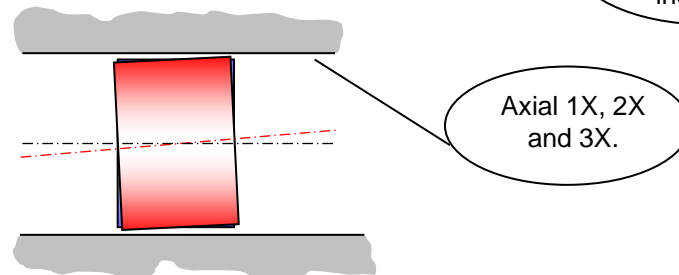- **Shaft manufacturing error**
- **Bearing housing overtorqued.**

Air gap

Shock pulse ↑

Damage frequencies ↑ envelope

Dirt
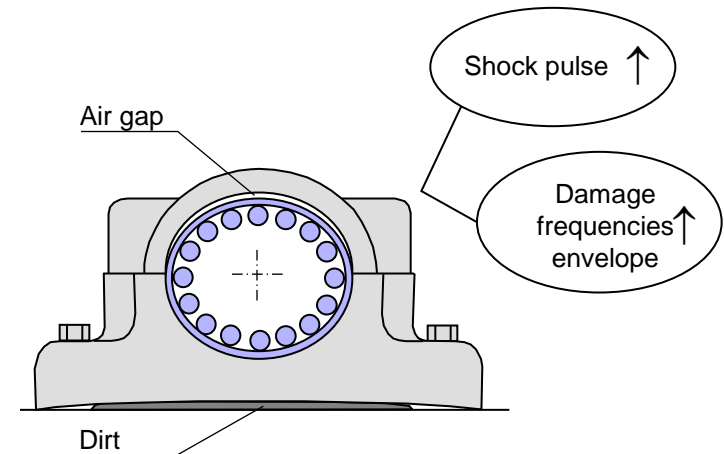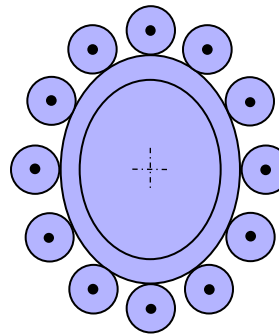
Bearing forces on floating bearing.

- **Incorrect installation**
- **Wrong housing calculation**
- **Manufacturing error in bearing housing**

Fixed bearing

Floating bearing

Severe vibration Bearing temperature increases
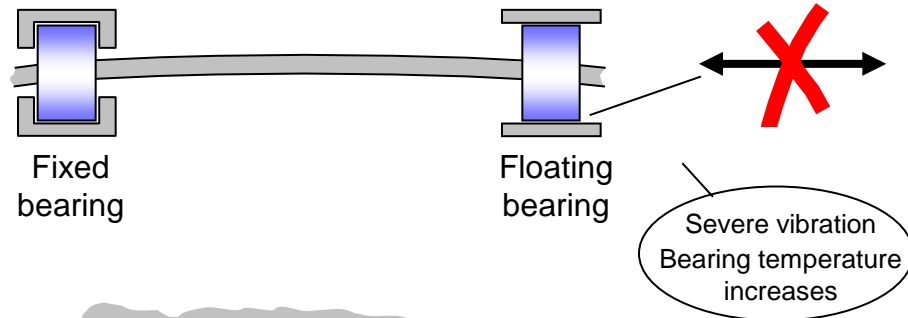
Cocked bearing.

- **Incorrect installation**
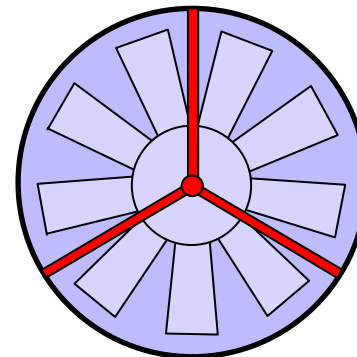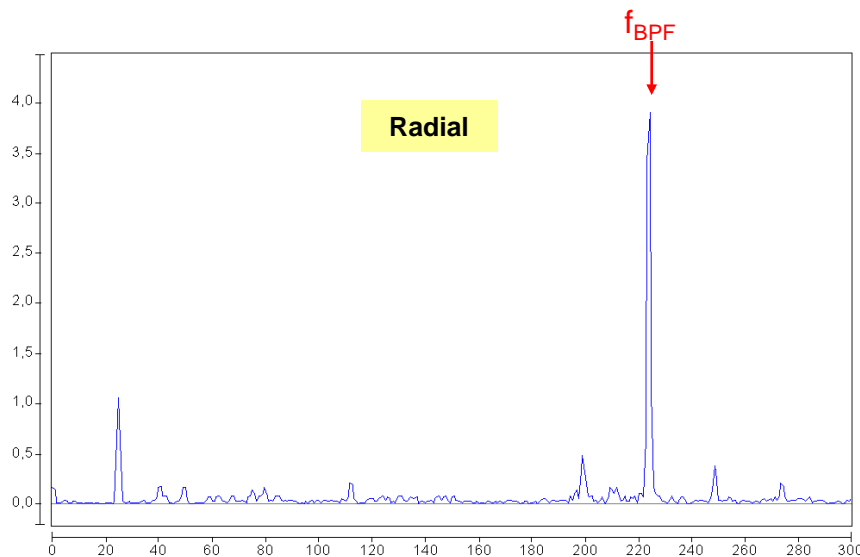
Axial 1X, 2X and 3X.

Blade and Vanes

MP

MP
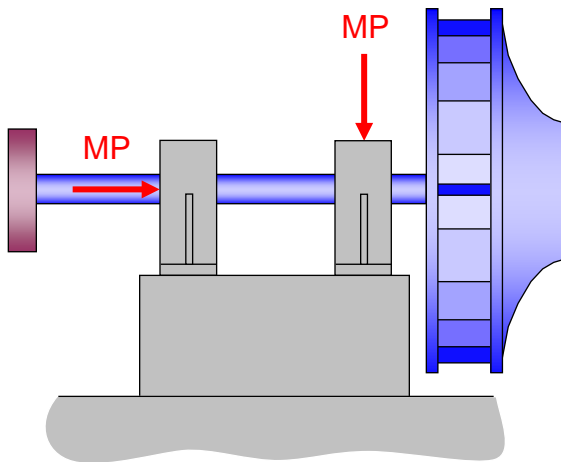
A blade or vane generates a signal frequency called blade pass frequency, f_{BP}:

$$f_{BP} = B_n \cdot N$$

B_n = # of blades or vanes
N = rotor speed in rpm

Identify and trend f_{BP}.

An increase in it and/or its harmonics may be a symptom of a problem like blade-diffuser or volute air gap differences.
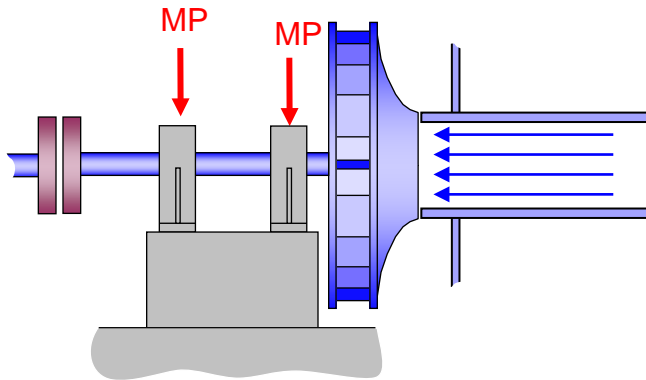
f_{BPF}

Radial

Example characteristic frequency:

3 struts in the intake; x=3.

9 blades; B_n=9.

$$f_{BP} \cdot x \ = \ N \cdot B_n \cdot x$$
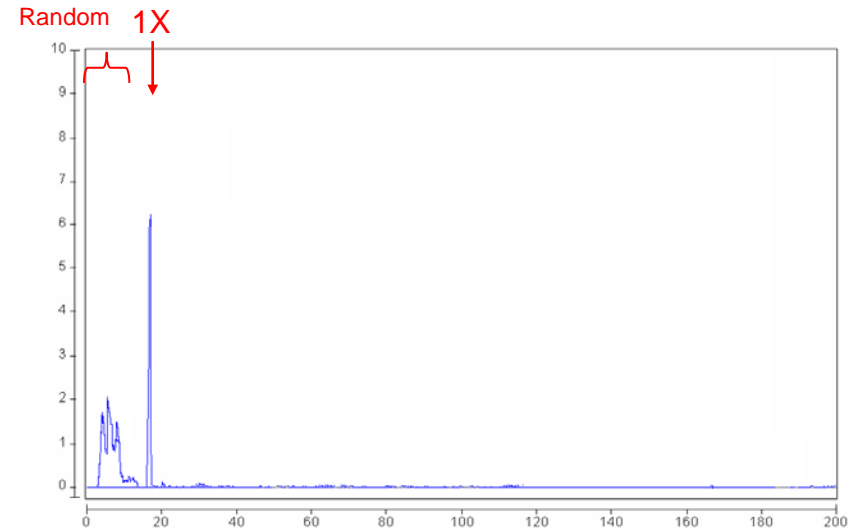
Characteristic frequency = $N \cdot 27$

Aerodynamics and Hydraulic Forces

MP   MP

There are two basic moving fluid problems diagnosed with vibration analysis:

- **Turbulence**
- **Cavitation**

Cavitation:

1X   f$_{BPF}$ Random

Turbulence:

Random   1X

Belt Drive Faults
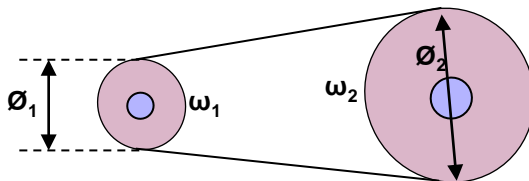
MP   MP

MP   MP
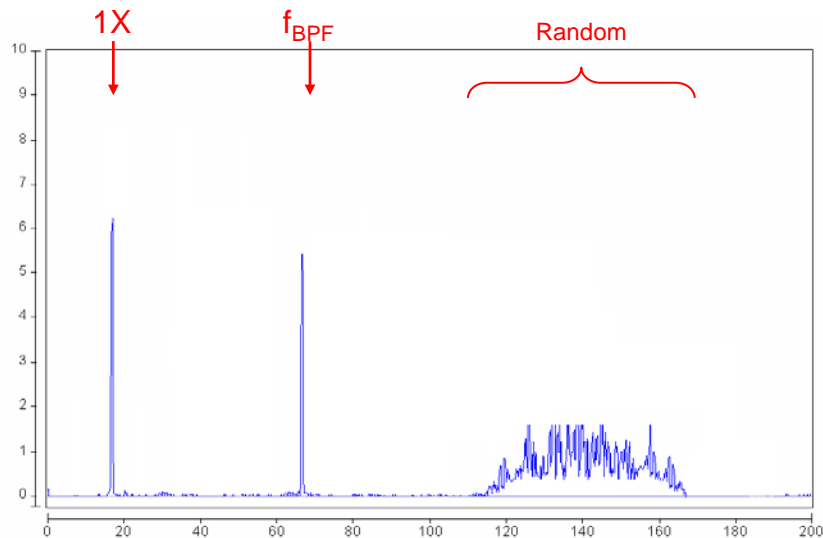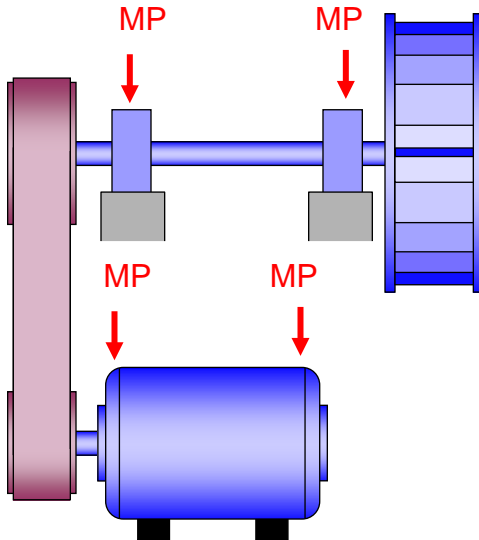
Belt transmission a common drive system in industry consisting of:

- **Driver Pulley**

- **Driven Pulley**

- **Belt**

The dynamic relation is: Ø$_1$ ω$_1$ = Ø$_2$ ω$_2$

Ø$_1$ ω$_1$ ω$_2$ Ø$_2$

Belt frequency:

$$f_B = \frac{3,1416\,\omega_1\phi_1}{l}$$

l: **belt length**

Belt Drive Faults

Belt Worn:

f_n

$1X, 2X, 3X f_B$
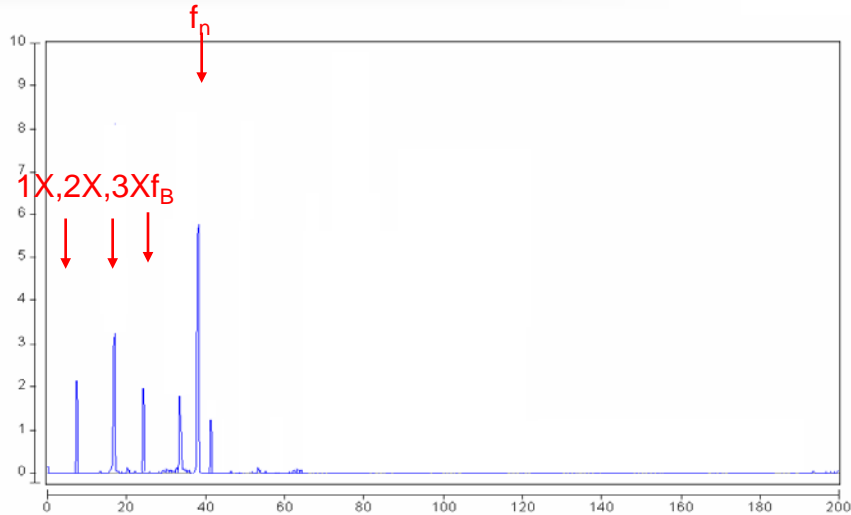
The belt frequency f_B and first two (or even three) harmonics are visible in the spectrum.

2 f_B generally dominates the spectrum

Pulley Misalignment:

Offset   **Angular**   **Twisted**
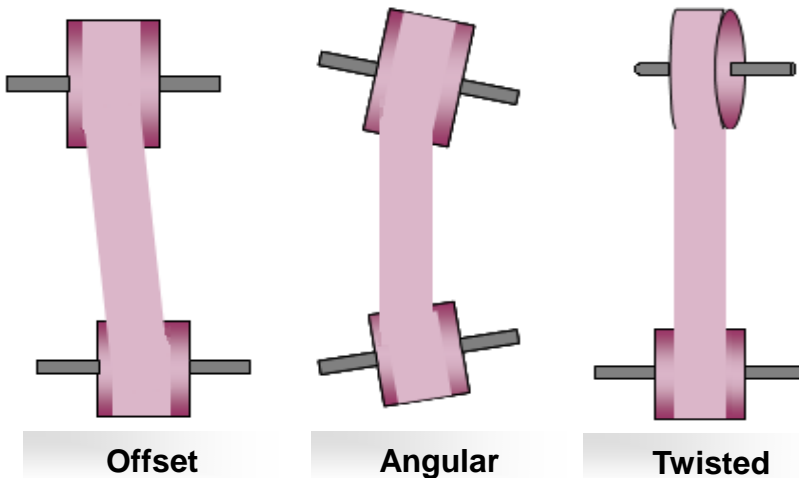
1X of diver or driven pulley visible and predominant in the axial reading.

Belt Drive Faults

Eccentric Pulleys:
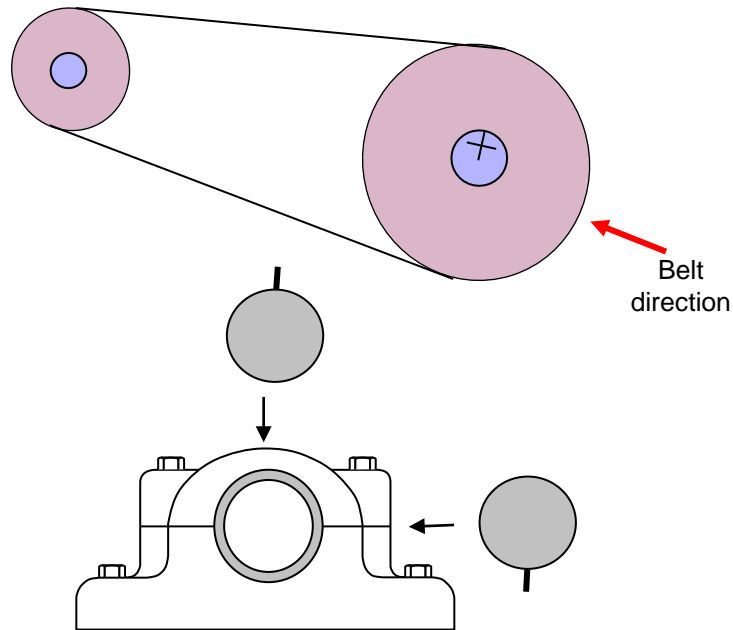
Belt direction

The geometric center doesn't coincide with the rotating center of the pulley.

High 1X of the eccentric pulley visible in the spectrum, predominant in the radial direction.

Easy to confuse with unbalance, but:

• Measurement phase in vertical an horizontal directions may be 0° or 180°.

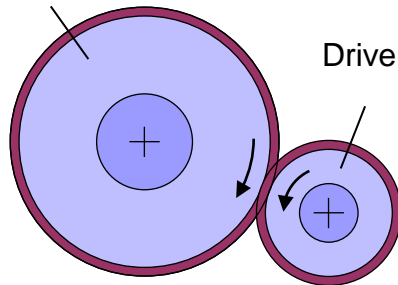**• The vibration may be higher in the direction of the belts.**

Belt Resonance:
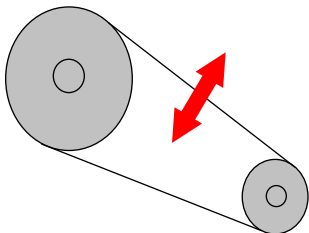
If the belt natural frequency coincides with either the driver or driven 1X, this frequency may be visible in the spectrum.

Gear Faults

Spur Gear:

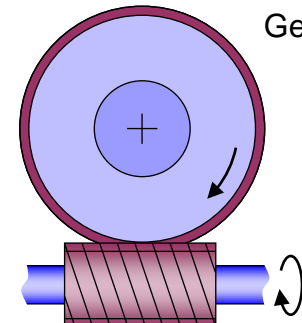Driving gear

Driven gear

Gear (wheel)   Pinion

gear wheel

gear wheel pair

gear train
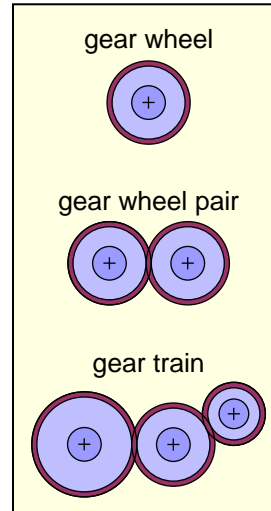
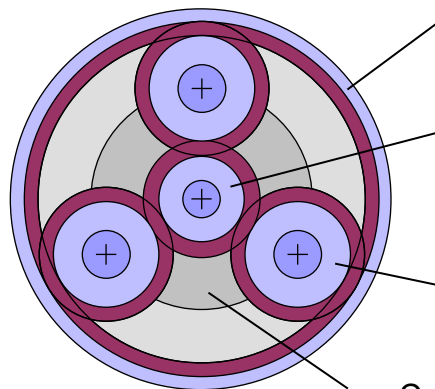**Worm Gear:**

Gear

Worm gear

Planet Gear:

Ring (cone)

Sun gear

Planet gear

Carrier

Bevel Gear:

Bevel gear

Bevel gear

Gear Faults

Gear Meshing:

Left flank

Working flank

End point of tooth meshing

85

89

Right flank

88

86

4

87

3

5

6

2

1

Starting point of tooth meshing

Pitch point

Non working flank

Pitch line
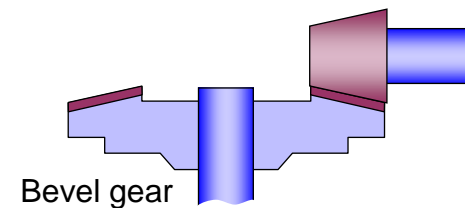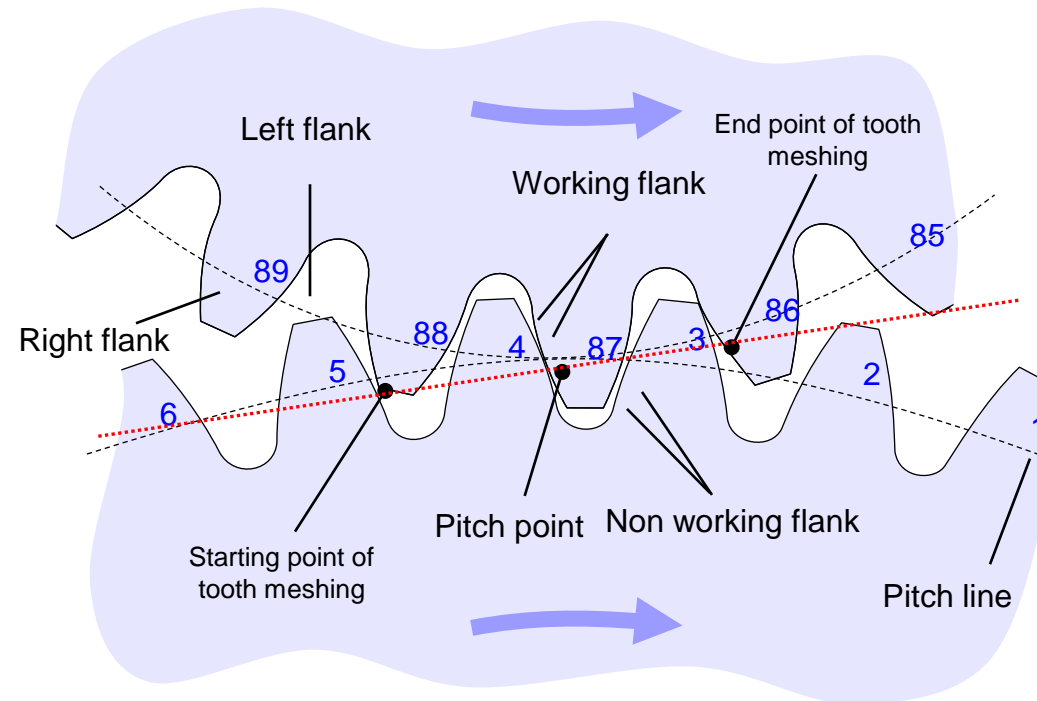
Gear meshing is the contact pattern of the pinion and wheel teeth when transmitting power.

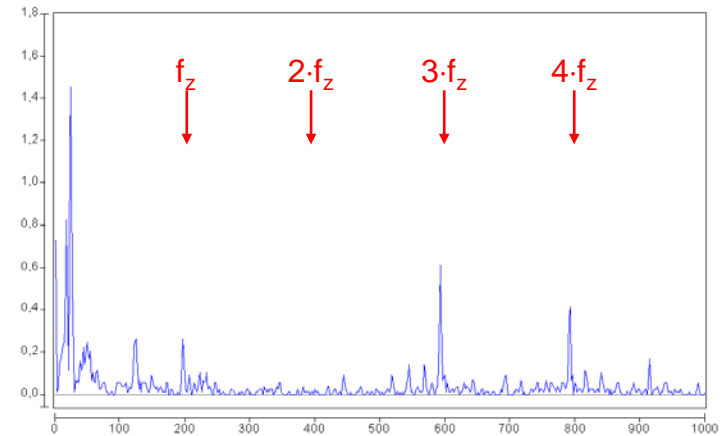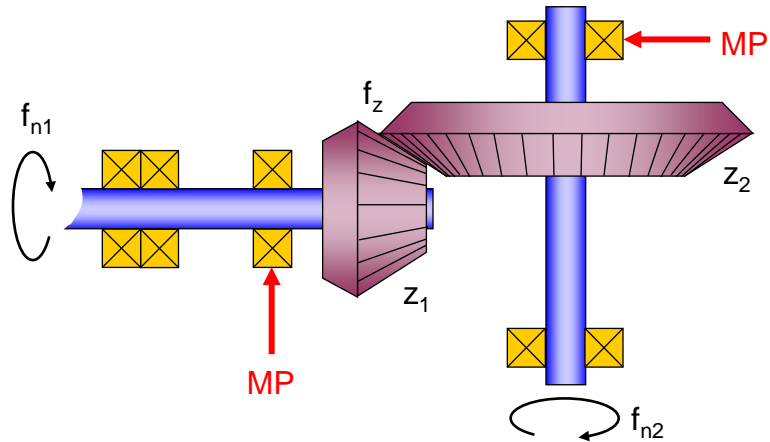**The red dotted line is the contact path where the meshing teeth will be in contact during the rotation.**

Flank line

Top land

Tooth space

Tip edge

Pitch surface

Tooth

Root flank

Root mantel flank

Flank profile

Gear mesh frequency f_Z can be calculated:

$$F_z = z\, f_n$$

Where z is the number of teeth of the gear rotating at f_n.

Gear Faults
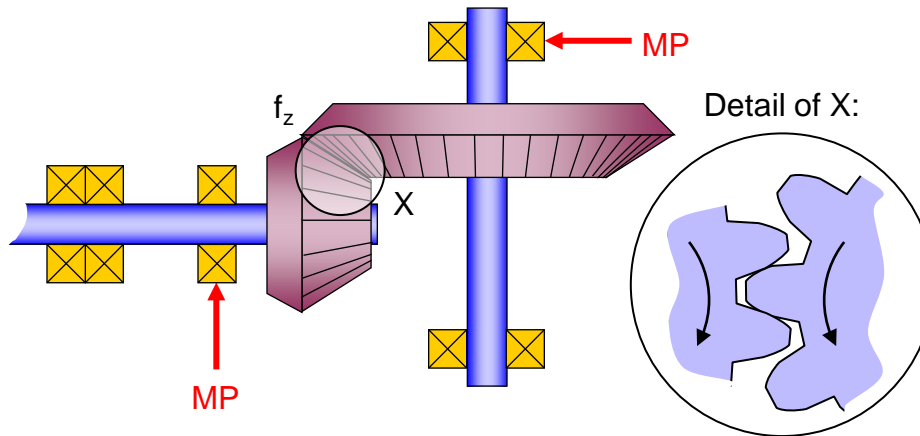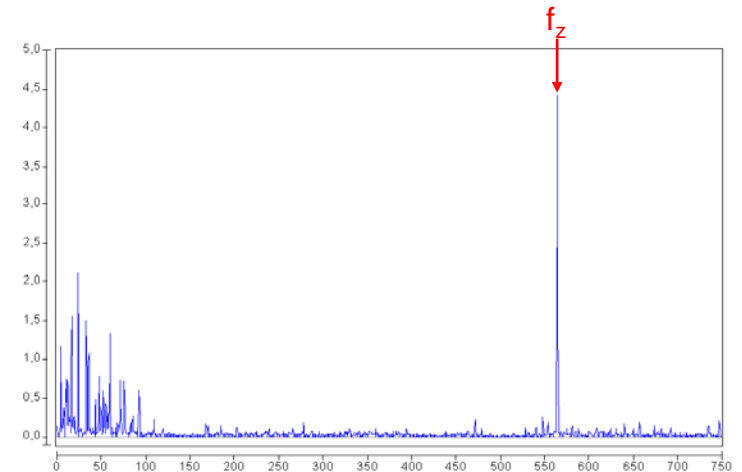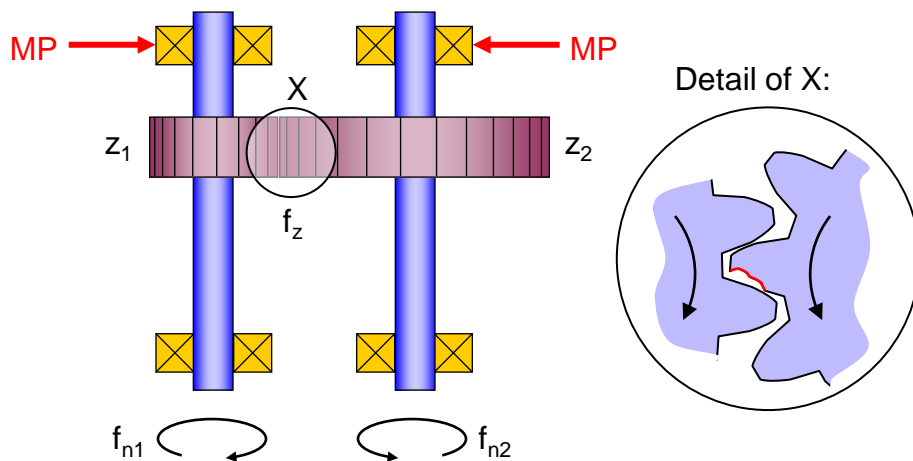
Gear Faults

Incorrect tooth shape

Detail of X:

Tooth break-out

Detail of X:

Gear Faults
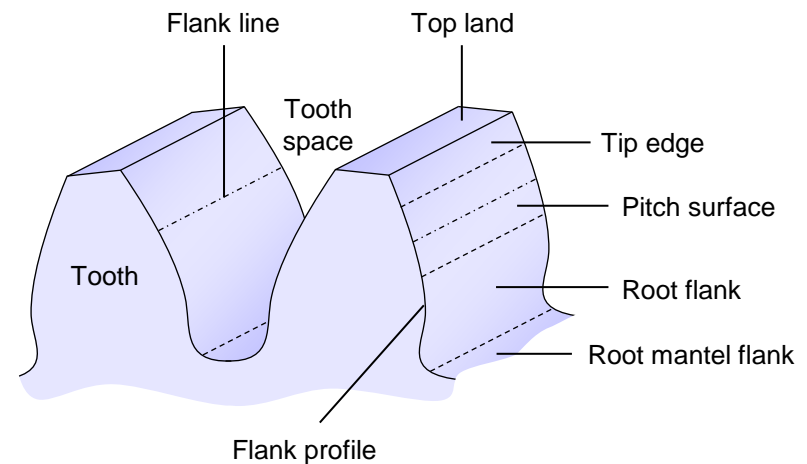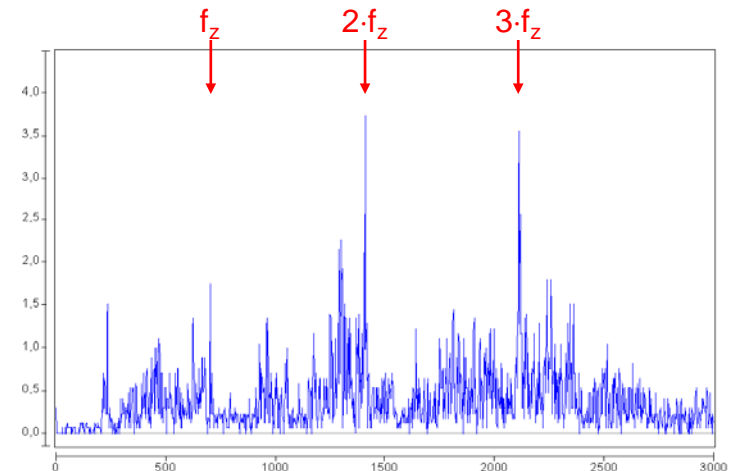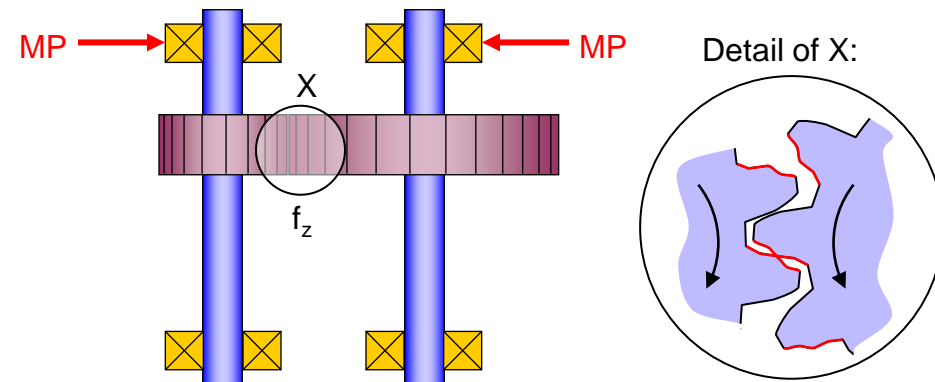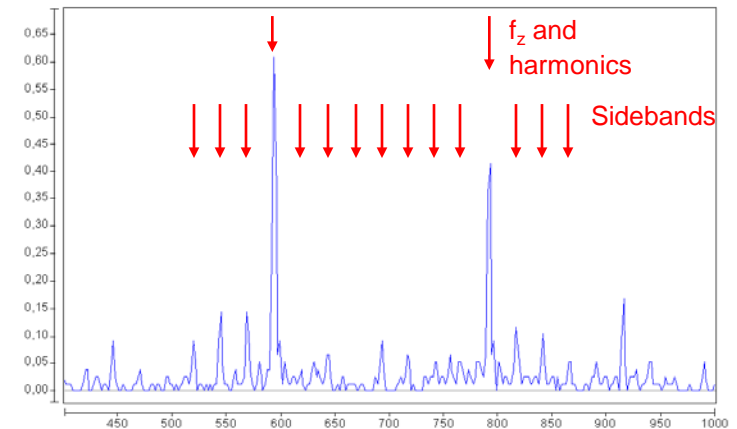
Eccentricity, bent shafts

MP → ← MP

X

f_z

Detail of X:

f_z and harmonic sidebands
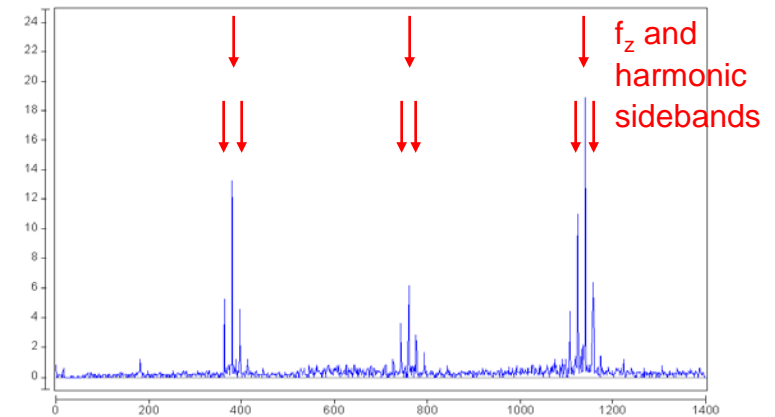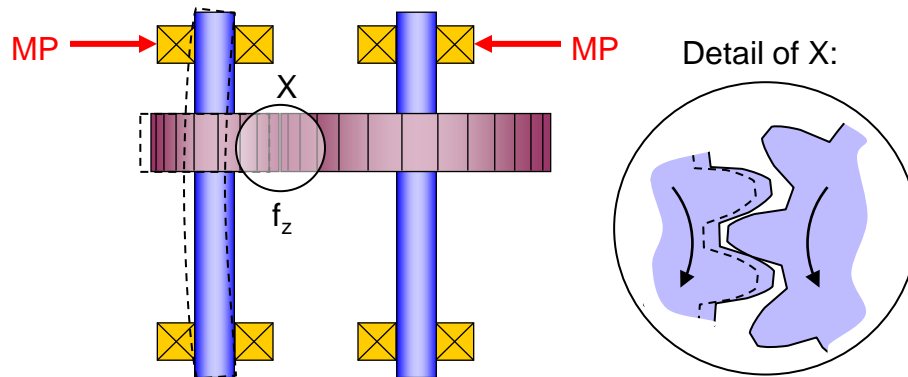
"Ghost frequencies" or machine frequencies

Gearwheel being manufactured

Cutting tool

Worm drive part of the gear cutting machine

z_M

f_Z f_M "Ghost frequency"

Electrical Motors
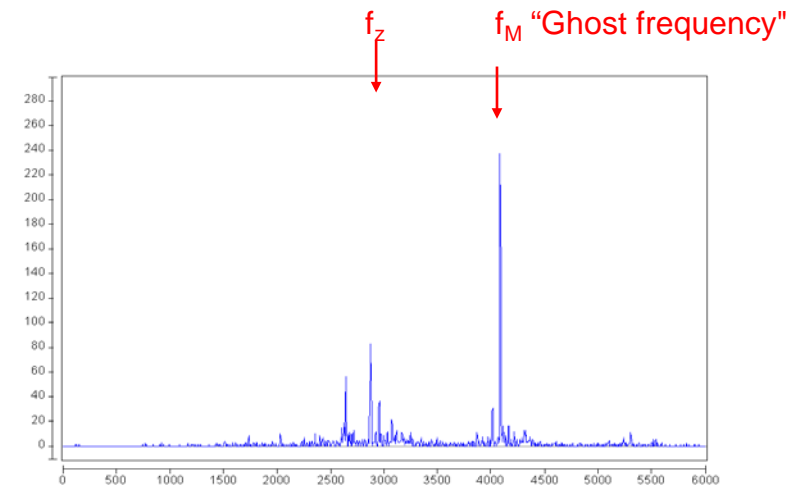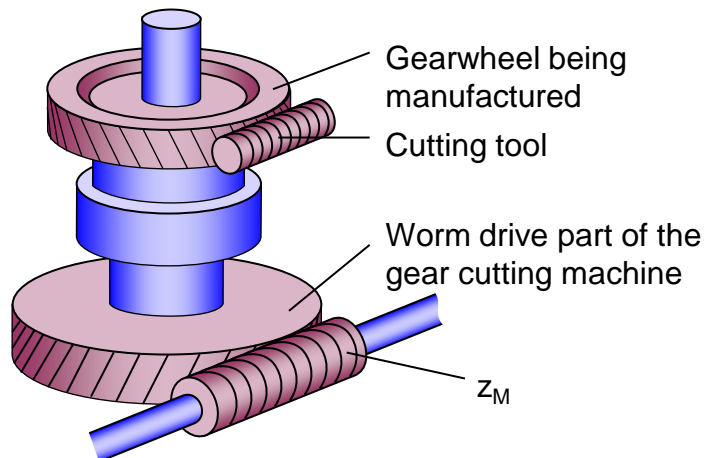
Electromagnetic forces vibrations:

Twice line frequency vibration: $2 \cdot f_L$

Bar meshing frequency: $f_{bar} = f_n \cdot n_{bar}$

Synchronous frequency: $f_{syn} = 2 \cdot f_L / p$

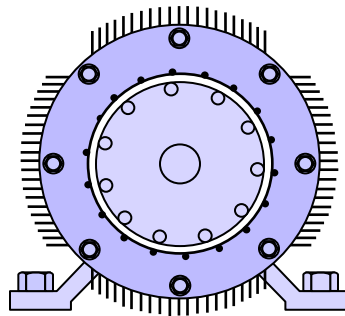**Slip Frequency: $f_{slip} = f_{syn} - f_n$**

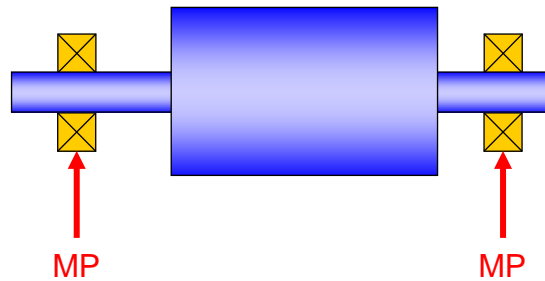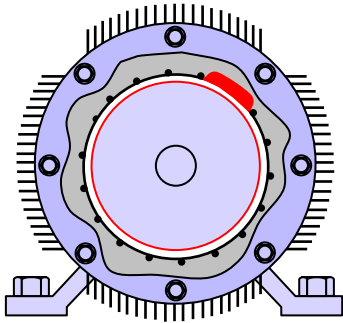**Pole pass frequency: $f_p = p \cdot f_{slip}$**

f_L: line frequency

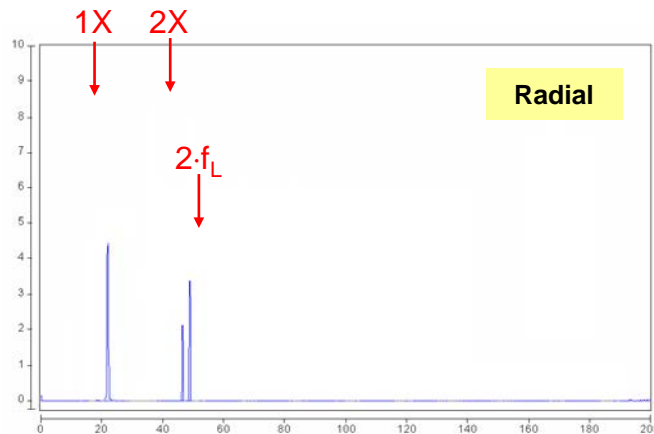n_{bar}:   number of rotor bars

p: number of poles

- **Stator eccentricity**
- **Eccentric rotor**
- **Rotor problems**
- **Loose connections**

Stator Eccentricity:

- **Loose iron**
- **Shorted stator laminations**
- **Soft foot**

MP   MP

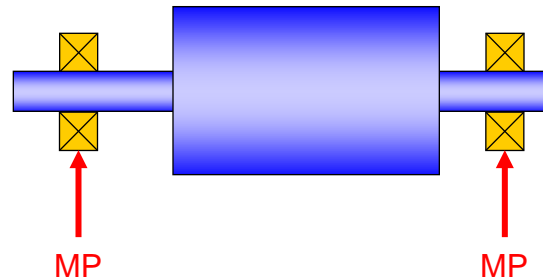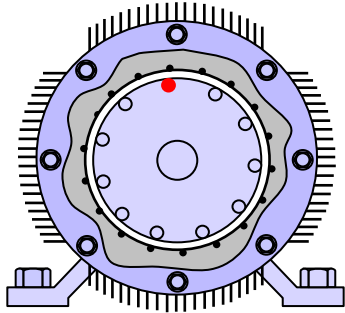

Radial

1X   2X

$2 \cdot f_L$

1X and 2X signals

f_L **without sidebands**

Radial predominant

High resolution should be used when analyzing two poles machines.

Electrical Motors

Eccentric Rotor:

- Rotor offset
- Misalignment
- Poor base
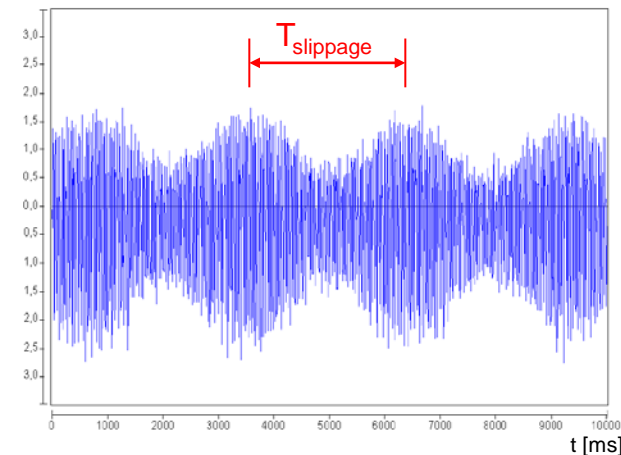
MP   MP
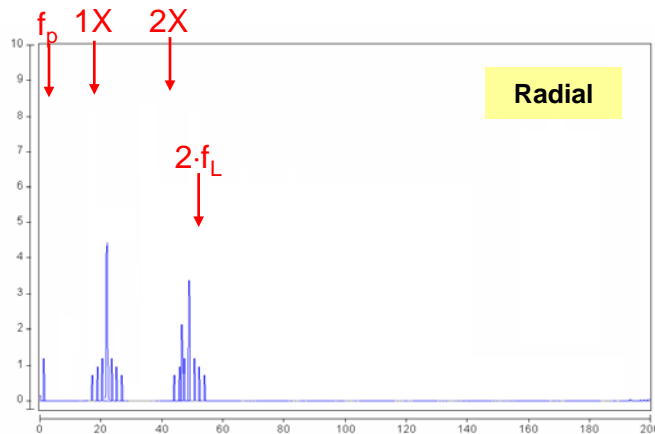
f_p, 1X, 2X and $2f_L$ signals.

1X and $2f_L$ with sidebands at f_P.
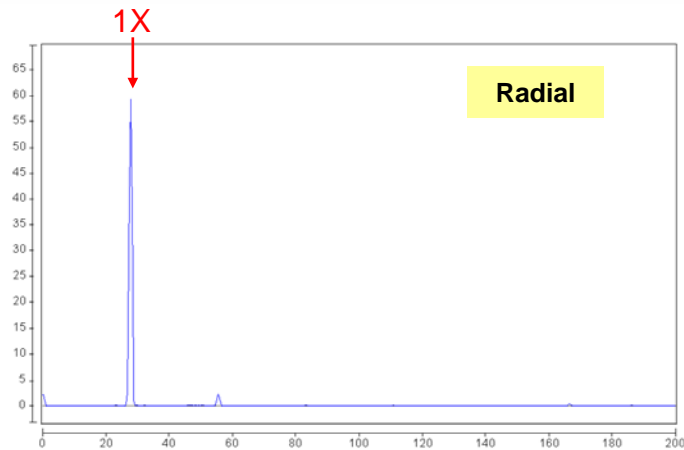
Radial predominant.

High resolution needed.

Modulation of the vibration time signal with the slip frequency f_{slip}
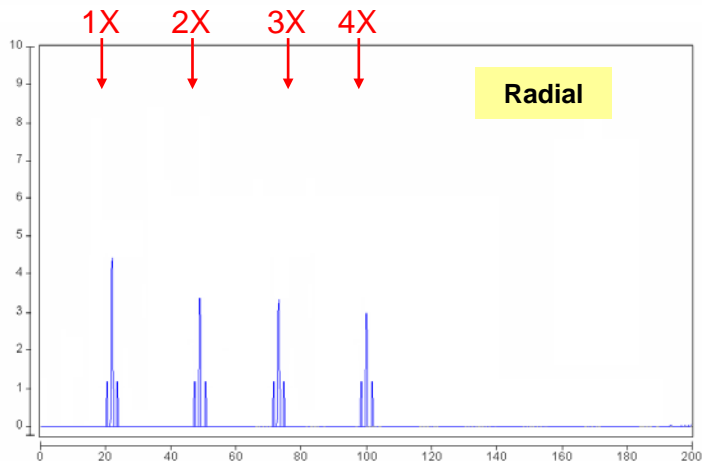
$T_{slip} \approx 2\text{-}5$ s

Electrical Motors

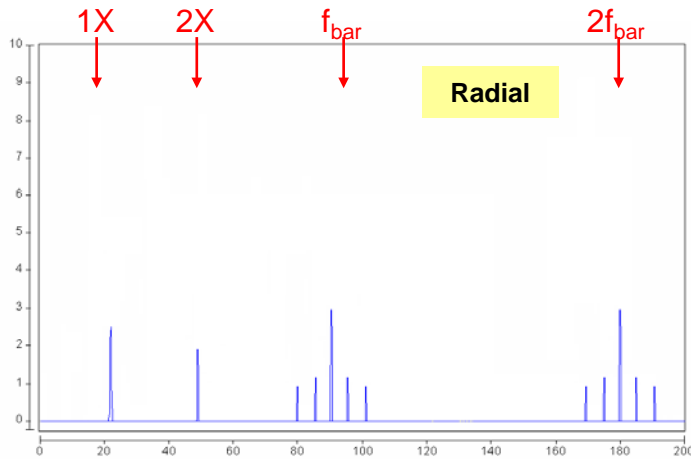**Rotor Problems:**

1. Rotor thermal bow:

- **Unbalanced rotor bar current**
- **Unbalance rotor conditions**
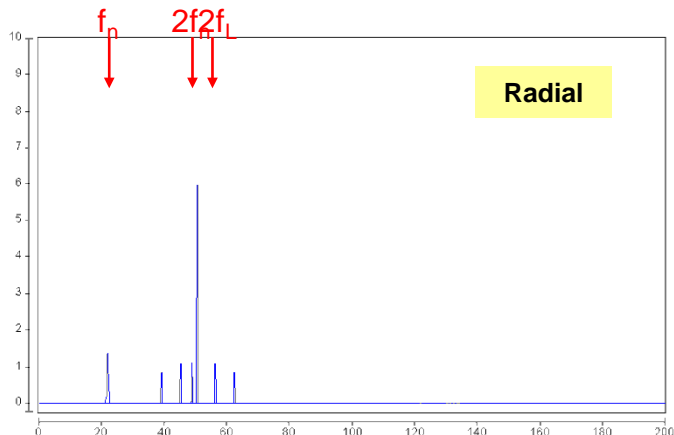- **Observable after some operation time**

2. Broken or cracked rotor bars:

- **1X and harmonics with sidebands at f_P**
- **High resolution spectrum needed**
- **Possible beating signal**

Electrical Motors

3. Loose rotor bar:

- f_{bar} and $2f_{bar}$ with $2f_L$ sidebands
- $2f_{bar}$ can be higher
- 1X and 2X can appear

Loose connections:

- $2f_L$ excessive signal with sidebands at 1/3 f_L
- Electrical phase problem
- Correction must be done immediately